

# Thermowell calculations



**Dirk Bauschke**

Engineering Manager

**David Wiklund**

Senior Principal Engineer

**Andrew Kitzman**

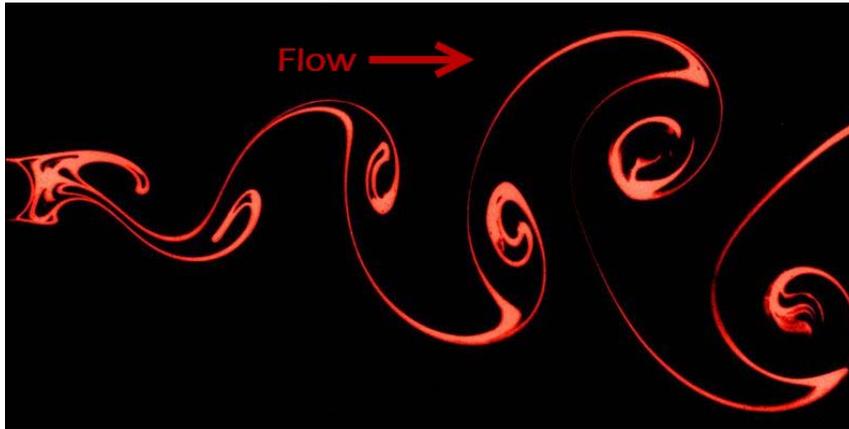
Mechanical Design Engineer

**Danjin Zulic**

Marketing Engineer

## 1.1 Introduction

Thermowells are essentially a circular cylinder installed like a cantilever into the process piping. They provide process condition protection and a process seal for temperature sensors. As a process fluid passes around the thermowell, low pressure vortices are created on the downstream side in laminar, turbulent, and transitional flow. The combination of stresses, generated by the static in-line drag forces from fluid flow and the dynamic transverse lift forces caused by the alternating vortex shedding, create the potential for fatigue-induced mechanical failures of the thermowell. Piping designers may use a variety of tools to predict and avoid thermowell failures in their systems, but ASME PTC 19.3-1974 had been the standard by which most thermowells were designed.



Color enhanced smoke trail showing von Karman Vortex Street in laminar fluid flow<sup>(1)</sup>

## 1.2 Brief history of ASME PTC 19.3

The standard dates back to 1957 when ASME (American Society of Mechanical Engineers) determined that the 1930's Supplement on Temperature Measurement was unsatisfactory because it did not include thermal and stress effects. ASME asked the Boiler and Pressure Vessel Committee to create a document, but it was deemed outside their scope. A stand-alone committee was then charged with all of temperature measurement with thermowell design as a section. The basis for ASME PTC 19.3-1974 was a paper authored by J.W. Murdock (1959).<sup>(2)</sup>

John Brock of the Naval Post Graduate School conducted some follow-on work in 1974 that uncovered several items that Murdock either assumed or ignored. Brock suggested such ideas as using a variable Strouhal Number rather than a fixed Strouhal Number, applying installation factors in the approximation of the natural frequency of the thermowell, and reviewing the frequency ratio limit of 0.8 to account for uncertainty in the natural frequency calculations<sup>(3)</sup>. Some of these demonstrated that there could be improvements made to ASME PTC 19.3-1974.

(1) Wikipedia [http://en.wikipedia.org/wiki/Vortex\\_induced\\_vibration](http://en.wikipedia.org/wiki/Vortex_induced_vibration) as of 5/20/2011

(2) Murdock, J.W., "Power Test Code Thermometer Wells" *Journal of Engineering for Power* (1959).

(3) Brock, John E., "Stress Analysis of Thermowells," *Naval Postgraduate School, Monterey CA* (1974).

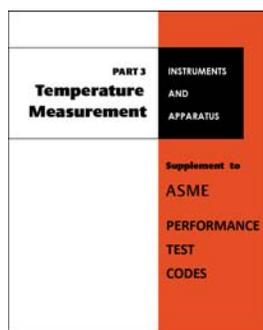
ASME PTC 19.3-1974 did not seem to account for all installations. An example of a high profile catastrophic thermowell fatigue failure came when the Monju (Japan) Fast Breeder reactor was shutdown due to a leak in a liquid sodium coolant system in 1995. The investigation revealed that the thermowell was designed in accordance to ASME PTC 19.3-1974 but the failure mode was due to the in-line resonance, which is not accounted for in the standard. The result was the development of the Japanese version of the standard, called JSME S012<sup>(1)</sup>. The reactor was eventually restarted in May of 2010 after years of investigation and legal battles.

For the most part, though, ASME PTC 19.3-1974 was used successfully in both steam and non-steam applications. Several key factors caused ASME to re-form the committee in 1999 to completely rewrite the standard; advances in the knowledge of thermowell behavior, a number of catastrophic failures (Monju among them) and the increased use of Finite Element Analysis for stress modeling. When combined, these factors caused many in the industry to move away from the rudimentary methods and simplified tables laid out in ASME PTC 19.3-1974 in favor of more advanced methods for predicting the thermowell natural frequency and calculating the forced frequency.

Rather than simply update the existing version of ASME PTC 19.3-1974, the committee decided to release a new standard due to the significant changes associated with the effort. The thermowell calculation portion of ASME PTC 19.3-1974 was 4 pages. By comparison, the new standard, known as ASME PTC 19.3 TW-2010 (“TW” for thermowell), is over 40 pages due to the explanations of theory and the sheer complexity of the process.

ASME PTC 19.3 TW-2010 was released in July 2010.

## 1.3 ASME PTC 19.3-1974 methodology



As previously stated, the 1974 standard is very brief. It allows few stem profiles and uses simplified equations to model the thermowell for natural frequency calculations. Even though it allows any attachment method that is approved by the ASME Boiler and Pressure Vessel and Piping Codes, the equations do not differentiate between common mounting style variations such as flanged, threaded, and socket weld, and ignores the effects of different stem profiles, such as straight, taper, and stepped. Bore dimensions not in the tables are not accounted for, so bores for 1/4-in. and 6 mm diameter sensors share the same constants in the equations and no constants are provided for 3 mm diameter sensor bores.

For all its drawbacks, though, ASME PTC 19.3-1974 does have a simple process for thermowell evaluation that helped make it widely accepted in the industry; gather the process data and the thermowell materials information, calculate the natural and Strouhal Frequency, compare the ratio to 0.8, calculate the bending stress, compare maximum pressure to process pressure, and check the maximum length to the desired length.

Gathering the process data and materials information is a straightforward step except that there is one piece of data that is no longer readily available. The “Ratio of frequency at process temperature to frequency at 70 °F” is not easily found.

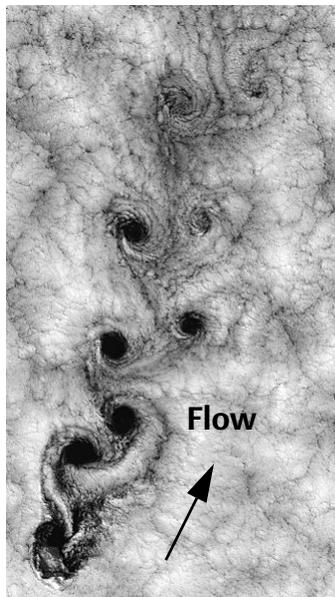
The method of calculating the thermowell natural frequency uses a simple equation, but some of the terms, such as  $K_f$ , are not well defined. If the thermowell U-length does not match one of

(1) Odahara, Sanoru, et al. “Fatigue Failure by In-line Flow-induced Vibration and Fatigue Life Evaluation,” *JSME International Journal, Series A, Vol. 48, No. 2 (2005)*.

those listed in the table, the designer should use the data for the length longer than the thermowell to be conservative. For an acceptable thermowell design, the ratio of the Strouhal frequency and the natural frequency “shall not exceed” 0.8.

The final step is an evaluation of the thermowell length based on the steady state stress. This determines the maximum length the thermowell can be in order to handle the bending stress. This length is compared to the desired length to determine if it is acceptable or if it must be shortened.

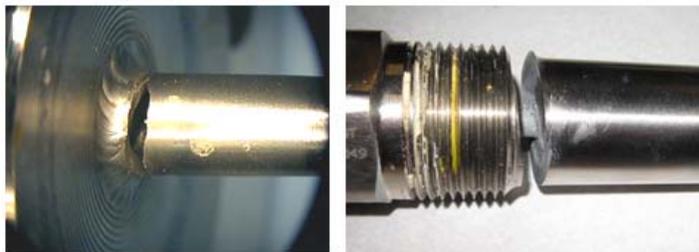
## 1.4 Vortex shedding theory (basis for ASME PTC 19.3 TW-2010)



When a fluid flows around a blunt object in its path, vortices are formed downstream of the object. This is commonly referred to as vortex shedding, Von Karman Vortex Street, or flow vortices. The vortices are low pressure cells that are created and shed downstream in an alternating pattern. The differential pressure due to the alternating vortices produces alternating forces on the object. This results in alternating stresses on the object as it deflects. This phenomenon is observed in nature as eddies in the current downstream of bridge piers, swirls in the clouds downwind of the peaks of mountains, or Aeolian tones heard as the wind passes around utility lines. While vortex shedding is useful for process flow measurements, thermowell designers should avoid it due to the potential for failure.

Landsat 7 image of a von Karman Vortex Street in the clouds off the Chilean coast near the Juan Fernandez Islands (15 Sept 99) <sup>(1)</sup>

Because the major cause of thermowell failure is fatigue due to resonance, the designer needs to understand vortex shedding in order to avoid its effects and predict the vortex shedding frequency. Since vortex shedding occurs at frequencies anywhere from about 50Hz to 1500Hz, the thermowell can experience a large number of cycles in a short amount of time.



Examples of thermowell failures due to fatigue

As the vortex shedding frequency, or Strouhal Frequency, approaches the thermowell natural frequency, the tip displacement and stresses are greatly magnified and the thermowell can fail due to the large amount of energy it must absorb. So, in addition to process conditions such as

(1) NASA Earth Observatory Website  
["http://www.dvidshub.net/image/842014/landsat-7-reveals-large-scale-fractal-motion-clouds-image-day"](http://www.dvidshub.net/image/842014/landsat-7-reveals-large-scale-fractal-motion-clouds-image-day)

pressure, temperature, and corrosion, the designer must account for the high cycle fatigue strength for overall suitability in the application.

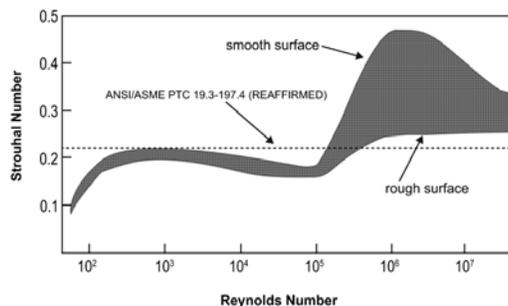
### 1.4.1 Minimum velocity

For slow flowing process fluids, there is not enough energy transferred from the process fluid to the thermowell to cause fatigue failure. If the following conditions are met, there is no need to conduct frequency limit calculations as the risk of thermowell failure is negligible.

1. Process Fluid Velocity,  $V < 0.64$  m/s (2.1 ft./sec)
2. Wall Thickness,  $(A - d) \geq 9.55$  mm (0.376 in)
3. Unsupported Length,  $L \leq 0.61$  m (24 in)
4. Root and Tip Diameter ( $A$  and  $B$ )  $\geq 12.7$  mm (0.5 in)
5. Maximum Allowable Stress,  $S \geq 69$  Mpa (10 ksi)
6. Fatigue Endurance Limit,  $S_f \geq 21$  Mpa (3 ksi)

Even so, these low velocities could still excite the in-line resonance and cause sensor failure due to the high vibration that exists at resonance. If these criteria are not met, or if there is a chance of stress corrosion or material embrittlement due to fluid interaction (which would cause a change to the fatigue endurance), the designer must fully evaluate the thermowell design.

### 1.4.2 Strouhal Number



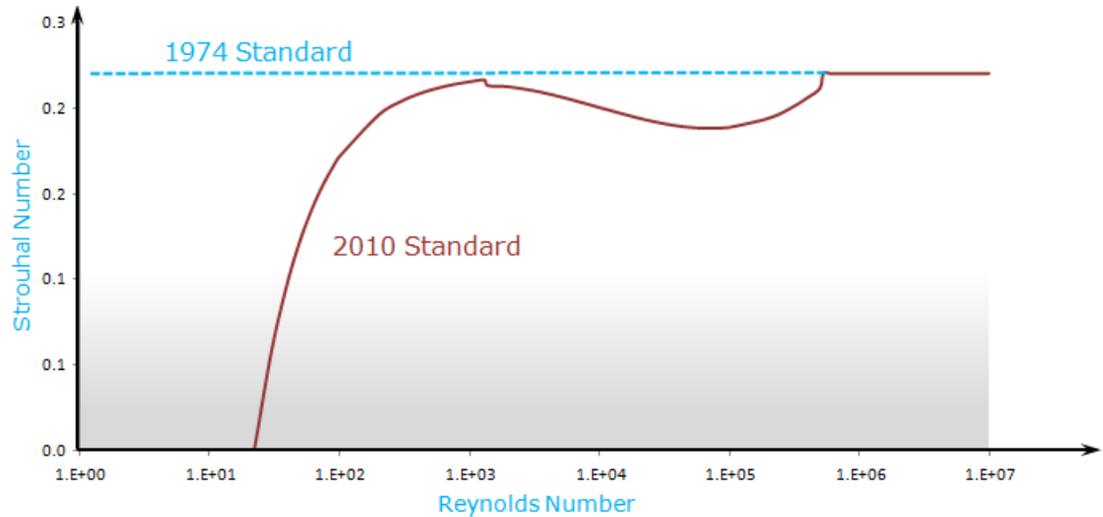
There has been much discussion on the topic of whether to use a fixed or variable Strouhal Number. ASME PTC 19.3-1974 used a fixed Strouhal Number of 0.22 while Brock recommended a variable Strouhal Number depending on the Reynolds Number. Many in the industry began to incorporate the variable Strouhal Number to the vortex shedding frequency equations within the framework of ASME PTC 19.3-1974 calling it “the Brock Method” or something similar.

Typical chart showing Strouhal Number as a function of Reynolds Number

The ASME PTC 19.3 TW-2010 committee reviewed the subsequent experiments before deciding on how to use the variable Strouhal Number. Two papers published in the JSME International Journal in 2001 showed interesting test results for machined straight and tapered cylinders that were similar to thermowells in form. The forces and vibration amplitudes were measured while the cylinders were immersed in a fluid flow. The conclusion was that the evidence of a high Strouhal Number in previous experiments was based on measurements of the vortex shedding and not of the actual forces on the thermowell.<sup>(1)(2)</sup>

- (1) Sakai, T., Iwata, K., Morishita, M., and Kitamura, S., “Vortex-Induced Vibration of a Circular Cylinder in Super-Critical Reynolds Number Flow and Its Suppression by Structure Damping,” *JSME Int. J. Ser. B.* 44, 712-720 (2001).
- (2) Iwata, K., Sakai, T., Morishita, M., and Kitamura, S., “Evaluation of Turbulence-Induced Vibration of a Circular Cylinder in Super-Critical Reynolds Number Flow and Its Suppression by Structure Damping,” *JSME Int. J. Ser. B.* 44, 721-728 (2001).

“Rough” surfaces were defined in the experiments as measuring in excess of 128 Ra. No thermowell in the process industry has a surface finish of more than 32 Ra and the stress limits and calculations in ASME PTC 19.3 TW-2010 are not valid for surface finishes rougher than 32 Ra.

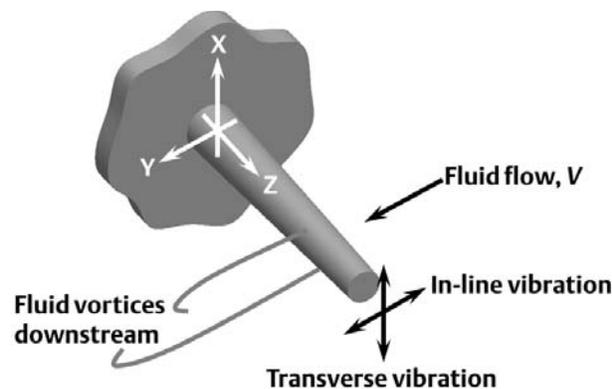


Based on this data, the ASME PTC 19.3 TW-2010 committee decided to incorporate a variable Strouhal Number defined by the rough cylinder curve. To simplify calculations, the designers are also allowed to conservatively approximate the Strouhal Number as 0.22. This is especially useful if the designer cannot establish the dynamic or kinematic fluid viscosity to determine the Reynolds Number.

### 1.4.3 Reynolds Number

In any fully immersed flow, a fundamental parameter is the Reynolds Number. The Reynolds Number is the ratio of the inertial forces to the viscous forces in the flow field. For the purposes of vortex shedding elements, the length input for the Reynolds Number is the width of the shedding element. In the case of thermowells, this is the tip diameter.

### 1.4.4 Thermowell natural frequency



ASME PTC 19.3 TW-2010 models the thermowell as a simple cantilever beam and applies a series of correction factors to account for the differences from the ideal beam by including added fluid mass, added sensor mass, non-uniform profile beam, and mounting compliance. For stepped stem thermowells, most all the correlations and calculations are more complex due to the geometry and stress concentration points.

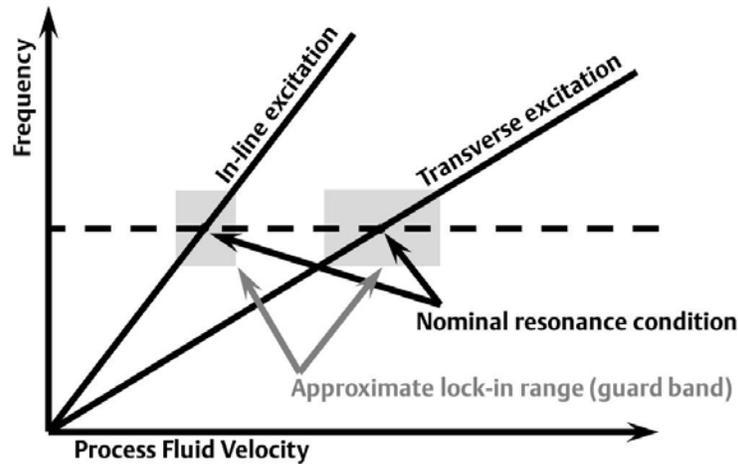
Fluid induced forces and assignment of axes for calculation of thermowell stresses

Because of this, ASME PTC 19.3 TW-2010 restricts the dimensional variation of stepped stem thermowells considered within the scope of the standard.

After all the correction factors are applied, the “in-situ”, or installed natural frequency,  $f_n^c$ , is calculated and used for the rest of the frequency analysis.

## 1.4.5 Critical velocities

Once the thermowell natural frequency has been established, the designer needs to set the margin of safety between the natural frequency and the Strouhal frequency.



In-line and transverse excitation schematic showing “lock-in” region

There are actually two modes of thermowell excitation. The transverse (lift) force that causes the thermowell to vibrate perpendicular to the flow while the in-line (drag) force causes the thermowell to vibrate parallel to the flow. The in-line vibration is approximately twice the frequency of the transverse. The in-line “velocity critical” (where the Strouhal Frequency equals the natural frequency) is approximately half the velocity as the transverse. ASME PTC 19.3-1974 does not address the in-line vibration, only the steady state bending stress.<sup>(1)</sup>

While the change in the shedding frequency is proportional to the fluid velocity, the thermowell locks-in to the resonance frequency very easily. It can also take a considerable change in velocity to get the thermowell out of shedding vortices at its natural frequency. Since the damping of typical thermowells is very low, it is vital to stay out of resonance. At resonance the forces and displacements are greatly magnified.

$$f_s < 0.8 f_n^c$$

The 20% guard band accounts for the significant variability due to:

- the non-linearity of the thermowell elastic response
- loose thermowell manufacturing tolerances
- material property information established to only 3 significant digits
- minor, routine variations in flow rate, temperature, density, or viscosity in the process

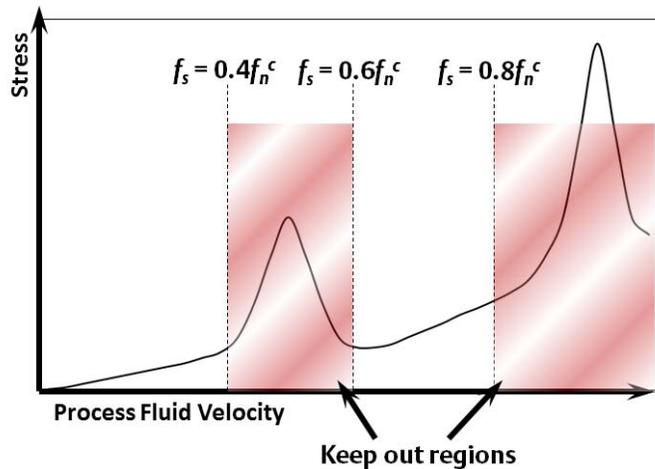
Since the in-line vibration occurs at roughly half the velocity of the transverse (or twice the frequency), liquids have further limitations.

$$2f_s < 0.8 f_n^c$$

(1) ASME Standard, Performance Test Codes 19.3-1974 (Reaffirmed 1998).

Viewing this a bit differently gives a wider perspective on where thermowell operation is allowed.

$$f_s \text{ (steady state)} < 0.4 f_n^c \quad \text{or} \quad 0.6 f_n^c < f_s \text{ (steady state)} < 0.8 f_n^c$$



Graph depicting the amplitude response of a thermowell to fluid-induced forces

ASME PTC 19.3 TW-2010 also contains a provision for “super critical” operation where the thermowell is operated above the thermowell natural frequency. Emerson strongly discourages operating thermowells in this region.

## 1.4.6 Scruton Number

New to the theory is the use of the Scruton Number, which represents the intrinsic damping of the thermowell. ASME PTC 19.3 TW-2010 takes a very conservative perspective and sets the damping factor to 0.0005 unless it is otherwise determined.

A Scruton Number less than 2.5 means that there is no intrinsic damping and the thermowell must be evaluated at the in-line resonance frequency and stay away from the transverse resonance frequency. As the Scruton Number increases, there is an increased level of intrinsic damping that reduces the deflections and, therefore the stresses. An acceptable level of damping will allow the thermowell to operate at the in-line and maybe even the transverse resonance frequencies.

If the conditions are such that the thermowell will be operating above the natural frequency, higher order resonances must be considered, but ASME PTC 19.3 TW-2010 does not provide any guidance in this and Emerson strongly discourages operating thermowells in this region.

## 1.5 Bending and pressure stress (as used in ASME PTC 19.3 TW-2010)

While it seems that there is a lot of attention being given to the vortex shedding theory and application, the stresses within the thermowell and forces applied are also critical to evaluating suitability for specific process applications. In contrast to the simple method in the 1974 version, ASME PTC 19.3 TW-2010 takes a much more detailed look at both the frequency and the stresses on the thermowell. This allows a wider variety of mounting styles, profiles, and bore sizes that reflect the offerings available in the industry today.

In total, there are 4 quantitative criteria in ASME PTC 19.3 TW-2010 for a thermowell to be found acceptable for a particular set of process conditions:

1. **Frequency Limit:** the resonant frequency of the thermowell must be sufficiently high so that destructive oscillations are not excited by the fluid flow.
2. **Dynamic Stress Limit:** the maximum primary dynamic stress must not exceed the allowable fatigue stress limit. If the design requires that the thermowell pass through the in-line resonance to get to the operating conditions, there is an additional fatigue check at resonance.
3. **Static Stress Limit:** the maximum steady-state stress on the thermowell must not exceed the allowable stress, as determined by the Von Mises criteria.
4. **Hydrostatic Pressure Limit:** the external pressure must not exceed the pressure ratings of the thermowell tip, shank, and flange (or threads).

In addition, the suitability of the thermowell material for the process environment must be considered. This means the designer must evaluate how corrosion and erosion affects the thermowell as well as how exposure to the process conditions affects material properties.

### 1.5.1 Frequency limit

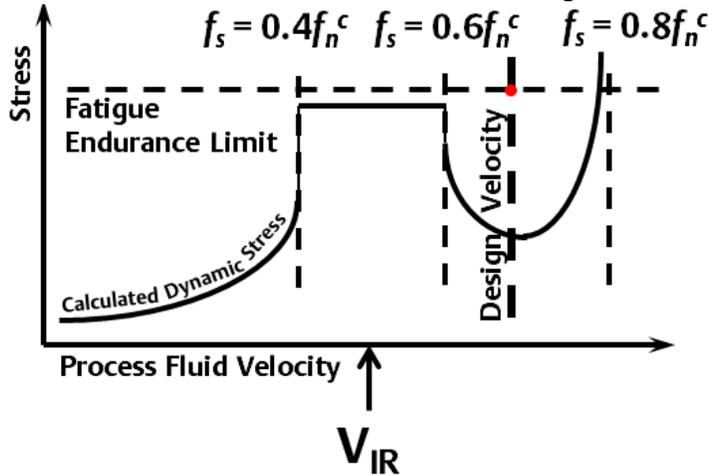
The vortex shedding theory section discusses the ASME PTC 19.3 TW-2010 method for Strouhal Frequency calculation. If the Strouhal Frequency is between the in-line critical frequency lock-in band and the transverse critical frequency lock-in band, and the Scruton Number evaluation indicates insufficient damping, the thermowell design must be modified unless all of the following conditions are met:

1. The process fluid is a gas.
2. The thermowell passes through in-line resonance only during start-up, shut-down, or otherwise infrequently during operation.
3. The peak stress at resonance is less than the fatigue limit of the material.
4. The process fluid does not cause the material properties to change (esp. fatigue resistance).
5. The consequences of thermowell failure are an acceptable risk.

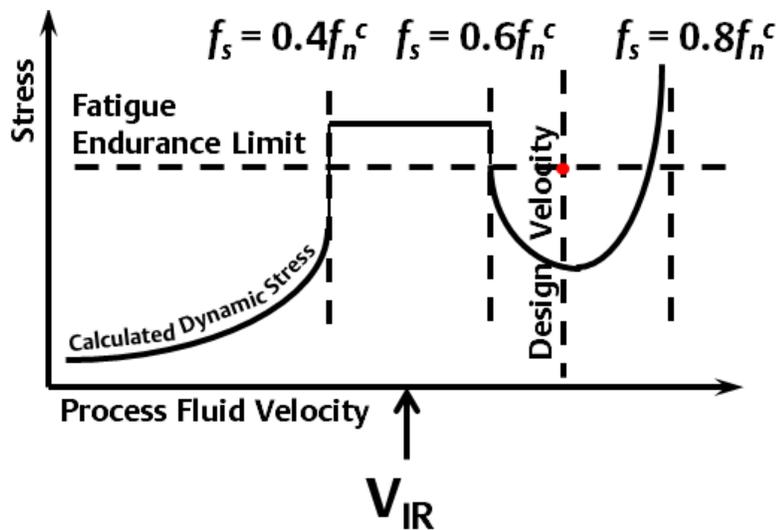
### 1.5.2 Passing through in-line resonance

If the thermowell Peak Oscillatory Bending Stress is less than the Fatigue Endurance Limit at the in-line velocity critical, then the thermowell may pass through the in-line resonance lock-in region on the way to the steady state design velocity. Steady state velocities within the in-line resonance

lock-in region are not allowed due to the high number of fatigue cycles imposed on the thermowell as well as the increased likelihood of sensor damage.



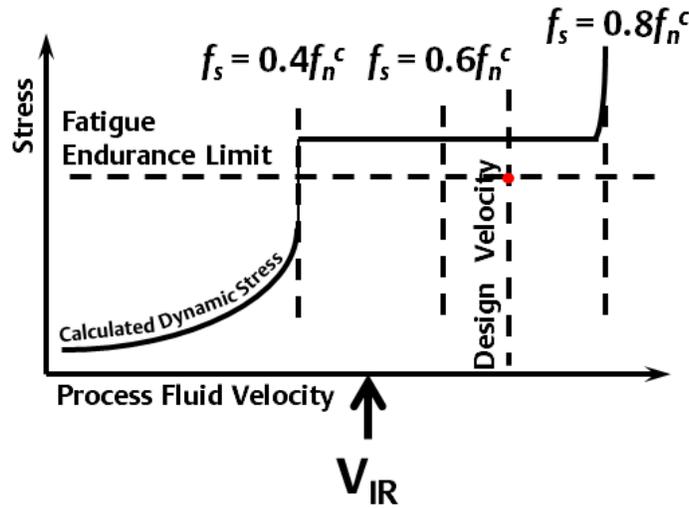
Example chart showing thermowell design that passes in-line resonance evaluation



Example chart showing thermowell design that does not pass in-line resonance evaluation (this design may be acceptable per ASME PTC 19.3 TW-2010)

If the thermowell Peak Oscillatory Bending Stress is greater than the Fatigue Endurance Limit at the in-line velocity critical, there is more ambiguity about whether the thermowell can operate above the in-line velocity critical. Theoretically, if the thermowell is passing quickly through the in-line resonance lock-in region, it is allowed to operate between  $0.6 f_n^c$  and  $0.8 f_n^c$ . Fatigue cycle count is cumulative over the life of the thermowell, so it is critical to know how long the thermowell is in

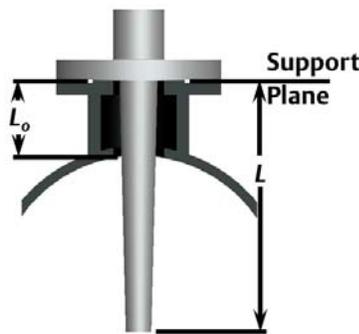
resonance. Since fatigue life is dependent on many factors, the longer the thermowell operates in resonance the less certain its lifespan.



Example chart showing how Emerson will interpret the in-line resonance evaluation (Emerson would find this design unacceptable)

Due to the fact that design details about ramp up speed are not known by the instrumentation providers, thermowells that do not pass the Peak Oscillatory Bending Stress evaluation and are operating above the in-line critical velocity will be reported as unacceptable by Emerson.

### 1.5.3 Thermowells partially shielded from the flow

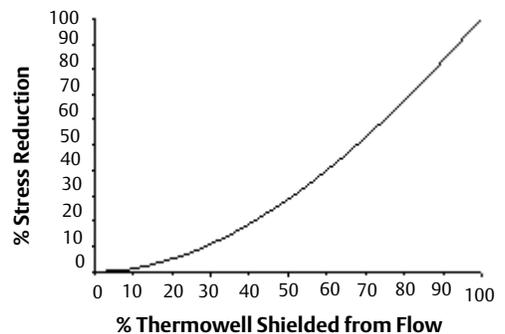


Tapered thermowell partially shielded from flow

Most thermowell installations are partially shielded from flow; the length of the thermowell exposed to the flow is not the full unsupported length and the equations for bending moment and bending stress need to be adjusted.

The effect of the shielding on a tapered thermowell is easily shown, but the effect of shielding on stepped stem thermowells is much more difficult to predict or model because the exposed surface is not a uniformly changing shape and there is a large discontinuity in the data. As a result, there are two sets of evaluations performed for stepped stem shielded thermowells based on the position of the step relative to the fluid flow. The stress calculations must also be performed twice to determine stresses at both the thermowell root and at the step.

Once the installation and process conditions are understood as well as where the Strouhal Frequency sits in the frequency domain, the analysis of the actual stresses applied to the thermowell can be performed. As previously mentioned, if the thermowell is intended to operate above the in-line velocity critical, there are cyclic stresses at the in-line resonance to consider as it passes through that region on the way to the design velocity. Also the steady state and dynamic stresses at the design velocity must be evaluated.



Effect of shielding on a tapered thermowell

## 1.5.4 Evaluation of in-line cyclical stress

The cyclic stresses, resulting from the in-line and transverse forces on the surface of the thermowell, are concentrated at the root. To account for the resonance conditions, the calculations must be performed at the in-line resonance velocity critical to see if the Peak Bending Stress at resonance is less than the Fatigue Endurance Limit of the material. Because this analysis is conducted at the in-line critical point, the magnification due to the in-line resonance overshadows the lift forces so the lift forces can be ignored to simplify the calculations. This evaluation need only be conducted if the Scruton Number evaluation indicates that the process conditions require it.

The in-line velocity critical is used to calculate the force per unit area applied to the thermowell. Since the process fluid velocity is given as an average rather than a velocity profile, the calculations also assume that the unit area is the entire exposed length of the thermowell. If some of the thermowell is partially shielded from the flow (as in the case of a standoff pipe), this must also be taken into account. For stepped stem thermowells, this analysis must be performed at both peak stress locations (root and base of stepped stem).

To ensure that the calculations are conservative, the intrinsic damping factor, is set to 0.0005. Stepped shank thermowells must be evaluated at two places to identify the highest stress of the two.

One of the major changes in the ASME PTC 19.3 TW-2010 is the use of a table to specify the Allowable Fatigues Stress Limits. The table groups materials together into a Material Class and cross references them to the installation method to determine the stress limit.

It is important to note that partial penetration welds are viewed as having less fatigue resistance than full penetration welds and are given lower values in this table. See Thermowell Construction Requirements below for more information.

## 1.5.5 Evaluation of steady state stress at design velocity

Thermowells must also be evaluated at the design velocity as well to ensure that they meet the demands of the process environment. The steady state stress is a combination of the external pressure from the process as well as the drag force. Again, these are calculated for the location of maximum stress, so if the thermowell is partially shielded, or if it is a stepped stem, the calculations should be performed with those installation considerations.

Once the Maximum Stress is calculated it can be used to determine if the Von Mises Criteria is met. The Von Mises Criteria is used to evaluate shear and pressure stress conditions in spheres and circular cylinders. It predicts the plastic yielding condition of materials.<sup>(1)</sup> Success in this evaluation means that the steady state stresses do not exceed the material fatigue strength and the thermowell can be used at the desired design velocity.

## 1.5.6 Evaluation of dynamic stress at design velocity

The dynamic stresses on the thermowell are attributed to the oscillating lift (transverse) and drag (in-line) forces. The magnification factor represents the exponential nature of the increase in forces as the Strouhal Frequency nears the thermowell natural frequency such as near the in-line velocity critical. If the Strouhal Frequency does not fall into the in-line or transverse natural frequency lock-in bands, then magnification factors are calculated and applied to the cyclical stress equations. The cyclic drag and lift forces need to be calculated at the design

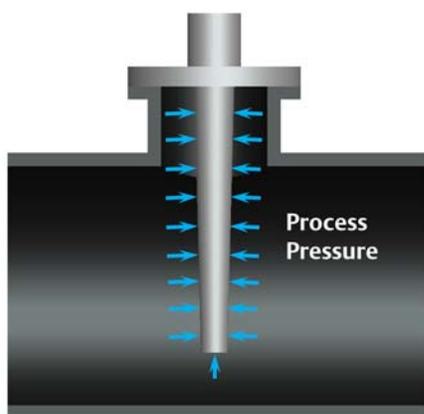
(1) Brock, John E., "Stress Analysis of Thermowells," Naval Postgraduate School, Monterey CA (1974)

velocity in the same way as the in-line cyclical stress evaluation was performed. Unlike the in-line cyclical stress evaluation previously performed, the lift forces are not zeroed out.

If the design velocity is greater than the in-line velocity critical, the thermowell might have to be treated as if it will operate at in-line resonance stress levels indefinitely. See section above on Passing Through In-Line Resonance.

Obviously there are a number of evaluations performed on the thermowell design, but with information such as the in-line velocity critical, the steady state, and dynamic stress evaluations, the designer can have a detailed picture about where the thermowell is operating in the frequency domain as well as how close it is operating to its fatigue limit. This information will allow the designer to decide what safety factors to maintain in their process.

## 1.5.7 Pressure stress evaluation



The final check necessary to see if the thermowell design is acceptable for the application is the pressure stress evaluation. This is often overlooked as it generally is not the cause of design unsuitability, but it is critical nonetheless. The pressure stress check must be performed on both the shank and the tip separately.

To calculate the pressure on the shank as a check for suitability there are two methods offered depending on the process pressure. For process pressures less than 103 MPa (15 ksi), ASME PTC 19.3 TW-2010 recommends using ASME Boiler Pressure Vessel Code (BPVC) Section VIII Paragraph UG-28,

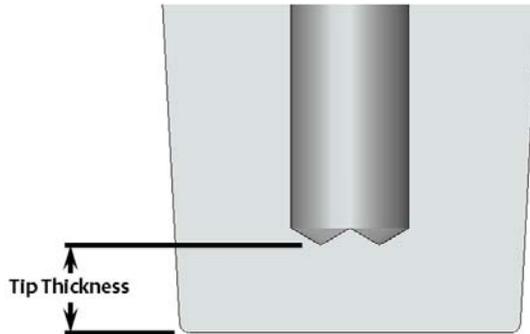
to calculate the allowable external pressure. The temperature restrictions listed in this section of the BPVC do not apply as most thermowells are designed under ASME B31.1 or ASME B31.3. Maximum allowable stress values should be sourced from either of those two standards instead. The reason the calculation from the BPVC is referenced in ASME PTC 19.3 TW-2010 is that the equation has a history of successful use and is relatively known in the industry.

In the event that the desired thermowell material is not in the BPVC or if a simpler method is desired, ASME PTC 19.3 TW-2010 provides an alternative simplified relationship. The drawback to using the simplified method is that the shank pressure determined by this method may be as much as 17% lower than the value calculated by UG-28 method for some materials at some temperatures. The benefit is a less complex calculation and an additional safety margin.

For high pressure (> 103 MPa (15 ksi)) applications, ASME PTC 19.3 TW-2010 points to ASME BPVC Section VIII Division 3 or ASME B31.3, Chapter IX for the calculation. Pressures this high (exceeding the pressure limits for 2500# flanges in ASME B16.5) will need to be carefully evaluated and not performed through an automated tool.

The tip thickness is the thinnest dimension from the outside tip to the furthest point of the drill. Since most thermowells are manufactured using gun drills, it is critical that the tip thickness used is the actual measure of the thinnest point. The peak dimension is used to calculate the

sensor length since the peak will contact the sensor, not the “valley.” When the gun drill is sharp, the valley can be as much as 0.060” [1.5mm] deeper (thinner) and becomes thicker as the drill wears.



Thermowell tip thickness detail

The maximum pressure that the thermowell can withstand is the lesser of the shank or tip pressure limit.

### ▲ CAUTION

Whether referring to ASME PTC 19.3 or ASME PTC 19.3 TW-2010, the pressure stress evaluation only refers to the stress that the thermowell stem (or shank) and tip can withstand, not what the thread or the flange can withstand. Process connection selection and pressure rating evaluation should be performed before the thermowell design is evaluated for vortex induced vibrations.

## 1.5.8 Materials information

The best engineering practice for materials information is to use reliable and standardized information whenever possible. Emerson only uses materials information from open source standards such as ASME Boiler and Pressure Vessel Code and ASME B31.1/B31.3. This information is generally conservative and industry accepted. In theory, Emerson could use vendor information to populate our materials database. This practice is discouraged, however, because Emerson cannot ensure that a specific batch of material is used on a particular thermowell to match a particular report. This is not a practical or reliable method of optimizing thermowell performance.

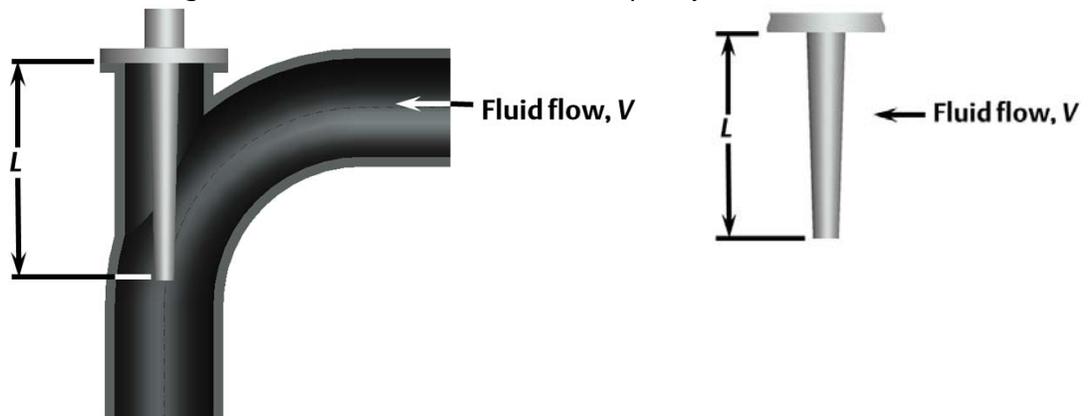
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## 1.6 Installation variations

The manner in which thermowells are installed in a process can have a significant effect on the thermowell stress calculations and the vortex shedding. The variations discussed here are beyond the “standard” installations such as flanged, threaded, and welded thermowells, or partial shielding of the thermowell.

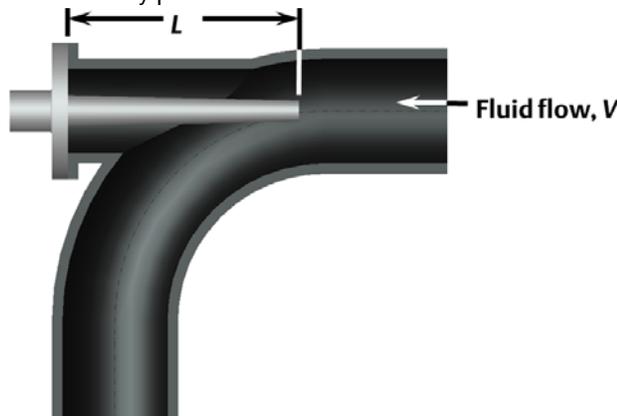
## 1.6.1 Elbow installations

ASME PTC 19.3 TW-2010 gives no meaningful guidance on the installation of thermowells in an elbow. Modeling the flow in an elbow is extremely difficult due to the turbulence and complexity. ASME PTC 19.3 TW-2010 suggests that to be conservative, consider the entire unsupported, unshielded length to be exposed to the flow with the forces acting perpendicular (i.e. “normal”) to the thermowell axis. To many, this is not an acceptable answer. Some comments in ASME PTC 19.3 TW-2010 and committee discussions yielded an alternative to this overly conservative view. If the tip is sufficiently upstream or downstream from the elbow such that the fluid flow is parallel to the thermowell axis at the tip, then the Strouhal Number is very small because the flow across the tip is negligible. ASME PTC 19.3 TW-2010 states that this is beyond the scope of the standard, while others in the industry maintain that this type of installation would be a simple solution for thermowell designs that are too close to the natural frequency.



Thermowell installed with tip facing downstream in an elbow

ASME PTC 19.3 TW-2010 suggests that the thermowell pointed in the upstream direction is the better installation because the amount and location of the flow stream applies a smaller moment arm and force to the thermowell and the flow over the tip is more laminar. If the tip is pointed downstream, the swirling of the fluid after passing around the thermowell could have some cross tip components, but this is extremely difficult to model. As with angled installations below, the moment arm calculation is complicated, therefore the changes in force, moment arm, and stress, are not easily predicted.



Thermowell installed with tip facing upstream in an elbow

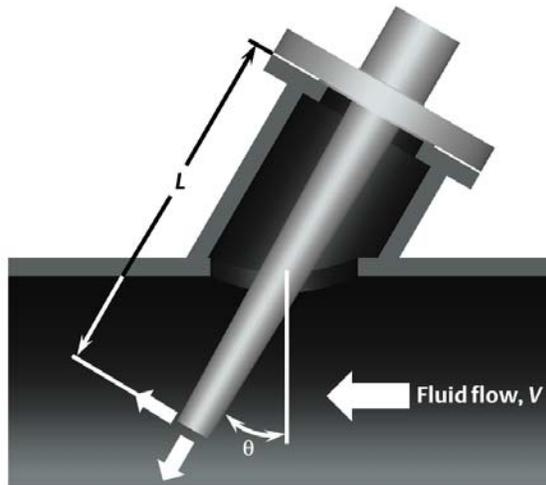
Emerson is considering a more extensive investigation into these installation methods to provide some justification for the benefits of these solutions.

## 1.6.2 Angled installations

Customers frequently install thermowells at an angle to the flow for accessibility, to reduce the forces acting on the thermowell, or to increase the exposure to the flow in smaller line sizes in order to obtain a more accurate temperature reading. The effect of the “yaw” angle on the tip velocity is not a matter of simple trigonometry. It also complicates the prediction of the stresses and forces acting on the thermowell.

As the installation angle changes, the velocity across the tip is reduced.

$$V_{\text{cross axial}} = V_{\text{flow}} \cdot \text{COS}(\theta) \quad \text{for } \theta \text{ up to } 30^\circ \text{ (1)}$$



Velocity vectors in angled installations

As the angle approaches 0° the velocity component across the tip approaches unity, or 100% of the flow velocity. This can be helpful in reducing the Strouhal Frequency for installations where stresses are not the limiting factor.

This same logic does not hold true for the forces acting on the thermowell. As the angle changes, the surface area increases, the length of thermowell increases thereby increasing frictional forces. The fluid flow around the thermowell is increasingly complicated to model, and therefore, the changes in force, moment arm, and stress, are not easily predicted. ASME PTC 19.3 TW-2010 states that this is beyond the scope of the standard, however, Emerson is

considering a more extensive investigation into this installation method to provide some justification for the benefits of this solution. There are difficult customer applications where this may be the only viable method of installation.

## 1.6.3 Square ducts vs. round pipes

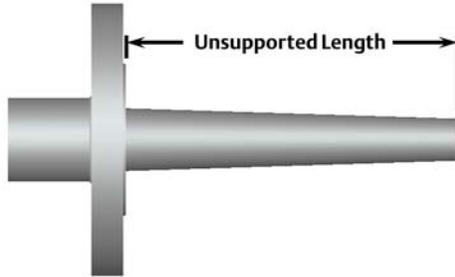
Since the new standard uses average velocity, it is not necessary to know or model the flow profile of square ducts and how it might differ from round pipes. The designer simply needs to know the mass flow rate, the density, and the cross sectional area to calculate the average velocity. See the Average Velocity section for more discussion on this.

## 1.7 Unsupported length definition

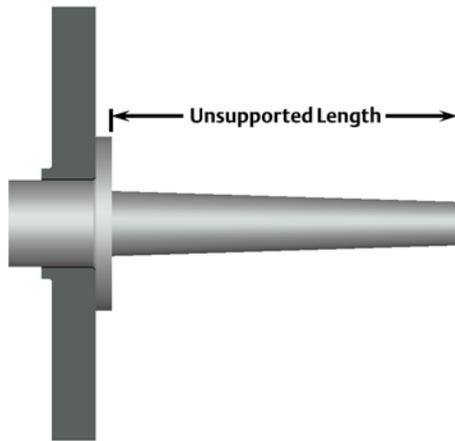
The definition of unsupported length has changed with the release of ASME PTC 19.3 TW-2010. The beam theory model that is used in ASME PTC 19.3 TW-2010 is very sensitive to changes in unsupported length, so it is necessary to make the definition clear and consistent.

(1) S. E. Ramberg, “The Effects of Yaw and Finite Length upon the Vortex Wakes of Stationary and Vibrating Cylinders,” *Journal of Fluid Mechanics* 128, 81-107 (1983).

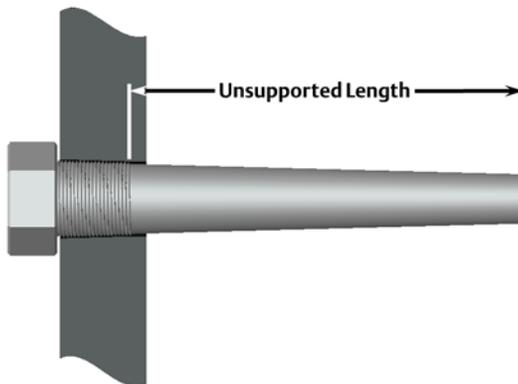
For flanged thermowells (including Lap Flange / Van Stone), the immersion length is the same as the unsupported length.



Flange Thermowell



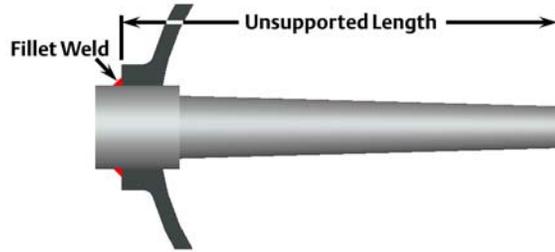
Lap Flange / Van Stone Thermowell



Threaded Thermowell

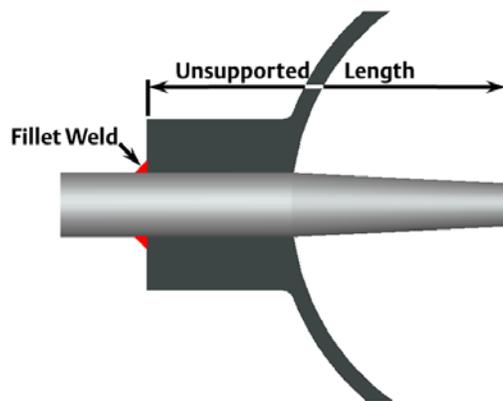
For threaded thermowells, the immersion length is the same as the unsupported length, but experimental results have shown that the unsupported length starts about 2 or 3 threads into the threaded section. This is accounted for in ASME PTC 19.3 TW-2010 by application of a correction factor for threaded thermowells.

The biggest change is seen in the welded thermowells. Use of the immersion length incorrectly shortens the unsupported length and raises the natural frequency of the thermowell.



Through-hole Welded Thermowell

The correct place from which to calculate the unsupported length is from the weld point, but since the weld point is not known exactly by Emerson, the weld point must be estimated for the calculation or the overall thermowell length must be used. This will ensure that no matter how the thermowell is installed, the calculation provided is on the safe, conservative side.



Welded Thermowell per DIN 43772

An alternative method would be to scribe a line onto the thermowell to show where it must be welded for the calculation to be accurate or conservative. This would be more involved and require knowledge of the socket dimensions.

A special case where the unsupported length must be clarified is for welded thermowells installed per DIN 43772. These are designed with a tight tolerance hole bored into a pipe and a tight tolerance machined barstock thermowell. Because there is a tolerance for installation, it is impossible to ensure an interference fit that would be required to move the unsupported length (see Velocity Collars section). In the absence of other information, the unsupported length for this

installation will be defined per the detail at left.

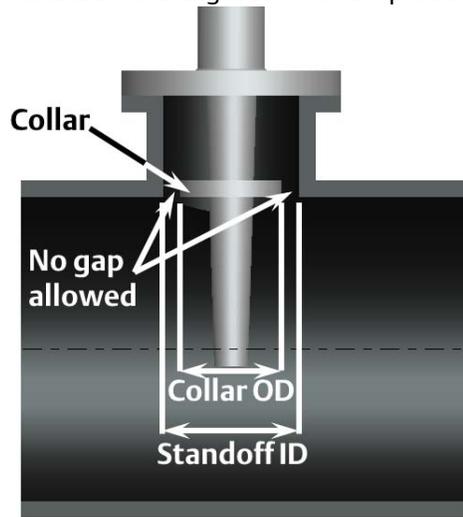
## 1.8 Velocity collars

The use of velocity (or frequency) collars is addressed in ASME PTC 19.3 TW-2010 as well. The standard states that collars are not recommended as a rigid support for the purposes of shortening the unsupported length. This can only be achieved through the use of an interference fit. This is because thermowell tip displacements are generally very small (less than 0.5 mm)<sup>(1)(2)</sup> and any gap between the collar and the standoff pipe inner diameter will render the collar ineffective at reducing the effective unsupported length. Also, like a hammer repeatedly striking a metal surface, the thermowell or the pipe will continue to deform and the gap will widen until there is no contact whatsoever. This could create stress risers in the thermowell due to the indentations, or it could eventually break welds that may be located near the collar. The collar must be an interference fit to

(1) Finch, P., Hamblin, M., and Constable, D., "In-situ Measurement of Thermowell Vibration during Production Train Pressurisation," Woodside Energy Ltd. Report (date unknown – post 2001 and pre 2010).

(2) Haslinger, K.H., Westinghouse Electric Company, "Flow-induced vibration testing of replacement thermowell designs," *Journal of Fluids and Structures* (2003).

be effective and since Emerson cannot ensure the final fit, we recommend using geometry or installations changes to meet the process conditions rather than the use of collars.

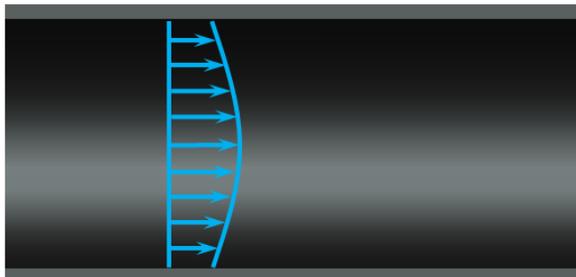


**Note**  
No gap is allowed between the collar and the pipe standoff ID.

Emerson will continue to provide thermowells with velocity collars built to customer's specifications, but we will decline to provide advice on velocity collar sizing and will not provide calculation reports for installations using velocity collars. Any other policy would give the appearance of the endorsement for that method. DIN 43772 welded thermowells will also be treated the same way as velocity collars as will other installations with similar intent.

## 1.9 Average velocity vs. velocity profile

One would think that the best way to calculate the forces on a thermowell would be to use the Finite Element Analysis (FEA) method to gain a substantial amount of detail and precision. The difficulty in using the FEA method is that the parameters for the inputs change continuously, so the precision generated far exceeds the usefulness from one moment to the next. Both ASME PTC 19.3-1974 and ASME PTC 19.3 TW-2010 standards use an average velocity for their calculations. The logic behind this is that the designer generally knows the mass flow rate and may know little about the flow profile across the pipe. The designer simply uses the mass flow rate, the fluid density, and the cross sectional area to calculate the average velocity. Using the average velocity also keeps the calculations simpler which fosters widespread user acceptance.



Example of a fully developed turbulent flow profile in a pipe

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## 1.10 Thermowell construction requirements

When calculating the in-situ natural frequency, ASME PTC 19.3 TW-2010 assumes that the thermowell is manufactured from solid barstock in order for the simple beam model to be used. Thermowell stems can be forged or roll formed material, but cannot be made from pipe or tube material. This restriction means that thermowells that are of 3-piece welded construction or those that are manufactured by drilling through the tip with welded cap are not covered by ASME PTC 19.3 TW-2010.

It is advisable to use full penetration welds for flanged thermowells to achieve the highest fatigue strength rating (see Allowable Fatigue Stress Amplitude Limits in ASME PTC 19.3 TW-2010). Forged thermowells accomplish the same goal without welds, but at a much higher cost. Van Stone style thermowells accomplish this without the cost, but many customers do not use this style of thermowell.

Another construction concern is the manufacturing tolerances of thermowells. Tolerances that are too loose can cause the beam equations to be inaccurate and can use up all safety margin built into the equations. But tightening the tolerances causes the thermowell manufacturing costs to increase. Corrosion allowance must also be considered for dimensions.



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**Emerson Process Management  
Rosemount Division**  
8200 Market Boulevard  
Chanhassen, MN 55317 USA  
T (U.S.) 1 800 999 9307  
T (International) 952 906 8888  
F 952 906 8889  
[www.rosemount.com](http://www.rosemount.com)

**Rosemount Temperature GmbH**  
Frankenstrasse 21  
63791 Karlstein  
Germany  
T 49 6188 992 0  
F 49 6188 992 112

**Emerson Process Management Asia  
Pacific Private Limited**  
1 Pandan Crescent  
Singapore 128461  
T 65 6777 8211  
F 65 6777 0947  
[Enquiries@AP.EmersonProcess.com](mailto:Enquiries@AP.EmersonProcess.com)

**Emerson Process Management**  
No. 6 North Street  
Hepingli, Dong Cheng District  
Beijing 100013, China  
T 86 10 6428 2233  
F 86 10 6422 8586

**Emerson Process Management  
Latin America**  
1300 Concord Terrace, Suite 400  
Sunrise Florida 33323 USA  
Tel + 1 954 846 5030