THERMOWELL VIBRATION INVESTIGATION AND ANALYSIS

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ABSTRACT

The current industry design practice for addressing vortex shedding-induced vibration in thermowells is to use the ASME Power Test Code 19.3, Part 3 (PTC) [1], which essentially requires the vortex shedding frequency to be less than the first natural frequency of the thermowell by a reasonable design margin.

The PTC also provides guidance for establishing the vortex shedding frequency and the natural frequency of the thermowell.

In a 1996 paper presented at the ASME Pressure Vessel and Piping Conference, Blevins, et al [2] published test results for the natural frequencies and damping coefficients of several standard design thermowells. Also presented were the classic formulations for the calculation of the Von Karman vortex shedding and the thermowell natural frequency. The Blevins data indicated that for certain types of thermowells there was a discrepancy between the measured thermowell natural frequency and the frequency calculated using the PTC method.

In this paper, the authors will review the basic calculations related to vortex shedding and thermowell natural frequency. This paper will also present Finite Element (FE) analyses of several thermowells from the Blevins paper and discuss the results of the FE analysis with respect to that paper's test results. Discrepancies between the natural frequency calculated by the PTC methodology and the thermowell natural frequency test data presented by Blevins, and the results of the FE analyses will be discussed. The authors also introduce a design technique using fatigue analysis to assess the likelihood of thermowell failure. Use of the FE-derived natural frequency information and the fatigue analysis techniques will improve the safety of thermowell applications and may extend the service velocity in which a specific thermowell can be used.

VORTEX SHEDDING

Thermowells that are used to measure the temperature of flowing fluids are subjected to a uniform loading from the fluid drag and a flow-induced varying force from Von Karman vortex shedding effects, as illustrated in Figure 1.

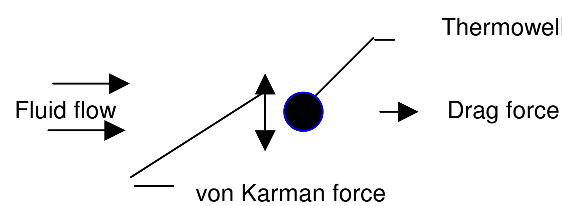

FIGURE 1

The reader is referred to the PTC for more information on other thermowell design aspects. For additional information related to formation and effect of Von Karman vortex, the reader is referred to Flow Induced Vibration, 2nd edition, by R. D. Blevins [3]. As indicated in Figure 1, the movement of a thermowell due to the Von Karman vortex shedding force is perpendicular to the fluid flow direction. The frequency of the vortex shedding is proportional to the fluid velocity and the diameter of the thermowell.

The PTC, paragraph 15, provides the classic vortex shedding frequency formulation as:

$$Fw = 2.64 * \frac{V}{B} \qquad (1.)$$

Where:

Fw = vortex shedding frequency, cycles per second
V = fluid velocity, ft per second
B = thermowell tip diameter, inches

Note: the constant of 2.64 is composed of a Strouhal number of 0.22 times the conversion of 12 inches/foot (0.22 x 12 = 2.64).

The PTC considers the Strouhal number to be constant for typical thermowell applications at a value of 0.22, which is a reasonable approximation for most industrial applications. Blevins [3] provides additional information on the variation in values of the Strouhal number.

The PTC vortex shedding frequency calculation is based on the tip diameter and is primarily applicable to a straight thermowell. Industry also utilizes thermowells that are tapered to increase the strength of the thermowell at the mounting region while maintaining a minimum diameter in the fluid flow region. This is done to increase the vortex shedding frequency.

There is no established standard for the rate of taper in thermowells. However, industrial thermowells typically have a minimum 0.625 inch tip diameter for a 0.26 inch bore and a 0.75 inch tip diameter for a 0.385 inch bore. Additionally, tapered thermowells typically have a maximum 1.0625 inch root (or mounting area) diameter.

At typical fluid velocities, the fluid flow profile will be reasonably uniform except in the region adjacent to the conduit wall, where viscous friction will reduce the velocity significantly. For calculation of the vortex shedding frequency it is reasonable to assume a uniform velocity based on the average conduit cross-sectional flow area and the total flow. It is also reasonable to use the average diameter for the length of a tapered thermowell that is in the flow region. For this assumption to be valid, the average diameter in the flowing fluid region should not exceed the tip diameter by more than a factor of 1.2. If the ratio is more than 1.2, it may be necessary to calculate the maximum and minimum vortex shedding frequency based on the minimum and maximum thermowell outside diameter exposed to the fluid flow. The PTC vortex shedding frequency calculation methodology uses only the tip diameter; for this reason, the PTC calculation results in a conservatively high shedding frequency.

THERMOWELL NATURAL FREQUENCY

The thermowell natural mechanical vibration frequency can be approximated by assuming that it is a simple cantilevered structure. The PTC uses a formulation that is based on a cantilevered beam with a constant that is used to adjust for the thermowell test data developed by ASME. Similar cantilevered beam calculations are presented in the technical papers by Blevins et al [2], Dozaki et al [4], and Bartran et al [5]. In the paper by Dozak, special note is made that the natural mechanical vibration frequency of a thermowell is affected by the mounting arrangement. A very rigid thermowell mounting, such as a pad type flange on a heavy walled vessel, will have a mechanical vibration frequency very nearly equal to the theoretical cantilever beam formulation of:

$$Fn = \frac{1.875^2}{2 * \pi * l^2} * \sqrt{\frac{E * I * g}{w}} \qquad (2.)$$

Where:

Fn = first natural frequency of thermowell, cycles/sec
1.875 = dimension factor - first mode
l = total length of well from root to tip, in
E = modulus of elasticity for thermowell material, lb/in^2
I = moment of inertia of thermowell at root, in^4
w = weight per unit length of the thermowell, lb/in
g = gravitational constant, 386.4 in/sec^2

The mounting of a thermowell on a nozzle on a thin walled pipe or vessel will result in a somewhat reduced first natural frequency because the mounting is somewhat flexible. The use of FE analysis to assess this type of mounting will provide more accurate frequency data, as discussed later in this paper.

The calculation format above does not take into consideration the effect of the fluid mass around the thermowell and is valid for most applications up to a fluid density of less than 20% of the density of the thermowell material. When the fluid density exceeds 20 % of the thermowell material density, the effect of the fluid is to reduce the frequency at which the thermowell becomes excited. In investigations of the thermowell natural frequency for the typical industrial thermowell application, it is not necessary to account for the fluid density.

The PTC uses the following formula to determine the thermowell first mechanical natural frequency (*Fqw*) regardless of whether the thermowell is straight or tapered (note: this formula is not listed in the PTC but is directly distilled from PTC formula # 5):

$$Fqw = \frac{Kf}{L^2} * \sqrt{\frac{E}{Den}} \qquad (3.)$$

Where:

Kf = ASME factor per table below:

Length of thermowell inches	Kf for 0.26 inch bore diameter thermowell	Kf for 0.385 inch bore diameter thermowell
2.5	2.06	2.42
4.5	2.07	2.45
7.5	2.07	2.46
10.5	2.09	2.47
16	2.09	2.47
24	2.09	2.47

E = modulus of elasticity for the thermowell material, lb/in^2)
Den = density of thermowell material, lb / in^3
L = length of thermowell from root to tip, in

FE MODELING

The thermowells discussed in the Blevins paper [2] have been modeled using finite element techniques. A separate model of each thermowell was constructed. One of these thermowell models is illustrated in Figure 2. The thermowell models were then combined with separate models of the restraining fixture used in the testing as illustrated in Figure 3.

FE Models

The models used in these analyses employed approximately 8500 nodes defining approximately 5000 solid brick elements.

Figure 2 – Basic thermowell model

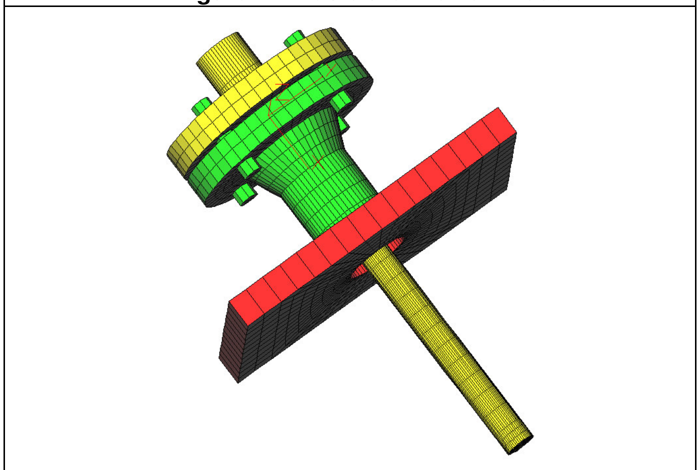

Figure 3 - Thermowell in test fixture

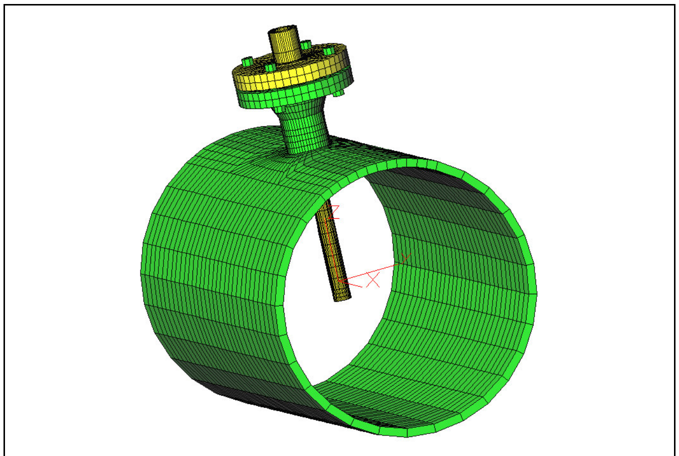

Figure 4 – Thermowell/pipe flange configuration

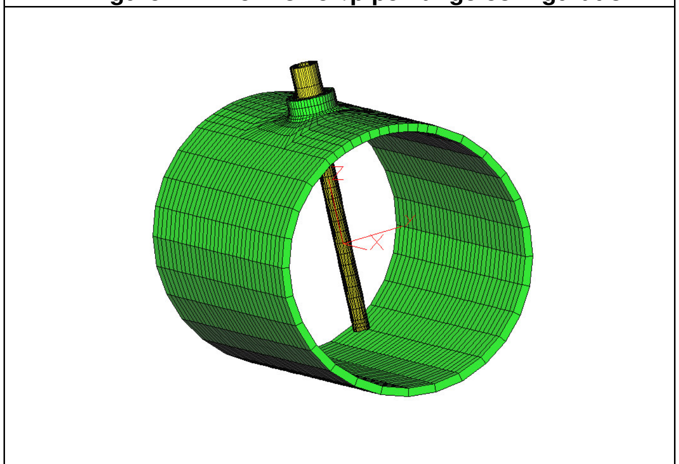

Figure 5 – Thermowell/pipe weld-o-let configuration

After the first model was constructed, it was a relatively simple task to parametrically modify the model to account for the various thermowell geometries. The computational effort was typically on the order of 15-20 minutes on a PC. Once the initial model was completed, the total time to analyze an addition thermowell configuration was on the order of 1-3 hours.

Computed Frequencies

The lowest modal frequency was determined for each thermowell/mount system. Those results are listed in Appendix A. It can be seen in Appendix A that the correlation between the test data and the FE-generated data is quite good. For the tapered thermowells, the FE approach correlates much better than the PTC calculation with the test data. The PTC calculation tends to significantly underestimate the natural frequencies of tapered thermowells.

MOUNTING EFFECT

In order to investigate the effect of the mounting type on the natural frequency of the thermowells, three of the thermowell models were combined with a model of a flange connection attached to a section of 8" dia. 3/8" wall pipe (Figure 4) and to a weld-o-let connection to the same pipe (Figure 5). The modal frequencies computed for these model combinations are also listed in Appendix A. It should be noted that the Blevins [2] test data

were developed using a relatively stiff mounting arrangement, as illustrated in Figure 3.

If we look at the results computed using different mounting configurations, several trends are evident:

1. When mounted in standard wall 8" pipe, the natural frequency of the thermowell is less than that computed or measured using the test fixture. The difference is more evident with the tapered thermowells than it is with the straight thermowell.
2. The computed frequency with the weld-o-let model is closer to the tested frequency than the flanged model.
3. The greatest deviation in computed frequencies is noted with the flanged model of the short (9") tapered thermowell vs. the other mounting conditions.

While it is always dangerous to extrapolate from such a limited number of examples, it would appear that the FE technique may provide a significantly better estimate of the natural frequency of a given thermowell than does the PTC calculation procedure. This is especially true for the tapered thermowell.

Additionally, in the case of a flanged connection to thin walled pipe, FE analysis or physical testing in place may be the best way to obtain an accurate estimate of the true natural frequency.

DISCUSSION OF RESULTS

Appendix A of this paper contains a summary of typical thermowell natural frequency data. From the data set, it is apparent that the PTC and FE results for a straight type thermowell agree with the test data published by Blevins. It is also apparent that for tapered type thermowells, there is considerable discrepancy between the PTC-computed frequencies and the test data. The FE-generated frequencies tend to agree well with the test data. The authors recommend that if the PTC approach indicates that a tapered thermowell is not acceptable for a critical service application, a re-evaluation should be made using either a physical test or FE analysis.

The 20% recommended design margin built into the PTC may not, in some cases, prove to be adequate. It is a known fact that most process plants are eventually operated at a rate higher than the original design capacity. Whether this is a result of a deliberate de-bottlenecking process or simply “pushing” the process, the net result is higher velocities than were used for the original design. The prudent engineer will keep these factors in mind during the thermowell selection process.

Additionally, consideration should be given to upset and other unusual events. For example, in some industries it is common to use a steam blow of the lines during the construction or startup process. This procedure can potentially result in velocities high enough to excite the second harmonic of the thermowell. This combination of high velocity and correspondingly high excitation force, along with possible high lock in frequency, can result in rapid fatiguing of the thermowell if the thermowell has not been designed to accommodate these conditions.

It should be noted that in the Blevins paper [2], he states that for low density fluids (less than 0.17 lb/ft^3) the fluid does not have enough mass density to produce any significant vibration of a typical thermowell at first natural frequency lock-in resonance.

FATIGUE ANALYSIS

The typical thermowell application should be limited to a non-vortex-induced vibration design. However, it is not always possible to completely avoid vortex-induced vibration. For some applications it is possible to achieve a design that can tolerate a limited duration of vibration under certain operational conditions.

When the thermowell natural frequency matches the fluid vortex shedding frequency, lock-in can occur. The thermowell will then achieve its maximum deflection and the resulting maximum fatigue stress. It is only necessary to investigate this condition if the calculated vortex shedding frequency is more than 80% of the thermowell’s natural frequency.

When lock-in occurs, the vortex shedding creates a force called lift on the side of the body perpendicular to the fluid flow (see Figure 1). The force that is developed by the vortex shedding effect is directly proportional to the square of the velocity of the flowing fluid. It is well documented (e.g. Blevins [3]) that vortex lock-in can occur when the Von Karman frequency is within approximately 20% of the mechanical frequency of the body it is forming around. Therefore, it is conservative to assume that the thermowell will lock-in with the fluid vortex shedding up to a frequency of 120% of the first mechanical natural frequency. If this maximum velocity exceeds the range of operating fluid velocity, the maximum design fluid velocity may be used to calculate the maximum force.

The thermowell deflection, at resonance, is determined by the force applied by the vortex shedding and the vibration damping ability of the thermowell. The self-damping ability of several typical thermowells was discussed and described in the technical paper publication by Blevins. The damping is significantly different for empty thermowells versus thermowells with a thermocouple installed. The placement of any solid item, such as a thermocouple, in the thermowell such that it is in contact with the thermowell bore will significantly increase the damping property of the thermowell.

The effect of the damping is to limit the maximum deflection for the applied harmonic force. The same applied harmonic force applied to the same thermowell without anything in the bore will have a significantly higher deflection (and accompanying cyclic stress) than a thermowell with something in the bore that produces damping.

Stress Calculation Procedure

The forcing effect of vortex shedding on the thermowell is given by Blevins as:

$$Fu = \frac{1}{2} * \rho * V^2 * D * C_L \qquad (4.)$$

Where:

Fu = force per unit length of thermowell in fluid flow, lb/in
ρ = flowing fluid density, lb/in^3
V = fluid velocity, in/sec
D = outside diameter of the thermowell, in
C_L = lift coefficient, (dimensionless, typically = 0.5)

The damping coefficient suggested in the technical paper by Blevins et al [2] is 0.002 for a thermowell with a thermocouple installed in the bore. A note of caution should be made here: if the designer uses the fatigue basis for thermowell design, then there must be assurance that the 0.002 damping coefficient is maintained by requiring the user to always have a suitable item installed in the bore of the thermowell.

For calculation purposes, it is easier to use a magnification factor calculated by:

$$Q = \frac{1}{2 * \zeta} \qquad (5.)$$

Where:

Q = magnification factor
ζ = damping coefficient

From this equation it is obvious that Q is equal 250 for the typical thermowell with a thermocouple in the bore. The best way to visualize the effect of the magnification factor is to consider a thermowell with a point force on the thermowell that produces a 0.001” deflection at the end of the thermowell. When that same force is exerted by a dynamic load (the vortex shedding at lock-in), the tip movement will be 250 time greater or, in this case, 0.25 inches.

The vortex shedding force will only exist on the portion of the thermowell that is in the flowing fluid. Therefore, it is necessary to determine the length of the thermowell that is extending into the fluid and consider that force to be acting at the mid-point of the length in the fluid flow. The total force is then calculated. The moment produced by this force at the thermowell root is the product of this force and mid-point length. The formula for the total force is then:

$$Ft = Fu * Q * Lf \qquad (6.)$$

Where:

Ft = force total applied to thermowell by vortex, lb
Fu = force per unit length from Equation 4, lb/in
Q = magnification factor from Equation 5
Lf = length of thermowell in fluid flow, in

The moment applied to the root of the thermowell can now be calculated using:

$$M = Ft * La \qquad (7.)$$

Where:
M = moment, in-lb
Ft = total force from Equation 6, lb
La = lever arm from root of thermowell to the mid-point of the force applied force region, in

Finally, the stress in the root of the thermowell is calculated using:

$$\sigma = \frac{M}{Sm} * Scf \qquad (8.)$$

Where:
σ = stress in root, lb/in^2
M = moment from Equation 7, in-lb
Sm = section modulus of root of thermowell, in^3
Scf = stress concentration factor at the root, usually taken as 1.2 for a typical thermowell

This stress must be considered to be fully reversing. That is, the stress range for fatigue evaluation is twice the computed stress from Equation 8. The stress calculated above is only one component of the stresses that may exist in the thermowell. The designer is cautioned to recognize that a high pressure or high drag thermowell application may need a complete stress vector analysis to establish the applicable cyclic stress. For typical piping and vessels which are under the ASME Section VIII and B 31.3 jurisdiction it is prudent to use the cyclic stress values from ASME Section VIII Division 2 for the design, based on the assumption that the total cycles will be greater than 10^7. ASME Section III also has similar fatigue values.

The calculated stress is typically considered fully cyclic and can be used as the total peak fully reversing stress (the pressure-induced stresses are usually very small in comparison to the vibration induced stresses). The reader is referred to the technical paper by Martens et al [6] for addressing the calculated stresses for cyclic determination for Section VIII applications.

RECOMMENDATIONS

- In addition to the PTC calculated values, the thermowell manufacturing industry should provide natural frequency data based either on actual test data or Finite Element analysis for various typical installation types.
- When the PTC thermowell natural frequency and vortex shedding calculation methods are considered to be too conservative, such as in the case of tapered thermowells, the frequencies may be determined using FE methods.
- The first natural frequency for a specific thermowell application, including its mounting structure, may be accurately calculated by Finite Element methodology.
- For typical thermowell applications, the vortex shedding frequency should not exceed 80% of the thermowell's first natural frequency in order to avoid lock-in resonance, as recommended by the PTC. This calculation should be at the maximum design fluid flow conditions, and consideratiion should be given for abnormal conditions such as upset and relief valve openings.
- When the PTC 80% separation rule cannot be maintained in abnormal, high fluid velocity conditions, it is possible to use the fatigue analysis approach to assure that the thermowell will not fail in fatigue during the abnormal condition. It is recommended that all such applications be fully reviewed and confirmed. Also it must be understood that if a thermowell is designed to accommodate vibration without failing, the temperature measurement thermocouple or other devices may be damaged during the vibrating condition.
- Where the fatigue analysis procedure is used, it is recommended that the thermowells be investigated for fatigue failure at fluid velocities up to least 133% over normal design flow and for all operational and startup conditions.

REFERENCES

1. ASME Power Test Code 19.3-174 Part 3, Temperature Measurement

2. Blevins, R. D., Tilden, B. W. and Martens D. H., 1996, "Vortex-Induced Vibration and Damping of Thermowells," PVP **328**

3. Blevins, R. D., 1990, *Flow Induced Vibrations,* 2nd edition, Van Nostrand

4. Dozake, K., Morishita, M., and Iwata, K., 1998, "Modification and Design Guide for Thermowell for FBR," PVP **363**

5. Bartran, D., et al, 1999, "Flow Induced Vibration of Thermowells," ISA Transactions **38**

6. Martens, D., and Hsieh, C.S., 1998, "Finite Element Investigation of a CBA Reactor for the Effects of Thermal Cycling," PVP **368**

APPENDIX A
TYPICAL THERMOWELL DATA
18-8 Stainless steel material

	Well # 1	Well # 2	Well # 5	Well # 6	Well # 11	Well # 8
Size/Type	1 ½” 150#	1 ½” 150#	1 ½” 150#	1 ½” 150#	1 ½” 150#	1 ½”150#
Length	7”	10”	10”	10”	9”	16”
Root Dia	0.752”	0.752”	0.878”	0.867”	0.100”	0.760”
Tip Dia	0.752”	0.752”	0.878”	0.633”	0.769”	0.760”
Shape	Straight	Straight	Straight	Tapered	Tapered	Straight
Bore Dia	0.26”	0.26”	0.375”	0.26”	0.375”	0.26”
Blevins Test Data Frequency	407 Hz	206 Hz	244 Hz	278 Hz	377 Hz	83 Hz
PTC Calculated Frequency	415 Hz	204 Hz	241 Hz	204 Hz	252 Hz	80 Hz
Frequencies Calculated with Finite Element Models						
Boundary						
Fixture	413 Hz	209 Hz	242 Hz	276 Hz	368 Hz	83 Hz
Flange		204 Hz		260 Hz	305 Hz	
Weld-o-let		205 Hz		268 Hz	356 Hz	