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THE USE OF ACOUSTIC COMB FILTERS TO AVOID CENTRIFUGAL COMPRESSOR DISCHARGE BLADE-PASS PULSATION INDUCED PIPING VIBRATIONS

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ABSTRACT

Centrifugal compressor blade-pass interaction periodic pulsation can cause acoustic resonance conditions and pipe vibrations in the suction and discharge lines of the compressor. A series of two or more progressively placed quarter wave resonators, called a quarter wave comb filter, can be designed to attenuate these pulsations and eliminate the risk of acoustically induced pipe vibrations. Comb filters have been used and demonstrated for over 50 years for a wide range of acoustic attenuation applications and their physics is well understood. This paper discusses the practical design of such a comb filter, its installation in the downstream piping near the discharge of the compressor, basic comb filter design rules, a step-by-step design process for comb filters, and provides a basic case study which quantifies its attenuation effectiveness. A number of parametric studies for key design variables, such as the number of circumferentially located comb filters, stub diameters, and number of filter stages versus pulsation attenuation effectiveness, are also presented. Results showed that two to three-stage comb filters can be effectively used to eliminate centrifugal compressor blade-pass excitation induced pipe radial resonances and associated vibrations.

INTRODUCTION

Periodic blade-pass pressure pulsation excitations from the suction and discharge of centrifugal compressors can cause acoustic resonance conditions and associated piping vibrations upstream and downstream of the centrifugal compressor. These high frequency pressure excitations are usually created by the translation of non-uniform jet-wake meridional flow of the first or last stage compressor impeller blade passages from the rotating impeller to the stationary suction/discharge piping frame. The excitation frequency of these blade-pass pressure pulses, relative to the stationary frame, is determined by the speed of the compressor multiplied by the number of impeller blades, and if this excitation frequency coincides with a flange radial acoustic piping mode or a short segment pipe axial mode, resonance and significant associated pressure force induced piping vibrations can result. These types of vibrations have previously been observed and are well documented in the literature Kushner, et al. (2002), Jungbauer, et al. (1997), and Price and Smith (1999) including their potential to cause structural damage to the pipe and pipe small bore connections.

Conventional approaches to address these structural vibrations have included changing pipe diameters to avoid radial acoustic resonant modes, stiffening the piping by increasing pipe thickness or adding stiffening clamps, and/or adding structural supports and mass. Attempts have also been made to reduce the blade-pass excitation pulsations using Helmholtz arrays in the discharge piping or circumferentially mounted quarter wave resonator tubes (Liu, et al. 2010).

Although both Helmholtz resonators and quarter wave absorbers are effective in reducing first order singular frequency pressure pulsations, they cannot remove or attenuate higher orders of these periodic excitations. They also effectively shift first order excitations to the second order, and if not carefully designed, can actually create damaging higher frequency acoustic resonances in the upstream/downstream piping. However, a series of two or more progressively 50% length-decreasing quarter wave resonators (i.e, wave length $\lambda/4$, $\lambda/8$, $\lambda/16$, etc.), also called a quarter wave comb filter, can be designed to shift excitation frequencies multiple orders higher such that their pulses superimpose to form a continuous steady waveform. This can be demonstrated to almost completely filter all excitations, even if several independent and functionally non-related periodic excitations occur. The advantage of a comb filter over Helmholtz arrays is that they (i) can handle the multiple frequencies and higher orders of complex wave forms that are typically seen from centrifugal compressor blade passing pulsation excitation and (ii) they are from a mechanical perspective significantly easier to install in compressor connected piping. Comb filters have been successfully utilized for finite frequency acoustic attenuation and their effectiveness has been demonstrated for a range of applications including recip compressors, race engines, and ship propulsion.

This paper discusses the practical design of such a comb filter, its installation in the downstream piping near the discharge of the compressor, simple design rules for comb filters, and provides modeling results and predictions of its effectiveness for various operating conditions. A step-by-step design methodology is presented, in addition to parametric studies for the number of circumferentially located comb filters and number of filter stages versus pulsation attenuation effectiveness. Finally, an example case study of a comb filter in a typical process gas centrifugal compressor is provided. The primary aim of this paper is not to demonstrate the functioning of this device (which is rather straight-forward physics), but rather to provide a clear, concise, and detailed step-by-step instruction on how to design an effective comb filter for a centrifugal compressor application.

BACKGROUND

Pressure Pulsations from Blade Interaction

The generation of pressure pulsation as the result of blade-to-blade, blade-to-vane, and synchronous blade-to-case interactions from turbomachinery, and especially from centrifugal compressors, has been well documented in the literature Liu, et al. (2010) and will only be briefly discussed herein. These localized periodic pressure and flow fluctuations are created when rotating blades interact either with stationary objects, such as vanes, struts, diffuser blades, etc., in the flow path or with other rotating blades. The frequency of these excitations is easily calculated by multiplying the speed of the rotating blades by the number of stationary flow path objects or obstructions. One should note that with all pulsations, frequency excitations of higher order multiples will also be created since the excitation is not necessarily sinusoidal in functional form. Specifically, the primary excitation frequencies from blade-to-blade and synchronous blade excitations are determined from

$$fpb = RPM * b / 60 \tag{1}$$

$$fp = RPM * nb /60 \quad \text{or} \quad RPM * nd /60 \tag{2}$$

$$finteract = RPM * nbd /60 \tag{3}$$

The amplitude of these excitations is more difficult to predict since it strongly depends on the fluid dynamics, impedance, geometry, and damping of the blade-pass interaction source. However, some "rules of thumb" have been developed and have been found to be reasonably accurate (Brun, et al. 2010). One should note that with all periodic pulsations, when they encounter an acoustic resonance due to changes of the system impedance or geometry, the amplification of the pulsation is possible. Thus, small excitations can result in significant pressure pulsations, especially in lightly fluid damped systems.

Acoustic Resonance Modes

The principles of acoustic resonance have been well studied over the last century because of the applicability of acoustic theory to the broad spectrum of noise-related engineering problems. Acoustic resonance modes are usually determined by the physical end geometry of the flow system with the principal modes being open-open, closed-closed, and closed-open. The corresponding modes for these conditions are standing waves, standing half waves, and standing quarter waves, respectively, or integer multiples of these. Piping resonances that are created from blade-pass excitations, due to their relatively high frequency, usually correspond to radial cross modes in the suction or discharge pipe and sometimes to axial modes in short pipe segments with varying cross-sectional areas. The mathematical description of the radial modes is beyond the scope of this paper, but their functional shape corresponds to the Bessel function solution of the 2D or 3D wave equation in a cylindrical coordinate system. These typical mode shapes, as shown in Figure 1, have been described in the literature and have been validated experimentally. Here, only the first six orders are shown, but a large number of multiples of half-wave resonance conditions and associated mode shapes are possible in the pipe's 2D radial space.

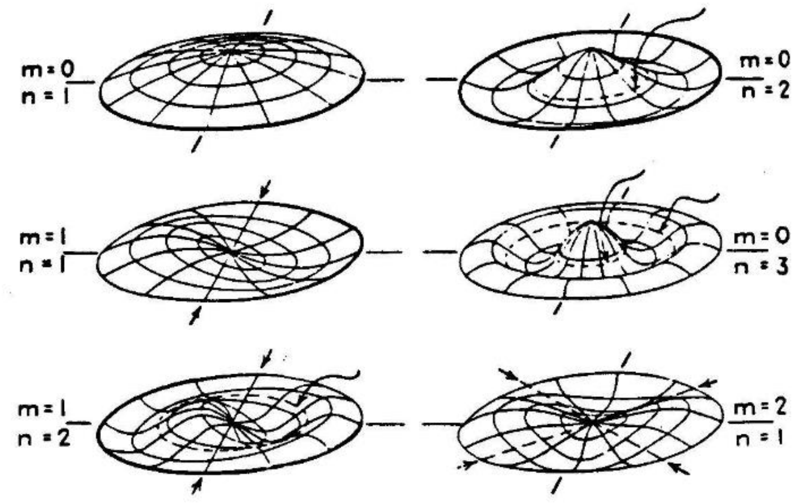


Figure 1: Radial Acoustic Pipe Modes

As a result, when analyzing the potential resonance impact of blade-pass excitation on radial acoustic piping modes, a number of modes must be analyzed. This can be shown in an interference diagram as shown in Figure 2.

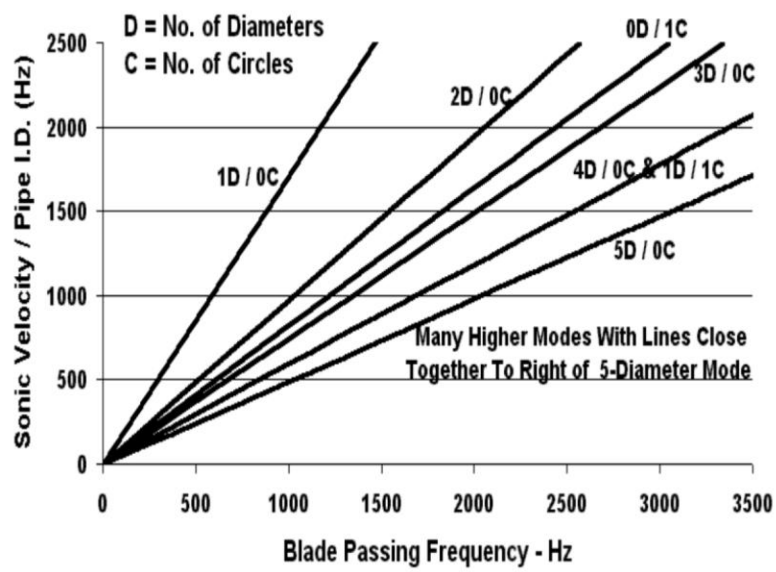


Figure 2: Potential Radial Pipe Modes versus Blade Pass Frequencies (Kushner, et al. 2002)

Pulsation Attenuation

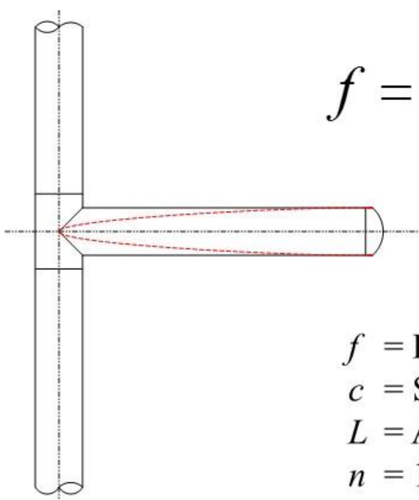
Pulsations may be attenuated or reduced through several hardware means (orifice plates, Helmholtz resonators or side-branch absorbers, or volume-choke-volume filters), but fundamentally the various conventional techniques rely on four basic attenuation mechanisms. These mechanisms are all physically interrelated but can fundamentally be divided into the following classes:

- Flow choking
- Wave shifting
- Gas compliance
- Viscous dissipation

Although flow choking, gas compliance, and viscous dissipation can theoretically be utilized for centrifugal compressor blade-pass pulsation attenuation, they are of limited practical use and are never applied in modern compressor stations because of size, effectiveness, and space constraints. Gas compliance (basically a very large bottle) reduces pulsation through diffusion and absorption of pressure pulsations into a large elastic gas volume, whereas viscous dissipation (mostly in applied in-pipe flow) reduces the amplitude of pulsation through intermolecular and boundary layer flow gradient friction. Viscous dissipation can be enhanced using flow mixers and in-pipe conditioners, but these also tend to increase the pressure losses in the flow. Similarly, choking requires flow constrictions or orifice plates with inherently large pressure drops which makes them undesirable for centrifugal compressor blade-pass pulsation control. Thus, the most commonly utilized pulsation attenuation mechanism for this application is wave shifting using Helmholtz resonators, Helmholtz bottles, or side branch absorbers. Obviously, simple stiffening of the pipe to reduce vibration is also used, but this addresses the symptom rather than the root cause. One should note that although wave shifting is the principal physical mechanism to reduce pulsations, some gas compliance and viscous dissipation also play a role. Generally speaking, flow choking techniques are more effective over a broad frequency range but have high pressure drops, while wave shifting methods, which have low pressure drops, are best applied to eliminate large amplitude pulsations at a single frequency (or smaller frequency band).

Wave Shifting
Wave shifting is the reduction of pulsations through a time-wave delay or phase shift effect. This occurs when a portion of the pulsating flow stream is diverted to a closed end in the piping system, which can be either a short stub (or quarter wave side branch) or a more elaborate closed end achieved through a side branch volume (Helmholtz resonator). Figure 3 shows the function of a simple quarter wave side branch absorber.

When multiple quarter wave side branches are arranged in an array, the configuration is also called a comb filter and functions to very effectively reduce pulsations at near zero pressure loss. Figure 4 shows a set of pulses traveling down a piping system and splitting into two lower amplitude sets (assuming equal pipe flow areas at the tee). The pulses in the stub branch travel down the length of the branch and then reflect back against the closed end. For the purposes of this example, the stub branch has a length equal to one-fourth of the wavelength.



$$f = (2n-1)\frac{c}{4L}$$

f = Response frequency
c = Speed of sound
L = Acoustic length of pipe span
n = 1,2,3,…

Figure 3: Quarter Wave Side Branch Absorber

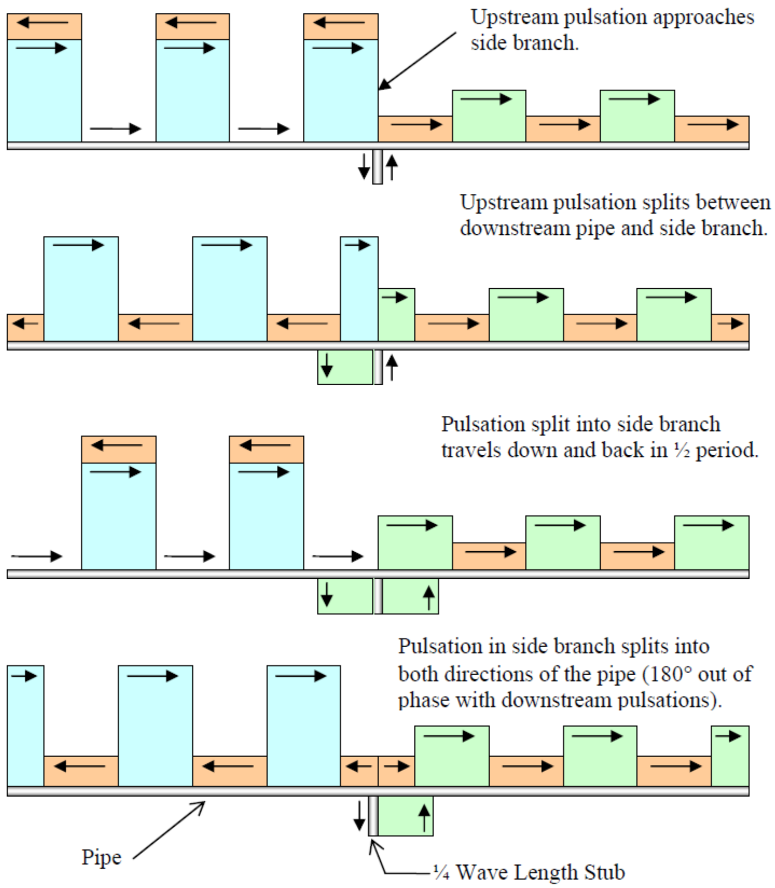


Figure 4: Pulsations through a Pipe with a Side Branch Attenuator (Tweten, et al. 2008)

When the reflected pulse rejoins the flow stream, its phase has effectively shifted over one c complete cycle. This results in tuning out the frequency corresponding to $\frac{c}{4 \cdot L}$. The shifted pulsation will create a new or higher amplitude pulse at a different frequency, upstream of the pipe stub. The higher frequency will depend on how the original pulsating flow stream combines with the reflected pulses coming out of the pipe stub. Although the pipe stub has beneficially reduced a particular frequency by shifting the energy to a different frequency, it may cause other resonance conditions to occur. These are often difficult to predict without advanced fluid models which capture the true behavior of the pulsating flow stream.

The shifting effect is also used in Helmholtz resonators, though the mode shape is more complicated in this case. The volume-choke geometry allows the pulsation attenuation to be accomplished in a smaller space than a side-branch stub. The classic Helmholtz resonator absorbs a particular frequency corresponding to the Helmholtz equation. In reality, the pressure pulse is shifted to a different point in time when it leaves the volume of the resonator. This effectively produces side bands on either side of the resonator primary frequency due to the shifted pulse.

The effectiveness of quarter wave and Helmholtz resonators for pulsation control have been tested and demonstrated, and the results have been extensively published in the literature Kinsler, et al. (1976) and Tweten, et al. (2008). Although much of this work was applied to reciprocating compressor pulsations, the basic physical principles and design rules are similarly applicable to centrifugal compressor blade-pass pulsation attenuation.

ANALYSIS METHOD

For the analysis of highly transient flow in complex piping systems, any "linearized" solutions of the transient wave equation or even transient perturbation transport solutions, such as those employing the method of characteristics or finite wave methods, are inherently not suitable, as they do not fully model the fluid flow and compressor physics. Thus, a full solution of the Navier-Stokes equation, coupled with physical compressor models, is the most appropriate solver to model the transient fluid flow and interaction of centrifugal compressors and their piping systems. The basic physics and methodology of this model were described in detail by Brun, et al. (2007) and is only briefly outlined below.

1D, unsteady compressible inviscid flow is governed by a system of three equations: continuity, x-momentum and energy. After a suitable non-dimensionalization, the system of equations can be written as Equation

$$\frac{\partial \overline{\boldsymbol{q}}}{dt} + \frac{\partial \overline{\boldsymbol{F}}}{\partial x} = 0 \tag{4}$$

where;

$$\overline{\boldsymbol{q}} = \begin{pmatrix} \varrho \\ \varrho u \\ \frac{p}{\gamma(\gamma-1)} + 0.5\varrho u^2 \end{pmatrix} \text{and } \overline{\boldsymbol{F}} = \begin{pmatrix} \varrho u \\ \varrho u^2 + \frac{p}{\gamma} \\ \left(\frac{p}{\gamma-1} + 0.5\varrho u^2\right) u \end{pmatrix} \tag{5}$$

A new 1D Lax-Wendroff method time-domain flow solver, applicable to any complex interconnected manifold and piping system, was developed to determine the highly transient fluid pulsations in centrifugal compressors and their associated piping system. This three-equation transient flow solver includes all terms of the governing equations, including fluid inertia, diffusion, viscosity, and energy dissipation. The solver moves forward in time and at each time step, boundary conditions at the segment end nodes are recalculated and enforced. A Peng-Robinson equation of state is utilized to determine gas physical properties at each node and time step.

Unlike previous solvers described in the literature Brun, et al. (2007), this new solver does not require equal-distant nodes in each system segment but rather relies on locally varying “artificial” time steps. Thus, although the methodology is still a time-domain solver, it eliminates the numerical accuracy and stability disadvantage of previous time-domain solvers. Specifically, this new solution algorithm was found to be significantly more robust and faster than previously described solvers. It is beyond the scope of this paper to provide a detailed description of the solver, but the basic description of the numerical approach can be found in Brun, et al. (2010) and Fletcher (1988).

CASE STUDY AND PARAMETRIC OPTIMIZATION

To demonstrate the viability and effectiveness of utilizing a comb filter to attenuate blade-pass induced pulsations in the suction and discharge of a centrifugal compressor, a basic case study is provided.

Comb Filter Design

Although the case study is based on a multi-stage compressor, only pulsations from the last stage in the compressor are analyzed and attenuated. Also, the compressor utilizes a vaneless diffuser such that only blade-pass frequency pulsations at the running speed are of relevance. In this particular case, the operating conditions of the compressor are:

- Speed: 7000 rpm (116 Hz)
- Impeller Tip Diameter / No. of Blades: 10 in (25.4 cm) / 16 blades (no splitters)
- Process Gas: Natural Gas (SG=0.62)
- Discharge Pressure / Temperature: 80 bara / 40°C
- Discharge Pipe Diameter: 9 in (20.3 cm)

Based on the above operating conditions, the primary fixed speed blade-pass excitation frequency is 1867 Hz, and the predicted periodic excitations at the compressor discharge are 0.2 bar. This excitation frequency coincides closely to an acoustic two-pipe diameter radial mode in the discharge pipe at 1875 Hz, and a radial pipe resonance would be expected. The excitation pressure trace was conservatively assumed to be a square wave to make sure the comb filter design properly attenuates higher order frequencies. However, one should note that due to viscous dissipation, some of the higher orders are damped out significantly by the time the waves reach the comb filter such that a more rounded, near sinusoidal time trace wave shape is often found.

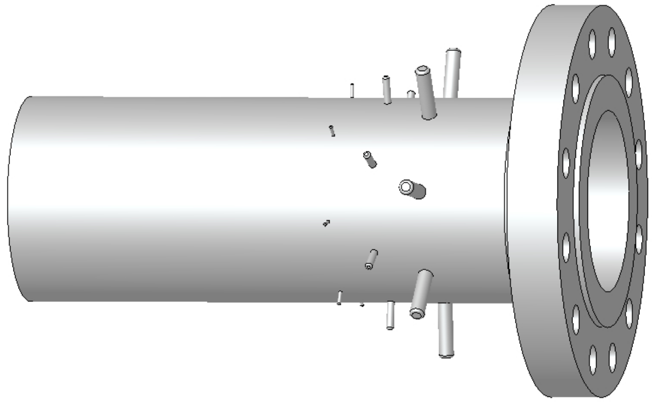


Figure 5: Comb Filter Design for Case Study

A quarter comb filter was designed based on the speed of sound of the process fluid at the discharge pressure and temperature (445 m/s) and the given blade-pass excitation frequency. Basic comb filter design rules are presented later on in this document. However, the resulting comb filter for the above case study was that for the first-stage quarter wave filter. Seven circumferential, evenly spaced stubs were placed 10 cm downstream from the compressor discharge flange, each 5.95 cm long and with a diameter of 1.49 cm. The second and third stage filters were offset, placed 5 cm and 10 cm downstream from the first stage and were 2.98 cm long / 0.74 cm diameter and 1.49 cm long / 0.35 cm diameter, respectively. Each row has seven quarter wave resonator stubs, resulting in a total of 21 stubs. Figure 5 shows a schematic of the comb filter design for this case study.

Case Study Analysis

For this specific excitation and comb filter design, the following four cases were numerically analyzed:

A. No Comb Filter
B. First Stage Quarter Wave Filter Only
C. First and Second Stage Comb Filter
D. First, Second, and Third Stage Comb Filter

For each of these cases, 1D transient analysis models were built in the aforementioned solver. As an example, Figure 6 shows a typical analysis model for Case B. Also, for each of these models, the frequency range was swept from 1750 Hz to 1950 Hz to capture the filter's effective band range.

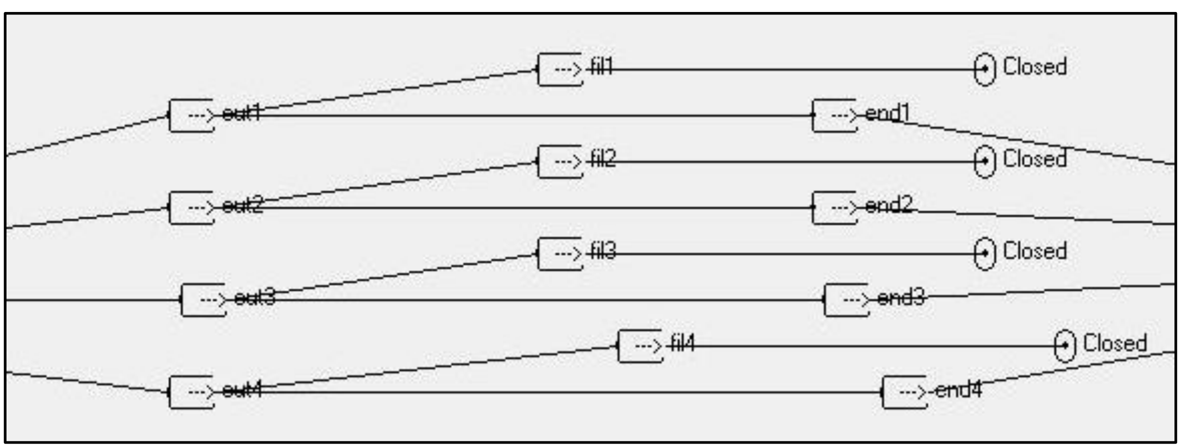


Figure 6: 1D Transient Model for Case B

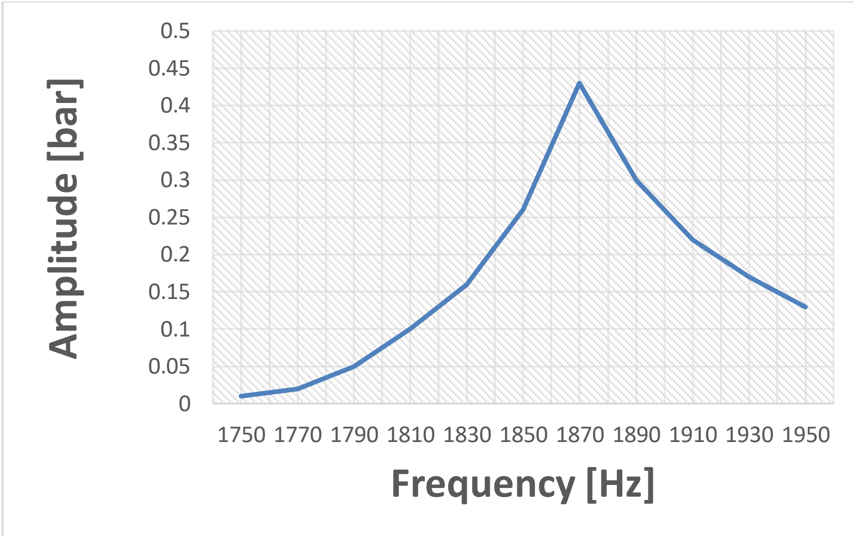


Figure 7: 1X Frequency Response Plot Sweep with No Comb Filter

Results from the case study are shown in Figure 7 and Figure 8. Figure 7 shows only the first order 1X frequency sweep results for Case A, i.e., the case without any filter applied. Figure 7 also shows that although the excitation amplitude at 1867 Hz was only 0.2 bara, the response in the pipe is significantly higher, reaching almost 0.44 bara due to the two-diameter pipe resonance at 1875 Hz.

Figure 8 shows the 1X (1867 Hz), 2X (3734 Hz), 3X (5601 Hz), and 4X (7468 Hz) pipe system response for all cases. The attenuation effectiveness of the different comb filter case design can easily be compared here.

As expected, Case A, with no filter, shows a typical frequency distribution of a mildly smoothed square wave. Case B shows that a single row of quarter wave resonators effectively halves the amplitude of the pulsations but also shifts some of the energy of these pulses to the second order. When a second row of quarter wave resonators is utilized, as shown for the Case C comb filter design, the second order pulsations have also been significantly reduced. Although in this case, some of the second order energy should have been shifted to the fourth order. These high frequency pulsations are naturally damped by the fluid's viscosity such that they are not seen to be significant. Similarly, Case D, with three rows of filters installed, effectively eliminates the pulsations. However, the pulsation attenuation improvement between the two-stage and three-stage comb filter is seen to be minimal in this case. In all studied cases, pulsations at the third and fourth order (i.e., above 5000 Hz) were not found to be relevant.

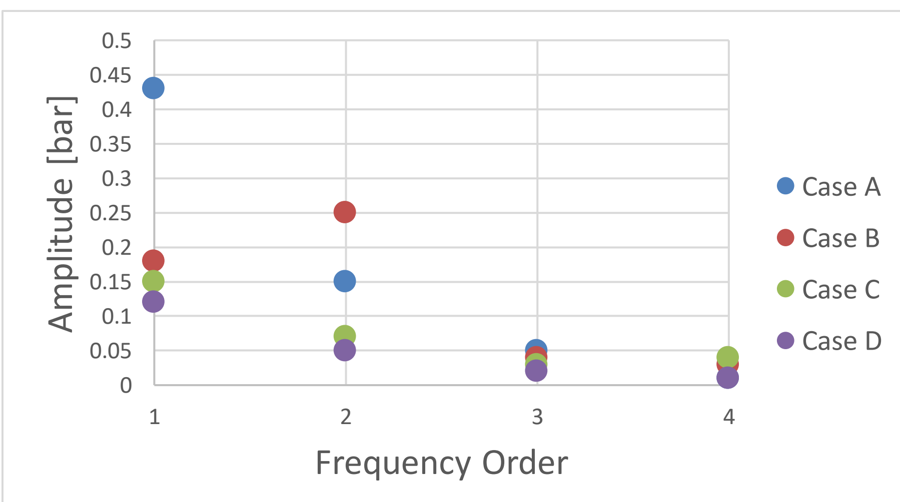


Figure 8: Pipe System Response for Cases A, B, C, and D

Parametric Studies
To evaluate the effect of various comb filter design options, such as number of stages, number of circumferentially placed resonators, and total resonator cross-sectional area, several parametric studies were undertaken. For these studies, the above discussed case study design was utilized as the baseline design, and design parameters were varied to determine their impact on the lower order pulsations. A number of additional parametric studies were undertaken for other baseline designs, but the results are not presented herein. Although these analyses were specific to the case study, some of the results can be generalized to develop basic comb filter design rules.

Figures 9 through 11 show results from some of the parametric studies. One should note that the amplitude results are normalized by the Case A baseline results (i.e., no filter) such that the resulting numbers effectively represent percent filter effectiveness. For example, Figure 9 shows the percent pulsation amplitude for the first, second, and third order pulsations as a function of the number of comb filter stages. As anticipated (and as previously seen in the above case study), a single quarter wave filter stage effectively attenuates 1X but amplifies 2X pulsations. A comb filter with two stages addresses this issue by removing most 2X pulsations. Very little difference is seen between a two, three or four-stage comb filter design. In several other cases and parametric studies, similar results were found such that it is safe to assume that in general, for most cases, a two-stage comb filter design adequately addresses pulsation resonance problems and further comb stages are unnecessary.

When circumferentially placing quarter wave resonator stubs, several design compromises have to be considered. These are:

- The total cross-sectional area of the resonator stubs should be sufficiently large to absorb pulsation energy to attenuate the excitation and resonance energy. This can be accomplished either through wider diameter stubs or by increasing the number of stubs.
- The aspect ratio of the stub should be high enough such that 3D acoustic effects are avoided.
- Stubs should be arranged circumferentially to avoid cross-radial acoustic interaction, i.e., the stubs should not be located directly radially opposing each other on the pipe. This can be achieved by making sure that the number of circumferentially placed stage stubs is based on prime numbers such as 3, 5, 7, 11, etc.
- For practical design reasons, the pipe diameter of the stub should be close to commercially available pipe size.

Thus, it is important to determine the appropriate number of quarter wave stubs and their diameter. Figures 10 and 11 provide case studies for these comb filter design variables with the above Case C as the baseline. Specifically, Figure 10 shows the impact of a number of stubs (keeping their diameter constant) on pulsation amplitude, and Figure 11 shows the impact of stub diameter (keeping their number constant at 7) on pulsation amplitudes. One should note that in Figure 10, the x-axis is total quarter wave stub cross-sectional area as a percent of total pipe cross-sectional area. As anticipated, both charts show similar results. The pulsation amplitudes decrease non-linearly with the number of stubs and the stub diameter, i.e., with total stub cross-sectional area. Figure 10 shows that beyond a 15% cross-sectional area, the improvement in pulsation attenuation is not significant. Similar results have been found in other case studies.

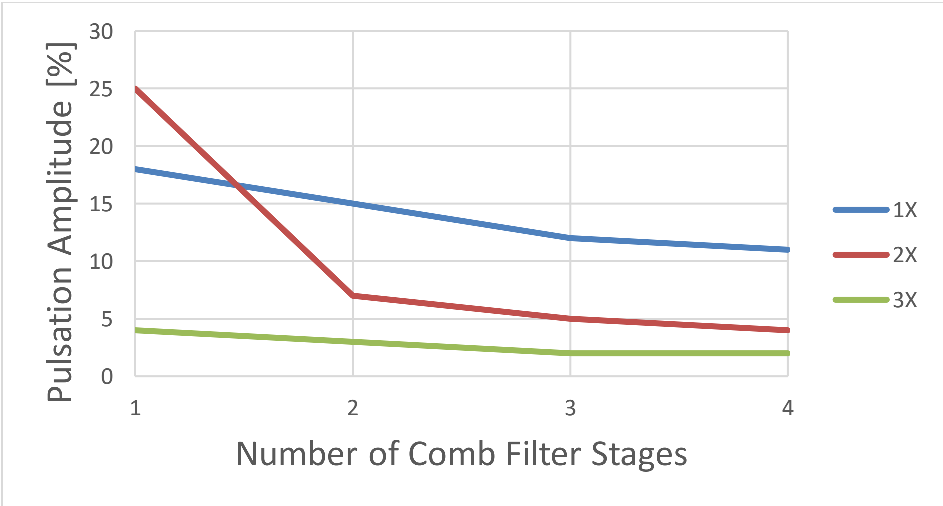


Figure 9: Pulsation Amplitude versus Number of Comb Filter Stages

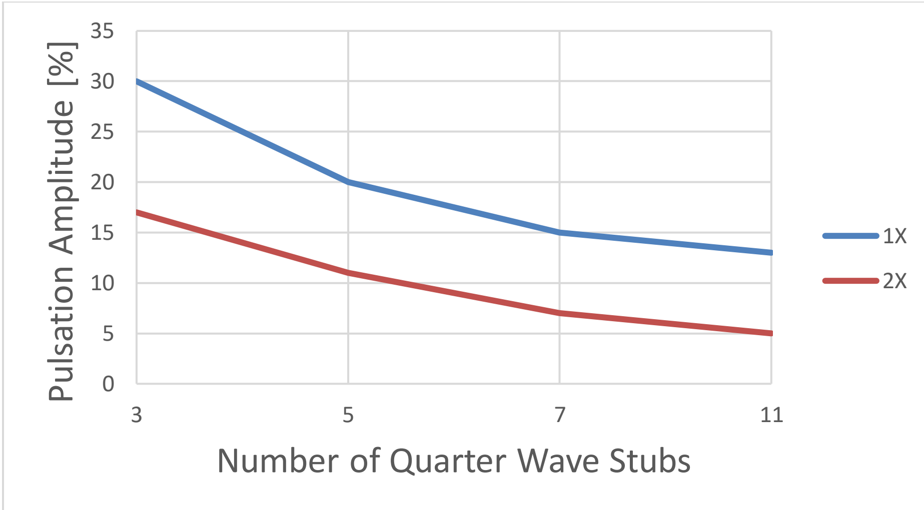


Figure 10: Pulsation Amplitude versus Number of Quarter Wave Stubs per Stage

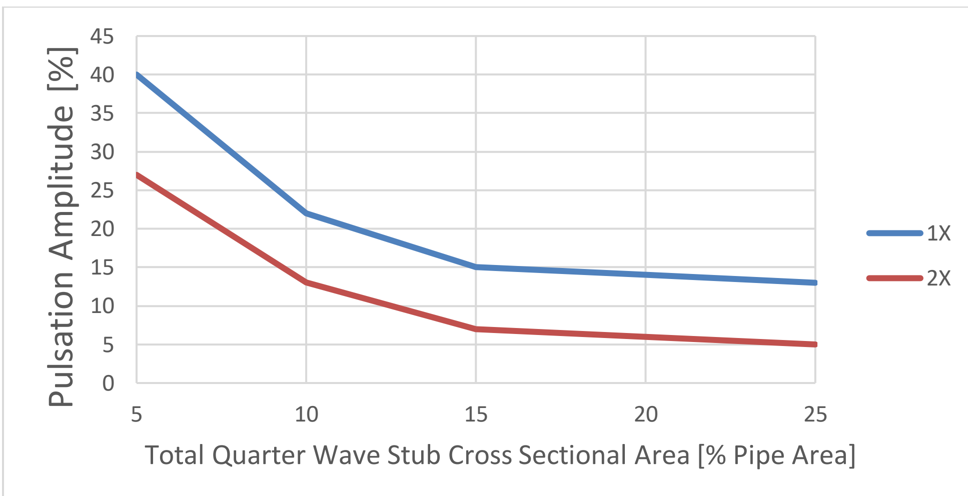


Figure 11: Pulsation Amplitude versus Total Quarter Wave Stub Cross-Sectional Area

COMB FILTER DESIGN

Based on the above described parametric studies, past analysis cases, acoustic filter design literature, and experience with testing quarter wave resonators, a number of comb filter design rules and a step-by-step methodology can be developed for centrifugal compressor blade-pass pipe radial resonance attenuation.

Design Rules

The following comb filter design guidelines are suggested. One should note that these were developed based on specific cases and may not always be applicable for all comb filter designs.

1. A minimum of three evenly circumferentially spaced quarter wave stubs per stage should be used.
2. The total cross-sectional area of first-stage quarter wave stubs should be at least 15% of the pipe cross-sectional area. For second-stage stubs, the cross-sectional area should be at least 7.5% of the pipe cross-sectional area.
3. The number of stubs per stage should be prime numbers, i.e., 3, 5, 7, etc. to avoid radially opposing close-end cross resonances.
4. Downstream quarter wave stubs should be circumferentially offset from upstream absorbers to avoid inducing resonances from upstream vortex shedding.
5. The first-stage comb filter should be as close to the compressor flange as possible (within reason from a design and maintenance perspective). Subsequent stages should not be more than 10 cm upstream/downstream from the previous stage.
6. The diameter of each quarter wave stub should be approximately 5-15% of the pipe diameter.
7. The quarter wave stub aspect ratio should not exceed 1:4 to avoid 3D effects.

8. The quarter wave stub aspect ratio limits the total cross-sectional area of each stub, and the total number of stubs is driven by the need to absorb sufficient pulsation energy.
9. Usually two rows of comb filter stages are sufficient, but analysis should be performed if there is a concern about very high frequency excitation.

Design Process

From the above rules, the basic comb filter design process is reasonably straight forward and can be summarized as follows:

1. Calculate the quarter wave stub length for the first, second, and third-stages based on the desired attenuation frequency and gas speed of sound. Specifically:

$$\text{Length [m]} = \text{Speed of Sound [m/s]} \;/\; (4 * \text{Frequency[Hz]}) \tag{6}$$

2. Determine the stub diameter by not exceeding a diameter-length aspect ratio of 1:4 and considering commercially available pipe sizes.
3. Calculate the number of stubs required to have the total cross-sectional area of all stubs be no less than 15% of the pipe's cross-sectional area. The number of stubs should be based on a prime number, with a minimum of three.
4. A minimum of two comb stages should be utilized to avoid second order pulsations.
5. The second-stage stubs should be half the length and half the diameter of the first-stage stubs.
6. For the second stage, the total cross-sectional stub area should be at least 7.5% of the total pipe cross-sectional area.
7. The second row does not have to have the same number of stages as the first row, but stubs should be radially offset (staggered) between stages.
8. A third row of comb-filter stages is usually not necessary unless there is a concern about very high frequency pulsations. In this case, the design rules of the second stage can be followed.

SUMMARY

Blade-pass pressure pulsation excitations from the suction and discharge of centrifugal compressors can cause acoustic resonance conditions and associated piping vibrations upstream and downstream of the centrifugal compressor. The excitation frequency of these blade-pass pressure pulses relative to the stationary frame is calculated from the speed of the compressor multiplied by the number of impellers blades, and if this excitation frequency coincides with a flange radial acoustic piping mode or a short segment pipe axial mode, resonance and significant associated pressure force induced piping vibrations can result.

A series of two or more progressively 50% length-decreasing quarter wave resonators (i.e, wave length $\lambda/4$, $\lambda/8$, $\lambda/16$, etc.), also called a quarter wave comb filter, can be designed to shift excitation frequencies multiple orders higher, such that their pulses superimpose to form a continuous steady waveform. This can almost completely filter out all excitations. The advantage of a comb filter over Helmholtz arrays is that they (i) can handle the multiple frequencies and higher orders of complex wave forms that are typically seen from centrifugal compressor blade passing pulsation excitation and (ii) they are from a mechanical perspective significantly easier to install in compressor connected piping.

This paper discussed the practical design of such a comb filter, its installation in the downstream piping near the discharge of the compressor, simple design rules, the design process for comb filters, and provides a basic case study modeling results and predictions of its effectiveness. A number of parametric studies for the number of circumferentially located comb filters, stub diameters, and number of filter stages versus pulsation attenuation effectiveness were also presented. Results showed that two to three-stage comb filters can be effectively used to eliminate centrifugal compressor blade-pass excitation induced pipe radial resonances and associated vibrations. The design and installation of comb filters on new or existing compressor installations is relatively straight forward since standard pipe components can be utilized. The goal of this paper was not to demonstrate the functioning of the comb filter but to provide a clear, concise, and detailed step-by-step instruction on how to design an effective comb filter for a centrifugal compressor application.

NOMENCLATURE

b	= Number of blades
c	= Speed of sound
d	= Number of vanes (diffuser)
f	= Response frequency
L	= Acoustic length of pipe span
m	= Length
n	= Integer 1, 2, 3…
p	= Pressure
u	= Velocity
ϱ	= Density
γ	= Specific heat ratio

FIGURES

APPENDIX

None

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