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REVIEW OF CAUSES AND MITIGATION OF CAVITY NOISE IN MACHINERY AND OTHER MECHANISMS

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ABSTRACT

Two significant causes of noise related to cavities are direct and indirect flow induced turbulence/vortex shedding mechanisms. Examples of induced noise can be found in many applications of both closed-flow and open-flow cavities – some with resonance of acoustic modes. An example is a flow valve with a cavity where flow along the cavity gives pulsations either trapped within the valve or exciting downstream piping acoustic modes. There are passive methods of mitigation besides detuning such as modification of the entrance to the cavity, blockage, and use of Helmholtz resonators. Natural frequencies of cavity acoustic modes can be irregular, but for many such as with circular, square, rectangular or axisymmetric shapes can give symmetry of modes. An example is a cavity at the sides of rotating disks, where transverse symmetrical modes having circular and diametric patterns are similar to structural vibratory modes for bladed disks. In the last decade it has been documented that for centrifugal compressors blade passing acoustic pressure pulsation due to Tyler-Sofrin spinning modes can add to alternating stress from non-uniform flow excitation, such as from stator wakes. Cavity acoustic mode excitation then has been termed “triple coincidence” or “triple crossing”, explaining rare documented impeller fatigue failures and likely a reason, at least partially, for some unexplained failures. A novel method described herein is to treat these and similar cavities as fluid-filled disks, then utilize or add blade-like elements within the cavities. The method described (patent application, PCT US1820880) to reduce response of these cavities is to intentionally mistune the elements as has been documented for bladed disk modes. Other applications of this method are possible for many other mechanisms. These modification(s) can alleviate concern for any mechanism having structural vibration excitation acoustically and/or for environmental noise issues.

INTRODUCTION

There are beneficial effects of cavities, namely musical instruments that involve enhanced coupling of structural and acoustic modes; e.g. violins per Gough [1]. Another example is optimization of trapped vortex combustors requiring specific designs of cavities. Helmholtz resonator cavity designs for noise reduction also are tested and evaluated to optimize designs. By contrast cavity acoustic modes can cause high noise and damage to structures, machinery, and many mechanisms. Perhaps the most referenced paper on flow passing over cavities “Rossiter waves” (also termed Rossiter–Heller tones) in-fact resulted from research for a noise problem in a wind tunnel. As Rossiter [2] had stated: *“It was found that the unsteady pressures contain both random and periodic components. The random component predominates in the shallower cavities (length/depth ratio> 4) and is most intense near the rear wall. The periodioc component predominates in the deeper cavities (length/depth ratio < 4) and may form standing wave patterns. It is suggested that the periodic component is due to an acoustic resonance within the cavity excited by a phenomenon similar to that causing edge-tones. It may be suppressed by fixing a small spoiler at the front of the cavity.”* Modified Rossiter – Heller formulae, Heller and Bliss [3], are very useful for rectangular shallow “open-flow” cavities at low compressible flow speeds. There are differences for very long shallow cavities (closed-flow: length to depth > 10) that fill with more fluid during the process of generating acoustic waves, and also have more adverse pressure gradients. Another much cited publication giving a summary of cavity types and issues is by Rockwell and Naudascher [4]. See Fig. 1 for their detailing types and variations.

As shown by Unalmis et al. [5], for very high flow speeds a “closed-box” formula can give better correlation depending on design of a covered opening for flow past a shallow cavity. In addition the authors provide the modified Rossiter-Heller equation, noting: *“This semi-empirical equation has been*

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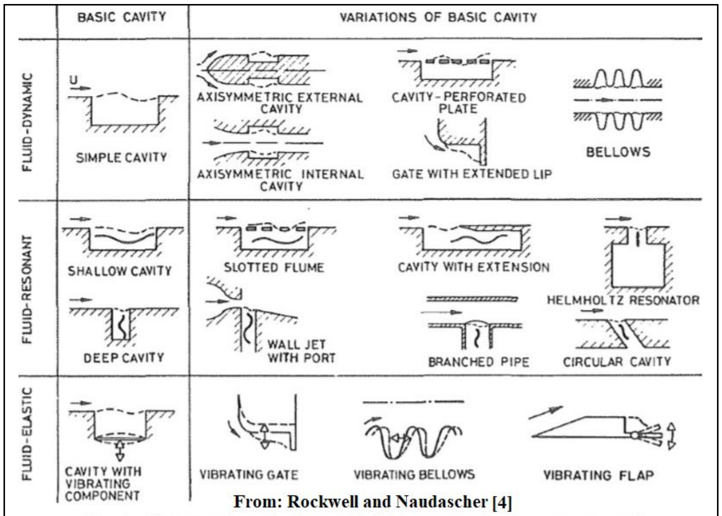
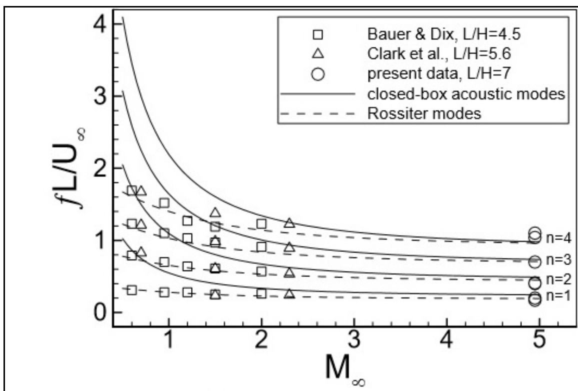
modified by Heller and Bliss such that the sound speed in the cavity is based on the recovery temperature of the flow." The very early references given above for verification equations discuss empirical constants verified by test data, showing a high dependence on length to depth ratio.



From: Rockwell and Naudascher [4]

Fig. 1 Matrix categorization of fluid-dynamic, fluid-resonant, and fluid-elastic types of cavity oscillations

Figure 1. Cavity Types and Potential Flow Oscillations.



From: Unalmis et al. [5]

Fig. 1 Variation of Strouhal numbers (based on cavity resonance frequencies) with Mach number and comparison with closed-box longitudinal acoustic and Rossiter modes (from Ref. 6).

Figure 2. Experimental Data Compared to Two Cavity Equations from Unalmis et al. [5].

Much research has been done for high speed flows as in Fig. 2, a summary for longitudinal modes frequency parameter, also termed Strouhal number, Vs Mach number. Per Roberts [6] there also can be a reduction in modal amplitude with increased supersonic Mach number as well as reduction in the broadband cavity noise. Roberts noted: *"However, there is no such simple and reliable method for the prediction of the peak magnitudes."* That opinion likely is mainly due to aerodynamic damping unknowns. Also see Roberts [6] for references giving possible calculation procedures. It also has since been reported by many such as Gloerfelt [7] that deeper cylindrical cavities in fact have higher modal radial and transverse modes that can cause even higher resonant response amplitudes, up to 10 times as high for side branches compared to the main pipeline. Also Ziada and Lafon [8] give analysis and experimental data for such a symmetric cavity, including radial modes.

The remainder of this paper is to mainly assist in improving calculations and response mitigation, emphasizing deeper cavities that are much more likely to have transverse diametral modes, also termed azimuthal or cross-modes. These modes, including those for axisymmetric designs, will be shown to be able to be intentionally mistuned to reduce response, as opposed to changing the frequency of excitation and/or responding modes, termed "detuning".

NOMENCLATURE

BPF – blade passing frequency
EO – engine order – i.e. harmonic of operating speed
n – number of nodal diameters for disk & transverse modes

Deep Cavities

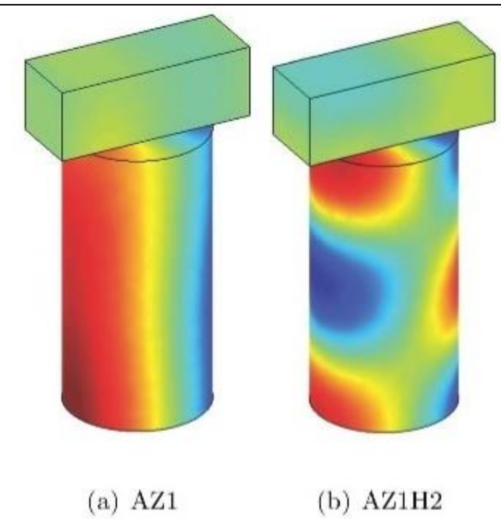


FIG. 5. (Color online) Numerical WEM results: normalized real pressure response of the cavity to monopole excitation at orifice opening of cavity to wind tunnel test section. Two acoustic resonances are shown: (a) first azimuthal mode AZ1 (He = 1.85, $A_{m,n,q} = A_{1,0,0}$); (b) second azimuthal-longitudinal combination mode AZ1H2 (He = 2.42 $A_{m,n,q} = A_{1,0,2}$).

From: Bennett et al. [9]

Figure 3. One-Diameter Cross Modes in a Deep Cavity.

Deep cavities are typically where depth to length ratio is greater than one. Bennett et al. [9] give a comprehensive summary for grazing flow exciting both longitudinal and azimuthal cross modes. As for shallow cavities, excitation is a function of Mach number. Fig. 3 gives their depiction of two one-diameter modes, the first mode and one with longitudinal nodal lines. Lock-on between these different resonant modes and shear layer excitation were measured by the authors and observed to occur when adjusting flow speed.

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Axisymmetric Cavities

Designs that are axisymmetric can also be shallow and generate Rossiter-type resonance. Often the more contentious having problematic longitudinal and also cross-modes are deeper cavities such as described by Barannyk [10], causing problems with piping and numerous mechanisms in contact with fluids.

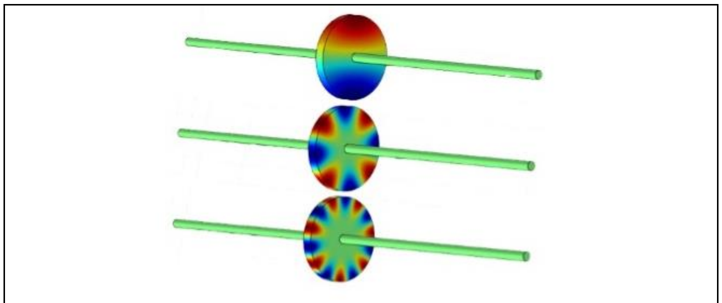

Fig. 4 Diametral acoustic mode shapes with the corresponding frequencies of f_1 = 992 Hz (top plot), f_4 = 2,891 Hz (middle plot) and f_7 = 4,618 Hz (bottom plot).
From: Oshkai [11]

Figure 4. One, Four, and Seven-Diameter Mode Shapes for Deep Axisymmetric Cavity.

Oshkai [11] also reviews deep axisymmetric cavities for pipeline-cavity systems, with excitation from compressible grazing flow across the opening. As depicted in Fig. 4, as number of diameters increase, amplitude for acoustic pulsation frequencies at the opening to the pipeline is much lower and thus less excitable by the flow source itself, as well as by acoustic components within the flow. For other designs the corresponding shear layer mode excitation of transverse modes is also from flow over the cavity. Besides those given by the Oshkai [11], his references also have many examples of such excitation for piping systems. Similarly flow over other cavities such as within valves can give excitation of transverse modes in downstream piping. Some modes can then become locked-in. Grazing fluid flow past cavities within internal components of a machine or mechanism can also occur. An example is a deep circumferential gap at casing joints. Besides direct grazing flow excitation there can be other sources outside and internally to excite radial and transverse acoustic modes in cavities, annular components, casings, etc. Asymmetry of stationary elements within a turbomachine annulus reduces noise besides direct excitation such as from wakes as explained in Kushner and Pettinato [12]. Another method for phase cancellation for reflected noise upstream and downstream is to change the distance to be one-half the wavelength between the noise source and reflecting surface. Alternatively there can be acoustic mode splitting and wave reflection upstream and downstream as explained by Zhao et al. [13], such that optimum aerodynamic damping can be obtained as shown to prevent fan blade flutter.

Pulsations at BPF (blade passing frequency) and harmonics from machinery such as in Fig. 5 can excite attached pipe acoustic modes along with those inside the casing. Cavities on side of disks are axisymmetric; for some designs diametral acoustic modes can match identical vibratory mode shapes of adjacent structures such as centrifugal impellers. Typical BPF noise levels are not excessive outside thicker casings compared to thinner, symmetrical piping, especially without vanes in diffusers.

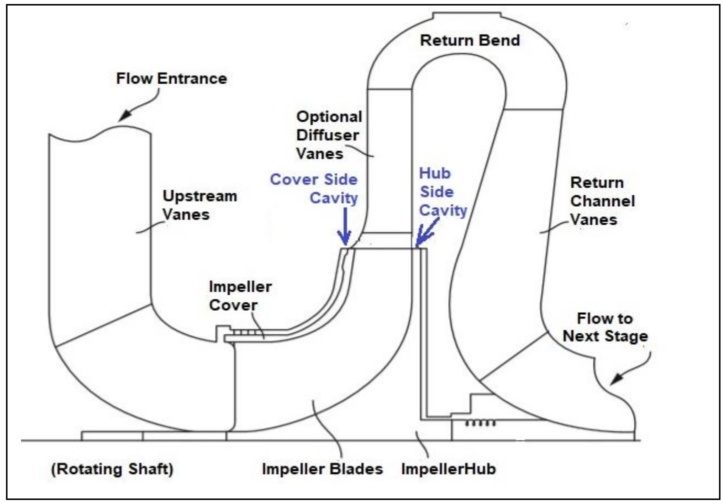

Figure 5. Centrifugal Compressor Stage with Covered Impeller.

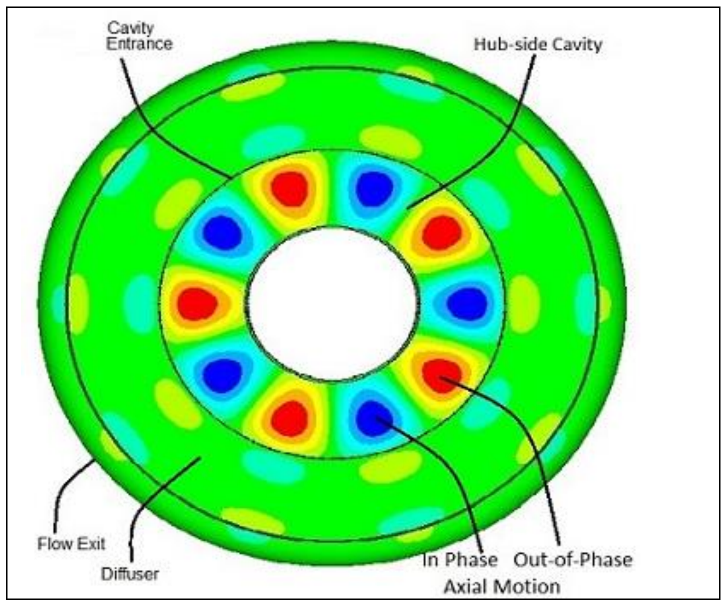

Figure 6. Five-Diameter Mode for Compressor Stage.

Shown in Fig. 6 is a cavity acoustic mode at the sides of an impeller; high amplitudes occur if resonant at BPF. By contrast to Fig. 4, centrifugal compressor cavities on the sides of impellers have high amplitudes at their opening where there is potential excitation from impeller exit flow sources containing BPF pulsations. Rather than grazing excitation such as for piping cavities, leakage flows into impeller side cavities so that flow effects can change cavity mode frequencies somewhat. Annex A below includes a summary for response of cavities at the sides of rotating compressor impeller disks, where

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transverse symmetrical modes having circular and diametric patterns can match structural vibratory modes for bladed disks. For some cases where there is a matching disk mode shape to that as shown in Fig. 6, axial excitation can then cause high alternating stress. In the last decade or so it has been documented that blade passing acoustic pressure pulsation with spinning modes, explained by Tyler and Sofrin [14], can add to alternating stress from non-uniform flow excitation, such as from stator wakes. Impeller side cavity acoustic mode excitation and response has been termed "triple coincidence" or "triple crossing", explaining rare documented impeller fatigue failures, and likely a contributing factor for some unexplained failures.

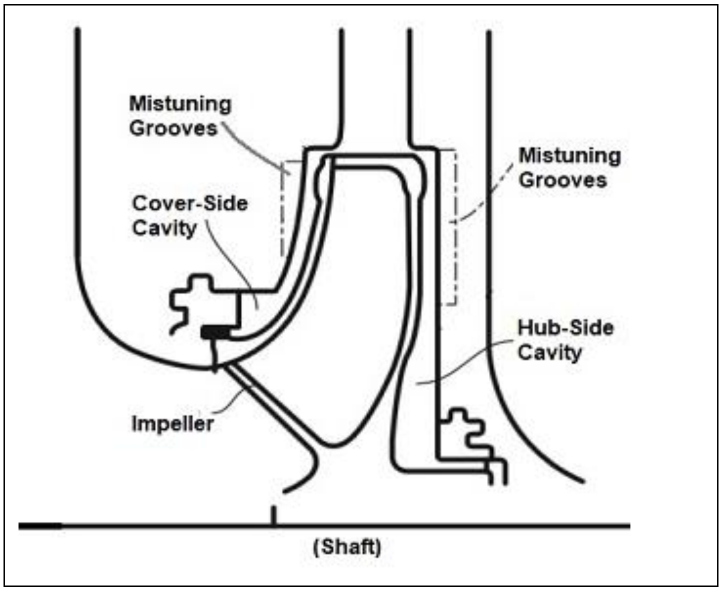

Figure 7. Mistuning Using Grooves at Cavity Stationary Casings.

A method described herein is to treat these and similar cavities as fluid-filled disks, then utilize or add blade-like elements, e. g. grooves or ribs within the cavities. The method is described within patent application, PCT US1820880, reducing response by intentionally mistuning the elements as has been documented for bladed disk structural modes – see Annex A. Intentional mistuning not only can prevent instability (blade flutter) but with certain harmonic patterns can reduce forced response of bladed disks. See for example Martel and Cuiping [15]. Hohl and Wallaschek [16] show amplitudes can be below tuned value with large intentional harmonic mistuning. The acoustic modes having diametral and circumferential nodal lines are in effect isolated fluid-filled disk modes. In lieu of utilizing detuning "triple coincidence" for a stage, this method utilizes specific harmonic mistuning of blade-like elements that separates the two natural spinning mode frequencies and also improves the damping. Depictions of grooves for mistuning impeller side cavities are in Figs. 7 and 8.

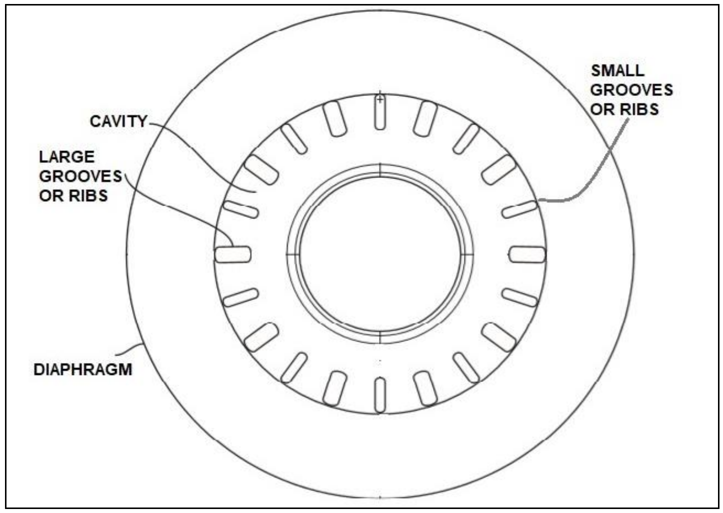

Figure 8. Mistuning Pattern with 10X Pattern for Mistuning 5-Diameter Acoustic Mode in Cavity.

TUNED DESIGN [Hz]	COVER & HUB Grooves 5 Large 5 Small [Hz]	% CHANGE	COVER & HUB Grooves 10 Large 10 Small [Hz]	% CHANGE	MODE SHAPE
1617.7	1616.5	-0.1%	1615.8	-0.1%	
1617.7	1619.5	0.1%	1620.5	0.2%	
2682.0	2554.9	-4.7%	2489.0	-7.2%	
2682.0	2704.5	0.8%	2670.0	-0.4%	
3028.9	2881.4	-4.9%	2816.5	-7.0%	
3028.9	3050.4	0.7%	3020.4	-0.3%	

Figure 9. Calculated 5-Diameter Mode Shape Frequencies with 10-Diameter Pattern of Mistuning Grooves.

Changes using mistuning grooves for three, 5-diameter mode frequencies using ANSYS finite element analysis are in Fig. 9. The tuned design has identical forward and backward spinning mode frequencies for each mode. The first mode at 1618 Hz is for the entire space, with high amplitude mainly within the vaneless diffuser. This mode has negligible changes with grooves in the casing walls. The use of five large grooves alternating with five smaller grooves has a 10-diameter harmonic component and changes the other 5-diameter cavity frequencies. Largest frequency changes for two cavity modes, 2682 Hz & 3029 Hz, are with 10 wide grooves alternating with 10 narrow grooves having a strong 10-diameter mistuning pattern. This pattern with a harmonic twice the number of nodal diameters has both forward and backward spinning wave frequencies reduced from the tuned case.

Test data using an internal splitter plate inside a cavity showing lower amplitudes when separating the two spinning mode tuned frequencies is in Barannyk [10], who also states: *"These results indicate that due to the presence of the splitter plate, the azimuthal behaviour of the diametral acoustic modes can switch from spinning or partially spinning to stationary. In*

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addition, the decrease in the maximum pressure amplitude indicate that the spinning modes are capable of producing more energy compared to stationary modes, as it was indicated in (Aly and Ziada, 2011)".

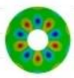
Tuned Design [Hz]	MODE SHAPE Hub Side	COVER & HUB Grooves 10 Large 10 Small [Hz]	% CHANGE	9 RIBS HUB SIDE [Hz]	% CHANGE	MODE SHAPE Hub Side
1617.7		1615.8	-0.1%	1535.3	-5.1%	
1617.7		1620.5	0.2%	1535.3	-5.1%	
2682.0		2489.0	-7.2%	2696.4	0.5%	
2682.0		2670.0	-0.4%	2696.4	0.5%	
3028.9		2816.5	-7.0%	3041.2	0.4%	
3028.9		3020.4	-0.3%	3041.2	0.4%	

Figure 10. Five-Diameter Mode Shape Frequency Changes From Tuned Design With 20 Grooves Vs Nine Ribs.

Other mistuning such as having nine equally-spaced grooves or ribs alter the 5-diameter mode frequencies equally, i.e. just due to effect of changes in cavity space area. That is shown in Fig. 10 where both forward and backward spinning mode frequencies are increased the same amount for the predominant cavity modes at 2682 Hz and 3029Hz. This proves that sufficient mistuning level with two times five, i.e. 10-diameter pattern can eliminate spinning modes and reduce acoustic response.

A similar intentional harmonic mistuning method is given in Hagen et al. [17] for mitigating flames interacting with circumferential acoustic modes in a combustion chamber, also explained by Banaszuk et al. [18]. These modes can give high noise termed screech and howl in jet engines that can also cause component fatigue. Flow with an intentional two times n-diameter pattern is used in their method to mistune acoustic modes, using variation in fuel flow and thus acoustic velocity due to temperature changes. There could be intentional mistuning patterns for other machinery and mechanisms using different flow variations.

MITIGATION OF CAVITY MODE RESONANCES

Cavity Longitudinal Modes

This paper providing methods for mistuning utilizing blade-like elements for diametral mode patterns may not be applicable for shallow cavities (length to depth > 1). There could be rare cases where a high frequency is found, such as a one or two-diameter mode that could be mistuned. Otherwise a comprehensive review of many potential methods to reduce amplitudes of Rossiter-type mode resonance for high speed flows is in the thesis by Roberts [6]. Use of Helmholtz resonators including design for multiple modes is explained in Roberts et al. [19].

Cavity & Piping Longitudinal Mode Standing Waves

Mistuning methods described above for an individual cavity are not normally applicable for typical low frequency longitudinal modes. See Ziada and Lafon [8] and Tonon et al. [20] for detuning besides other passive methods such as baffles. There could be multiple cavities that interact with each other, and add to excitation of diametral modes in a main pipe for example. Also, mistuning of patterns could be devised for main piping diametral modes that match side-branch longitudinal modes, as did the external loudspeaker in tests by Li et al. [21]. Simons et al. [22] show test data compared to older references as do Ziada and Lafon [8] who provide passive methods for reducing piping side-branch standing waves that could be applied to many designs, stating: *"These include changing the dimensions of the sidebranch, using antivortex inserts, detuning the side-branches by making them of different lengths, and adding spoilers at the upstream corner of the cavity. Additional remedial methods have been discussed within the scope of the presented industrial examples of acoustic resonance."*

Cavity Transverse (Cross-Mode) Acoustic Modes

Intentional harmonic mistuning with two times number of nodal diameters, summarized above for compressor impellers, could be considered for other mechanisms. The novel method described above is to treat these and similar cavities as fluid-filled disks, then utilize or add blade-like elements within the cavities. The method reducing response of these cavities is described in patent application PCT US1820880, intentionally mistuning the elements as has been documented for bladed disk modes. The acoustic mistuning could be applied to the cavity, annulus etc. itself; e.g. at passive elements at the entrance, or adding grooves, ribs etc. acting as blade-like elements, or else applied to some of the system connections such as piping near the source. In addition the affected structure could also be intentionally mistuned for the same mode shape to further reduce response. The blade-like elements could be on the side of the structure side of the cavity intentionally mistuning both acoustic and structure modes. Designs for multiple cross-modes with respective intentional mistuning patterns could also be considered.

Other potential applications besides gas compressors for the mistuned blade-like element arrangement to reduce response of acoustic modes include fluid-handling, including air, mechanisms such as engines, machinery (e.g. pumps, turbines, fans), fuel cells, piping, ducts, diffusers, nozzles, valves, silencers, mufflers, seals, heat exchangers, airframes, tires and wheels, rockets, combustion chambers, vehicles, speakers, and double-pane windows. Some examples are reviewed below:

Pumps

Fluid damping is much higher for liquid pumps and turbines and thus reduces likelihood of fatigue. Some do have vaned diffusers having much more risk. Franke et al. [23] show BPF interaction with downstream vanes, Ala. Tyler-Sofrin pulsation patterns for compressors.

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Acoustic resonance at BPF is less of an issue for pumps. Dickau and Perera [24] describe an acoustic resonance with data taken inside a crossover pipe at the second harmonic of blade passing frequency. There also was torsional excitation of the rotor at 1 x BPF so the pump design was suspect and designs with seven instead of six blades was the solution.

Valves

Many types of valves have caused high noise within the design as well as emanating into attached components especially thin-walled piping, at changes in high flow conditions besides when throttling flow. Residential noise at plant boundaries often requires reviews. Control valves can have instabilities downstream of the seat with possible damage as described and solved with geometry modifications by Hardin et al. [25]. Data for a high pressure bypass valve is given in Aly and Ziada [26]. Testing is described by Simons et al. [27] for a gate valve with Rossiter-Heller modes. High noise level was remedied by just inserting the original plug until an optimum location was found. Excitation of pulsations within piping for this case was thus reduced. Other valve testing and calculations is in Lacombe et al. [28]. Scale model test of seat modifications for a 28-inch diameter valve is described in Janzen et al. [29]. Intentional harmonic mistuning of the piping entrance for specific modes could be contemplated whenever the frequency is such that it easily is transmitted (above cut-off frequency).

Vehicles, Aircraft Fuselage, Landing Gear, Etc.:

Testing and analyses are needed for applications including stowing weapons in cavity-type bays to increase stealth, controlling the flow fields; e.g. exposed openings of airborne vehicles and observatories. For these and other applications subsonic, transonic and supersonic conditions alter the empirical data and calculation procedures; see aforementioned Unalmis et al. [5]. As can be excited in cylindrical designs (wheel wells etc.), some rectangular cavities could have higher frequency responsive symmetrical modes; refer to Bolduc et al. [30]. An alternate calculation procedure by Schmit et al. [31] that includes boundary effects due to doors, flaps etc. is promising. The authors use a modification of Lighthill's three-dimensional acoustic equation in analysis of hydrodynamic / acoustic wave interaction. Optimized size and weight baffles, ribs, grooves etc. could be used to apply specific harmonic mistuning as for deep axisymmetric cavities. A caution for calculations and testing cavities and/or their scale models is to insure all final structural elements are actually located inside that could affect the space, flow effects and thus modal frequencies.

SUMMARY

Details within this paper describe acoustic modes that can exist in many mechanisms and can be mitigated to eliminate fatigue and/or environmental issues. For those with problematic transverse modes, intentionally mistuning specific mode(s) is an option to consider, either using elements within the cavity or adding blade-like elements. The method is also an option to add to or replace methods using fluid flow mistuning, as well as to supplement intentional mistuning of affected structures such as rotating bladed disks. Other applications of this method are possible for many other mechanisms. These modification(s) can alleviate concern for any mechanism having structural vibration excitation acoustically and/or for environmental noise issues.

Note: The product and designs disclosed in this paper are protected by U.S. Patent No. 9,581,034 and patent application PCT US1820880, both owned by Elliott Company.

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ANNEX A

COMPRESSOR ACOUSTIC RESONANCE REVIEW

Vortex shedding has caused flow instabilities, e.g. at blade tips, as well as excitation of acoustic modes within cavities, both in circular and annular spaces. For an annular space Case A-2 in Kushner [32] is a summary of surge and noise issues with industrial axial compressors with blade fatigue failures, where high noise was also noted for an atmospheric inlet, 12-stage compressor. With original inlet casing vanes, only at certain flows vortex shedding frequencies locked on to acoustic modes in the flow annulus, giving concern acoustic pulsations were causing the fatigue failures. Strain gauge testing in the field using radio telemetry showed that was not the cause. Response was very low for the blades using 12% chromium stainless steel material, having beneficial material damping at resonance with lower harmonics of speed. There were no signs of significant excitation from the acoustic resonance. Rather, poor surge controls and continuous surging was proven to be the blade failure root cause. A modification of strut trailing edges was still implemented that greatly reduced noise level at the inlet pipe.

Camp [33] described spinning acoustic modes within a 6-stage low-speed axial compressor. Blade vibration with acoustic resonance for a transonic compressor stage also due to vortex shedding was measured by Holzinger et al. [34]. Axial compressor blading has also been prone to excitation due to pulsations at rotating BPF. As dynamic pressures are low for very high noise levels, it requires a resonance to have excitation high enough applied to rotating blades to be of concern. An example is given by Hellmich and Seume [35]. An acoustic 2-diameter mode response was described by Eisinger and Sullivan [36] for a centrifugal fan impeller. This disk may have been excited from the extreme turbulence of high flow rate, i.e. overload. Eckert [37] describes tests of a covered, radial fan impeller where diffuser vane excitation was considered and eliminated. The cause of fatigue failures was an instability coupled to gas acoustic pulsations on the shroud side inside a large axial cavity having tip seals; removal of the large cavity eliminated the problem.

Interaction structural resonance between rotating blades and stationary vanes is reviewed by Kushner [32], including effects of mistuning. In a similar manner rotating acoustic pulsations at BPF (blade passing frequency) can cause significant excitation of compressor impellers, albeit in rare cases and likely only at the extremes of operation near surge and perhaps overload. For those conditions, noise level increases and thus gives higher excitation within a stage, including inside the cavities on the sides of the impeller. Cases for noise at BPF are given by Kushner [32] and Kushner et al. [38], including other references on effect on piping noise. In addition see a more detailed example of piping failure resolution in Kushner et al. [39]. Design procedures are most important when there are vaned diffusers in close proximity to the impeller tip. Rotor-stator interaction for pressure pulsations has long been known using the equation as explained by Tyler and Sofrin [14] for number of rotating blades, B and number of stationary vanes, S:

$n = i \times B \pm j \times S;$ where i and j are integers, typically 1 or 2.

A positive value for “n” has modal shape with diametral nodes rotating in the same direction as the rotor; negative values give counter rotation.

Some recent papers brought the potential of acoustic interaction with impeller structural modes to a new understanding. Petry et al. [40] confirmed analysis given in their earlier paper, Konig et al. [41], while also reviewing swirling leakage flow effects. Their papers give analytical and confirming test results for two high-pressure compressor cases where modes of failed impellers were resonant at harmonics of stationary vanes – not at rotating blade-passing frequency. It was shown for two different 17-bladed impellers there was a 5-diameter disk mode with EO 22, along with a side cavity acoustic 5-diameter mode resonance at EO 17 from BPF. Thus “triple coincidence”, also termed “triple crossing”, could give significant bladed disk mode response at a specific speed for both failures:

- Tyler-Sofrin acoustic spinning mode with backward spinning 5-diameters; as n = 17 blades – 22 vanes = -5
- 5-Diameter impeller disk mode = EO 22
- Cavity gas acoustic mode with 5 diameters = EO 17

A triple coincidence is needed, the same structural and acoustic mode shape, along with the correct number of stationary vanes for Tyler-Sofrin spinning acoustic modes at BPF, exciting the cavity acoustic mode. The modification given by Konig et al. [40] for both failure cases was to “detune”; changing number of stationary vanes, from 22 to 25 for Case A and from 22 to 19 for Case B. For both modifications, disk mode frequencies did not change and blade-passing frequency remained the same, leaving the number of impeller blades at 17.

There thus is conclusive proof that rotating BPF is not a concern if it matches frequency of a disk diametral mode as some have claimed. Rather, noise at BPF excites at traveling wave frequency, one of the dual disk diameter modes, plus or minus number of diameters times speed. Depending on spinning mode direction, damping can be higher or lower. A suggestion is whenever there is an impeller failure, especially for a high-pressure “dry” gas compressor, acoustic analysis of gas modes at the sides of the cover and hub disk should be considered in root cause failure analysis, while verifying the two other required factors for “triple coincidence” are met. The acoustic analysis described is similar to the disk interaction resonance equations where phase must be matched. The fluctuating acoustic pressure mode shape is a function of the number of

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rotating blades and the number of stationary vanes. The question is answered – high noise at BPF can excite bladed disks, but for complex phase effects rarely causes impellers to fail.

Richards et al. [42] utilized strain gauge data to confirm potential impeller disk modes coincident with side cavity acoustic resonance. Their report did not show a "triple-crossing" match, but did find that there was BPF pulsation pattern caused due to a combination of numbers of upstream vane wakes and downstream vane acoustic reflections. In this case there was no corresponding serious acoustic resonance of the gas in the side cavities due to Tyler-Sofrin spinning modes, so it was concluded that dynamic strains were not of concern.

Besides normal large increase for designs interacting with vanes in diffusers, acoustic pulsations at BPF and at harmonics of BPF can increase for flows at the extreme ends of compressor performance maps. Often reports such as Petry et al. [40] do not give details on operation of the compressors that may have been the underlying cause. They noted many similar compressors had the same possible triple coincidence and had not failed. That agrees with the author's experience; in one case there was no doubt that there was extensive inadvertent operation near and in problematic surge, giving higher noise at BPF and affecting aerodynamic damping. Other similar failure cases could also have operated too long near or in surge, or with excessive incidence angles at or approaching choke (overload). Besides acoustic resonance one must consider other flow related excitation such as given by Kushner [32], Kushner et al. [38] and others, e.g. Ju et al. [43].

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