A Review on Numerical Study of Acoustic Waves for an Unsteady Flow past a Circular Cylinder

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***Abstract* - The study of flow past single circular cylinder is one of the areas where extensive research has been carried out especially for the vortex shedding. But the acoustic related phenomenon for the same case is not much well understood. Characterizing the problem of fluid structure interaction is very difficult. In this review the term acoustic wave is more focused. The second important term that is most related is the vortex wake. In case of getting converged numerical solution, there must be separate computation of flow and noise. The assumed source of noise may be the vortex itself as especially when the flow becomes unsteady. Very few numerical studies have been done that could show the interaction or the coupling mechanism of acoustic wave and vortex wake. The generation of noise is due to many reasons such as vortices in the wake region, turbulent nature and so on. But the challenging task is to measure the relative intensity between various parameters like sound pressure level, vortex shedding frequency, etc. LES could be the good turbulence model as the variation in turbulence results in variation in acoustic wave nature.**

***Index Terms* - Vortex shedding, Sound pressure level, Reynolds Number, unsteady flow, LES, Strouhal Number**

I. INTRODUCTION

Flow-induced noise is becoming a major issue in different types of industries. It is the result of fluid structure interaction. The assumed sources of flow noise can be fluctuation in mass flow, fluctuating pressure on the surfaces cylinder where the flow becomes unsteady) and turbulent wakes that sheds downstream the cylinder. For flow over circular cylinder, acoustic waves are generated that may be due to several reasons such as 1) incident turbulence 2) small scale vortices in the wake 3) transition of turbulent boundary layer 4) flow separation.

Important quantities for characterizing acoustic waves are acoustic pressure, particle velocity, particle displacement and sound intensity. The speed of acoustic waves is the same as speed of sound which depends on the medium they are passing through.

Acoustic wave can be reflected by a solid surface. If a travelling wave is reflected, the reflected wave can interfere with the incident wave causing a standing wave in the near field. As a result, the local pressure in the near field is doubled, and the particle velocity becomes zero and flow becomes stagnant. Attenuation causes the reflected wave to decrease in power as distance from the reflective material increases. Particular to numerical computation of noise, mainly two models are available named Ffowcs-Williams & Hawkings acoustic model and FW-H model.

A. Background

Studies of acoustic resonance in heat exchangers have produced a wealth of knowledge. The research focused in four general areas are 1) flow excitation mechanisms, 2) damping factors, 3) sound pressure level predictions, and 4) factors that affect the acoustic resonance phenomena.

The majority of this work has been experimental. This research has produced 1) Strouhal number maps for the different cylinder array configurations at acoustic resonance and non-resonant conditions, 2) empirical damping factors that try to predict if an acoustic resonance will develop, 3) empirical relations to determine sound pressure level for different array configurations, and 4) general understanding of the phenomenon.

There are, however, different issues that have not been addressed. There are no empirical or fundamentally based models that address the different relevant parameters responsible for the noise generation and attenuation of bluff bodies inside ducts. The current empirical damping models do not work in many situations and are only applicable when there is a coincidence condition, that is, when the noise frequency produced by the array matches an acoustic natural frequency of the duct. Additionally, these models in general do not account for the effects of cavity size, number of cylinders in the array, and other relevant parameters that affect the noise produced in the duct. Damping models such as the ones available today do not give information on the sound pressure levels produced at resonance and non-resonance conditions.

B. Description of flow-induced acoustic resonance phenomena in heat Exchangers

Flow-induced noise in heat exchangers and in particular acoustic resonance is a phenomenon that has been a problem for a long time. Acoustic resonance problems in heat exchangers can be present in a variety of applications, including chemical process exchangers, air heaters, power-generation boilers, marine boilers, conventional power plants, nuclear power plants, and heat-recovery heat exchangers. Other equipment in which acoustic resonance problems caused by the flow of gases over bluff bodies inside a chamber occur include turbojet engine compressors, turning vanes of wind tunnels, plates in a wind tunnel, and the combustion chambers of rocket engines.

The flow-induced noise is caused when gases flow transversely to both an array of bluff bodies' typically circular cylinders and a container cavity. Acoustic resonance has been observed for several different tube array configurations inside cavities of different shapes. The most common heat exchanger configurations containing tube arrays are rectangular and cylindrical ducts.

Flow-induced noise in heat exchangers is a very complex phenomenon. The reasons of complexities are the noise generation and attenuation mechanisms of this type of sound source and by fluid-acoustic-structural coupling effects. To predict the flow-induced noise from bluff bodies inside a duct, it is necessary to know the type of aero acoustic source created by the flow and its strength and directionality. Additionally, acoustic damping and sound attenuation mechanisms inside the duct must be considered. In certain cases the sound can also influence the vortex generation process.

II. A LITERATURE REVIEW

In heat exchangers, such as boilers for commercial use, acoustic resonant noise is occasionally generated in the ducts when gas is flowing laterally with respect to the axis of the tubes. The acoustic resonant noise generated from heat exchangers is usually caused by the resonance of acoustic modes inside the boiler and vortex shedding from the tube banks. Many studies have been published on the excitation mechanisms causing acoustic resonance in the tube banks in cross-flow.

Hanson & Ziada [1] have investigated the effects of acoustic resonance on the dynamic lift force acting on the central tube. There were two effects detected of resonant sound field includes generation of "sound induced" dynamic lift because of resonant acoustic pressure distribution on the tube surface and its interaction with vortex shedding. Sound enhancements coefficients and sound induced lift force development is carried through numerical solution. Hanson et al., [2] investigated aero acoustic response of two side-by-side cylinders against cross flow. It is concluded that acoustic resonance synchronizes vortex shedding and eliminates bistable flow phenomenon. There is a constant strouhal number noticed for Vortex shedding is which excites acoustic resonance.

Eisinger & Sullivan[3] have noticed strong acoustic resonance with acoustic pressure reading 165 dB for package boiler at n full load, suppression of resonance (lower frequency) through baffle covering with downstream section and the development of another resonance (higher frequency) in the upstream section that was without baffle. Feenstra et al., [4] carried out experimental investigation of the effects of width of test section for measuring the acoustic resonance with a small pitch rates staggered tubes. The conclusion was that it over predicts the maximum acoustic pressure versus input energy parameter. The acoustic frequency can be predicted by the mode number (a dimensionless integer), and shell diameter. The lowest acoustic frequency is achieved when mode number is 1 and D the characteristic length is the shell diameter. The acoustic frequencies of an exchanger can be excited by either vortex shedding or turbulent buffeting.

Barrington [5] indicated that the exciting frequencies are within 20% of an acoustic frequency; a loud sound was detected. Acoustic vibration becomes destructive when it is in resonance with some component of exchanger. That showed some energy lost in presence of acoustic vibration. The acoustic frequencies of shell can be changed by inserting a detuning plate parallel to the direction of cross-flow to alter the characteristic length.

Hamakawa & Fukano [6] focused vortex shedding in relation with the acoustic resonance in staggered tube banks and observe three Strouhal number (0.29, 0.22 and 0.19). There was no resonance inside tube banks for the last rows of tube banks. The vortices of 0.29 and 0.22 components alternatively irregularly originated.

Hamakawa & Matsue [7] investigated the relation between vortex shedding and acoustic resonance in a model (boiler plant) for tube banks to clarify the interactive characteristics of vortex shedding and acoustic resonance. The change in velocity due to vortex shedding was noticed inside the tube banks at the Reynolds number (1100-10000) without acoustic resonance and vortex shedding frequency of low gap velocities. Kumar et al. [8] in their review stated that controlling the vortex induced vibrations is of much importance in practical applications where active or passive control could be applied.

Oengoren and Ziada [9] have investigated the coupling mechanism between the acoustic mode and vortex wake, which may occur near the condition of frequency coincidence. They have investigated the system response both in the absence and in the presence of a splitter plate, installed at the mid-height of the bundle to double the acoustic resonance frequencies and therefore double the Reynolds number at which frequency coincidence occurs. They have investigated the effect of row number on vortex shedding and have carried out flow visualization in Reynolds number range of ≤ 355000.Vorticity shedding can cause tube resonance in liquid flow or acoustic resonance of the tube bundles or acoustic resonance of the tube bundles' containers in gas flows, Oengoren & Ziada [10] The vortex shedding frequency can become locked-in to the natural frequency of a vibrating tube even when flow velocity is increased Blevins, [11].

Aditya Kumar Pandey, L.A.Kumaraswamidhas and C.Kathirvelu [12] have investigated on the flow induced vibration phenomenon of vortex shedding and acoustic resonance in horizontal Low Temperature Super heater (LTSH) tube bundles in utility boilers at full load. CFD is used to perform the flow analysis. These results are used to predict the occurrence of vortex shedding and acoustic resonance phenomenon and to calculate the amplitude of vibration. It has been noticed that LTSH receives varying flow distribution along the depth of boiler. Tube vibrations can be reduced to within limits by either reducing the cross-flow velocity or increasing the tube natural frequency or suppressing the standing waves. Reduction in the cross-flow velocity of flue gas is directly related to desired steam output parameters and needs careful considerations. The provision of additional supports and baffles in between tubes, the natural frequency of tubes can be changed and acoustic standing waves can be suppressed. Flue gas has been taken as the fluid medium. The 3D model has been tested and analyzed with velocity inlet and pressure outlet boundary conditions and k-ε has been used as the turbulence model. Comparing vortex shedding frequency according to Chen's criterion with tube natural frequency and acoustic frequency it has been found that tube vibration is possible in 2nd mode. It could increase further at higher load and decrease if the boiler is operated at lower loads. Acoustic resonance is possible at higher nodes. According to Fitzhugh [13] criterion tube vibrations may occur in 2nd mode and acoustic resonance is possible at higher node. To address this non-uniform, the velocity contour profile has been captured at the above five locations along the depth of boiler from the analysis. These velocity profiles served as inlet boundary condition to the model of cross-section test model.

Kunihiko Ishihara, TaisukeTamehira, Masanori Tsuji, Masashi Ichimiya [14] have found out that the natural frequency of the tube decreases and the onset flow velocity of the self-sustained tone increases by inserting the baffle. Therefore, it is concluded that the natural frequency of the duct and the suppression of the self-sustained tone are irrelevant. Secondly, the baffle plate insertion in the entire tube bank is effective for the suppression of the self-sustained tone. The flow velocity of the self-sustained tone when the baffle plate is inserted on the upstream side is different from the case where it is inserted downstream. Two peaks, one for Karman vortex shedding and the other related to the excitation flow fluctuations can be seen in the tube bank. Furthermore, the excitation flow fluctuation is observed in the entire tube bank under the condition that the self sustained tone has been generated. It is not observed under the condition that the self-sustained tone has not been generated. Here the Strouhal Number is 0.13-0.19.For single cylinder, the vortex shedding Strouhal number is a constant with a value of about 0.2. Vortex shedding occurs for the range of Reynolds number 100<Re<10^5 and >10^6 whereas it dies out in-between. This gap is due to a shifting of the flow separation point in vortices in the intermediate transcritical Reynolds number. Vortex shedding can excite tube vibration when it matches with the natural frequency of the tubes.

H. hamakawa et al.[15] has given clear understanding of the effect of tube pitch ratio in tube banks on acoustic resonance as the acoustic mode increased, the acoustic damping ratio decreased for tube pitch ratio of 1.33 - 1.67 in in-line arrangement. As the pitch ratio decreased, the acoustic damping ratio increased. The acoustic damping ratio of 1.44 in in-line arrangement agreed well with that of the staggered arrangement. Secondly, for same tube pitch ratio, it is easy to generate the acoustic resonance for in-line tube banks compared with that of staggered tube banks. The acoustic resonance did not occur for staggered tube banks although the acoustic resonance of 3rd and 4th mode in the transverse direction occurred in in-line tube banks. The vortices of weak periodicity were formed inside the staggered tube banks at small tube pitch ratio. The acoustic resonances occurred at in-line tube banks for tube pitch ratio of 1.33 - 1.67. The vortex shed at broad-band frequency for small tube pitch ratio. The multiple acoustic modes for lower acoustic damping were occurred within the vortex shedding frequency. Acoustic resonance of lower-order modes occurred at the higher gap velocity. As the tube pitch ratio increased, the velocity of acoustic resonance decreased and the SPL increased overall.

Fitzpatrick and Donaldson [16] indicated that both the vortex shedding and turbulent buffeting theories are available to predict the behavior of the flow phenomena in in-line tube arrays. The results of their research did not identify a particular mechanism but pointed to the possibility that acoustic resonance could be the result of vortex shedding, turbulent buffeting, broadband turbulence, or a combination of the three.

Chen [17] noted that the flow for in-line arrays passes directly in the gaps between tubes. The shedding of vortices is controlled by the jet in the flow lane. This jet is disrupted by the presence of the downstream tubes. That is why the main parameters that affect the flow phenomena for this type of arrays should be linked to the flow lane width and the tube spacing in the longitudinal direction.

A very detailed investigation for in -line arrays with geometries covering the full range of tube spacing was performed in a series of papers by Ziada and Oengören, Ziada et al. and Oengören and Ziada [18]. What follows is a synthesis of the most important findings of their work. The findings represent the current state of understanding of the flow phenomena relevant to acoustic resonance for in-line arrays.

Ziada et al. [19] performed tests on staggered and in-line heat exchangers in an air tunnel and in a water tunnel for the in-line heat exchanger. For the in-line case, three Strouhal numbers were detected behind the first tube row: S1 = 0.92, S2 = 0.64 and S3 = 0.46. Initially, they were able to detect only S1 and S3 in the air tests, but after testing in the water channel and detecting S2, they went back to the air test and were able to identify S2, but only by using a small number of samples during averaging and for only some flow rates. S2 was found to be the one responsible for the acoustic resonance present in the air tunnel tests. With the in-line array configuration that they used (L/D) =1.35 and T/D=1.6 they were able to excite the third acoustic mode. No flow periodicity was able to excite the first two modes, and the third mode was excited by the very faint periodicity represented by S2. They present a detailed flow visualization study to try to explain the flow phenomena. This study presents a clearer picture of the phenomena. The flow periodicity present for this closely packed in -line array was a symmetric jet instability that could be seen in both the flow lanes and behind the tubes.

Oengören and Ziada [20] presented results of resonance tests of in-line arrays. They showed flow visualization techniques, the flow structure of the vortex formation during resonant conditions. This vortex formation was very different from what was found at non-resonant conditions. The vortex formation was synchronized throughout the array with the tubes shedding vortices in phase. An additional proof that another excitation mechanism was present is that the Strouhal number determined at non-resonant conditions does not predict the onset of resonance.

Ziada and Oengören [21] present detailed results of work for the flow instabilities. The first test performed in air showed that in this case the flow instability detected at off-resonant conditions was responsible for the acoustic resonance! These results differ from the results for intermediate and closely packed arrays described above. When water tests were made, it was found that the turbulence level played a key role in the flow instability phenomena. Two different types of instabilities were found: "global jet mode" and a "local wake mode," each having their own characteristic Strouhal number and not occurring simultaneously. For low turbulence levels upstream of the tube array, a symmetric jet instability similar to the one found for the intermediate spaced array was seen. For higher turbulence levels, a local wake instability mode was present. In this mode, alternate vortex was shed from the wake of the cylinders of the first few rows, and this was independent from cylinder to cylinder. After the fifth row, a narrow band turbulent excitation at a different frequency was present. If Strouhal number is determined for in-line arrays with large tube spacing under non resonant conditions, there will be uncertainty as to which instability mode is present at the time of the measurement, and thus, acoustic resonance predictions might fail. If an acoustic Strouhal number is determined at resonance, then, if this Strouhal number is used in the prediction of tube vibration for which symmetric vortex formation is relevant, it might produce erroneous results.

John Mahon and Craig Meskell [22] have noticed the effect of acoustic resonance on the static fluid forces on a stationary cylinder and the second the effect of acoustic resonance on the time delay between tube motion and the resultant flow reorganisation close to the measurement cylinder. They have found that acoustic resonance did not modify the fluid force. Acoustic resonance is seen to modify the time delay between tube motion and flow field around the cylinder.

Sanjay kumarsamy, Richard A. Corpus and Jewel B. Barlow [23] have computed noise generated due to flow over a circular cylinder. The sound pressure level corresponding to the dominant frequency over predicted in case of 2D and there was a reduction of 10dB in 3D results.

III. Conclusions

Based on these literature review and discussions, the following could be concluded:

- The possible sources of flow noise can be fluctuation in mass flow, turbulent wakes, fluctuating pressure, transition of boundary layer
- LES can be good turbulence model while doing acoustic simulation
- The turbulent boundary layer is less vulnerable to adverse pressure gradients. Therefore wake is much narrower and the pressure drag is greatly reduced. In this case acoustic power is increased but still it is not a major noise source.
- The fluctuating pressure near the cylinder wall increases the surface acoustic power and that is the major noise source in an unsteady flow over cylinder
- The computation of noise generated by the unsteady flow over a circular cylinder highly depends on the accuracy of the CFD numerical method employed. Mostly the hybrid approach of FVM and FEM could be better for numerical solution

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