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### IDENTIFICATION OF FLOW INDUCED VIBRATION IN STEAM EXHAUST HOOD-DUCT SYSTEM

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#### ABSTRACT

Vibration and noise of concern were observed at the hood-duct of the exhaust steam turbine at a power generating facility. The field personnel noticed this problem after a startup of the unit which followed the maintenance replacement of the expansion joints. The author was contacted by the field personnel and asked to formally evaluate this problem and advise on the steps needed to eliminate it. Several similar units were scheduled for the same maintenance work.

The analysis carried out was based on the acoustic and structural dynamic modeling of the critical parts of the hood-duct system in question. Both the flow induced excitations and mechanical structural responses were considered. The results of the analysis combined with field measurements indicate a vibration generation mechanism which is flow induced in nature and is coupled with the acoustic field and enhanced by the high stream-wise flow velocities. Consequently, there are two categories of solutions identified to the observed problem. The first considers addressing the problem at the source and the second suggests path treatment. Addressing the problem at the source by means of aeroacoustic optimization of the flow path is recommended. Detuning the mechanical response from the flow excitation(s), i.e. the vibration path treatment, involves an aspect of greater or lesser trial and error since the responses of the studied structure depend on many variables; therefore, even the most accurate calculations might not be sufficient to give the expected results. Both recommendations are discussed.

#### INTRODUCTION

The purpose of this study was to identify and direct the field personnel to solve exhaust hood-duct vibration in the auxiliary steam turbine ducting system. The duct system consisted of the rectangular hood connected on one side to the steam turbine exhaust and on the other side to the steam condenser as shown in Figure 1. The hood constituted a part of the duct conveying steam as well as provided a support for the elevated duct-pipe section leading to the condenser. The elevated duct section was made up of three expansion joints, bellows (two 4-convoluted and one 8-convoluted); see Figure 2. In between the bellows, the NPS 60 pipe spool pieces and a butterfly valve were installed. The bellows were welded to the spool pieces and connected together by means of tie-rods. The bellows and the spool pieces constituted a pressure balancing duct element. In addition, the bellows had the possibility to accommodate thermal displacements in the axial, lateral and torsional directions in the elevated duct. The duct segments, after the bellows, were again expanded into the rectangular shaped duct segments of cross section width equal to 2300mm and 660 mm height. The rectangular part of the duct led directly into the steam condenser at which the duct was welded into the condenser's inlet. Please note that the rectangular part of the duct as depicted in Figure 2 is of an annulus shape for the simplification of the plotting routine.

The vibration problems became predominant after the original bellows (including bellows of 6-5-5 convolutions), was replaced with the new bellows (bellows with 8-4-4 convolutions). It should be noted that the detailed design information for either the new or for

the old bellows was not available. However, every effort was made by the site personnel and the author to obtain the most pertinent information which could have been gathered and which might contribute to resolving the problem. Still, some information had to be assumed or calculated. These were the thermodynamic steam properties, steam quality, steam velocities as well as the detailed bellows design. These are given in the Tables 1 through 3 and Figure 3. The discussed data was judged by the field operators to be representative of the actual process conditions. The steam flow information was inferred from the measurements of steam flow capacity at the installed control points located before the turbine inlet and the steam take-offs. Consequently, the accuracy of flow rates was estimated to be between  $\pm 10\%$  to  $\pm 15\%$ .

## ANALYSIS PROCEDURE

First, the bellows vibration modes and the frequency response predictions were carried out. The longitudinal (accordion) and transverse bellows vibration modes were studied [1, 2]. These calculations were carried out in spite of the fact that the bellows under consideration was covered with a liner. However, the liner was said to be short and one of the leading edges and possibly the trailing convolution could have been exposed to the mean flow, causing vortex induced excitation.

In the next step, the bellows' responses were calculated with added fluid mass and the bellows' natural frequencies were determined [3, 4]. Later the entire hood-duct system was structurally modeled by means of a finite element program. During the investigation, the tie-rod stiffness and its influence on the dynamic behavior of the entire duct was studied.

In the next step, the actual flow velocities throughout the duct system were enveloped. Based on those the possibility of vortex induced vibration (VIV) was studied at the bellows' convolution, at the sharp elbow edge and at the tie rods protruding into the steam flow path. Also, a small step gap created by a short liner was investigated.

In the next step, the study re-focused on the 'sharp' 90° elbow and possible flow patterns in this part of the steam hood-duct section. The actual local flow velocities were relatively high, near 37 m/s, (versus a 21 m/s average), hence an unusual abrupt turn in the duct flow-path (90° sharp elbow) was suspected of significantly contributing to the aero-acoustic noise emanating from the hood-duct. Due to time and scope limits, the author used a former example of computer fluid dynamic (CFD) model to illustrate possible flow phenomena generated in the abrupt 90° elbow conduit system, see Figure 4.

This model was supplemented by an acoustical model representing the entire exhaust-duct system. As a result, the acoustical responses of this system were predicted and are given in Figure 5. The responses were limited to

predictions of the acoustical natural frequencies in the system and their sensitivity to the studied flow rates. The calculated cut-off frequency, for the validity of this study, was determined to be from 0.0 Hz up to 170.0 Hz.

Finally, the acoustical excitations were matched with the structural responses of the system and subsequent conclusions were drawn based on the acoustical and structural results. In addition, the author compared the field measurements to the predicted results. A good agreement between the predictions and the measurements was found.

## RESULTS

Table 1 of this report gives the pressure, temperature and thermodynamic information on the steam. Only 100% quality steam is listed in this table. After comparison of different quality steam versus pressure and temperature, it was decided that the values depicted in Table 1, when averaged, adequately represented the process steam conditions. Table 2 shows the estimate of the flow conditions at the turbine exhaust. Table 3 depicts the pertinent bellows information for 8-4-4 convolutions.

The following table, Table 4, consists of the predicted natural frequencies for the bellows with a number of convolutions which are given in the table. These predictions are based on the single 'building block' bellows geometry illustrated in Figure 3. The results include the bellows' material mass loading as well as fluid loading.

Table 5 describes different types of vortex induced vibration phenomena and quantifies the regions of their existence in the frequency bands of vibratory or acoustical responses. The tables are followed by Figures 1 through 3 describing the analyzed system. Next, Figure 4 depicts AS FOUND local flow velocities and a solution to the problem. The local velocity lines in a flow through the 90° elbow range from near 0 to 68 m/s. Also a reversed flow (secondary flow) in the "dead leg" is observed where the local velocities are perpendicular to the mean stream-wise velocity.

Figure 5 shows the predicted acoustical natural frequencies for the system. The following figure, Figure 6 (noise spectrum sample), illustrates the Fast Fourier Transform (FFT) of the measured sound pressure level in the field. This information is given by the raw Voltage adequately characterizing the sound pressure levels. Figure 7 describes a set of vibration measurements for the AS FOUND and MODIFIED systems. The MODIFIED results are from the test after the wooden blocks between the tie-bars and the pipe spool pieces were added by the field personnel. Both vibration results are compared.

Figure 8, depicts a 6 second time record of the characteristic 'coming and going away' low-audible

swishing noise measured near the 90° turn in the hood-duct system.

Figure 9 shows a sample of plausible vibratory responses of the hood-duct system. Again, the agreement between the predicted and measured results is excellent. This figure not only describes the frequencies of vibration but also depicts modes of vibration which could be identified when measured on site.

## DISCUSSION

When the bellows is excited by a flowing fluid, at least two groups of fluid forces should be considered: (i) vortex shedding forces, and (ii) passive fluid forces [2] which are present regardless of excitation. In this study they are considered separately. The passive fluid forces contribute added mass, damping and spring-like terms to the bellows description. Since the variation of damping and spring-like terms is small when compared to the fluid added-mass, only the latter is considered in detail. Therefore, the fluid added-mass is represented by the total amount of fluid trapped between adjacent convolutions. This makes sense for a few lower vibratory modes where trapped fluid moves along with the bellows. For higher modes, when the fluid becomes squeezed-in-and-out from between the convolutions, the acceleration of the fluid results in an alternating pressure force which is interpreted as an added-mass effect to the bellows vibration. These forces were calculated and the mechanical natural frequencies for the bellows are given in Table 4.

There are many mechanisms of vortex generation which could exist in the studied system. Based on experience, the author focused only on the most plausible ones. Even these mechanisms, of a practical nature, are difficult to prove or predict by numerical methods without elaborate modeling and measurements which are impractical or not possible in a production type facility. In the field, some methods to identify them indirectly are used, for example by means of vibration or noise. However, in the end, only laborious laboratory or in-situ vortex studies could confirm these phenomena to exist.

Consequently, the following approach was undertaken: several mechanisms of vortex excitation were considered. They are depicted in Table 5. Critical variables, namely mean flow velocity as well as shear layer velocity in the areas containing flow obstacles, were taken into account.

All of the generation mechanisms studied here, with the exception of the bellows, were considered further. The bellows' frequency was above 330 Hz and it was not likely to contribute to the vibration of interest. In addition, the bellows was to be covered by a liner, as per design information, which should eliminate it as a source of the problem. Other periodic vortex shedding frequencies

defined by Strouhal number and flow were indirectly observed in the vibration and noise spectra and confirmed by approximate spreadsheet calculations. This vibration was likely occurring at or near natural frequencies of the structure and could potentially cause failures.

Several predominant components of vibration were observed in the AS FOUND measurement results. These were related to the predicted flow-induced vibration, see Figure 7. In Figure 7 (as in others not shown here for the article page limit) the vibration amplitudes were noticeably reduced in the MODIFIED system. Also, several vibration response components lower than 100 Hz were diminished. However, there was still a noticeable vibration peak near the frequency of 97 Hz, which was present throughout the spectra.

From the above observation the following was deduced: the turbulence generated in the abrupt 90° elbow(s) at higher velocities allowed for transfer of a part of the flow energy into duct vibration.

Many studies show that the turning flow in a 90° elbow increases the local stream velocities drastically. This happens due to local fluid acceleration caused by a deviation of the stream velocities from the mean, i.e. primary, flow. When such a deviation happens, for example due to a vortex, the fluid particle has to cover a longer physical distance, hence the particle velocity must increase in order to conserve flow energy. The deviation from the mean flow velocity lines and stream lines is depicted in detail in Figure 4 where the local velocities can increase twofold or more from the mean/design parameters.

The next mechanism considered was a separation of the flow from the leading edge and generation of a vortex [5, 6, 8]. When a vortex is generated and its frequency of generation is close to any local acoustic natural frequency of the system, the vortex does not dissipate but it is fed by the acoustical energy of the local field. Consequently, this vortex triggers other vortices which are generated at the period of the resonant frequency of the acoustical field. They persist and 'lock-in' into the acoustical field. This is the likely source of the vibration and noise problem observed in the facility.

Another excitation could exist due to a mere flapping of the shear layer originating from a single point of flow separation in the confined backward-facing step geometry (e.g. gap in the liner), as opposed to the case of vortex roll-up from two points of separation in the bluff-body geometry. The flapping of the separating shear layer in the case of the backward-facing step is accompanied by Kelvin-Helmholtz instability leading to a relatively small-scale vortex roll-up in the near-field of the step. Such instability can be present in the studied case as well. Frequency response can range from 102 Hz to 166 Hz as predicted in Table 5. Also, there is a clear vortex generation present by the tie-rod immersed in the

steamflow path. Its frequency is related to the peaks near 54 Hz, see Table 5.

Generally, to attenuate or prevent harmful flow-induced vibration the actual and local flow velocities must be lowered. This can be achieved by an adequate aero-acoustic design of the flow path in the hood-duct system. An example solution to such a problem is depicted in Figure 4 (bottom part). The solution consists of optimization of the flow path and elimination of 'swirls' thus significantly reducing local flow velocities and secondary flows [7, 8]. In addition, for the same flow capacity the mean velocity is slightly reduced by shortening the actual flow path.

To support the above claims of vibration generation the author devised an acoustical model which predicted acoustical natural frequencies of the studied system. These are depicted in Figure 5. It is evident that the acoustic natural frequencies correspond directly to several above-discussed vortex generation mechanisms. It is however difficult to pinpoint the predominant cause. The separation of high local flow at the elbow(s) and vortex generation enhanced by the local acoustic field seems to be the main vibration cause. The indication of the existence of such a mechanism is confirmed by a low audible swishing noise periodically coming and going away. The sound pressure measurements, depicted in Figure 6, further support its existence. In this figure, it is worth noticing the large increase of acoustical energy and turbulence contents after each of the 90° turn locations (both near the turbine and after the duct's tee).

In addition, a signature of acoustical resonance near 97 Hz is confirmed after the butterfly valve. This acoustical excitation could be generated by the turbine 1x running speed but had to be reinforced by the valve or system acoustics. For example, the steam flow passing the edge of the pivotal disc in the valve generates a flow turbulence to form an unsteady large wake downstream of the valve. This unsteady wake could be the source or trigger of a noise exciting acoustical field near 97 Hz, see Figures 6 and 7. Figure 8 depicts one of the noise time histories including the low tone noise measured near the 90 degree elbow.

To further extend the study, the mechanical natural frequencies (MNFs) and vibration mode shapes of the entire hood-duct system were predicted. These are depicted in Figure 9. Again, they are surprisingly close to the frequencies of the predicted acoustic and measured vibration responses. Consequently, they substantiate the mechanism of vibration where vibration is excited by the high local flow velocities, enhanced by coupling with the acoustic field and reinforced by the system mechanical response (MNFs). Therefore, considering the origin of vibration the following mitigating actions are suggested: Address the problem *at the source*[8] by the aero-acoustic flow path design optimization (90° elbow) and its

implementation (i), or Detune the mechanical response from flow excitation by adding stiffness, damping or mass (or a combination of all the above); (ii); the *path treatment* method.

If applied by a qualified engineering firm, proposed approach (i) guarantees problem resolution. It would likely involve internal flow path modification by installation of thin plates directing the flow. Such action will significantly lower the local flow velocities and improve the turbine efficiency by 1% or more. Such an approach will limit the transfer of energy from the mean flow into vibration. The perceived comparably high cost of such a solution could be a deterrent, but the solution has numerous advantages, including low maintenance costs and significantly fewer component failures.

Proposed approach (ii) is the solution which could end up with a trial and error method. The reason for this is that there is no anchored structure available to significantly influence the mechanical responses of the vibrating hood-duct. The tie-rods which are used to attenuate vibration by means of wooden blocks through stiffening-up the duct spool pieces and delivering some damping are not the best design. They act positively when wooden blocks or similar components are attached to the tie-rods or duct without adding any strain to the structure. If the tie-rods are bent or stressed then they would not provide adequate protection for the bellows. Consequently, changing the mechanical natural frequencies of the hood-duct system should include lighter tie-rods with much greater stiffness, for example, NPS 4 or NPS 6 aluminum alloy pipes or equivalent symmetric profiles. In addition, currently installed wooden blocks should be replaced by a material providing high damping; for example, FAB-EPM or FabsorbFabreeka or a similar material. The damping rate of FAB-EPM is 7 to 15% of critical damping.

Path treatment approach is relatively cheap but it is classified a 'band aid' solution since it involves a trial and error approach and does not directly address the source of vibration. In addition, it may not provide the expected results.

## CONCLUSIONS

- 1) The replacement of the original 5-5-6 set of convolution bellows into a 4-4-8 set of convolution bellows drastically changed the mechanical response of the hood-duct system.
- 2) When the bellows is excited by a flowing fluid, at least two groups of fluid forces should be considered: fluid passive forces and the vortex shedding forces.
- 3) In this study, direct acoustic excitation by flow passing the bellows was excluded. By design, the bellows' liner prevented any contact of corrugation with the flow.

- 4) There are many mechanisms of vortex generation which could exist in the studied system.
- 5) Critical variables, namely mean flow velocity as well as shear layer velocity in the areas containing flow obstacles, were taken into account. Indirect methods of finding excitation phenomena signatures were employed, for example, vibration and noise measurements.
- 6) From the study of the prediction and measured results (vibration and noise) it was deduced that the turbulent flow at higher flow velocities generated mechanisms which transferred a part of the flow energy into vibration. Secondary flows in a 90° elbow drastically increase local stream velocities through longer particle paths.
- 7) It is evident that acoustic natural frequencies correspond directly to several considered vortex generation mechanisms. It is however difficult to pinpoint the predominant cause.
- 8) There are two main solutions to the problems observed in the field. One of them is to eliminate the source of vibration and noise. This approach may be thought to be costly at first but benefits would be seen in short order thanks to lower maintenance costs and significantly fewer component failures. This solution would equally and positively influence the turbine as well as the duct. In addition, an increase of turbine efficiency is expected. The second solution, the so-called 'path treatment', would address only the symptoms and may transfer the problem somewhere else.

## REFERENCES

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- 6) **Motriuk, R.W.**, 2003, 'A Perforated Conical Strainer as an Example of an Acoustic Noise Generator', ASME Pressure Vessels and Piping Conference, Cleveland, 2003.

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## TABLES AND FIGURES

**Table 1: Steam Conditions and its Properties**

| P              | T     | $\rho$               | Z        | cp/cv   | cp       | Visc       | Th.Cond.   | Soud Speed | MW        | Isent Exp |
|----------------|-------|----------------------|----------|---------|----------|------------|------------|------------|-----------|-----------|
| [kPa]abs       | [C]   | [kg/m <sup>3</sup> ] | [1]      | [1]     | [kJ/kgK] | [Pa s]     | [kJ/msK]   | [m/s]      | [kg/kmol] | [1]       |
| 13.6226        | 52.0  | 0.090537             | 0.998068 | 1.32939 | 1.87885  | 0.00000907 | 0.00002111 | 445.783    | 18.15301  | 1.32681   |
| 28.8509        | 62.0  | 0.141663             | 0.997188 | 1.32967 | 1.88500  | 0.00000935 | 0.00002190 | 452.235    | 18.15301  | 1.32592   |
| 33.9724        | 72.0  | 0.214118             | 0.996021 | 1.33025 | 1.89236  | 0.00000963 | 0.00002271 | 458.493    | 18.15301  | 1.32493   |
| 51.3417        | 82.0  | 0.314959             | 0.994509 | 1.33119 | 1.90109  | 0.00000991 | 0.00002355 | 464.544    | 18.15301  | 1.32384   |
| 75.6143        | 92.0  | 0.452027             | 0.995696 | 1.33259 | 1.91141  | 0.00001019 | 0.00002440 | 470.373    | 18.15301  | 1.33259   |
| 101.3250       | 102.0 | 0.590593             | 0.990895 | 1.33371 | 1.92121  | 0.00001047 | 0.00002526 | 476.148    | 18.15301  | 1.32146   |
| Average Values |       |                      |          |         |          |            |            |            |           |           |
| 50.7878        | 77.0  | 0.30065              | 0.995396 | 1.33113 | 1.89832  | 0.00000977 | 0.00002316 | 461.267    | 18.15301  | 1.3259    |

**Table 2: Estimated Turbine Flow Conditions**

| Approximate Steam Flow Conditions<br>During the October 23 Test |     |                     |              |
|---|-----|---------------------|--------------|
| Turbine Inlet   |     | Imp                 | Metric       |
| Mass Flow   | 90  | [KLH]               | 11.34 [kg/s] |
| T <sub>in</sub>   | 658 | [°F]                | 347.8 [°C]   |
| P <sub>in</sub>   | 196 | [psi <sub>g</sub> ] | 1351.7 [kPa] |
| Turbine Outlet  |     | Imp                 | Metric       |
| T <sub>out</sub>  | 125 | [°F]                | 51.7 [°C]    |
| P <sub>out</sub>  | 15  | [psi <sub>g</sub> ] | 103.4 [kPa]  |

**Table 3: Vendor's Expansion Joint Information**

| EJS Unreinforced Bellows      |          |           |                   |                       |                       |
|-------------------------------|----------|-----------|-------------------|-----------------------|-----------------------|
| Design And Bellows Dimensions |          |           |                   |                       |                       |
|                               | Imp      | Metric    | Spring Rates      | Imp                   | Metric                |
| Design Pressure               | 15 psig  | 103.4 kPa | Axial             | 537 lb/in             | 934.4 N/cm            |
| Design Temperature            | 500 °F   | 260 °C    | Lateral           | 633 lb/in             | 1101.4 N/cm           |
| Bellows ID                    | 61.25 in | 1556 mm   | Angular           | 4699 lb-in/deg        | 531 N-m/deg           |
| Bellows OD                    | 6.25 in  | 1657 mm   | Torsional         | 38.39E6 lb-in/deg     | 4338070 N-m/deg       |
| No. of Convolutions           | 4+4      | 4+4       |                   |                       |                       |
| Thickness                     | 0.024 in | 0.61 mm   |                   |                       |                       |
| No. of Piles                  | 2        | 2         |                   |                       |                       |
| Roll Size                     | 0.8 in   | 20.3 mm   | Bellows Weight    | 58.46 lb              | 1274 N                |
| Convoluted Length             | 7.0 in   | 178 mm    |                   |                       |                       |
| Bellows Pitch                 | 1.75 in  | 44.5 mm   | Moment of Inertia | 0.379 in <sup>4</sup> | 15.78 cm <sup>4</sup> |
| Convoluted Height             | 1.952 in | 49.4 mm   |                   |                       |                       |
| Skirt Length                  | 1.25 in  | 31.8 mm   |                   |                       |                       |

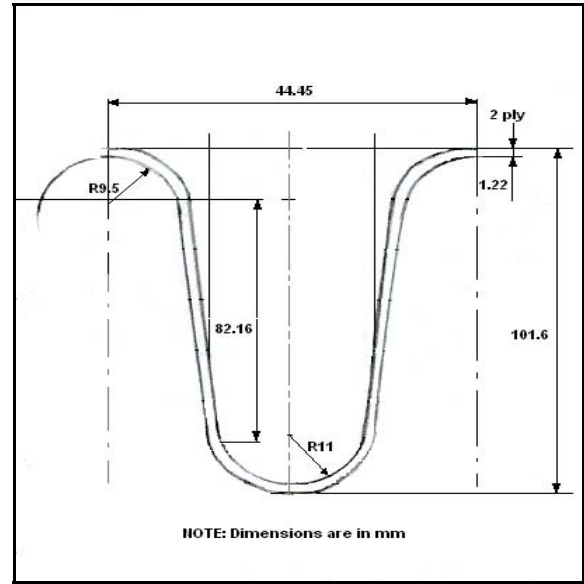


**Table 4: Predicted Acoustical Eigen Frequencies**

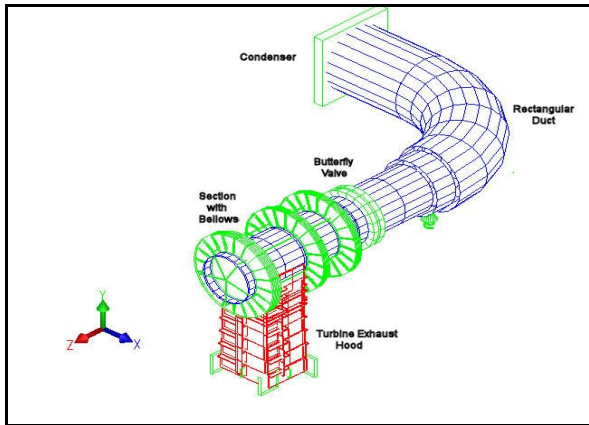
| BELLOWS NATURAL FREQUENCIES |       |      |      |         |          |
|-----------------------------|-------|------|------|---------|----------|
| Convolutions                | 4     | 8    | 12   | EJMA 8* | EJMA 8** |
|                             | [Hz]  | [Hz] | [Hz] | [Hz]    | [Hz]     |
| <b>Axial Response</b>       |       |      |      |         |          |
| Mode 1                      | 26.0  | 12.0 | 8.1  | 11.4    | 29.6     |
| Mode 2                      | 53.0  | 24.0 | 16.2 | 22.3    | 57.9     |
| Mode 2                      | 80.0  | 37.0 | 24.2 | 32.4    | 84.3     |
| Mode 4                      | 107.0 | 49.0 | 32.3 | 41.2    | 107.3    |
| <b>Lateral Response</b>     |       |      |      |         |          |
| Mode 1                      | 46.6  | 15.9 | 7.9  | 7.2     | 18.8     |
| Mode 2                      | 66.6  | 23.2 | 12.1 | 19.9    | 51.7     |
| Mode 2                      | 89.1  | 31.3 | 16.7 | 38.7    | 100.8    |
| Mode 4                      | 119.6 | 42.3 | 22.7 | 64.3    | 167.5    |

**Table 5: Investigated Vortex Mechanisms**

|               | Different Mechanisms of Vortex Induced Vibration (VIV) |                   |                   |                |                  |                  |                   |
|---------------|--|-------------------|-------------------|----------------|------------------|------------------|-------------------|
|               | PROTRUSION INTO FLOW                                   |                   |                   |                |                  |                  |                   |
|               | Rectangle  | Rectangle         | Flat Plate        | Flat Plate     | Bellow           | Backward Step    | Tie-Rod           |
| H - Height    | 12.7 mm  | 6.4mm             | N/A               | N/A            | N/A              | 5 mm             | N/A               |
| Length or Dia | 66 mm  | 66 mm             | 66 mm             | 1202 mm        | 20.44 mm         | N/A              | 0.146.1 mm        |
| St number     | 0.132  | 0.132             | 0.16              | 0.16           | 0.18             | 0.01358 to 0.022 | 0.212             |
| Velocity      | 37.7 m/s   | 37.7 m/s          | 37.7 m/s          | 37.7 m/s       | 37.7 m/s         | 37.7 m/s         | 37.7 m/s          |
| Frequency Hz  | 75.4 (±10 to 15%)                                      | 45.7 (±10 to 15%) | 91.4 (±10 to 15%) | 5 (±10 to 15%) | 332 (±10 to 15%) | 102 to 166       | 54.7 (±10 to 15%) |



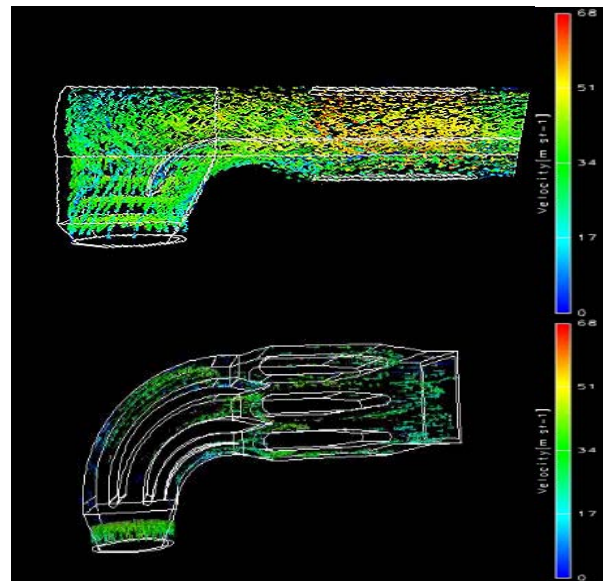
**Figure 3: Details of Bellows Geometry**



**Figure 1: Schematic of the Analyzed System**



**Figure 2: Part of the Studied System – Hood-Bellows**



**Figure 4: Local Velocity Lines for Un Optimized (Top) and Redesigned (Bottom) Flow Path**

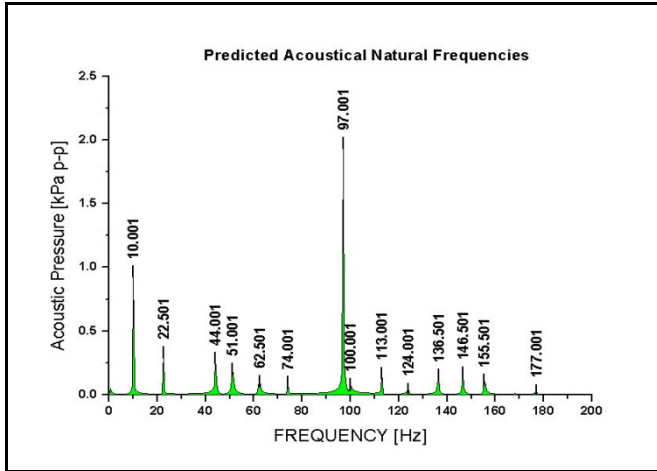
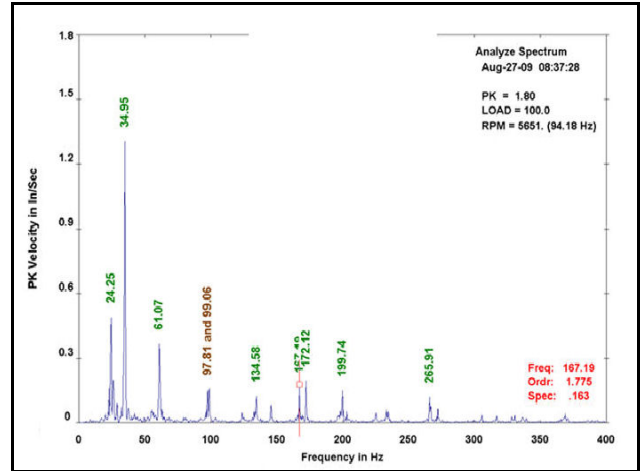


Figure 5: Predicted Acoustical Natural Frequencies for the Hood-Duct System



(a)

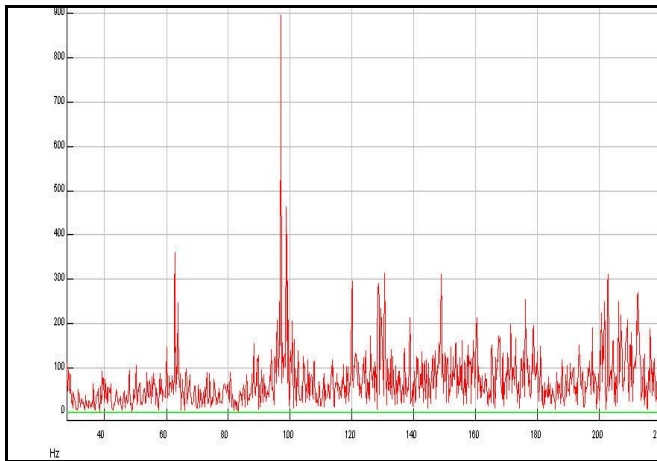
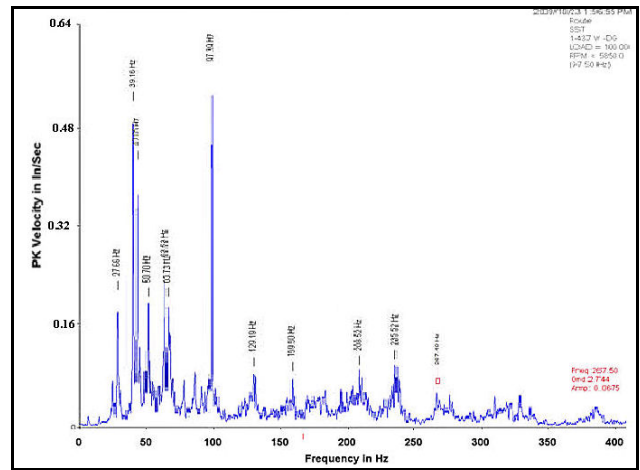


Figure 6: Noise Spectrum Measured near 90 deg Turn



(b)

Figure 7: Vibration Spectra Taken at the Same Location for (a) AS FOUND and (b) MODIFIED Systems

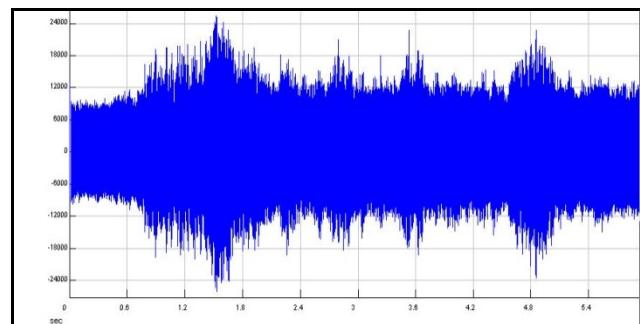
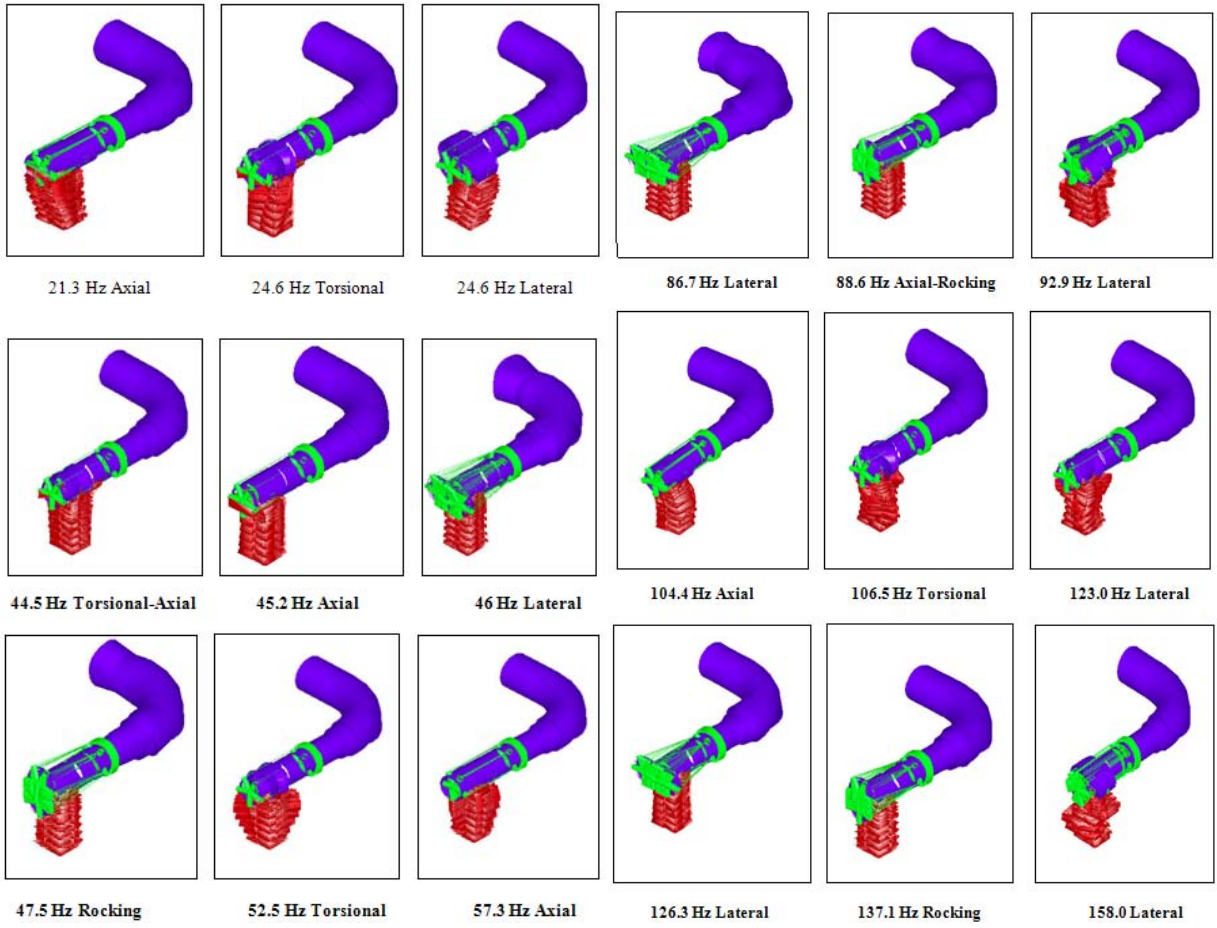


Figure 8: A Sample of Noise near 90 degree Turn at Hood Connection



**Figure 9: Predicted Mechanical Natural Frequencies**