# THE EFFECT OF INLET FLOW DISTORTION ON THE PERFORMANCE OF CENTRIFUGAL FANS FOR UTILITY APPLICATIONS

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#### ABSTRACT

This paper provides an overview of published literature on the effect of inlet Flow Distortion on the performance of centrifugal fans, with an emphasis on a recently completed testing program [8] on the effect of inlet distortion on the performance of airfoil bladed Centrifugal fans, at various inlet vane positions. The results of this testing program [8] show that a model fan, when equipped with variable inlet vanes, is generally subject to lower levels of performance degradation, for a given level of inlet distortion, than the same model when the variable inlet vanes are not present. This indicates that the variable inlet vanes are acting to condition the distorted flow prior to entry into the fan wheel in a beneficial manner. The reduction in performance degradation seems to occur when the variable inlet vanes are wide open (parallel to the fan shaft axis), and does not seem to improve with closure of the variable inlet vanes. The maximum reduction in Total Pressure development and Total Efficiency measured for fans without variable inlet vanes in this test was -8.4% and -6.1% respectively. The maximum reduction in Total Pressure development and Total Efficiency measured for fans with variable inlet vanes in the wide open position was reduced to -2.9% and -3.5% respectively. In fact, the beneficial effects of the variable inlet vanes appear to be greater with the vanes in the wide open position, and are possibly less beneficial when the inlet vanes are partially closed. (The maximum reduction in Total Pressure development and Total Efficiency measured for fans with variable inlet vanes in a partially open position was -2.5% and -4.1% respectively.) Refer to Table 2 for a summary of the Maximum Distortion Observed for the testing program.

An unanticipated result of the tests conducted [8] is the discovery that the effects of distortion seem to vary with the position of the rating point along the fan curve, with distortion effects being greater at higher flow conditions and lower at lower flow conditions. An actual field case study on the negative effects of inlet distortion on a centrifugal Induced Draft (ID) fan is also presented. Recommended steps to take in designing a fan system to minimize the potential for any negative effects on fan performance from inlet distortion are also presented. The conclusion compares the results of previously conducted tests [1, 2] and recently conducted tests [8], and concludes that the recent tests do not support many of the earlier findings. This leads to authors to believe that the actual impact of inlet distortion is highly dependent on the specific geometry of the airfoil-bladed centrifugal fan involved, and the relationship to various fan design parameters is not currently well understood.

## TERMINOLOGY

The previous testing reports [1, 2] on this subject established some conventions for definition of inlet flow distortion, which have been used in subsequent publications [3, 4, 8] and will also be used in this report. In the discussion of inlet distortion, the term "transverse" refers to the direction perpendicular to the fan shaft. The term "axial" refers to the direction parallel to the fan shaft. These are illustrated in Figure 1. The terms "transverse distortion parameter" and "axial distortion parameter" were also defined in the previous references [2, 3, 4, 8] and have the same meaning in this report.



#### Figure 1 Illustration of Transverse and Axial Definitions

#### NOMENCLATURE

The following nomenclature is used in this report, (as defined in references [2, 8]):

#### Variables

M = Number of measurements in the traverse grid in the "a", (axial) direction

N = Number of measurements in the traverse grid in the "t", (transverse) direction

 $P_L$  = Local values of total pressure loss, mm H<sub>2</sub>O

 $P_T$  = Fan total pressure rise, mm H<sub>2</sub>O

Q = Volume flow rate, m<sup>3</sup>/s

V = Inflow velocity, m/s

 $\overline{V}$  = Averaged value of V, m/s

 $\hat{\mathbf{V}}$  = Normalized distortion parameter, dimensionless

TE = Fan Total Efficiency, %

 $t_{d\ 1\text{-}1}$  and  $t_{d\ 1\text{-}2}$  = Dry-bulb temperature measurements at the traverse plane

## **Symbols**

= Efficiency

- = Denotes difference between distorted and reference levels
- $\Phi$  = Pitch angle for directional probe
- $\psi$  = Yaw angle for directional probe

#### Subscripts or Superscripts

- a = Axial direction
- d = Denotes distorted inflow case
- i = Index in the transverse direction for velocity survey
- j = Index in the axial direction for velocity survey
- o = Denotes reference level
- T = Denotes total quantity, pressure or efficiency
- ^ = Denotes a normalized distortion parameter
- = Denotes an averaged value

# **Test Point Numbering and Axes**

Figure 2 defines the test point numbering and axes used in the excel data files for this project.



Figure 2 Definition of Test Point Numbering and Axes

#### INTRODUCTION

The most comprehensive references for the impact of inlet flow distortion on the performance of centrifugal fans is found in references [1] and [2]. These papers summarize the laboratory test conducted on a centrifugal airfoil fan. The conclusion of references [1] and [2] were:

- "Even moderate levels of distortion 10 to 15 percent velocity variations or 10 to 15 percent lateral offset of the flow entering the inlet box – can lead to significant penalties in performance – 5 percent in efficiency and pressure rise. Relatively severe distortion – 30 to 50 percent velocity variation or flow offset – can cause serious deterioration in fan performance – 10 to 12 percent reduction in efficiency and pressure rise."
- 2. Axial distortion has less impact than transverse distortion. For 20% distortion the impact on pressure is 3% for transverse distortion and 0.5% for purely

axial distortion. The testing and following conclusions are in reference to transverse distortion.

- 3. The impact of distortion on efficiency was essentially the same with and without inlet guide vanes installed.
- 4. The impact of distortion on total pressure rise without inlet guide vanes depends on the distortion relative to rotation. With inlet guide vanes the impact is independent of rotation and is less in the worst case.

All of the testing conducted in references [1] and [2] with inlet guide vanes was conducted with the vanes wide open. The results of references [1] and [2] are compared to the resulting of this testing program in the conclusion.

This report summarizes the results of a recently conducted testing program [8] developed by the Electric Power Research Institute (EPRI), FlaktWoods, and the American Society of Mechanical Engineers Performance Test Code (ASME PTC-11) Committee members to investigate the effect of various types and levels of distorted inlet flow on the performance of a typical airfoil-bladed centrifugal fan which is equipped with variable inlet vane (VIV) control. This type of fan and means of control is very common for main service fans used in fossil-fuel Utility Power Generation stations for many applications, including Forced Draft (FD), Primary Air (PA), Secondary Air (SA), Induced Draft (ID) and ID Booster applications. Testing programs conducted in the past [1, 2] have shown that even relatively low levels of distorted inlet flow have a negative effect on the performance of airfoil bladed centrifugal fans. Due to spatial constraints, and other system design considerations, distorted inlet flows are quite common in the applications described above, and system designers, architect-engineers, end users and fan manufacturers have struggled for years to determine what level of inlet distortion might be acceptable for a given application. The purpose of this testing program [8] was to determine whether the presence of variable inlet vanes in such fans has a mitigating effect on the negative effects of inlet distortion, and how this effect varies with the position of the variable inlet vanes for various types and levels of distortion.

#### **APPROACH**

The goal of the testing program [8] was to develop new information on the effect of various types of distortion on the performance of a particular airfoil bladed centrifugal fan model, both with and without variable inlet vanes, and at various closures of the VIV's, when exposed to specific types of Transverse and Axial Distortion of varying magnitudes. All testing was conducted in a laboratory accredited by the Air Moving and Control Association (AMCA) for fan testing. The distortion measurements were made using a five-hole directional probe connected to a computerized data acquisition system that is capable of determining the angularity of the flow in two planes. The flow measurements were made in an AMCA 210 Figure 12 test setup, using the flow nozzles in the accredited chamber. This report contains curves plotting the calculated results of the tests [8], using the calculation formulas

presented in this report for the analysis of distortion as well as those from AMCA 210 for the calculation of the tested model fan performance parameters.

# BACKGROUND

The performance of fans for Utility of Industrial applications is normally based on model tests conducted to a specific test code under laboratory conditions. Test codes commonly used for this are published by ANSI/AMCA [5] and ISO [6]. Common to all such test codes is the establishment of a relatively uniform velocity profile at the fan inlet but not perfectly uniform due to laminar boundary effects and turbulence. The results of these model tests are used by fan engineers to establish the performance of full size custom-designed fans using the laws of geometric and dynamic similitude, commonly called the "fan laws", that state that the performance of a full-sized fan can be predicted from the test results of a model fan as long as similitude is maintained between the full size fan and the model. This process is described in detail in reference [7]. This means that the performance of all custom-designed fans for Utility of Industrial Applications is predicated on the fan having a relatively uniform velocity at the inlet to the fan. Unfortunately, when fans are installed in actual Utility and Industrial applications, it is quite common to have non-uniform or distorted flow patterns at the inlet of the fan, due to the upstream ductwork configuration for ducted fans or the geometry of the space in which the fan is installed for open inlet fans. Previous model testing [1, 2] has shown that even moderate levels of distortion can have significant effects on the performance of an airfoil-bladed centrifugal fan for such applications. This information has lead to the establishment of inlet flow distortion limits in some fan field test codes [3], and the recognition of the detrimental effects of distortion in other fan field test codes [4]. During the meetings held to develop the current version of ASME PTC-11-2008, discussions were held on the concept of whether the effects of inlet distortion would be the same for a centrifugal fan with an inlet box if the fan were equipped with variable inlet vanes or did not have variable inlet vanes, with some members feeling that the presence of variable inlet vanes should help make the fan less sensitive to the effects of inlet distortion. This discussion lead to a draft of a testing program designed to explore the topic of the effect of inlet distortion on an airfoil bladed centrifugal fan equipped with variable inlet vanes. EPRI subsequently became interested in this project and provided funding to allow a limited test program to be conducted to explore this topic [8], based on the initial test program envisioned by the ASME PTC-11 group, using a pre-existing model fan supplied by FlaktWoods to minimize the cost of the testing program. This report describes this recent test program and results [8].

# **TEST PLAN**

Although not typical of most Utility Fan installations, the test program was based on using an Arrangement 1, single width, single inlet (SWSI) airfoil-bladed centrifugal fan to reduce the volume of data necessary to be collected (Figure 3).



Figure 3 The Fan Model Prior to Adding the Inlet Box

It should be noted that previous testing [1, 2, 8] conducted also used a similar SWSI type fan. It was felt that the nature of this investigation would not be greatly enhanced by using a double width, double inlet (DWDI) model, and the extra data necessary to be collected for a DWDI model would reduce the number of actual tests that could be completed. The model had a 1 meter tip diameter airfoil-bladed impeller [8] (as compared to a 0.762 meter fan used for previous testing) [1, 2] and is a medium specific speed fan (the previously tested fan was a wider and somewhat higher specific speed fan) [1, 2]. The model [8] was equipped with an inlet box and with (12) flat plate individually adjustable variable inlet vanes mounted in the inlet bell, and supported on each end of the vane, which were manually set in position for each different VIV setting to be tested. The VIV position identified as 100% open consisted of the VIV's being positioned parallel to the fan shaft (commonly described as a 90 degree position). The inlet box had a splitter plate in box on the side opposite the inlet. The flow path into the model consisted of a bell mouth entry from free space into the distortion generator, which was connected to the inlet box with a spool piece (Figure 4). The inlet pressure was measured in the spool piece.



Figure 4 Fan Model Inlet Configuration

The distortion generator consisted of a series of perforated plates supported by thin rods running axially across the face of the distortion generator frame (Figure 5).



Figure 5 The Distortion Generator

Variations in distortion were created by various combinations of the perforated plates being employed.



Figure 6 Transverse Distortion Test Configuration

Figure 6 illustrates a configuration for creating transverse distortion and Figure 7 shows a configuration for creating axial distortion.



Figure 7 Axial Distortion Test Configuration

The distorted flow was measured in the spool piece just upstream of the fan inlet box flange using a five-hole directional probe, so that any angularity in the distortion could be identified.



Figure 8 Measurement Plane Location Employing Directional Probe

Total pressure, static pressure and velocity pressure were all measured at this plane (Figure 8).

The dimensions of the model fan inlet box were 0.4996m x 1.938m [8], so a 10 port wide (along the transverse face of the inlet box perpendicular to the fan shaft) and 6 point deep traverse grid was used at the fan inlet box plane to measure the distorted flow. This inlet box has a somewhat lower aspect ratio than the 0.3 m x 1.4 m inlet box used in previous testing [1, 2]. As it turned out, the method used to create the distortion resulted in very little angularity of the distorted flow into the fan, and the quality of the results would probably not have varied in any measureable way if the distortion measurements had been made with a non-directional probe. The model was set up for testing in an AMCA 210 Figure 12 configuration (Figures 9 & 10), which involves discharge of the fan through approximately 2.5 equivalent duct diameters into a chamber where flow and outlet pressure are measured using calibrated nozzles and chamber piezometer rings [5].



Figure 9 AMCA 210 Figure 12 Test Configuration

The test laboratory is accredited by AMCA for this test configuration, and the tests were conducted in accordance with all instrument calibrations required by AMCA 210.



Figure 10 AMCA 210 Figure 12 Test Configuration

To properly evaluate the effect of distortion on the performance of the model, it was necessary to take several measurements along the fan curve (three measurement points were used for this testing program) to define a section of the performance curve. The effects of distortion are then defined along a given constant parabolic system line (pressure varying as the square of flow). Variations of the operating point for the model along its performance curve were created by varying a discharge damper and exhaust fan at the discharge end of the test chamber. Fan input power was measured using a calibrated torque sensor mounted on the input shaft (Figure 11) [8].



Figure 11 Fan Input Power Torque Sensor

All data was collected using a computerized data acquisition system that sampled the data at frequent intervals.

An important point in this program was the recognition that the perforated plates used to create the distortion into the fan also created a pressure drop that the model fan needed to overcome. To account for this, a calculation of the massaveraged total pressure at the traverse plane was made and this was added to the fan pressure measured in the chamber to get the fan total pressure being generated by the model at each operating point, with the distortion generator being considered part of the fan system rather than a part of the fan model itself. A similar approach was used in previously referenced distortion testing. [1, 2, 8].

The testing program involved of a series of tests, with each test consisting of three operating points on the model fan performance curve taken at points spanning the peak efficiency for the model at a given VIV position and distortion type and magnitude. The three operating points on each fan curve identified three system curves, which were used to quantify performance changes due to the inlet flow distortion generated for subsequent tests for the same VIV settings, as well as target the three operating points on the performance curves for different VIV settings (Low Flow, Medium Flow and High Flow). The performance of the model was measured at two different VIV settings, 100% open, 55% open and with no VIV, with 11 different inlet flow distortions, described as follows [8]:

- 1. No distortion
- 2. A single lobed transverse distortion parameter of approximately 5-10% with an offset to generate flow spin with rotation of the impeller
- 3. A single lobed transverse distortion parameter of approximately 25-35% with an offset to generate flow spin with rotation of the impeller
- 4. A single lobed transverse distortion parameter of approximately 45-55% with an offset to generate flow spin with rotation of the impeller
- 5. A single lobed transverse distortion parameter of approximately 5-10% with an offset to generate flow spin against rotation of the impeller
- 6. A single lobed transverse distortion parameter of approximately 25-35% with an offset to generate flow spin against rotation of the impeller
- 7. A single lobed transverse distortion parameter of approximately 45-55% with an offset to generate flow spin against rotation of the impeller
- 8. A single lobed axial distortion parameter of approximately 15% with an offset toward the inboard side of the inlet box
- 9. A single lobed axial distortion parameter of approximately 50% with an offset toward the inboard side of the inlet box
- 10. A single lobed axial distortion parameter of approximately 15% with an offset toward the outboard side of the inlet box
- 11. A single lobed axial distortion parameter of approximately 50% with an offset toward the outboard side of the inlet box

The transverse distortion parameter and axial distortion parameter are described in the references [2, 3, 4].

It was decided that conducting a series of tests without the VIV was important in being able to determine whether the presence of the variable inlet vanes themselves, regardless of the position of the VIV's, would have a mitigating influence of inlet distortion on the performance of the model [8]. This test program, therefore, consisted a total of 33 combinations of VIV position and/or presence (No VIV, 100% VIV and 55% VIV) each with the 11 distortions described above, with three operating point measurements per distortion (Low Flow, Medium Flow and High Flow) requiring a total of 33 x 3 = 99 individual performance tests to capture all the required data.

Table 1 describes the final test plan [8]:

Table	1	Final	Test	Plan
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Test Number	VIV Position	Distortion Type	Distortion Offset	Approx. Distortion Parameter %
T0901- 008	100%	None	NA	NA
T0901- 021	100%	Transverse	With Rotation	~ 50%
T0901- 022	100%	Transverse	Counter Rotation	~ 50%
T0901- 012	100%	Transverse	Counter Rotation	~ 30%
T0901- 013	100%	Transverse	With Rotation	~ 30%
T0901- 014	100%	Transverse	With Rotation	~ 10%
T0901- 015	100%	Transverse	Counter Rotation	~ 10%
T0901- 020	100%	Axial	Inboard	~ 50%
T0901- 019	100%	Axial	Outboard	~ 50%
T0901- 017	100%	Axial	Outboard	~ 15%
T0901- 018	100%	Axial	Inboard	~ 15%
T0901- 023	55%	None	NA	NA
T0901- 024	55%	Transverse	With Rotation	~ 50%
T0901- 025	55%	Transverse	Counter Rotation	~ 50%
T0901- 026	55%	Transverse	Counter Rotation	~ 30%
T0901- 027	55%	Transverse	With Rotation	~ 30%
T0901- 028	55%	Transverse	With Rotation	~ 10%
T0901- 029	55%	Transverse	Counter Rotation	~ 10%
T0901- 030	55%	Axial	Inboard	~ 50%
T0901- 031	55%	Axial	Outboard	~ 50%
T0901- 032	55%	Axial	Outboard	~ 15%
T0901-	55%	Axial	Inboard	~ 15%

Test Number	VIV Position	Distortion Type	Distortion Offset	Approx. Distortion Parameter %
033				
T0901- 034	No VIV	None	NA	NA
T0901- 035	No VIV	Transverse	With Rotation	~ 50%
T0901- 036	No VIV	Transverse	Counter Rotation	~ 50%
T0901- 037	No VIV	Transverse	Counter Rotation	~ 30%
T0901- 038	No VIV	Transverse	With Rotation	~ 30%
T0901- 039	No VIV	Transverse	With Rotation	~ 10%
T0901- 040	No VIV	Transverse	Counter Rotation	~ 10%
T0901- 041	No VIV	Axial	Inboard	~ 50%
T0901- 042	No VIV	Axial	Outboard	~ 50%
T0901- 043	No VIV	Axial	Outboard	~ 15%
T0901- 044	No VIV	Axial	Inboard	~ 15%

# CALCULATIONS

The following calculations have been used to quantify distortion as shown in the figures of this report:

The mean velocity for an N x M Traverse Grid (as defined in [2, 3, 8]):

$$\overline{V} = \frac{\sum_{j=1}^{M} \sum_{i=1}^{N} V_{i,j}}{N * M}$$

The mean velocity at each of the N traverses (as defined in [2, 3, 4, 8]):

$$\overline{V}_{i} = \frac{\sum_{j=1}^{M} V_{i,j}}{M}$$

The mean velocity at each of the M grid stations (as defined in [2, 3, 4, 8]):

$$\overline{V}_{j} = \frac{\displaystyle{\sum_{i=1}^{N} V_{i,j}}}{N}$$

Transverse Distortion Parameter (as defined in references [2, 3, 4, 8]):



Figure 12 illustrates a typical transverse distortion pattern created during this testing program.



Figure 12 Typical Transverse Distortion

Axial Distortion Parameter (as defined in references [2, 3, 4, 8]):

$$\hat{V}_{a} = \frac{\sqrt{\frac{\sum_{j=1}^{M} (\overline{V}_{j} - \overline{V})^{2}}{\frac{M}{\overline{V}}}} *100$$

Figure 13 illustrates a typical axial distortion pattern created during this testing program.



Figure 13 Typical Axial Distortion

Besides introducing distortion into the fan inlet flow, the distortion generator also creates a pressure drop which the fan must overcome. As mentioned previously, the performance of the fan and the distortion generator together was measured in an AMCA 210 Figure 12 configuration. This means that it is necessary to credit the fan performance with the work associated with the distortion generator. The total pressure drop across the distortion generator was measured at the traverse plane where the inlet velocity profile was measured. Using these measurements, a mass averaged estimate of the total pressure drop across the distorter was calculated using the following formula, and the fan performance was credited with this pressure drop:

Mass averaged estimate of the total pressure across the distortion generator (as defined in reference [2, 8]):

$$P_{L} = \frac{\sum_{j=1}^{M} \sum_{i=1}^{N} P_{i} * V_{i,j}}{N * M * \overline{V}}$$

To determine the reduction in fan performance due to the effects of inlet distortion, it was necessary to take several measurements along the fan curve (three measurement points were used for this testing program) to define a section of the performance curve. The effects of distortion are then defined along a given constant parabolic system line (pressure varying as the square of flow). The dependent variables selected to represent the effect of inlet distortion on performance, were

 $P_{Td}/P_{To}$  representing the change in pressure rise and  $_{d^{\prime}}$   $_{o}$  representing the change in total efficiency. The tested fan performance with distortion is compared to the test with corresponding inlet condition with no distortion, as shown in Figure 14 which is an example illustrating the effects of 50% traverse distortion parameter on the fan with no inlet vanes.

Degradation in Total Pressure due to Distortion (as defined in [8]):

$$\frac{P_{Td}}{P_{To}} = \frac{(P_{Td} - P_{To})}{P_{To}} * 100$$

Degradation in Total Efficiency due to Distortion (as defined in [2, 8]):

$$\frac{d}{d} = \frac{(d - d)}{d} * 100$$



Figure 14 Typical Comparison of Distorted Versus Undistorted Performance Curves

#### **FINAL TEST RESULTS**

As described above, a total of 33 combinations of VIV position and/or VIV presence, and distortion type and magnitude, were conducted for the final test results. This required a total of  $33 \times 3 = 99$  individual final performance tests to capture all the required data. Graphical presentation of the results of these tests are presented in the form of 24 graphs of various parameters (refer to Appendix A, Figures A-1 through A-24) which, for transverse distortion illustrate the effects of with rotation and counter rotation distortion, various VIV effects, and low, medium and high flow conditions on a single graph. For axial distortion, these graphs illustrate the effects of inboard and outboard distortion, various VIV effects, and low, medium and high flow conditions on a single graph.

# Observations from the Graphical Presentation of the Data

#### Transverse Distortion

Without the VIV, counter rotation distortion greatly increases pressure development (ramming the wheel) and with rotation distortion greatly decreases pressure development (prespin) [8]. The presence of VIV's in the 100% position eliminates most of these effects [8]. Partial closure of the VIV's does not prove beneficial beyond the effects of adding the VIV at 100% position [8]. Without the VIV, transverse distortion in both directions causes a net decrease in efficiency [8]. The presence of VIV's in the 100% position results in an improvement in efficiency (with respect to No-VIV operation) in the counter rotation direction (eliminates the effects of ramming) and a reduction in efficiency with rotation (prespin effect removed) [8]. Partial closure of the VIV's is not beneficial in either direction [8].

#### Axial Distortion

Without the VIV, distortion in both directions greatly decreases pressure development [8]. The presence of the VIV's in the 100% position reduces these effects [8]. Partial closure of the VIV's does not appear to be beneficial with the possible exception of the low flow case [8]. Without the VIV, axial distortion in both directions causes a net decrease in efficiency [8]. The presence of VIV's in the 100% position reduces this effect [8]. Partial closure of the VIV's is not beneficial in either direction [8]. With or without VIV's, Axial Distortion has a more negative effect on efficiency than transverse distortion [8]. This is opposite to previously reported testing results [2].

#### Overall effect of distortion

Other than pressure development for fans with transverse distortion in a counter rotation direction (ramming), all fan performance decreases with increasing levels of distortion [8]. Adding VIV's to an airfoil-bladed centrifugal fan appears to make the fan significantly less sensitive to the effects of all types and magnitudes of inlet distortion [8]. This reduction in sensitivity is most effective with the VIV's in the 100% open position, and decreases with further closure of the inlet vanes [8]. With respect to position of the operating point on the fan curve, in general, the higher the flow is beyond peak pressure, the greater the effect of distortion [8]. This variation of the effect of distortion along the fan's performance curve was an unanticipated, but logical, additional finding of this testing program, which has not been reported prior to the most recent testing [8].

# Summary of Recent Testing Program [8] Results

The results of this testing program [8] show that a model fan, when equipped with variable inlet vanes, is generally subject to lower levels of performance degradation, for a given level of inlet distortion, than the same model when the variable inlet vanes are not present. This indicates that the variable inlet vanes are acting to condition the distorted flow prior to entry into the fan wheel in a beneficial manner. The reduction in performance degradation seems to occur when the variable inlet vanes are wide open (parallel to the fan shaft axis), and does not seem to improve with closure of the variable inlet vanes. The maximum reduction in Total Pressure development and Total Efficiency measured for fans without variable inlet vanes in this test was -8.4% and -6.1% respectively. The maximum reduction in Total Pressure development and Total Efficiency measured for fans with variable inlet vanes in the wide open position was reduced to -2.9% and -3.5% respectively. In fact, the beneficial effects of the variable inlet vanes appear to be greater with the vanes in the wide open position, and are possibly less beneficial when the inlet vanes are partially closed. (The maximum reduction in Total Pressure development and Total Efficiency measured for fans with variable inlet vanes in a partially open position was -2.5% and -4.1% respectively.) Refer to Table 2 for a summary of the Maximum Distortion Observed for the testing program.

An unanticipated result of the tests conducted [8] is the discovery that the effects of distortion seem to vary with the position of the rating point along the fan curve, with distortion effects being greater at higher flow conditions and lower at lower flow conditions.

# **Additional Comments**

The uncertainty in the test measurements result in a 0.7% estimated fan characteristic uncertainty (as defined in reference 5) [8]. In examining the test results, it is important to keep uncertainties in mind when comparing the test results to help determine which sort of differences in the various curves presented are significant and which differences may not be significant. It should be noted that the results shown in this report represent the effects of inlet distortion on one particular medium specific speed airfoil bladed centrifugal fan, in Single Width, Single Inlet (SWSI) configuration, when exposed to a very specific type of distortion. The type of distortion generator used in this series of tests purposely resulted in flow that was very uniform in nature, other than in the specific direction where distortion was intended to take place. While this is useful in being able to accurately define and quantify the type of distortion involved, it is not representative of a real world situation, where inlet flow distortion will involve combinations of axial and transverse distortion with components that are not perpendicular to the inlet plane.

Test Number	VIV Position	Distortion Type	Distortion Offset	Approx. Distortion Parameter %	Degradation in TP due to Distortion	Degradation in TE due to Distortion
T0901-035	No VIV	Transverse	With Rotation	~ 50%	-8.40%	-1.50%
T0901-036	No VIV	Transverse	Counter Rotation	~ 50%	5.70%	-2.80%
T0901-041	No VIV	Axial	Inboard	~ 50%	-4.20%	-5.30%
T0901-042	No VIV	Axial	Outboard	~ 50%	-4.60%	-6.10%
T0901-021	100%	Transverse	With Rotation	~ 50%	-1%	-1.50%
T0901-022	100%	Transverse	Counter Rotation	~ 50%	-1.20%	-1.30%
T0901-020	100%	Axial	Inboard	~ 50%	-1.4	-1.7
T0901-019	100%	Axial	Outboard	~ 50%	-2.9	-3.5
T0901-024	55%	Transverse	With Rotation	~ 50%	-2.50%	-2.70%
T0901-025	55%	Transverse	Counter Rotation	~ 50%	-1.70%	-2.70%
T0901-030	55%	Axial	Inboard	~ 50%	-1.50%	-2.6
T0901-031	55%	Axial	Outboard	~ 50%	-2.20%	-4.1

 Table 2
 Maximum Distortion Observed for High Flow Condition

Indicates Maximum Value for a given VIV configuration

#### **DISTORTION CASE STUDY**

The following case study was initiated to determine the effects of combined axial and transverse inlet distortion on a single inlet, single width fan controlled with inlet box dampers. The actual fan experiencing the inlet distortion has an airfoil blade design with an overall diameter of 127" and a total width of 22.75". The fan is a primary air fan that was installed as part of the balanced draft conversion of an existing Southern Company pulverized coal fired boiler.

Several months after initial operation of the fan, bearing vibration levels increased to the point that the fan was taken out of service. Based on the inlet damper position of the fan at full load, the initial suspicion was that the fan was oversized on flow and was running left of peak on the fan curve in an unstable region of the fan curve where the fan would be running in stall and experience high levels of vibrations. Inspection of the fan found multiple cracks in the fan hub, presumable from the aerodynamic forces experienced by running in stall. The fan hub was repaired and the fan was brought back into service to perform a fan system performance test. The performance test revealed that the system flow requirements were per the specified full load duty point, but the fan did show high levels of axial and transverse inlet distortion. Based on the high levels of inlet distortion, it was hypothesized that the inlet distortion could be causing the aerodynamic forced vibrations that led to the fatigue failure of the hub. Modifying the fan inlet ductwork to eliminate the inlet distortion was impossible, so the decision was made to model the fan to observe the effects of the inlet distortion on the fan and to determine what modifications to the fan could be done to mitigate the effects on the fan.

The fan model was an approximate 1/3<sup>rd</sup> scale. The model wheel diameter was 40", while the actual fan wheel diameter is 127". The model scaling was per AMCA 210 and 802 guidelines. The fan inlet and outlet isolation dampers were also modeled as shown in Figure 15.



Figure 15 Primary Air Fan Model

The fan inlet distortion measured on the actual fan was simulated by placing screens and boards across the fan inlet box as shown in Figure 16.



Figure 16 Inlet Screens

The following charts show a comparison of the inlet profile on the actual fan and the simulated inlet distortion on the model fan.



Figure 17 Actual Fan Inlet Profile – Axial Distortion 36.8%, Transverse Distortion 14.7%



Figure 18 Model Fan Inlet Profile – Axial Distortion 34.0%, Transverse Distortion 12.1%

The actual and model fan inlet profiles are similar, but not exact. Both fall outside of the AMCA 803 acceptance criteria to not exceed 10%. The actual fan performance test was administered using a 5-hole pitot tube, which measured some fan inlet flow non-parallel to the duct. The model fan was administered with a standard pitot tube that in not capable of measuring yaw and pitch angles.

The following charts depict a comparison of the fan performance with and without distortion. The fan model was credited for the pressure drop created by the inlet screens and boards. The first is the flow versus static pressure rise curve. The second is the flow versus power curve. The data points taken with uniform inlet conditions are in blue, while the distorted inlet conditions are in red. The model fan performance has been corrected to the actual fan size using the fan laws.



Figure 19 Fan Curve, Pressure vs. Flow – Comparison of Distorted and Non-Distorted Inlet Conditions



Figure 20 Fan Curve, Power vs. Flow – Comparison of Distorted and Non-Distorted Inlet Conditions

The effects of inlet distortion on fan performance were relatively insignificant and fell within the test uncertainties of 2% on flow, 2% on pressure and 3% on power. The static pressure rise and power consumption was slightly higher at the 60% inlet damper position. The pressure vs. flow and power vs. flow were almost identical for the 20% and 100% inlet damper positions.

Ultimately the aerodynamic forces that caused the fatigue failure of the actual fan hub were caused by running the fan in rotating stall for an extended period of time. By modeling the fan, the concerns about the effect of inlet distortion were address and the actual rotating stall regions on the fan curve were observed, which were well within the normal expected range of operation of the fan. Based on these observations, the fan was retrofitted with variable inlet vanes that are less susceptible to rotating stall in the low flow, low pressure area of the fan curve. The additional inlet restrictions of the variable inlet vanes were also assumed to help any impact from inlet distortion.

#### **Recommendations to Minimize Distortion Effects**

Properly sized and functioning main service fans are essential to the operation of a fossil-fuel Utility Unit and to many industrial applications as well. These fans are also some of the largest consumers of energy in such plants. As such, it is important to understand the relationship that proper inlet flow patterns to such fans may have on the ability of the fan to meet the required performance of the Unit or process, and the impact that may result to the fan's flow, pressure, and required input power from distorted inlet flow. The duct layout upstream of the fan inlet is the most common source of inlet flow distortion. The cost of adding additional duct length to ensure a favorable flow profile can be prohibitive in new installation and often is not feasible in existing installations. For a new installation, or an existing installation with new fans and modification to the ducts, flow modeling using either scale physical models or CFD is recommended. Such modeling can be effective and can

provide guidance for improving the flow profile for a minimal cost using turning vanes.

Flow modeling can be included in the scope of the fan supplier, or can be conducted by an engineering consultant. If is important for all parties involved to work together to develop cost effective turning vanes to create an acceptable inlet flow profile. When fan systems are analyzed using such techniques to minimize inlet distortion, most fan performance problems related to inlet distortion can be avoided. When situations arise where higher levels of distortion cannot be avoided, information such as that presented in this report can be used to attempt to determine what level of distortion might be acceptable for a given fan application, fan type and means of control, through an estimation of the effect of the type and magnitude of the distortion on the performance of the fan. In taking these steps, the system designer can help assure that the fan and associated system operate as intended, with the lowest reduction in fan output and efficiency due to inlet distortion, resulting in the most efficient fan installation possible, given the system design constraints.

#### CONCLUSIONS

Very little factual information exists on the effect of inlet distortion on fans. The information that does exist [1, 2] does not thoroughly address the effects of distortion on the performance of centrifugal fans equipped with variable inlet vanes. Therefore, this report and the referenced testing [8] adds new information to the body of public information on the important topic of the negative effects of inlet distortion on centrifugal fans.

Returning to the major findings of references [1] and [2] outlined earlier in this paper, the recently conducted testing [8] shows major differences, and, in fact, several findings that are the opposite of previously concluded, as described below.

- "Even moderate levels of distortion 10 to 15 percent velocity variations or 10 to 15 percent lateral offset of the flow entering the inlet box – can lead to significant penalties in performance – 5 percent in efficiency and pressure rise. Relatively severe distortion – 30 to 50 percent velocity variation or flow offset – can cause serious deterioration in fan performance – 10 to 12 percent reduction in efficiency and pressure rise." The testing from Reference [8] showed less sensitivity to distortion effects, as can be seen in Figures 21 and 22.
- Axial distortion has less impact than transverse distortion. For 20% distortion the impact on pressure is 3% for transverse distortion and 0.5% for purely axial distortion. The testing and following conclusions are in reference to transverse distortion. The testing from Reference [8] showed the opposite effect Axial distortion had greater impact than transverse, and had a much more significant impact than Reference [1] and [2], as illustrated in Figures 23 and 24.

- 3. The impact of distortion on efficiency was essentially the same with and without inlet guide vanes installed. The testing from Reference [8] indicated the opposite effect.
- 4. The impact of distortion on total pressure rise without inlet guide vanes depends on the distortion relative to rotation. With inlet guide vanes the impact is independent of rotation and is less in the worst case. This is the only major finding from references [1] and [2] that was supported by the testing from Reference [8].

The data presented in the case study presented in this paper also varied from both references [1.2] and [8]. The main difference between each of these tests was the fact that they were done on different model fans. This leads to authors to believe that the actual impact of inlet distortion is highly dependent on the specific geometry of the airfoil-bladed centrifugal fan involved, and the relationship to various fan design parameters is not currently well understood. Significant additional testing would be required to determine what fan design parameters influence the relative sensitivity of fan performance to the effects of inlet flow distortion.



Figure 21 Comparison of % Change in TP vs. Transverse Distortion Parameter – High Flow



Figure 22 Comparison of % Change in TE vs. Transverse Distortion Parameter – High Flow



Figure 23 Comparison of % Change in TP vs. Axial Distortion Parameter – High Flow



Figure 24 Comparison of % Change in TE vs. Axial Distortion Parameter – High Flow

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#### **APPENDIX A**

# **Graphical Presentation of Test Results [8]**



Figure A-1 % Change in TP vs. Transverse Distortion Parameter and VIV Condition – High Flow



Figure A-2 % Change in TE vs. Transverse Distortion Parameter and VIV Condition – High Flow



Figure A-3 % Change in TP vs. Transverse Distortion Parameter and VIV Condition – Medium Flow



Figure A-4 % Change in TE vs. Transverse Distortion Parameter and VIV Condition – Medium Flow



Figure A-5 % Change in TP vs. Transverse Distortion Parameter and VIV Condition – Low Flow



Figure A-6 % Change in TE vs. Transverse Distortion Parameter and VIV Condition – Low Flow



Figure A-7 % Change in TP vs. Axial Distortion Parameter and VIV Condition – High Flow



Figure A-8 % Change in TE vs. Axial Distortion Parameter and VIV Condition – High Flow



Figure A-9 % Change in TP vs. Axial Distortion Parameter and VIV Condition – Medium Flow



Figure A-10 % Change in TE vs. Axial Distortion Parameter and VIV Condition – Medium Flow



Figure A-11 % Change in TP vs. Axial Distortion Parameter and VIV Condition – Low Flow



Figure A-12 % Change in TE vs. Axial Distortion Parameter and VIV Condition – Low Flow



Figure A-13 % Change in TP vs. Transverse Distortion Parameter and Flow – No VIV



Figure A-14 % Change in TE vs. Transverse Distortion Parameter and Flow – No VIV



Figure A-15 % Change in TP vs. Transverse Distortion Parameter and Flow – 100% VIV



Figure A-16 % Change in TE vs. Transverse Distortion Parameter and Flow – 100% VIV



Figure A-17 % Change in TP vs. Transverse Distortion Parameter and Flow – 55% VIV



Figure A-18 % Change in TE vs. Transverse Distortion Parameter and Flow – 55% VIV



Figure A-19 % Change in TP vs. Axial Distortion Parameter and Flow – No VIV



Figure A-20 % Change in TE vs. Axial Distortion Parameter and Flow – No VIV



Figure A-21 % Change in TP vs. Axial Distortion Parameter and Flow – 100% VIV



Figure A-22 % Change in TE vs. Axial Distortion Parameter and Flow – 100% VIV



Figure A-23 % Change in TP vs. Axial Distortion Parameter and Flow – 55% VIV



Figure A-24 % Change in TE vs. Axial Distortion Parameter and Flow – 55% VIV