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EARLY DETECTION OF ROTATING STALL PHENOMENON IN CENTRIFUGAL COMPRESSORS BY MEANS OF ASME PTC 10 TYPE 2 TEST

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ABSTRACT

A centrifugal compressor presenting rotating stall can exhibit many problems, as limitation on its operational range, high level of subsynchronous vibrations, reduced efficiency and premature bearing wear.

There are many design criteria and standard practices for aerodynamic design of impellers and diffusers of centrifugal compressors. Nonetheless, the inception of rotating stall in new projects is still a reality. Therefore, an early detection of this phenomenon in a compressor is highly desirable.

This paper proposes a way to detect the rotating stall problem during an ASME PTC10 Type 2 Test and how to scale these results (amplitude of vibration and inception point) as an estimative to site conditions.

A comparison between the results of the proposed extrapolation from ASME PTC10 Type 2 Tests to the full load/density (almost an ASME PTC-10 Type 1 test) conditions is also presented. Scaling up guidelines for vibrations from similarity test to full density conditions are presented.

Finally, an acceptance criterion for the amplitude of subsynchronous vibrations is proposed.

INTRODUCTION

In general, rotating stall in a vaneless diffuser will occur only at very low flows, close to the compressor surge line. However, in some cases, rotating stall can occur well within the compressor operating range.

The presence of this phenomenon may result in a significant drop in aerodynamic performance, increase on the vibration levels and, in some cases, a reduction of the operational range. This is very critical for compressors working on high pressure levels, since the forces involved are much larger due to the high density of the working fluid. The high vibration levels can lead to high financial losses in operation and maintenance due to the impossibility to operate, therefore an early detection of this phenomenon during the compressor's test campaign is highly desirable.

Even when the standard test campaign described at the section 4 of API 617 is successfully accomplished, the rotating stall phenomenon may not be detected and still be present during operation.

This paper proposes a methodology to detect this phenomenon during the compressor factory tests.

The different types of tests and their characteristics will be discussed in order to determine how tests can be carried out to detect the rotating stall problem. Any modification shall focus in minimizing the impact on the current tests program.

The modified tests to detect rotating stall phenomenon and an acceptance criterion for subsynchronous vibration are presented.

A comparison between the results from this methodology with full density conditions is presented.

ROTATING STALL

Rotating stall is a periodic unstable flow phenomenon consisting of recirculating cells rotating in the same direction of the impeller rotation. Typically formed at low flows, stall occurs when the radial velocity component of the flow is insufficient to permit the gas to proceed along the diffuser. In other words, the gas has not enough dynamic energy to win the adverse pressure gradients present in the diffuser.

In three dimensional analysis, the boundary layer separation at the walls of the diffuser is pointed as the starting cause of rotating stall phenomenon.

Before the phenomenon has taken place, the flow is very stable. With its reduction the flow pattern becomes unsteady, with cells moving in the direction of the rotational speed.

These cells, which are associated with pressure fluctuations, act as an excitation source of vibration for the shaft, mainly through the impeller. This excitation source allied with the rotordynamic characteristics (damping, stiffness) of the structure and machine will result in rotor vibration. Kita, et al. (2008) initiated a study to predict the amplitude of subsynchronous vibrations (SSV) caused by rotating stall.

The typical frequency range for the pressure fluctuations (consequently the vibrations) is normally between 5 to 20 percent of impeller rotational speed, but it can reach up to 35 percent. The detection of subsynchronous vibration on this frequency range is the first alert to the occurrence of rotating stall phenomenon.

By further reduction of flow, rotating stall can be a factor on leading the compressor to surge. During the rotating stall, in contrast with surge, the net flow rate through the compressor remains roughly constant. Surge, on the other hand, will cause the whole flow to oscillate.

Senoo and Kinoshita (1978) proposed a project criterion to define a critical flow angle for onset of rotating stall. Kobayashi, et al. (1990) improved the calculation of this critical angle.

POSSIBLE SOLUTIONS

There are many engineering and academicals solutions whenever the Rotating Phenomenon is presented. Most of them aim to reduce tangential velocity or to raise the radial velocity (or both) until the stall cells are eliminated. Indeed, inception point of rotating stall is related to a critical flow angle (Senoo, Nishida-Kobayashi etc.). Another well-known solution is to guide the flow in the diffuser.

Pinching consists of width reduction on the gas path at the diffuser. The manufacturing of pinch rings and their installation is normally the fastest reaction to a stall situation. There are many reports of successful elimination of rotating stall using this method, i.e. as reported by Fulton and Blair (1995).

Low Solidity Vaned Diffusers (LSVD) redirect the flow in order to optimize the radial and tangential velocities, suppressing the rotating stall phenomenon. Many optimizations of the vanes geometry and location have been studied.

Diffuser Roughness can play a role on the stall suppression as shown by Ishida, et al. (2001). A rougher wall can suppress the three-dimensional separation from boundary layer.

Radial Grooves, as numerically demonstrated by Gao, et al. (2009), can eliminate the reverse flow from diffuser walls. Grooves position and format can be optimized by CFD.

A Gas Jet can suppress or amplify the rotating stall phenomenon depending on the flow angle of the jet. Tsurusaki and Kinoshita (2001) explained the control effect using the conservation law of the moment of momentum.

FACTORY ACCEPTANCE TESTS

The factory acceptance tests are the last chance to detect the rotating stall problem in the manufacturer installations.

The requirements that should be met in the tests in order to enable the detection of rotating stall phenomenon are:

1) Vibration measurement including full spectrum analysis;

2) Sweep of the compressor curve (low flow region investigation);

3) Compressor aerodynamics close to field conditions or at flow similarity conditions.

MRT - MECHANICAL RUNNING TEST

The mechanical behavior of the machine is analyzed. The equipment is operated from zero to the maximum continuous speed. Normally a lighter gas as Helium (MW 2g/mol) or vacuum is used to save power and eliminate the hazard of a combustible gas.

The conditions 2) and 3) described above are not fulfilled during the MRT and therefore the detection of rotating stall phenomenon is not possible at this test.

PT1 - ASME PTC 10 TYPE 1 PERFORMANCE TEST

As the type 1 performance tests are almost at the field conditions, they are the most appropriate tests to detect the phenomenon of interest of this paper. This test is normally run as a complete unit test (CUT) with suction and discharge densities close to site conditions and fulfills the three listed conditions for rotating stall detection.

The disadvantages of using PT1 test for rotating stall identification are:

- due to the high costs, this test is not usually specified by the majority of the users;
- when specified by users, this is normally carried out in only one machine. Being the last test of the campaign, it is often too late to detect and remove the rotating stall phenomenon.

FSFL - FULL SPEED FULL LOAD TEST

The full speed full load tests are a good alternative to the type 1 tests. The modified type 1 test, as proposed by Miranda and Noronha (2007), focuses on the mechanical behavior and rotordynamic stability.

Depending on the agreed test procedure, some adaptation on the original test to match conditions 2) and 3) may be needed.

This kind of test faces the same disadvantages as those described to the PT1 tests.

PT2 - ASME PTC TYPE 2 PERFORMANCE TEST

This test fulfills the conditions 2) and 3), but not completely the condition 1).

Although not specified by the standards, the vibration measurement (including real time spectral analysis) at performance tests is usually available on vendors test bench.

The PT2 and MRT are normally specified by purchasers, in opposite of the more expensive PT1 and FSFL. The main disadvantage in detecting rotating stall phenomena is the relatively low rotational speed used and the low densities achieved during the test.

DECISION

Based on the comparison presented between the tests, the ASME PTC type 2 Test may be the best opportunity to detect the rotating stall phenomenon during the manufacturing phase.

The decision was based on the optimization between efforts to adapt the test and the temporal sequence of manufacturing / testing.

DETECTION OF ROTATING STALL IN THE ASME PTC TYPE 2 TEST

INSTALLATION

The adaptation needed on the normal PTC type 2 test installation is only the close monitoring of vibration data. That should not be an issue as the mounting positions for the MRT and PT are normally the same. Therefore, vibration probes and acquisition system are usually available.

DETECTION

As the SSV in the frequency range from 0.05 - 0.20 X are detected on the PTC type 2 test, the phenomenon shall be further investigated, as other phenomena, such as structural resonance, can also generate SSV in this frequency range.

Typical evidence of structural resonance is that it remains at a fixed frequency. In order to determine the excitation source, tests at different speeds should be carried out. At least three different test speeds are recommended.

Whenever the dominant frequency of SSV varies with test speed, the structural resonance hypothesis can be discarded. Ahmed and Gadalla (2009) demonstrated that frequency of stall varies in linear proportion with impeller speed. This is particularly true for a single stage machine where stall is always on the same component, yet it tends to deviate in multistage machines where stall can move from one stage to another as a function of the speed and Mach number.

A further confirmation from the aerodynamic source of SSV can be obtained from an additional test. This new test will be called “density tests” on this paper and consists in SSV measurement at different gas densities (at least three). Whether the direct link between SSV recorded amplitudes and gas density can be demonstrated, the aerodynamic source of excitation is confirmed.

SCALING UP

The vibration measured at the test will not be equal to the vibration measured at full density conditions in the site.

The amplitude of vibration caused by the rotating stall phenomenon is dependent on the gas density. Hence a correction shall be applied to transpose the test-bench results to the field conditions.

The Figure 1 shows the correlation between sub-synchronous vibration amplitude and different average gas densities, recorded in a series of PT2 tests.

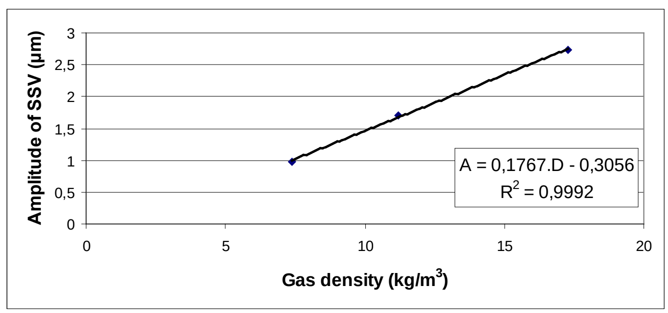

Figure 1: SSV amplitude versus gas densities

The linear behavior of amplitude of vibration with the gas density is a realistic hypothesis within the typical range of Oil&Gas compressors applications. In that way, an extrapolation of curve obtained from the density tests could be a good estimative of the amplitude of SSV in site conditions.

For calculation purposes the mean density across the compressor shall be used.

Although this curve changes with each individual compressor, as the stiffness, damping and excitation forces may vary considerably, it is reasonable to assume one curve per compressor type.

Whether, for any reason, the density tests could not be carried out, it is possible to use a density relation to scale the SSV observed during testing up to operating conditions. This is demonstrated on Equation 1.

$$A_F = A_T \times \frac{D_F}{D_T}$$

Equation 1: SSV amplitude up-scale

Where:

A_F = Amplitude of vibration at the field;
A_T = Amplitude of vibration at the test;
D_F = Density at field;
D_T = Density at test.

This approach has a bigger inherent error in comparison with the extrapolation results.

ACCEPTANCE CRITERIA

The establishment of an acceptance criterion is the most difficult task on this methodology. Generally, rotating stall becomes a problem if:

- Amplitude of subsynchronous vibration is relevant, or up-trending;
- Inception point of rotating stall is within operating range of the compressor.

The sub-synchronous vibration amplitude is the symptom of rotating stall. As long as the amplitude of vibration remains at low levels and no up-trend is developing, it is possible to operate with the problem.

However, during a factory acceptance test, it is impossible to foresee if a trend is going to establish. Also, determining the amplitude of sub-synchronous vibration threshold beyond which the machine is threatened is a hard task.

A first approximation can be based on the item 4.3.1.3.2. from the API 617 Standard 7th Edition:

"4.3.1.3.2 While the equipment is operating at maximum continuous speed, or other speed required by the test agenda, vibration data shall be acquired to determine amplitudes at frequencies other than synchronous. This data shall cover a frequency range from 0.25 – 8 times the maximum continuous speed. If the amplitude of any discrete, non synchronous vibration exceeds 20% of the allowable vibration as defined in 2.6.8.8 of Chapter 1, the purchaser and the vendor shall mutually agree on requirements for any additional testing and on the equipment's acceptability."

The value of 20 percent of allowable vibration for non-synchronous speeds can be adopted for the expected SSV amplitude at operating conditions. To estimate this value, the correction described in section *SCALING UP* of this paper shall be applied.

However, the frequency range specified in the above item needs to be adapted. As described, rotating stall can occur between 5 and 20 percent of shaft speed, hence, the frequency range proposed for the modified test should be: "0.05 - 8 times the test speeds".

This criterion shall be applied in all the operating range of compressor, from the surge control line (SCL), which correspond to a flow 10% higher than the guaranteed surge line, to the maximum flow, and at multiple speeds.

Normally, the highest amplitude of SSV will be present at the low flow region. In some cases, the vibration amplitude may decrease and change frequency when reducing gas flow, this occurs when the rotating stall phenomenon is transitioning from a single rotating stall cell to two (or even more) rotating stall cells, which reduces the resultant force on the shaft.

CASE STUDY

This methodology arose from problems faced in manufacturing of the compression system for two FPSOs with capacities of 6 millions standard cubic meters of natural gas per day. Each one of the six compression modules (three per FPSO) is composed by three sections in two casings. The low pressure (LP) compressor has two sections back-to-back and the high pressure (HP) compressor has a section with 7 impellers in a straight-through arrangement.

The low pressure compressor presented low amplitude sub-synchronous vibrations during the performance tests. After testing each one of two sections separately, it was determined that SSV was originated on the section 2 of the compressor. In this section the diffuser pinching is ranging from about 35% to 45% (ratio between diffuser width and impeller exit width) and the diffuser ratio is about 1.4 (ratio between radius at end of diffuser vertical wall and radius at impeller exit).

As the HP compressor has the same impellers / diffusers concept of the second section of LP compressor, tests were carried out for HP compressor as well. Although in a minor amplitude, SSV were also detected for the HP compressor.

The second section of LP compressor is constituted of 4 impellers (2D), with no drains in each individual impeller. Therefore, it was not possible to determine the exact source of the subsynchronous vibration by the installation of pressure probes, as described by Sorokes, et al (1994). Even by analyzing the critical angles for this project, it was not possible to determine the exact stalled stage, given that all of them are close to the Nishida-Kobayashi criterion. The Table 1 shows the margin to Nishida-Kobayashi criterion for the LP compressor for the FPSO 1 and Table 2 shows the same parameter for FPSO 2.

	Stage 1	Stage 2	Stage 3	Stage 4
Nishida-Kobayashi critical angle	13.49º	13.45º	13.42º	13.35º
Actual Flow Angle	15.72º	15.63º	15.48º	14.93º
Margin versus Nishida-Kobayashi criterion	**2.23º**	**2.18º**	**2.06º**	**1.58º**

Table 1: Margin to Nishida-Kobayashi criterion (FPSO 1)

	Stage 1	Stage 2	Stage 3	Stage 4
Nishida-Kobayashi critical angle	13.49º	13.45º	13.42º	13.35º
Actual Flow Angle	14.83º	15.20º	15.48º	15.28º
Margin versus Nishida-Kobayashi criterion	**1.34º**	**1.74º**	**2.06º**	**1.93º**

Table 2: Margin to Nishida-Kobayashi criterion (FPSO 2)

For the HP compressors the minimal margin to Nishida-Kobayashi criterion is 3.78º in the first diffuser. All other stages have a margin greater than 5.5°. This clarifies why the HP compressors, even working with higher densities, were affected in a much minor grade by SSV in comparison to the LP compressors.

The suggested methodology of this paper was applied to the LP compressor of FPSO 1. After the SSV detection, the test was repeated with different speeds as shown in Figures 2, 3 and 4.

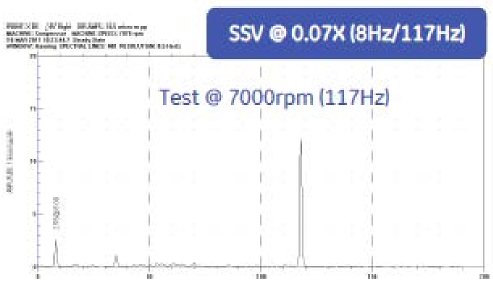

Figure 2: Vibration Spectrum at 7000 rpm

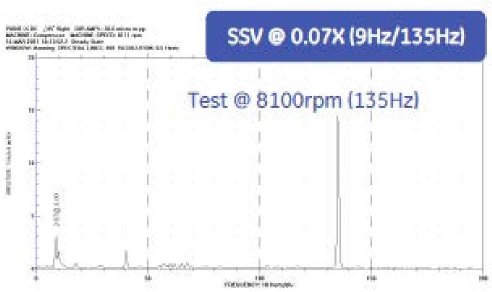

Figure 3: Vibration Spectrum at 8100 rpm

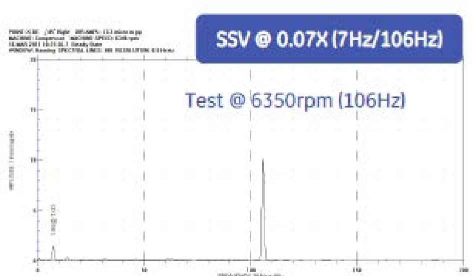

Figure 4: Vibration Spectrum at 6350 rpm

After the confirmation of a speed-related phenomenon, test was repeated with three different gas densities. The results are shown in Figure 5.

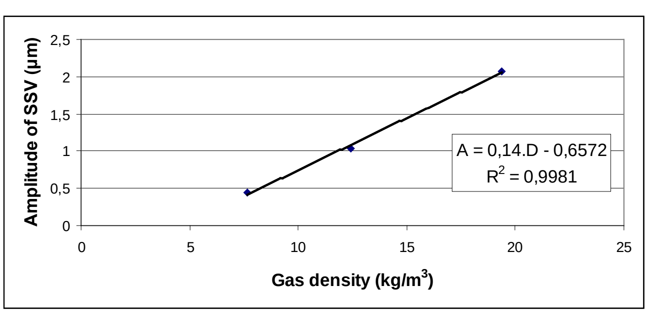

Figure 5: Gas density and SSV amplitude relation

The extrapolation formula was applied using the mean density of gas (28.95 kg/m^3) at the field. The foreseen amplitude of subsynchronous vibration at surge control line was 3.40µm, which is within acceptance criterion (5µm or 20 percent of the allowable overall amplitude 25µm). Using the densities relation as described in Equation 1, the predicted value was 2.37µm.

The next step was the confirmation of this criterion on a full load, full density test.

The mean value of SSV amplitude found in FSFL was 3.99µm, when the field density was almost achieved (96.2 percent). The Figure 6 shows a vibration spectrum at the surge control line.

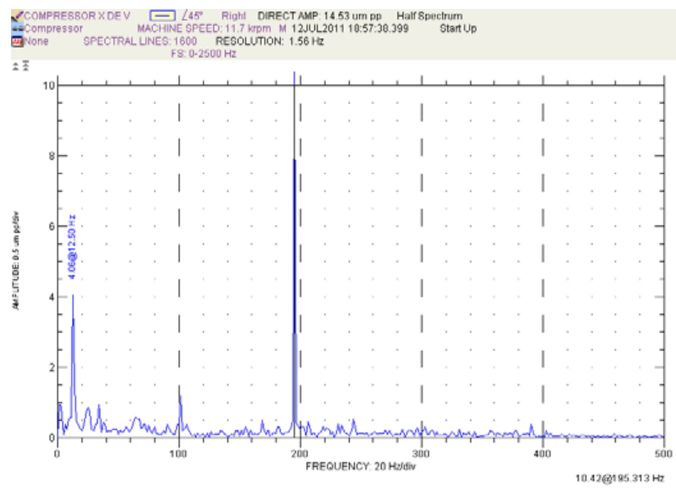

Figure 6: Vibration spectrum at SCL.

The agreement using the extrapolation technique was better than that using the Equation 1.

The comparison between the frequency of excitation in the PT2 and FSFL tests is showed at Table 3, which confirms the reproduction of phenomenon on the PT2 tests.

Speed Tests (rpm)	8111	7070	6348	11713
Stall Frequency (Hz)	9	8	7	12,5
Percent of synchronous speed	0.067X	0.068X	0.066X	0.064X

Table 3: Stall frequencies

Despite of the good agreement between the extrapolation from PT2 and FSFL tests, only one test is not enough to validate the proposed methodology.

In the same project three more FSFL tests were carried out, two for the high pressure compressors and one for the low pressure compressor.

Due to schedule constraints, the density tests could not be realized. The comparison between PT2 and FSFL was done only by means of densities ratio (as described by Equation 1).

The first high pressure compressor did not present SSV in the operational range during neither the PT2 nor the FSFL tests. In this case, rotating stall was only observed to the left of surge control line.

Tables 4 and 5 show the comparison between SSV amplitudes observed in FSFL and PT2. The PHI column is a dimensionless coefficient for the volume flow, allowing direct comparison of the inception point of rotating stall phenomenon.

PHI	Full Density (µm)	PT2 Upscaled (µm)
0,0185	3,73	n.a.
0,0176	6,77	2,68
0,0159	n.a.	7,49

Table 4: Second LP compressor (FPSO 2)

PHI	Full Density (µm)	PT2 Upscaled (µm)
0,0130	1,13	n.a.
0,0129	6,30	n.a.
0,0122	n.a.	0,84
0,0109	n.a.	7,79

Table 5: Second HP compressor (FPSO 2)

At these tests, the agreement between the two conditions was not satisfactory, as the inception point of rotating stall at full density occurred at a higher flow than that of PT2. The uncertainty of flow measurement (approximately 2 percent) used on the tests may be responsible for part of the observed difference of 5 to 7 percent at the rotating stall inception point.

As demonstrated for the first LP compressor, the use of extrapolation of many density tests results in a better agreement with full density conditions. Nevertheless it should be noted that, at least on these cases, the observed SSV amplitude observed in the FSFL test was lower than predicted by upscaling the PT2 results, although it occurred at a higher flow.

Some extra-head was observed during the FSFL tests, what can induce lower flow angles at impeller exit and explains the anticipation of rotating stall inception point.

Van Den Braembussche, et al. (1980) applied a Reynold's Number correction to the critical angle defined by Senoo and Kinoshita (1978) in order to match theory and experimental data. That can be a way forward to improve the methodology described on this paper.

CONCLUSIONS

The detection of rotating stall in centrifugal compressors can be anticipated using a modified ASME PTC 10 type 2 test. The additional effort is the close monitoring of vibration during the performance test. Once a subsynchronous vibration is detected, some additional tests are required.

An acceptance criterion based on an API 617 modified item was proposed. The decision process to accept or reject a machine with the proposed acceptance criterion depends on the specific conditions of the project. In some cases, the introduction of a solution (i.e. LSVD) can bring more risk (i.e. vanes passing frequency resonance) than the rotating stall itself. Therefore the adoption of this acceptance criterion shall be an agreement between purchaser and vendor.

The comparison between the values extrapolated from the PT2 and FSFL tests demonstrated a good agreement on the SSV amplitude. In some tests the inception point was observed to be at higher flow during the FSFL test than in the PT2, but this may be partially due to flow measurement uncertainties and differences on Reynolds number.

The methodology was developed during testing phase of compressors manufacturing, hence the amount of possible tests was limited by a tight schedule and controlled costs.

For industrial applications the use of this methodology may be sufficiently accurate. The methodology shall be improved with more tests in a controlled environment as a test rig.

NOMENCLATURE

2D -	= Bi-dimensional impellers
CFD	= Computational Fluid Dynamics
FSFL	= Full Speed Full Load Test
HP	= High Pressure
LP	= Low Pressure
LSVD	= Low Solidity Vaned Diffusers
MRT	= Mechanical Running Test
MW	= Molecular Weight
PHI	= Suction Flow Factor
PT1	= ASME PTC 10 TYPE 1 PERFORMANCE TEST
PT2	= ASME PTC 10 TYPE 2 PERFORMANCE TEST
SCL	= Surge Control Line
SSV	= Subsynchronous Vibrations

REFERENCES

Ahmed, S. A., and Gadalla, M., 2009, An Experimental Investigation of the Effects of Vaneless Diffuser Geometry on the Stall Characteristics, @ Proceedings of the ASME 2009 Fluids Engineering Division Summer Meeting, FEDSM2009-78262.

API 617 7th Edition, 2001, Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services, Seventh Edition, American Petroleum Institute, Washington, D.C..

Ferrara, G., Ferrari, L., Mengoni, C. P., De Lucia, M., Baldassarre, L., 2002, Experimental Investigation and Characterization of the Rotating Stall in a High Pressure Centrifugal Compressor: Part I: Influence of Diffuser Geometry on Stall Inception @ Proceedings of ASME TURBO EXPO 2002 3-6 June 2002.

Ferrara, G., Ferrari, L., Mengoni, C. P., De Lucia, M., Baldassarre, L., 2003, Experimental Investigation and Characterization of the Rotating Stall in a High Pressure Centrifugal Compressor: Part III: Influence of Diffuser Geometry on Stall Inception and Performance (2nd Impeller Tested) @ Proceedings of ASME TURBO EXPO 2003 16-19 June 2003.

Ferrara, G., Ferrari, L., Mengoni, C. P., De Lucia, M., Baldassarre, L., 2002, Rotating Stall in Centrifugal Compressor Vaneless Diffuser: Experimental Analysis and Phenomenon Characterization @ The 9th of International Symposium on Transport Phenomena and Dynamics of Rotating Machinery

Fulton, J. W. and Blair, W. G., 1995, Experience with Empirical Criteria for Rotating Stall in Radial Vaneless Diffusers, @ Proceedings of the Twenty-Fourth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 97-

106.

Gao, C., Gu, C., Wang, T. and Yang, B., 2009, Passive Control of Rotating Stall in Vaneless Diffuser with Radial Grooves: Detailed Numerical Study, @ Proceedings of ASME Turbo Expo 2009: Power for Land, Sea and Air, GT2009-59323.

Ishida, M., Sakaguchi, D., and Ueki, H., 2001, Suppression of Rotating Stall by Wall Roughness Control in Vaneless Diffusers of Centrifugal Blowers, @ ASME Journal of Fluids Engineering, 123 (1), pp. 64-72.

Kita, M., Iwamoto, S., Kiuchi, D. and Kawashita, R., 2008, Prediction of Subsynchronous Rotor Vibration Amplitude caused by Rotating Stall, @ Proceedings of the Thirty-Seventh Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 97-102.

Kobayashi, H., Nishida, H., Takagi, T., Fukoshima, Y., 1990, A Study on the Rotating Stall of Centrifugal Compressors (2nd Report, Effect of vaneless diffuser inlet shape on rotating stall), @ Transactions Japan Society of Mechanical Engineers (B Edition) 56 (529): pp. 98-103.

Miranda, M. A. and Noronha, R. F., 2007, ASME PTC 10 Modified Test for Mechanical Assessment of Centrifugal Compressors, @ Proceedings of the Thirty-Sixth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 87-94.

Senoo, Y. and Kinoshita, Y., 1978, Limits of Rotating Stall and Stall in Vaneless of Centrifugal Compressors, ASME paper 78-GT-19.

Tsurusaki, H. and Kinoshita, T., 2001, Flow Control of Rotating Stall in a Radial Vaneless Diffuser, @ ASME Journal of Fluids Engineering, 123 (1), pp. 281-286.

Van Den Braembussche, R., Frigne, P., and Roustan, M., Rotating Nonuniform Flow in Radial Compressors, in Centrifugal Compressors, Flow Phenomena and Performance, AGARD CP 282, Paper 12, May 1980.

Sorokes, J. M., Kuzdzal, M. J., Sandber, M. R., and Colby, G. M., 1994, ““Recent Experiences in Full Load Full Pressure Shop Testing of a High Pressure Gas Injection Centrifugal Compressor”, Proceedings of the Twenty-Third Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 3-18.

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