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HOW TO DESIGN AND OPERATE QUIET CENTRIFUGAL COMPRESSOR SYSTEM

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ABSTRACT

Historically, designs of centrifugal compressor systems focused on the aerodynamic and performance aspects. Noise, pulsation, and vibration phenomena were rarely considered. Recent applications of high flow and high power centrifugal compressors require that this approach be changed. Several transmission system failures, in different gas transmission companies, were documented. They included fatigue failures of the compressor components, piping attachments, and, in some instances, pipework shell failures. As a result, numerous investigations were carried out. While the compressors were adequately designed from the aerodynamic performance point of view, they appeared to act as dynamic generators, producing excessive noise, pulsation, and vibration levels even when operated well within their design parameters. It was found that neither the designers nor equipment users had a clear understanding on how to practically analyse and mitigate such dynamic phenomena.

The objective of this study is to briefly explain possible sources of the observed problems in the hope that such explanation might provide a means for preventing or minimising noise and pulsation generation in centrifugal machines. The study is based on the author's experience in mitigating pulsation/noise and vibration problems mainly in the single stage natural gas centrifugal compressor systems. The study briefly describes differences in operation between vaned and vaneless diffuser compressors. It considers pipework and its influence on the compressor dynamic performance, and addresses some aspects of the compressor design in both aerodynamic and acoustic areas. Furthermore, it gives several practical methods to mitigate high frequency pulsation and vibration problems. Most of the approaches suggested here were implemented in the field and evaluated either by the author or others.

INTRODUCTION

To these days, many gas transmission system designers, some working for world renown consulting firms, do not consider it necessary to dynamically design compression facilities incorporating centrifugal compressors. They reason that such machines cannot cause unacceptable pulsation and vibration problems similar to those observed in reciprocating compressors. Hence, a standard opinion prevails that the use of centrifugal compressor eliminates acoustical vibration problems. How wrong this statement is, only the owners and operators of high flow gas transmission systems can testify.

Frequently, numerous operational problems and equipment failures begin to surface at a compression facility within 100 to 2000 hours of operation. They include failure of piping or compressor components, unscheduled outages as well as noise related environmental complaints (Kormann et al., 1994), (Marscher and D'Orsi, 1996), (Zhou and Motriuk, 1996), (Motriuk and Harvey, 1998). Such integrity problems are usually followed by solutions forced by government regulatory bodies or technical, safety, and economic requirements. Frequently, the compressor operation is restricted to comply with the noise regulations or to prevent structural failures, thus limiting its throughput and operation flexibility. On the other hand, costly modifications to the piping, such as installation of acoustic cladding, modification or addition of supports, as well as introduction of internal changes to the compressors take place to achieve expected reliability and performance. Such modifications are

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carried out by the facility owners and significantly increase the cost of operation and maintenance.

There are many examples of such situations in gas transmission companies around the world. The author is aware of several cases in which no adequate acoustical studies were carried out, hence, many integrity and operational problems could not be prevented. The observed cases included several cracked discharge pipe shells, see Figure 1a, and fatigued piping attachments which resulted in dangerous gas leaks into compressor buildings and in fire hazards. Other severe failures were also observed. They occurred in the impeller, see Figure 1b, in the compressor diffuser vanes, see Figure 1c, and in the inlet guide vanes (IGV's), see Figure 1d. It should be emphasised here that while the compressors were thought to be adequately designed, they appeared to act as dynamic generators producing excessive noise, pulsation, and vibration even when operated well within their design parameters.

RATIONALE FOR DYNAMIC ANALYSIS

The pulsation and noise generated by centrifugal compressor impellers could be a limiting factor in industrial application of these machines. A dynamic analysis of such compressors and the attached piping systems could lead to a significant reduction in noise, pulsation and vibration. Moreover, if applied at the design stage, it would result in the lowest cost of the design, construction and operation. There are numerous operational and case study examples, e.g. (Rogers, 1991), which recommend some form of dynamic analysis for all but the smallest compressor units (less than 200 HP). Such analyses would eliminate the need for after-start-up troubleshooting studies and system modifications.

Components of Noise and Pulsation Generated by Piping System

Noise and pressure oscillations in piping systems are caused by flow turbulence, increased turbulence due to discontinuities in flow path (flanges, thermowells, valves), internal acoustic field response, and transmission of mechanical vibrations into the piping through its walls. However, the most powerful sources of noise and pulsation in centrifugal compressor piping are close-end side branches. For certain flows, periodically shed vortices at the mouth of a closed branch tune themselves to the branch acoustical natural frequencies. When such a coincidence happens, the 'lock in' mechanism efficiently transfers mean flow energy into the resonating branch and is responsible for sustained vibration and noise (Jungowski et al., 1989). Also, high gas velocities at pressure reducing valve constrictions generate noise and pulsation which are predominantly broad band in nature and at frequencies from 0 to 1000 Hz (Carruci and Mueller, 1982). In these instances, higher order acoustic modes could dominate the structural response and the external noise radiation.

Components of Noise and Pulsation Generated by the Compressor

Two general components of noise and pulsation can be distinguished in a centrifugal compressor: i) a discrete frequency component occurring at blade passing frequency and its harmonics, and ii) a broadband component. During normal operation of a centrifugal compressor, discrete components originate from periodically fluctuating forces caused by interaction between rotating and stationary elements of the impeller blade and cutwater systems (Mofrey, 1964). At the same time, quasi periodic peaks may be generated by vortices shed from the trailing edges of the impeller blades. Simultaneously with the periodic forces, broad band noise and pulsation are formed by randomly fluctuating forces exerted on the impeller. These are caused by turbulent boundary layers, turbulence in the inlet flow, and by secondary flow cells in the impeller. All the above excitations might excite local acoustical natural frequencies of the compressor casing resulting in noise and vibration.

Recently, it was shown experimentally (Japikse, 1996) that broadband noise and pulsation in centrifugal machines are predominantly generated within impeller itself by partially stalled portions of the impeller blades. In centrifugal machines, flow separation over a large area of the impeller blade is difficult to avoid because of inherently complicated impeller geometry and its complex flow characteristics. Moreover, swirling flows are frequent whenever there is a combination of an axial and radial component of gas flow velocities. Hence, there are swirling flows present in compressors where the axial flow is collected and re-directed towards (or from) the main impeller axis, for example in radial inlet compressors. The swirling flow is susceptible to a vortex breakdown process which rapidly transforms a highly organised and undisturbed state of a swirl into a highly turbulent flow region (Halsey, 1986), thus generating large scale fluctuations at the suction or discharge of the compressor.

Centrifugal System Impedance and Its Influence on Pulsation/Noise Sources

It is well known that the power radiated by an acoustic source depends on impedance against which it operates. The power radiated in the open environment is drastically different from the power radiated into a duct or a pipe. At low frequencies, the ducted quadrupole acoustic source (U^8) is changed by the duct to the equivalent dipole source (U^6) radiating into a free field, (refer to any acoustic textbook for a definition of the quadropole and dipole acoustic sources). On the other hand, at high frequencies, the same ducted source is equivalent to a quadrupole source radiating into the free field. This is because the impedance of the duct/pipe at high frequencies changes into the characteristic impedance of the free field (Davis and Ffowcs, 1968). In his study (Mofrey, 1964) independently came to the same conclusion. He has shown that at low frequencies, the dipole sources (representing fluctuating momentum sources in the impeller blade raw) radiated sound (pulsation) which significantly depended on upstream and downstream impedances of the compressor volute and the attached to it pipework. Such a behaviour of acoustical systems was observed and studied by the author in many centrifugal compressor stations by means of measurement and dynamic analyses. Numerous predicted and some measured results have shown that the centrifugal compressor could 'amplify', 'attenuate' or be 'transparent' to the low frequency plane wave pulsations. For example, it was observed that a relatively low level vortex generated pulsation at side branch or constriction was amplified by the modelled centrifugal compressor to relatively high pulsation levels. When the centrifugal compressor was located in the acoustical system at a given pulsation resonant mode and near the point of low acoustic impedance, close to the vortex shedding frequency as depicted in Figure 2, then a weak acoustic resonance present in the system was significantly amplified. A gain of tens of kilopascals in pressure oscillations was possible, see Figure 3. Frequently, in cases where the reciprocating and centrifugal compressors were operating in parallel configurations, the already

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filtered reciprocating compressor established a weak standing acoustical wave but the centrifugal compressor highly amplified it. Pulsations grew to such levels that they could 'push' the centrifugal compressor into surge, although the unit was originally operating in the safe envelope, near its surge control line. For high frequencies, the influence of the attached piping on the compressor was significantly diminished, (Motriuk and Harvey, 1998). This is explained in (Mofrey, 1964).

MITIGATING PULSATION-NOISE SOURCES

It seems that counteracting all of the dynamic sources in centrifugal compressor systems is impossible. In fact, such a task can be successfully accomplished if co-operation between designers and acoustic engineers takes place. Then, the necessary actions could be undertaken to mitigate and optimise the pulsation, noise and vibration sources. These actions are categorised in the following groups: i) minimum acoustic design requirement, ii) proper operation of the installed equipment, and iii) design changes to the compressor internals.

Acoustic Design Requirement

The first type of pulsation-noise control can be implemented in centrifugal compressors which are thermodynamically and acoustically balanced, i.e. the acoustical aspect of the compression process for a given machine has been reviewed and optimised during the machine design process. This approach includes a relatively straightforward acoustic analysis of the centrifugal compressor station with the yard piping. Steady state operating conditions are considered and they include all possible flow configurations as per designed compressor performance and its operation philosophy (bypass flows, recycle flows, in parallel or series flow configurations, etc.). Quasi-steady operating conditions for the control valve, protecting compressor from surge, are considered as well. The analysis determines sources of noise, their strengths, and locations. Also, the system acoustic impedance curve is predicted and the compressor location on this curve is determined, Figure 2. At this stage, the acoustic system impedance curve can be changed with relative flexibility, for example, by moving piping elements such as side branches, orifice plates, bends or reducers. This step is followed by necessary acoustic modifications to eliminate all significant vortex shedding sources such as acoustically tuned closed end branches, noisy valves, and discontinuities in the flow path. Such actions can be accomplished by increasing or shortening the side branch lengths, by installing vortex spoilers, by installing valve mufflers (whisper trims) or by moving the sources away from the compressor. Full details of the calculation procedures are given in (Rogers, 1991). This type of noise and pulsation control takes care of low frequency problems.

At the same time, a three dimensional acoustic and structural response analysis of the piping immediate to the compressor is carried out. Several boundary conditions are considered to minimise the boundary condition errors. A set of different shell response frequencies are obtained and compared to three dimensional excitations calculated for open-open and open-close acoustical boundary conditions. If a match between the excitation and response frequencies is present, further analysis is performed assuming a conservative pressure pattern inside the pipe. Based on these calculations, a change of piping wall thickness is recommended and acoustic cladding specifications are made.

Then, all structures which could fail as a result of high cycle low stress fatigue are investigated. For example, the termowells for process and compressor temperature protections can be installed at the booster discharge nozzles with adequate protrusions (Zhou and Motriuk, 1996).

Proper Operation of Compression Facilities

Misuse of centrifugal machines frequently causes considerable noise, vibration, and pulsation problems in transmission systems as well as system failures. A common example of misuse of centrifugal machines is the operation of such compressors, with vaned diffusers, across their performance map. Since centrifugal compressors are frequently installed with diffuser vanes to increase their efficiency of compression process by two to four percent and expand the diffuser window of operation, most of the diffusers are the channel type (economics) and low solidity (range) diffusers. These diffuser designs should increase the compressor range of operations but, in fact, they do not fulfil their expected performance. Numerous laboratory and field tests (Petela and Motriuk, 2000) show that vaned diffusers are useful only near the design point and cause severe flow separations, efficiency losses and blockage at high gas flows, see Figure 4. For wide regimes of operations, the following adverse effects of vaned diffusers in centrifugal compressors were noted: i) The compressor operating range was narrowed, moving surge limit at every compressor speed, ii) Maximum efficiency was about 3 to 4% higher than in the vaneless diffuser configuration, however the region of efficient operation constituted only ~15% of the operating range and was close to the surge limit. In comparison, an efficient operation range for an impeller with vaneless diffuser extends for almost 50% of the operating range, iii) There was performance deterioration at high flow rates, which was primarily caused by a significant static pressure loss at the vaned diffuser channel entrance. Pressure recovery in the diffuser was not sufficient to balance this loss and, as a result, the pressure rise across the stage was smaller than the pressure rise across the impeller alone, Figure 5, iv) Surge was rapid and violent, in contrast to vaneless compressors which experience gradual transition through rotating stall, see Figure 6. It was observed that a removal of vanes from the diffuser stabilised vibration and noise. Since vaned diffusers perform to their specifications only near the machine design point, for wide flow regimes, the vanes should be removed. During such a modification, the diffuser should be 'pinched' and the forces acting on the rotor bearings should be analysed to preserve the rotor stability. If, at certain flow conditions, the bearings become overloaded, a partial rather than complete removal of diffuser vanes should be considered. The results of diffuser vane removal are illustrated in Figure 7, based on overall vibration-noise reduction. It should be noted that the surge line location and compressor performance characteristics are considerably modified when the parameters of aero assembly are changed, see Figure 8. For details of the performance map and efficiency changes refer to (Motriuk and Harvey, 1998).

When a compressor is operated near extreme flow conditions, or near surge (stall) or choke lines, there could be separations in the compressor volute, see Figure 9 and reference (Flathers and Bache, 1996). Consequently, the noise generation is considerably increased at these operating points. An example of a 'rumble' noise generation is shown in Figure 6 which depicts a three-cell rotation stall for a vaneless diffuser compressor operating near the surge line. A change in the angle of inlet guide vanes, which pre-swirl the compressor inlet

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flow, can effectively alter the compressor surge line location, and even eliminate the stall. However, making such changes must always include testing the surge protection system.

Direct and Indirect Intervention at the Source of Pulsation-Noise

The third type of pulsation-noise control focuses on two approaches. They are i) a direct intervention at the source of noise and pulsation by modifying the centrifugal compressor parameters, and ii) a passive disruption of the acoustic field by installing acoustic spoilers, mufflers, and by local pipe stiffening.

- **Direct Intervention**

Generally, high amplitudes of noise, pulsation and vibration observed in an adequately designed centrifugal compressor, operating within its design envelope, signify acoustical design problems. A permanent and significant reduction of noise can be achieved by using direct methods (Cumpsty, 1989) (Japikse, 1996). They include optimisation of several parameters in the aero-assembly. Since most of the noise is generated by the impeller exit, (Japikse, 1996), the inlet and outlet impeller velocity triangles have to be closely examined and optimised. The optimisation process has to depend on the compression train (i.e. jet, power turbine, gear box if any) capabilities. The interpretation of velocity triangles is of utmost importance, see Figure 10. The impeller quality can be adequately evaluated by the exit velocity triangle. This triangle is drawn according to the fundamental principle of vector addition. It comprises the relative velocity vector W_{exit}, the blade velocity or impeller speed U_{exit}, and the absolute velocity C_{exit}. The meridional velocity component C_{mexit} is governed by the conservation of mass relationship, and the absolute flow angle β depends on the relative flow angle and the impeller speed. The meridional vector divides U_{exit} into $C_{\theta exit}$ and $W_{\theta exit}$ components. These vectors have the following physical meaning: C_{mexit} represents flow rate and $C_{\theta exit}$ represents kinetic energy delivered to the system ('swirl velocity'). Large $C_{\theta exit}$ is undesirable for compressor applications since it generates swirl which significantly impairs compression process. Also, $C_{\theta exit}$ represents the amount of kinetic energy which should be minimised. Therefore, at the design stage, an optimisation of this vector (e.g. through a back sweep angle) is necessary. Consequently, the level of kinetic energy delivered by the impeller could be lowered and, as a result, much less noise/pulsation would be generated in the machine.

Another optimisation should include the impeller exit mixing region where the relative flow velocity vectors should be monotonous and uniform, i.e. any sharp velocity profiles such as square wave shapes must be avoided.

Other elements which require optimisation are described in literature Cumpsty (1989), (Japikse 1996), (Motriuk, 1996). They include the following: the clearance between impeller blades and cutwater, the angle between blades and cutwater, the cutwater slope and shape, the blade slope (rake), the diffuser vane rake, the impeller blade slots and the vortex devices.

- **Passive Disruption**

Further action can be taken to disrupt or attenuate an acoustic field by means of passive elements. These include an installation of acoustic spoilers, see Figures 11a and 11b, tuned to high frequency acoustical wave modes. The spoilers are inserted in the discharge end of the compressor and they destroy or disrupt high frequency pressure patterns by breaking them up. As a result, the patterns either decay exponentially or are divided into even higher frequency patterns. The spoiler paths introduce a wave phase difference, so that when the mismatched waves meet, they dissipate further. Performance of such spoilers is depicted in Figures 12a and 12b and described in detail in (Motriuk, 1996).

The noise generated by coincidence of internal higher order acoustic waves with resonant shell flexural modes can be attenuated by using a stiffening clamp. The clamp is designed to increase shell stiffness and provides additional damping, see Figure 13. Details can be found in (Motriuk, 1996).

There are also reactive and absorptive silencers which, when inserted into compressor flow paths, dissipate acoustic energy. The absorptive silencers have a form of perforated liners. This kind of silencers frequently generate an adverse noise called singing. The noise is caused by the vortex shedding from the flow over a perforated surface. Therefore, prior to an installation of a liner silencer, the following facts have to be considered: i) silencer location; the noise in centrifugal compressor is mostly produced at the exit of the impeller. It is then carried and intensified in the diffuser area. It may be further reinforced by the volute-collector and the compressor nozzle-'trumpet'. Hence, the compression and noise generation mechanism is very complex and cascade in nature, for example, any disturbance created at one element is transferred to another and influences its performance. ii) the system acoustics; the noise generation process involves three-dimensional acoustic propagation of pressure disturbance in the compressor volute. The noise strength depends on the response modes of both the volute and the attached piping.

Various perforation sizes of liner based silencers were extensively tested in the aero-industry throughout the years (Mangiarotty, et al., 1968) (Mayer et al., 1958). The tests showed that liners frequently generated 'singing' noise. However, while it was possible to counteract amplitude of the 'singing' noise in aero-industry applications by reducing the liner hole diameters, such an approach might not be possible in natural gas applications due to gas impurities which may clog the perforations. Clogging can make such a device ineffective.

CONCLUSIONS

1) Pulsation, vibration and noise are directly or indirectly generated in the compressors and their pipework. The levels depend mainly on the system's acoustics, piping layout and geometry, compressor design details/internals, and gas flow parameters.

2) Successful control or elimination of pulsation, vibration and noise in centrifugal compressor facilities should incorporate assessment of the system acoustic impedance, proper location of the compressor on the impedance curve, and elimination or control of the acoustic sources. Such a control is achievable through performing appropriate dynamic analyses which include interaction of the compressor with its pipework.

3) Design capabilities of a vaned or vaneless centrifugal compressor and its operating range should be described by the designer/vendor and well understood by the operators. Improper operation of the compressor causes noise, pulsation, vibration which lead to loss of performance and frequent failures.

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4) During the design or troubleshooting processes, an option to use attenuation devices such as local spoilers, clamp-dynamic absorbers, compressor interior silencers, and acoustic cladding should be considered. The attenuators installed in the flow paths should not significantly limit the efficiency of the compression process.

5) Any changes to the compressor aero-assembly could influence the compressor pulsation, vibration and noise performance. For example, removal of the diffuser vanes stabilises vibration and noise but it can change the surge line location at every speed of the rotor or cause uneven static pressure distribution in the compressor volute which, in turn, can result in unacceptable side loads on the rotor bearings.

5) Optimisation of several parameters in the aero-assembly is critical to pulsation, vibration and noise generation. The interpretation of the impeller exit velocity triangles is of utmost importance because by reading them correctly any impeller can be evaluated and its design changed to minimise the noise generation.

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REFERENCES
Cumpsty, N.A., 1989, "Compressor Aerodynamics, John and Wiley &Y Sons, Inc., New York

Carruci, V.A., and Mueller, R.T., 1982, Acoustically Induced Piping Vibration in High Capacity Pressure Reducing Systems", ASME Paper No. 82-WA/PVP-8.

Davies, H.G., and Ffowcs,W.J.E., 1968, "Aerodynamic Sound Generation in a Pipe", Journal of Fluid Mechanics, 32, pp. 765-778.

Flathers, M.B., and Bache, G.E, 1996," Aerodynamically Induced Radial Forces in a Centrifugal Gas Compressor - Part 2: Computational Investigation", International Gas Turbine and Aeroengine Congress and Exhibition, Binmingham, UK, June 10-13

Halsey, D.M., 1986 "The character of Swirl in Turbulent Pipe Flow with Reference to Its Effect on Flowmeters", Cranfield Institute of Technology, Ph.D. Thesis, U.K.

Japikse, D., 1996, Centrifugal Compressor Design and Performance", Concepts ETI, Inc., Vermont, USA.

Jungowski, W.M. et al., 1989, Cylindrical Side-Branch as Tone Generator', Journal of Sound and Vibration, 131(2), pp. 265-285.

Korman, J.P. et al., 1994,' Thermowell Design Review" NGTL Internal Report, Calgary, Alberta, Canada

Lyon, R.H., !975, "Statistical Energy Analysis of Dynamic Systems: Theory and Applications" MIT Press.

Mangiarotty, R.A., et al., 1968,"Duct-Lining Materials and Concepts", NASA SP-189, pp. 29-52.

Marscher, W. and D'Orsi, N., 1996, Technical Memorandum No. 484 "Aero/Pulsation/Vibration Testing of a Compressor at Vetchland Booster Pipeline Station", CONCEPTS ETI, Vermont.

Meyer, E., et al., 1958, "Experiments on the Influence of Flow on Sound Attenuation in Absorbing Ducts", Journal of the Acoustical Society of America, 30, pp. 165-174

Mofrey, C.L., 1964, "Rotating Pressure Patterns in Ducts: Their Generation and Transmission' Journal of Sound and Vibration (1964) 1, pp. 60-87.

Motriuk, R.W., 1996, "Verification of Two Methods to Mitigate High Frequency Pipe Shell Vibration", ASME Flow Induced Vibration Conference, PVP FIV Volume 328, pp. 405-413

Motriuk, R.W., and Harvey, P.D., 1998, "Centrifugal Compressor Modifications and Their Effect on High Frequency Pipe Wall Vibration", Proceedings of the 1997 ASME PVP Conference, Orlando, Florida, USA, Vol. 355, pp. 21-29 (re-print in Journal of Pressure Vessel Technology, August 1988, Vol., pp. 276-282.

Petela, G., and Motriuk, R.W., 2000, "Effect of Vaned Diffuser on Performance and Onset of Pressure Instabilities in Centrifugal Compressors", Proceedings of the 7th FIV 2000 International Conference in Lucerne, Switzerland, Flow Induced Vibration, pp.733-742.

Rogers, L.E., 1991, "Design Stage Acoustic Analysis of Natural Gas Piping Systems in Centrifugal Compressor Stations", ASME Gas Turbine and Aeroengine Congress and Exposition (92-GT-238), Orlando, June 3-6.

Zhou, Z.J., and Motriuk R.W., 1996, "Influence of Tapered Thermowell Length on Temperature Measurement", Proceedings of the 1996 ASME PVP Conference, Montreal, Canada, Vol. 333, pp. 97-104.

FIGURES

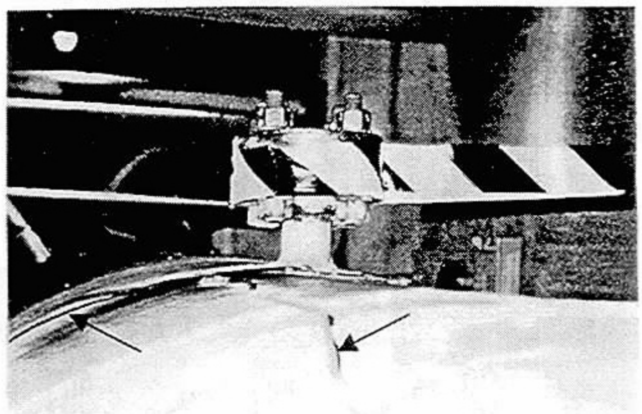

Figure 1a: Example of the Failed Compressor Discharge Pipe

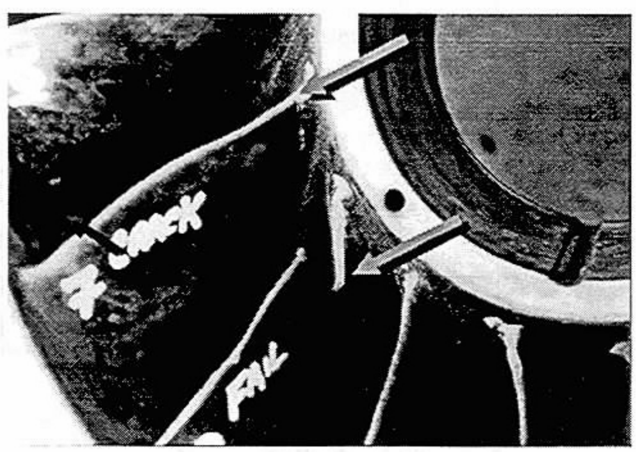

Figure 1b: Example of Severe Impeller Failure

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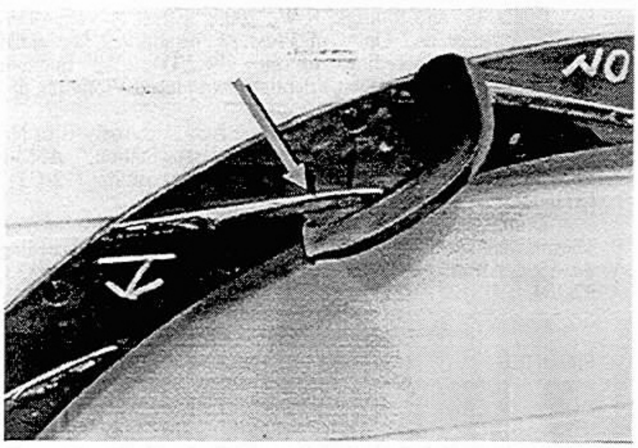

Figure 1c: Example of Diffuser Vane Failure

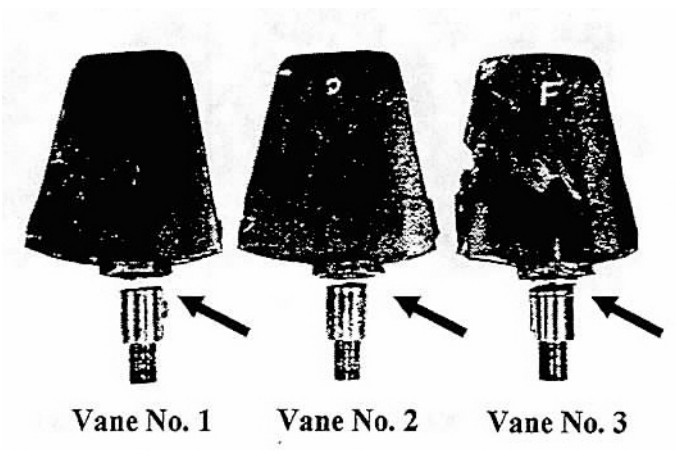

Figure 1d: Example of Inlet Guide Vane Failure

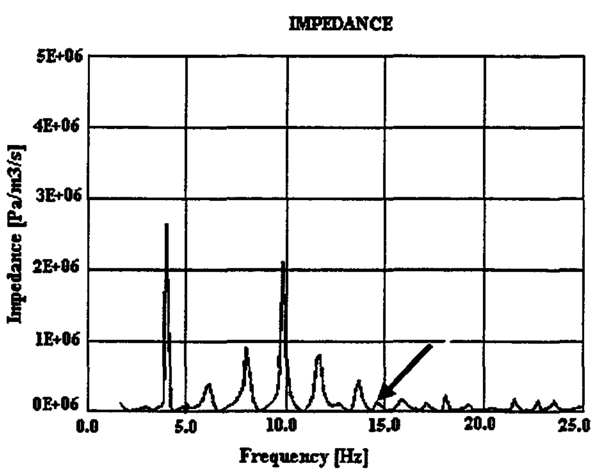

Figure 2: Acoustic System Impedance Curve

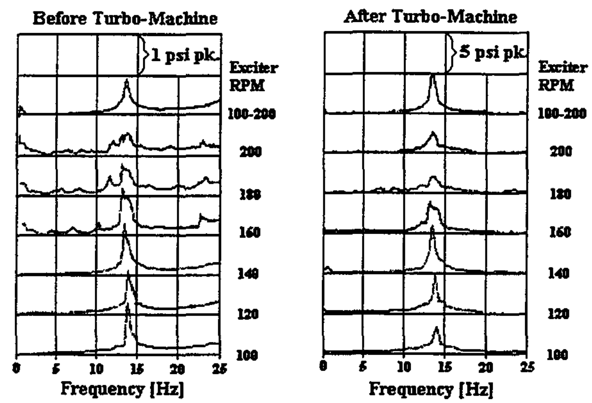

Figure 3: Amplification of System Resonance by Centrifugal Machine (Measurement)

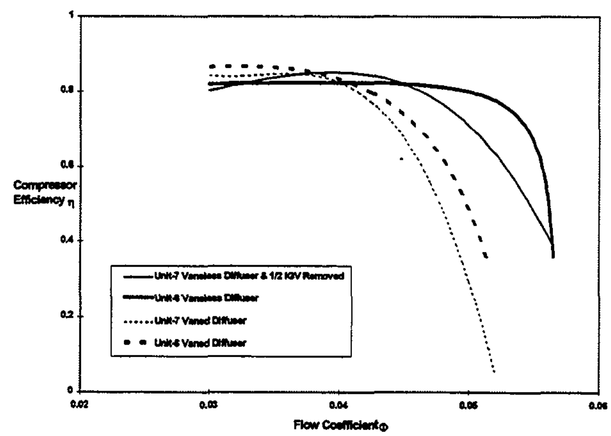

Figure 4: Change in Compressor Efficiency after Diffuser Vane Removal

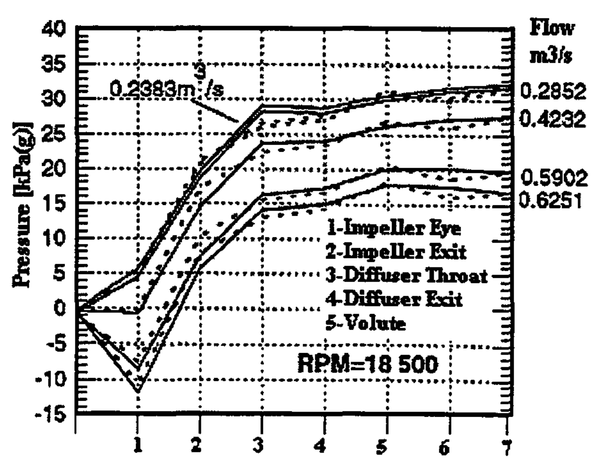

Figure 5a: Rise of Pressure across Compressor (Vaneless Diffuser)

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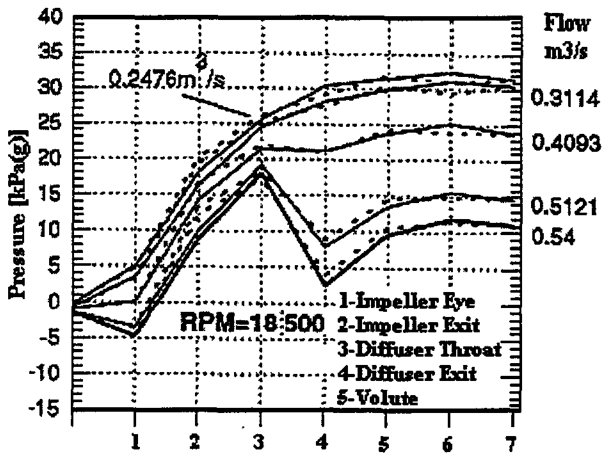

Figure 5a: Rise of Pressure across Compressor (Vaned Diffuser)

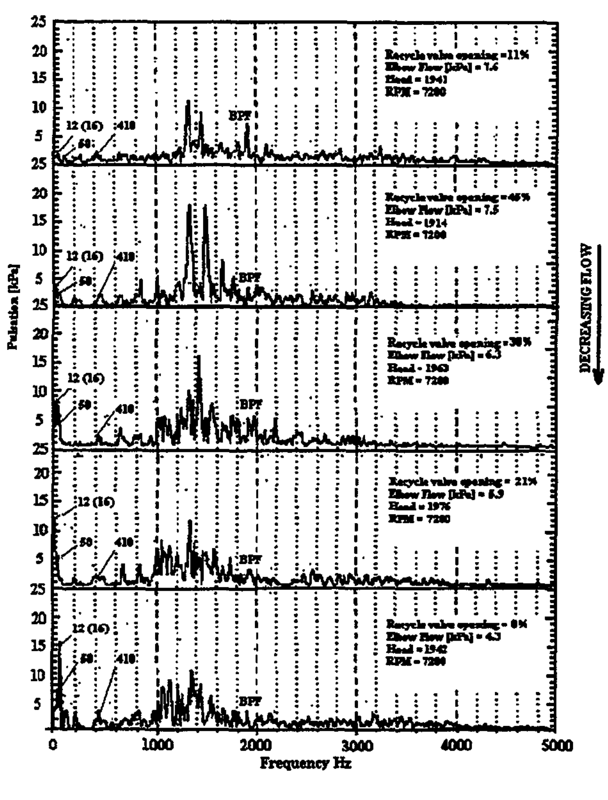

Figure 6: Example of Rotating Stall in the Compressor without Diffuser Vanes

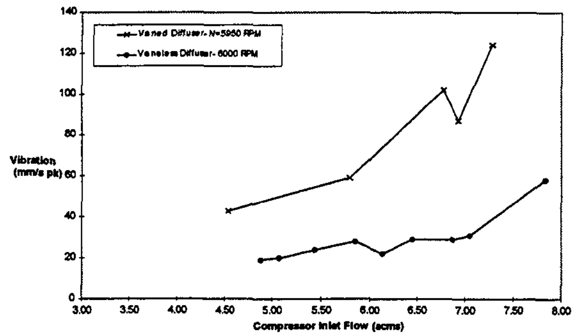

Figure 7: Effect of Removing Diffuser Vanes on Vibration (Noise) Levels

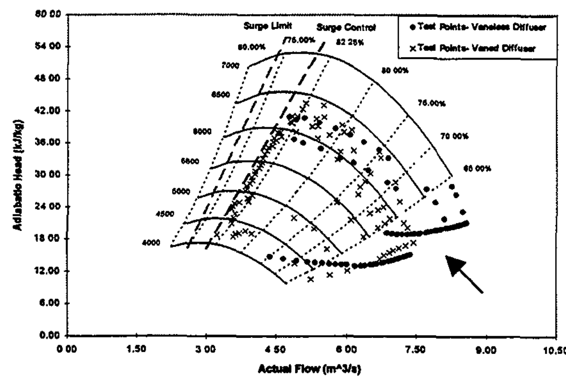

Figure 8: Effect of Removing Diffuser Vanes on Compressor Performance Map

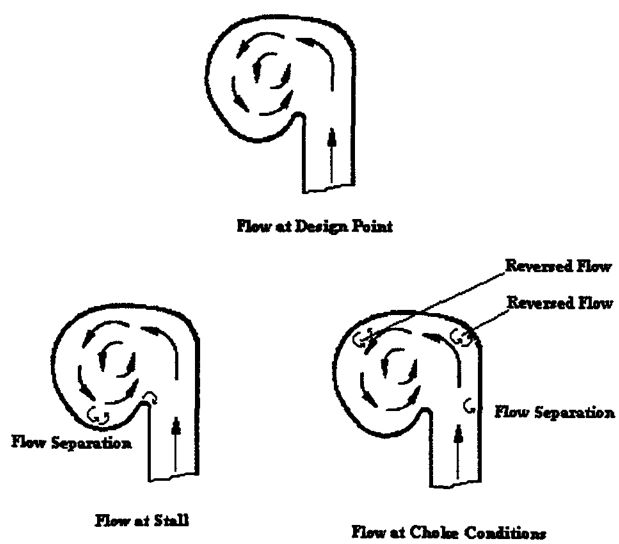

Figure 9: Flow Separations and Secondary Flows for Compressor Operating at the Design Point, Stall and at the Choke Limit

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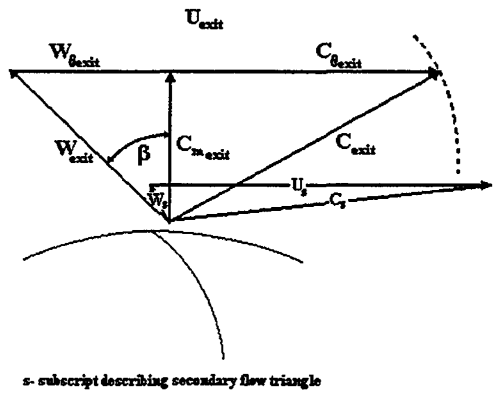

Figure 10: Impeller Exit Conditions Prior to Mixing -Velocity Vector Representation

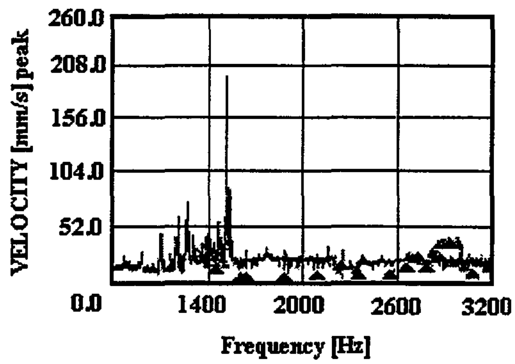

Figure 12a: Performance of Spoiler Type 1 (black triangles)

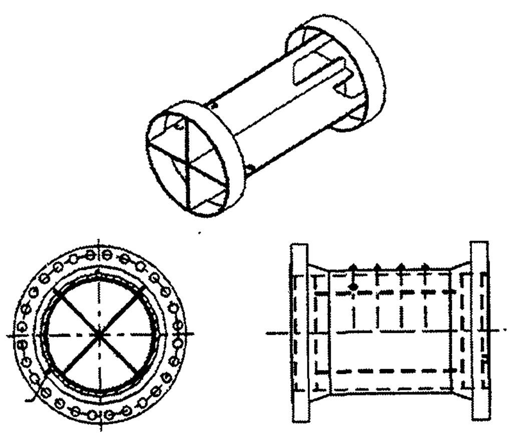

Figure 11 a: Acoustical Spoiler - Type 1

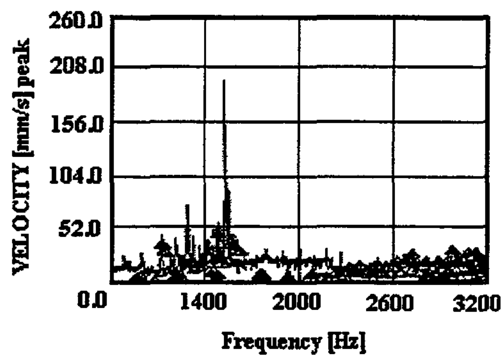

Figure 12b: Performance of Spoiler Type 2 (black triangles)

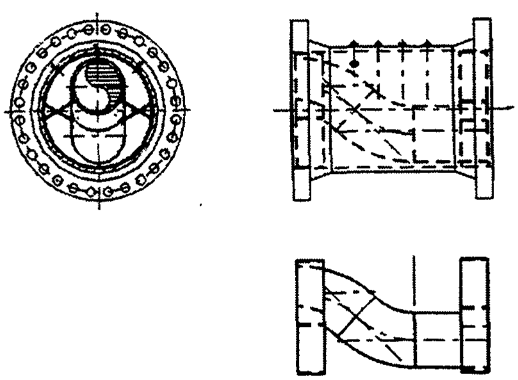

Figure 11 b: Acoustical Spoiler - Type 2

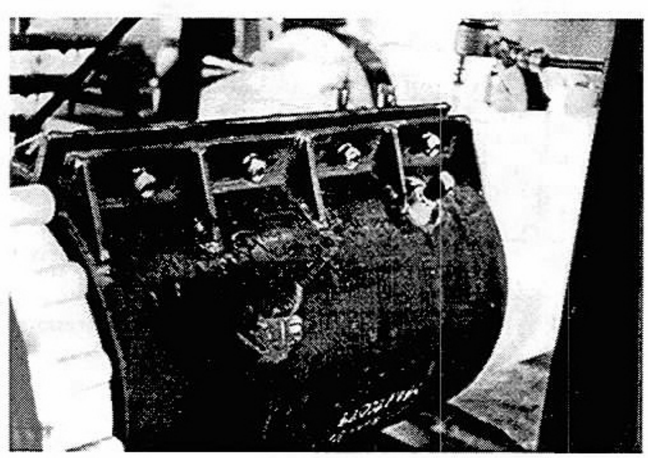

Figure 13: Stiffening Clamp

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