THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47th St., New York, N.Y. 10017

98-GT-68

MULTISTAGE CENTRIFUGAL COMPRESSOR SURGE ANALYSIS

PART I: EXPERIMENTAL INVESTIGATION

G.L. Arnulfi
University of Udine
Dipartimento di Energetica e Macchine
Udine, Italy

P. Giannattasio
University of Udine
Dipartimento di Energetica e Macchine
Udine, Italy

C. Giusto
University of Genova
Istituto di Macchine e Sistemi Energetici
Genova, Italy

A.F. Massardo
University of Genova
Istituto di Macchine e Sistemi Energetici
Genova, Italy

D. Micheli
University of Trieste
Dipartimento di Energetica
Trieste, Italy

P. Pinamonti
University of Udine
Dipartimento di Energetica e Macchine
Udine, Italy

ABSTRACT

This paper reports an experimental investigation on centrifugal compressor surge. The compression system consists of a four-stage blower with vaned diffusers and a large plenum discharging into the atmosphere through a throttle valve. Measurements of unsteady pressure and flow rate in the plant, and of instantaneous velocity in the diffusers of the first and fourth compressor stage are performed during deep surge, at several valve settings and three different rotation speeds. Additional tests have been carried out on a different system configuration, i.e., without plenum, in order to obtain the steady-state compressor characteristics and to collect reference data on stall in surge-free conditions. In this configuration, a fully developed rotating stall was detected in the compressor diffusers, while during surge it affects only a limited part of the surge cycle.

The goal of the present experimental work was to get a deeper insight into unstable operating conditions of multi-stage centrifugal compressors and to validate a theoretical model of the system instability to be used for the design of dynamic control systems.

NOMENCLATURE

A_i	inlet compressor area
C_1	flow velocity in the 1st stage diffuser
C_4	flow velocity in the 4th stage diffuser
f	frequency
f_H	Helmoltz frequency
$\dot{m}$	mass flow rate
p_0	ambient pressure
p_c	compressor delivery pressure
p_p	plenum pressure
t	time
T_H	Helmoltz period
U	impeller tip velocity
VAR	Valve Area Ratio
$\varphi = \dot{m}/\rho_0 U A_i$	flow coefficient
$\psi_c = 2(p_c - p_0)/\rho_0 U^2$	compressor pressure coefficient
$\psi_p = 2(p_p - p_0)/\rho_0 U^2$	plenum pressure coefficient
ρ_0	air density at ambient conditions

INTRODUCTION

Recently, much effort has been devoted to improving the performance of centrifugal compressors by using both numerical techniques and experimental investigations. Although higher efficiencies were obtained with improved aerodynamic design, the working range of the compressor, where the compression system operates in stable conditions, remains narrow. The surge inception line is commonly adopted as the limit of the stable operating range.

Surge is a dynamic phenomenon consisting of large amplitude low-frequency oscillations of flow rate. It can occur in compression systems, involving at least a compressor, a resistance and a volume of sufficiently high capacity, when the flow rate is lower than the design value. Operation during surge is unacceptable for industrial applications, since it can cause serious damage to the plant.

The existence of two different kinds of surge is well known:

- a slight oscillation without any net back-flow, with frequency close to the Helmoltz resonance of the system, i.e., "mild surge";
- a strong oscillation including complete breakdown and reverse flow through the compressor, with frequency substantially lower than the Helmoltz frequency, i.e. "deep surge".

Mild surge may occur in systems where the compressor operates near its instability limit, and it is recognised as a trigger of the deep surge inception. The deep surge is a damaging condition that must be

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Stockholm, Sweden — June 2–June 5, 1998
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1998

avoided. Therefore, a deep understanding of the phenomenon is required to minimize its effects and to design proper control systems. In particular, experimental investigations on pressure and velocity fields and on flow-rate oscillations in the compression system are needed to attain a better understanding of flow instability and to validate theoretical models.

As far as industrial centrifugal compressors are concerned, several experimental works were carried out to investigate surge (e.g., Toyama et al., 1977, Abdelhamid et al., 1979, Hansen et al., 1981) and, more recently, to study the interaction between surge and rotating stall (Ribi and Gyarmathy, 1993, Jin et al., 1994). Furthermore, experiments were performed to investigate the influence of the impeller and diffuser geometry on system instability (Jin et al., 1994, Cyrus et al., 1995, Ribi and Gyarmathy, 1995). These works are concerned with high pressure single-stage centrifugal compressors, with tip impeller diameter between 100 and 400 mm and rotation speed between 10000 and 50000 rpm. Usually, tests were carried out during surge inception, detecting mild surge and limiting the observation of deep surge to few consecutive cycles in order to avoid damages.

The present research considers a compression system based on a low pressure multi-stage centrifugal compressor, suitable for operation in conditions of completely developed deep surge. This compressor offers the possibility of varying its configuration, so allowing the influence of the geometry on surge features to be investigated. Detailed measurements of pressure, flow rate and instantaneous velocities in the compressor diffusers have been performed in order to understand the instability behaviour in more details and to collect experimental data to be used for validating the theoretical model of the compression system described in the part II of the present work (Arnulfi et al. 1998).

EXPERIMENTAL PROCEDURE

Test Rig

Experiments have been carried out at the Laboratory of the Dipartimento di Energetica - University of Trieste on a modular centrifugal blower plant (Fig. 1). Different configurations of this multi-stage compressor were investigated in previous works, by performing flow velocity measurements both in steady conditions (Arnulfi et al., 1995) and during rotating stall (Arnulfi et al., 1996).

In the present investigation, a blower configuration with four stages and vaned diffusers has been considered. All the stages have the same geometry; the shrouded impellers have 16 backswept blades (35° inlet angle and 67° outlet angle, both referred to the tangential direction, 160 mm inlet diameter, 465 mm outlet diameter, 23 mm inlet width, 8 mm outlet width); the vaned diffusers have parallel straight walls (470 mm inlet diameter, 570 mm outlet diameter, 10 mm axial width) and 27 straight blades (18° inlet angle and 40° outlet angle); the vaned return channels have 8 curved blades.

The compressor is driven by a DC motor connected to a speed increasing gear. The blower operates in an open circuit: air enters a radial bellmouth inlet duct, flows through the blower, is fed to a cylindrical plenum by a connecting pipe and is discharged into the atmosphere through a motorized throttle valve.

The plenum volume is 3.132 m^3, which is large enough to allow surge to arise with different configurations of the blower, namely, when varying the number of stages and the diffuser type. With the present blower configuratimn, deep surge occurs at stall inception practically at all rotor speeds.

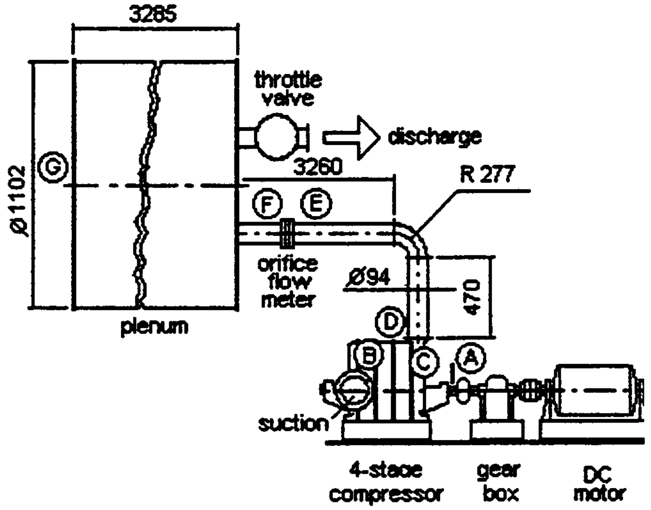


Fig. 1 - Test rig.

Measuring Technique

Measurements were performed to determine both steady-state compressor characteristics and pressure and flow rate oscillations in the plant during surge. Furthermore, the instantaneous flow velocity was measured at the outlet of the first and last stage diffusers, in order to investigate the interaction between surge and rotating stall.

The instrumentation system is shown in Fig. 2, where the capital letters refer to the transducers location of Fig. 1. Pressures were measured by variable inductance transducers, temperature by type K-thermocouples, rotational speed by a magnetic pick-up.

The plenum pressure was measured by means of a single transducer mounted on the bottom lid, since the pressure can be considered as uniform throughout the volume at any time instant. It is due to plenum dimensions which are much smaller than the wavelength of an acoustic wave having a frequency on the order of that typically associated with surge.

The mass flow rate was measured by an orifice flow meter mounted in the pipe; such a kind of measurement technique was preferred to the one proposed for unsteady flow measurements by Greitzer (1976) because the present plant allows the orifice flow meter to be properly located between compressor and plenum, while reducing the number of high accuracy measurements to be made. On the other hand, the use of an instrument suited for steady flow measurements was considered acceptable because of the slow dynamics of surge (few hertz).

Velocity measurements were performed by means of two constant temperature hot-wire Dantec anemometers, with single straight miniature probes (type 55P11, with tungsten wire of 5 µm diameter and 1.25 mm length). A stationary hot-wire technique was used to measure the intensity of the velocity vector in order to detect stall disturbances; the probes were located radially 8 mm upstream the exit of the diffusers, in the middle of the blade-to-blade channel. Both probes had their wire oriented parallel to the rotor axis, i.e., normal to absolute velocity, the axial velocity being negligible due to the geometry of the diffusers.

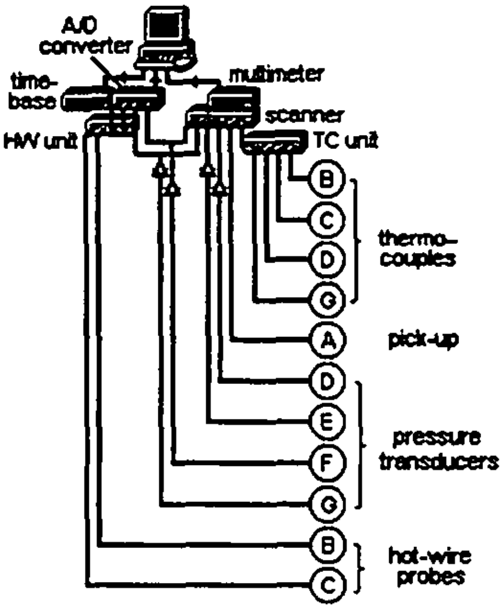


Fig. 2 - Schematic of the Instrumentation system.

The voltage signals of velocity, flow rate and plenum pressure were sent to an A/D converter board, and the trigger signal was provided by a plug-in time-base card. Measurements were performed at a sampling rate of 9 readings/revolution for almost 300 revolutions to detect stall and 4 readings/revolution for over 1000 revolutions to detect surge, which values appeared to provide a resolution suited to the frequency of the phenomena under investigation. As the other measurements are concerned, only average values were stored.

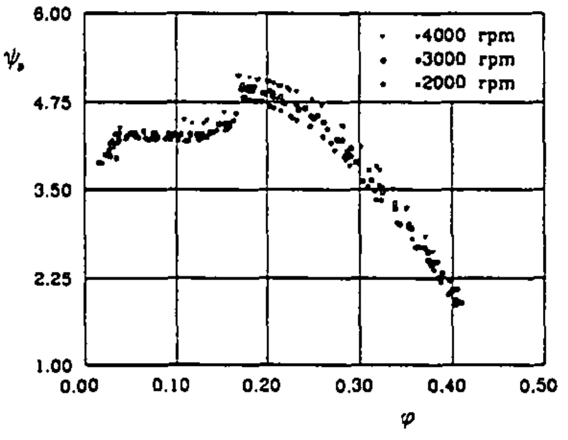


Fig. 3 - Steady-state compressor characteristics.

Operating Conditions

The tests were carried out at three compressor speeds, i.e., 2000, 3000 and 4000 rpm, and at different settings of the throttle valve which were kept fixed during each test. Two different plant configurations, i.e., without and with plenum, were tested. In the former, the throttle valve was directly mounted at the exit of the compressor outlet pipe. Such a configuration was used to obtain the complete steady-state characteristics of the blower, surge being absent also at very low mass flow rates, and to measure flow velocity in the diffusers in condition of fully developed rotating stall. On the contrary, the plant configuration including the plenum was employed to characterise the compression system operation during surge.

EXPERIMENTAL RESULTS

Tests without plenum

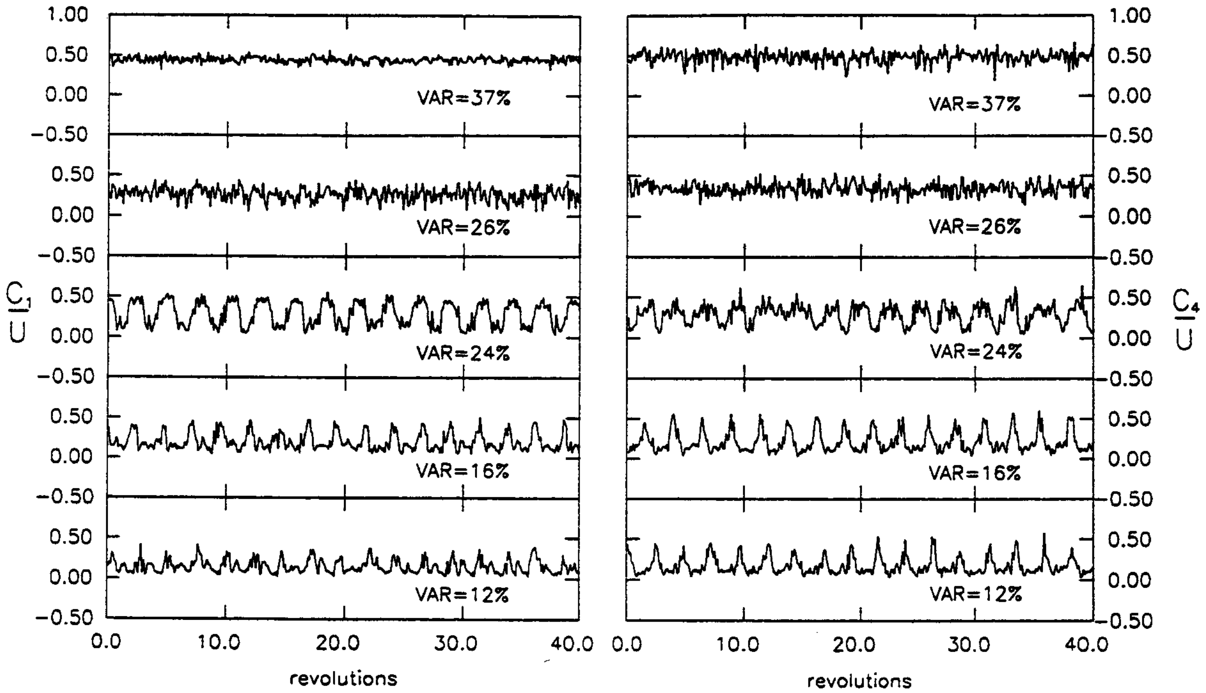
The blower steady-state characteristics are shown in Fig. 3. The steep drop in pressure coefficient suggests an abrupt stall inception at about $\varphi = 0.17$. The characteristic curves were obtained by means of a double sweep of the mass flow rate, both opening and closing the valve, in order to detect stall hysteresis, if any. An accurate evaluation near the stall limit showed no evidence of such a phenomenon.

Hereafter, the results of the hot-wire measurements are reported with reference to significant flow rate and compressor speed conditions. The different valve settings are expressed as Valve Area Ratio (VAR), i.e., the ratio of actual to maximum valve flow area.

Figure 4 shows the flow velocity in both diffusers during 40 consecutive revolutions, at 3000 rpm and different valve settings. The corresponding spectra, obtained by means of a FFT analysis, are plotted in Fig. 5 as a function of the ratio between the signal frequency and the impeller rotation frequency. Rotating stall is observed in both diffusers when VAR is 24% or lower. At this valve setting, the velocity signals in both diffusers appear to be quite sinusoidal, while new harmonics arise as the valve is closed. Figure 5 shows a stall frequency of about 20 Hz, which is 40% of the impeller rotation frequency. The same percentage value was found at the other compressor speeds. The present measurement technique, which employs a single probe, does not allow to detect the number of stall cells, but previous tests on the same blower with different number of stages or vaneless diffusers always showed a single stall cell (Arnulfi et al., 1996).

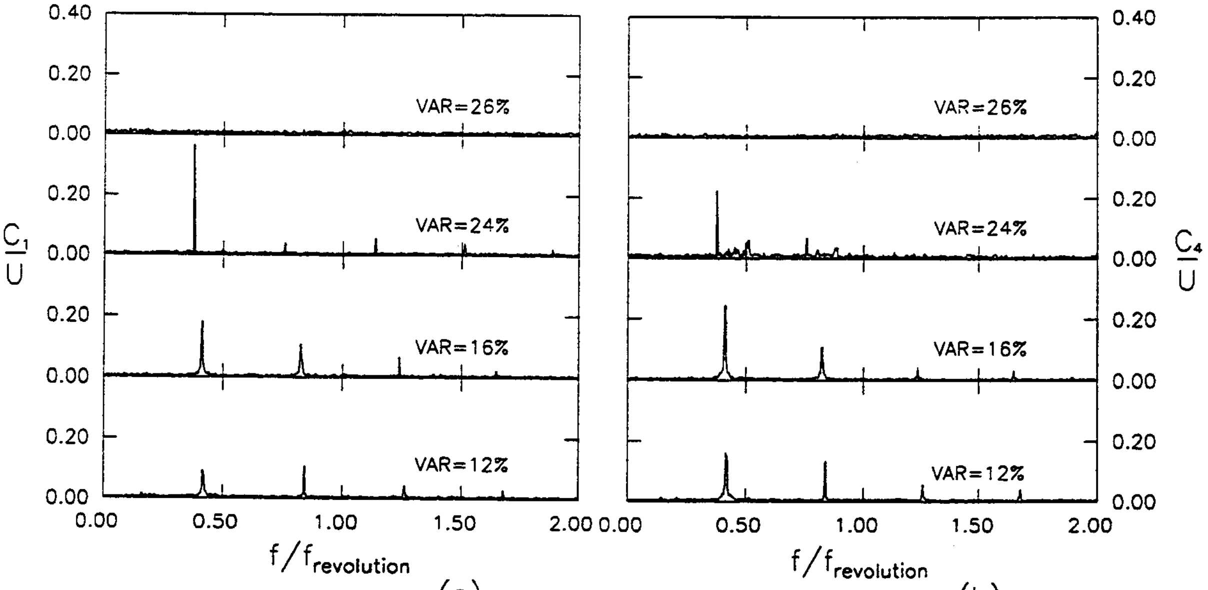
At the highest rotational speed (4000 rpm) and low VAR values, surge was detected even in this plant configuration. Figures 6 and 7 show the time histories of velocity and flow coefficient together with velocity spectra at two different valve settings (VAR=15% in Fig. 6 and VAR=13% in Fig. 7). At VAR=15% an oscillation at the stall frequency is observed in both velocity signal and flow coefficient trace, which latter shows also limited regions of reverse flow. At VAR=13%, the oscillation frequency has fallen to 16% of the impeller rotation frequency and reverse flow is much more pronounced, so denoting the occurrence of surge instability.

It has to be pointed out that the present hot-wire technique allows only the magnitude of the flow velocity to be measured, not its direction. However, when a reverse flow occurs in the discharge pipe it occurs also in the vaned diffusers, so that the sign of the velocity was changed according to the sign of the flow rate signal.



(a) (b)

Fig. 4 - Flow velocities in the diffusers of the first (a) and fourth (b) compressor stage at different valve settings and 3000 rpm.



(a) (b)

Fig. 5 - Flow velocity spectra in the diffusers of the first (a) and fourth (b) compressor stage at different valve settings and 3000 rpm.

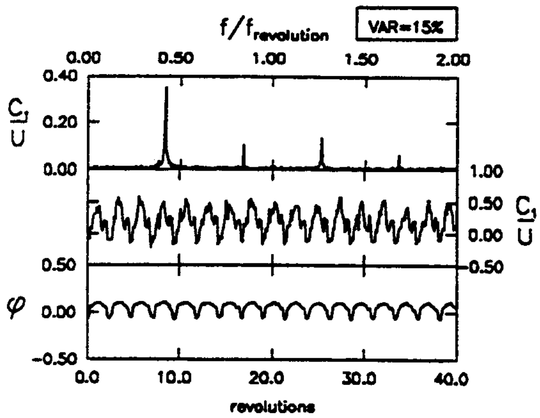


Fig. 6 - Velocity spectrum and time history of velocity and flow coefficient in the diffusers of the first compressor stage, at 4000 rpm and VAR=15%.

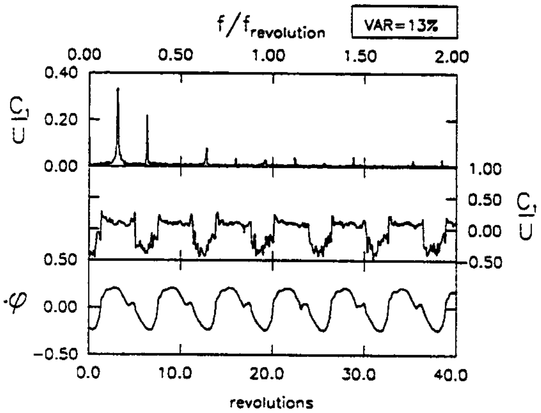


Fig. 7 - Velocity spectrum and time history of velocity and flow coefficient in the diffusers of the first compressor stage, at 4000 rpm and VAR=13%.

Tests with plenum

By connecting the plenum to the discharge pipe, conditions of deep surge were observed at all the compressor speeds.

In Fig. 8 the surge cycles observed at 2000 rpm are shown for several valve settings in terms of plenum pressure coefficient ψ_p vs. flow coefficient φ. In order to provide a clearer representation of the surge cycles, the signals have been filtered by cutting off frequencies higher than 30 Hz. In the same figure, the steady-state compressor characteristic referred to the plenum pressure is plotted for reference.

Figure 9 shows the time evolution of flow coefficient, plenum pressure coefficient and flow velocities in the first and fourth diffuser at the same conditions of Fig. 8. In these plots, time is non-dimensionlized by using the Helmoltz period of the compression system. Surge oscillations are observed when VAR $\leq$ 26%, but at VAR values of 26% and 24% the phenomenon is not periodic, any reverse flow is absent and the instability oscillations of the flow velocities are very small when compared with the flow rate and pressure ones. As it is demonstrated in Part II (Arnulfi et al., 1998), such conditions correspond to virtually stable operating points, the equilibrium steady-state flow coefficients being slightly greater than the stall limit value. However, a small perturbation is sufficient to cause a pronounced, but temporary, system instability. Such a situation occurs randomly and results in a non-periodic behaviour, rather than in a limit cycle. A similar phenomenon has been described by Lin (1996). When the valve is closed further on (VAR$\leq$20%), the system behaviour becomes periodic, resulting in true surge cycles, and the velocity signals appear modulated at the same frequency of the flow rate oscillations. The limit cycles are typical of deep surge, since they exhibit two slow phases, which develop along both stable and negative branches of the compressor characteristic, and two much faster phases during which large changes in flow rate are observed at almost constant pressure. As the valve is closed, surge cycles shift to higher pressures; at the same time the duration of the positive flow phase reduces while the length of the negative flow one increases. In fact, when the valve flow area is reduced, the storage of elastic potential energy into the plenum becomes faster, while the subsequent blow-down process mainly occurs as a reverse flow through the compressor rather than a positive flow through the valve. It results in an increased flow resistance when the plenum is emptying and, consequently, in a higher mean value of the cycle pressure.

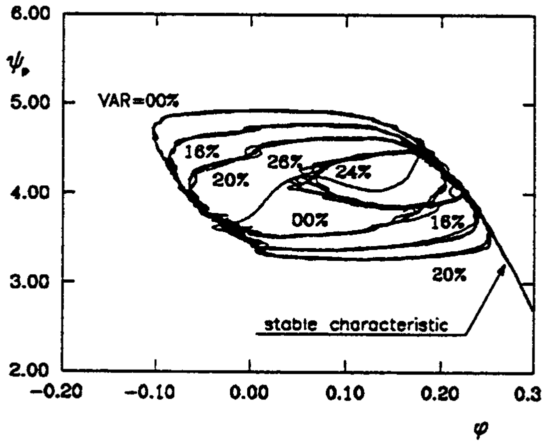


Fig. 8 - Surge cycles at different valve settings and 2000 rpm.

At constant speed, the frequency of surge oscillations varies slightly with valve opening, a maximum value being reached at an intermediate throttle setting between surge inception and valve shut-off. Figure 10 shows the results obtained at 2000 rpm by means of a FFT analysis of the periodic signals reported in Fig. 9. A maximum surge frequency of 0.75 Hz, corresponding to 80% of the Helmoltz frequency, occurs at VAR=16%, while it falls to about 0.70 Hz (74% of the Helmoltz frequency) when VAR is 0% or 30%. A similar behaviour has been observed at higher compressor speeds. Figure 10 also shows that the flow rate, pressure and velocity signals concerning the same valve setting contain the same number of harmonics of significant amplitude.

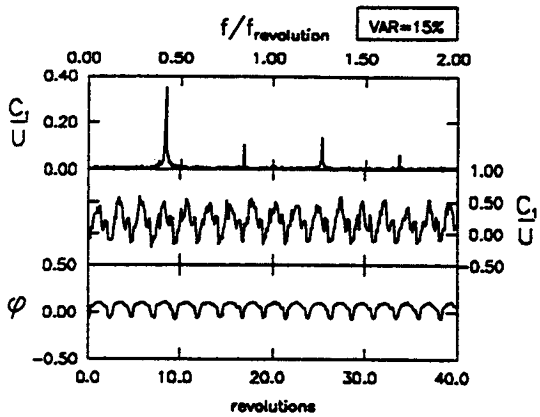


Fig. 6 - Velocity spectrum and time history of velocity and flow coefficient in the diffusers of the first compressor stage, at 4000 rpm and VAR=15%.

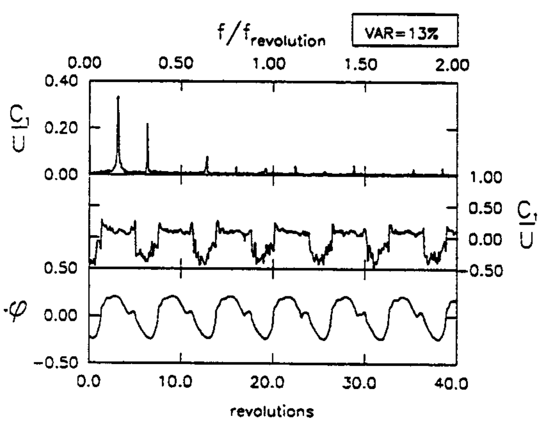


Fig. 7 - Velocity spectrum and time history of velocity and flow coefficient in the diffusers of the first compressor stage, at 4000 rpm and VAR=13%.

Tests with plenum

By connecting the plenum to the discharge pipe, conditions of deep surge were observed at all the compressor speeds.

In Fig. 8 the surge cycles observed at 2000 rpm are shown for several valve settings in terms of plenum pressure coefficient ψ_p vs. flow coefficient φ. In order to provide a clearer representation of the surge cycles, the signals have been filtered by cutting off frequencies higher than 30 Hz. In the same figure, the steady-state compressor characteristic referred to the plenum pressure is plotted for reference.

Figure 9 shows the time evolution of flow coefficient, plenum pressure coefficient and flow velocities in the first and fourth diffuser at the same conditions of Fig. 8. In these plots, time is non-dimensionlized by using the Helmoltz period of the compression system. Surge oscillations are observed when VAR≤26%, but at VAR values of 26% and 24% the phenomenon is not periodic, any reverse flow is absent and the instability oscillations of the flow velocities are very small when compared with the flow rate and pressure ones. As it is demonstrated in Part II (Arnulfi et al., 1998), such conditions correspond to virtually stable operating points, the equilibrium steady-state flow coefficients being slightly greater than the stall limit value. However, a small perturbation is sufficient to cause a pronounced, but temporary, system instability. Such a situation occurs randomly and results in a non-periodic behaviour, rather than in a limit cycle. A similar phenomenon has been described by Lin (1996). When the valve is closed further on (VAR≤20%), the system behaviour becomes periodic, resulting in true surge cycles, and the velocity signals appear modulated at the same frequency of the flow rate oscillations. The limit cycles are typical of deep surge, since they exhibit two slow phases, which develop along both stable and negative branches of the compressor characteristic, and two much faster phases during which large changes in flow rate are observed at almost constant pressure. As the valve is closed, surge cycles shift to higher pressures; at the same time the duration of the positive flow phase reduces while the length of the negative flow one increases. In fact, when the valve flow area is reduced, the storage of elastic potential energy into the plenum becomes faster, while the subsequent blow-down process mainly occurs as a reverse flow through the compressor rather than a positive flow through the valve. It results in an increased flow resistance when the plenum is emptying and, consequently, in a higher mean value of the cycle pressure.

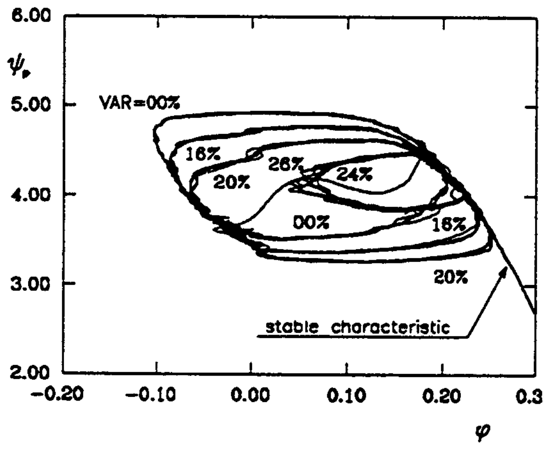


Fig. 8 - Surge cycles at different valve settings and 2000 rpm.

At constant speed, the frequency of surge oscillations varies slightly with valve opening, a maximum value being reached at an intermediate throttle setting between surge inception and valve shut-off. Figure 10 shows the results obtained at 2000 rpm by means of a FFT analysis of the periodic signals reported in Fig. 9. A maximum surge frequency of 0.75 Hz, corresponding to 80% of the Helmoltz frequency, occurs at VAR=16%, while it falls to about 0.70 Hz (74% of the Helmoltz frequency) when VAR is 0% or 30%. A similar behaviour has been observed at higher compressor speeds. Figure 10 also shows that the flow rate, pressure and velocity signals concerning the same valve setting contain the same number of harmonics of significant amplitude.

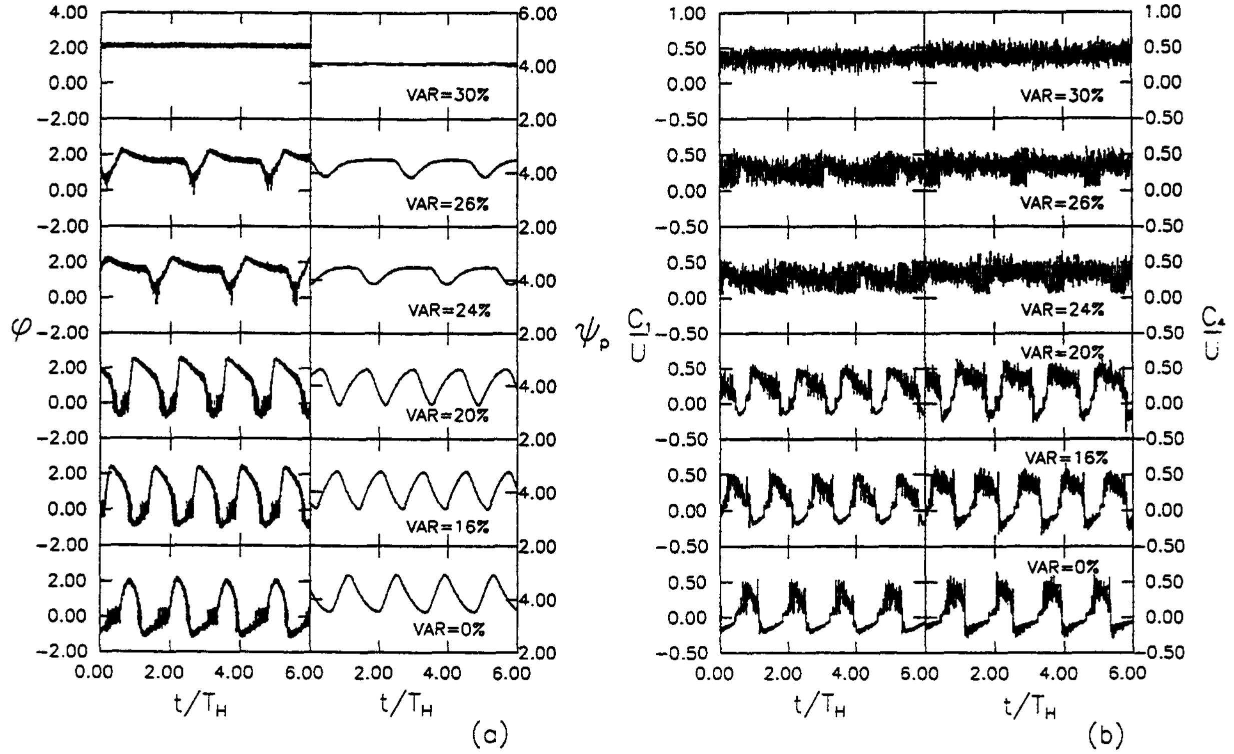


Fig. 9 - Time histories of flow coefficient and plenum pressure coefficient (a), and of velocities in the diffusers of the first and fourth compressor stages (b), at different valve settings and 2000 rpm.

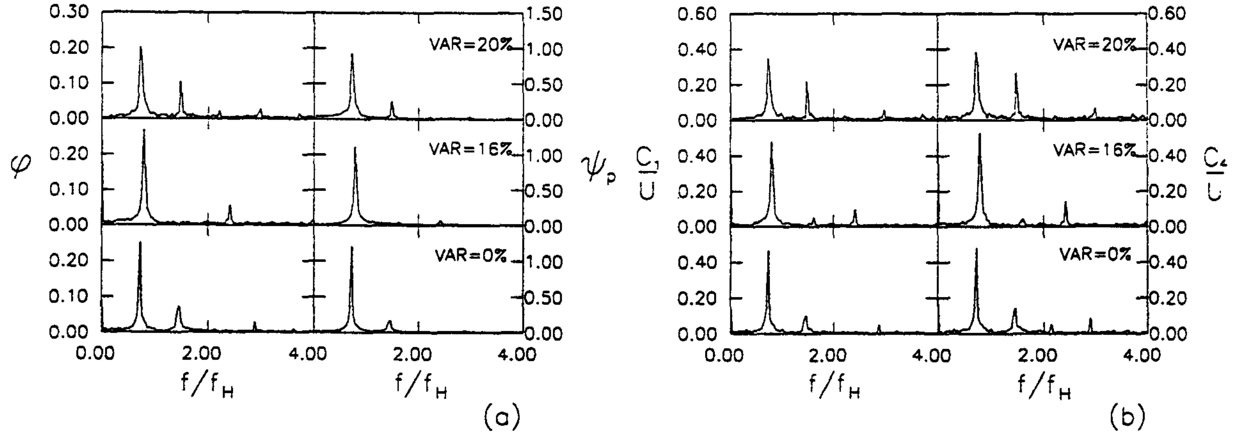


Fig. 10 - Spectra of flow coefficient and plenum pressure coefficient (a), and of velocities in the diffusers of the first and fourth compressor stages (b), at different valve settings and 2000 rpm.

As far as the influence of the compressor speed is concerned, the surge frequency turns out to decrease when the speed is increased, as observed by Jin et al. (1994) in the case of a single stage compressor. Figure 11 shows the velocity signals in the fourth diffuser and the corresponding spectra obtained at different speeds and VAR=16%. The surge frequency turns out to decrease almost linearly from 80% of Helmoltz frequency at 2000 rpm to 54% at 4000 rpm.

The present hot-wire measurements confirm that rotating stall can occur during part of surge cycle. This is shown in Fig. 12, where the flow velocities in the fourth diffuser at 2000 rpm, already reported in

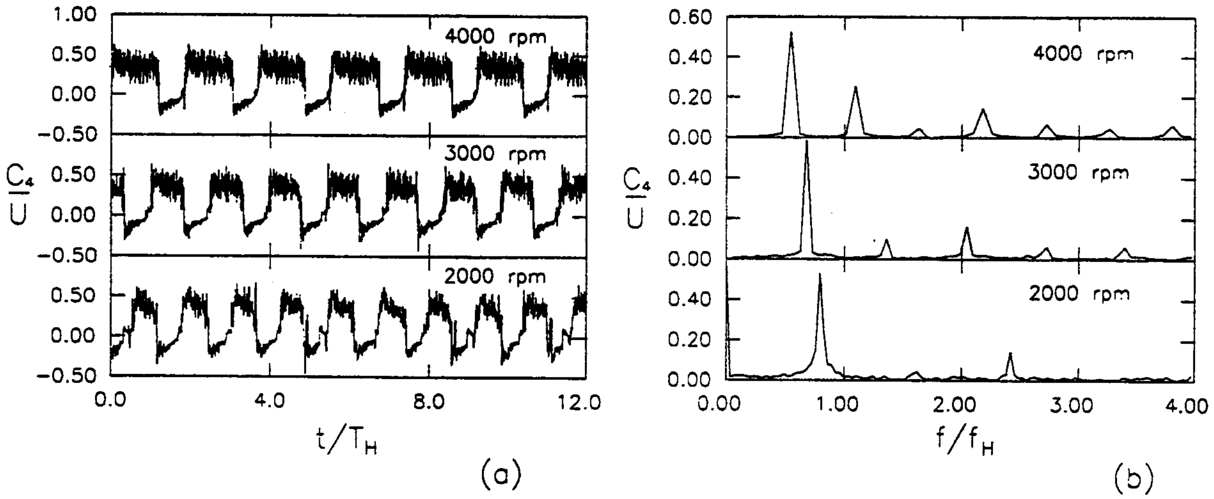


Fig. 11 - Time history (a) and spectra (b) of flow velocity in the diffuser of the fourth compressor stage at VAR=16% and different rotation speeds.

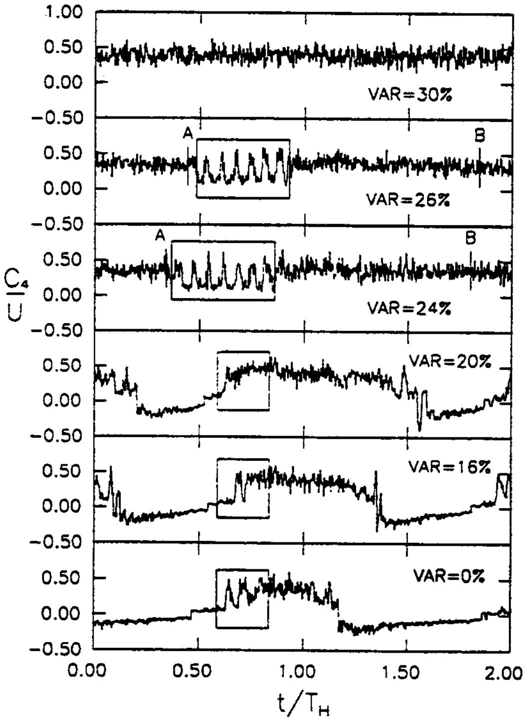


Fig.12 - Flow velocities in the diffuser of the fourth compressor stage during a surge cycle, at different valve settings and 2000 rpm. Frames point out stalled regions.

Fig. 9, are plotted in a larger scale (time origins are not preserved) in order to point out stall oscillations. These latter are framed in the figure and are quite evident at VAR=26% and VAR=24%, which correspond to temporary instability conditions of the system rather than to real surge. However, stall oscillations are not always present because the "surge-like" cycle produces continuous changes in the flow features by moving the instantaneous working point through successive conditions of stability, stall inception and full instability. Indeed, reference points A and B in Fig. 12, which denote the beginning and the end of the system instability, respectively, allow the rotating stall to be located over the whole cycle portion at decreasing pressure. When deep surge occurs, i.e., VAR$\leq$20%, stall is detected from the end of the reverse flow phase to an instant shortly before the stable branch of the compressor characteristic is reached, as already pointed out by Bammert and Mobarak (1976). Really, in deep surge conditions, Fig. 12 shows further velocity oscillations, which might have the stall frequency, occurring at 4 to 6 impeller revolutions after the surge line is crossed, some time before the flow reversal. In order to investigate on this matter, a FFT analysis has been performed on windows of the flow velocity signals. Harmonics having a typical frequency of rotating stall were detected only on the low pressure portion of the surge cycle. On the contrary, the little number of velocity oscillations and the large noise to signal ratio which characterise the inversion from positive to negative flow did not allow an accurate frequency analysis to be performed in this region and any conclusion to be drawn about the presence of rotating stall in the high pressure side of the surge cycle. Figure 13 shows the results of the FFT analysis concerning the windows where rotating stall was detected. In part (a) of the figure the stalled portion of the deep surge cycle at VAR=0% is marked, as an example, while in part (b) the velocity spectra show frequencies close to 40% of the impeller rotation frequency. This stall frequency turns out to be the same obtained by the tests without plenum, as also found by Greitzer (1976).

CONCLUSIONS

Detailed measurements in surge instability conditions were performed on a low-pressure compression system. The analysis of the flow rate and pressure measurements in the plant and of the flow velocities signals in the diffusers of the multistage centrifugal compressor allowed the unsteady system behaviour to be fully characterised.

Tests carried out on the plant configuration without plenum showed a high frequency velocity oscillation in the diffusers due to rotating stall, while surge was detected only at the highest rotation speed. By connecting the compressor delivery pipe to a large plenum, typical low frequency oscillations due to surge instability were observed at low flow rates and at all the tested compressor speeds.

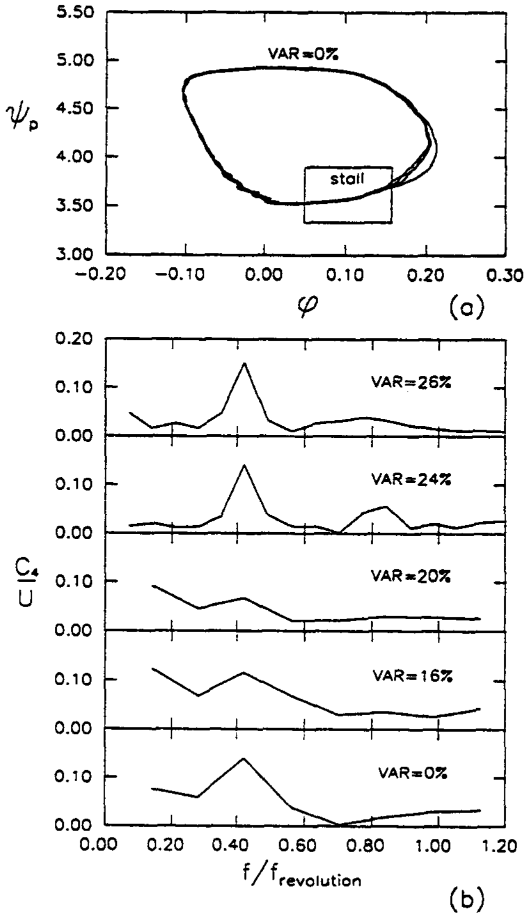


Fig.13 - Stalled portion of the surge cycle at VAR=0% (a) and velocity spectra of the framed signals of fig.12 (b).

The surge features were examined in detail at different operating conditions of the compression system and significant differences in the measured values were observed when varying both valve setting and rotation speed. In particular, non-periodic oscillations were detected when the steady-state equilibrium flow rate was set at a slightly higher value than the stall limit one. In addition, rotating stall was observed in the blower diffusers on the low pressure portion of the surge cycle.

The aim of the present stage of the research was to get a deeper insight into surge instability of multistage centrifugal compressors and to obtain systematic experimental data for validating theoretical models of the compression system. These latter will be used to design both passive and active control systems capable of widening the stable operating range of the compressor through suppression of surge (Arnulfi et al., 1998).

ACKNOWLEDGMENTS

The authors would like to thank MURST (Italian Research Ministry) for the financial support granted to this research.

REFERENCES

Abdelhamid, A.N., Colwill, W.H., Barrows, J.F., 1979, "Experimental Investigation of Unsteady Phenomena in Vaneless Radial Diffusers", *ASME Journal of Engineering for Power*, vol.101, March, pp. 52-60.

Arnulfi, G.L., Giannattasio, P., Giusto, C., Massardo, A.F., Micheli, D., Pinamonti, P., 1998, " Multistage Centrifugal Compressor Surge Analysis. Part II: Numerical Simulation and Dynamic Control Parameter Evaluation", *ASME Turbo Expo Conference*, Stockholm (S), June.

Arnulfi, G.L., Micheli, D., Pinamonti, P., 1995, "Velocity Measurements Downstream of the Impellers in a Multistage Centrifugal Blower ", *ASME Journal of Turbomachinery*, vol.117, October, pp. 593-601.

Arnulfi, G.L., Micheli, D., Pinamonti, P., 1996, "Experimental Investigation on Rotating Stall in a Centrifugal Blower with Two and Four Stages and Vaneless Diffusers", ASME Paper 96-GT-171.

Bammert, K., Mobarak, A., 1976, in discussion of : Greitzer, E.M., "Surge and Rotating Stall in Axial Flow Compressors. Part II: Experimental Results and Comparison with Theory", *ASME Journal of Engineering for Power*, vol.98, April, pp. 213-215.

Cyrus, V., Sprinc, M., Paroubek, J., Kyncl, J., 1995, "Nonsteady Flows in Centrifugal Compressor Stages with Low Flow Coefficient", VDI Berichte 1186 - *Turbomachinery Fluid Dynamic and Thermodynamic Aspects Conference*, Erlangen (D), March, pp.389-404.

Greitzer, E.M., 1976, "Surge and Rotating Stall in Axial Flow Compressors. Part I: Theoretical Compression System Model. Part II: Experimental Results and Comparison with Theory", *ASME Journal of Engineering for Power*, vol.98, April, pp. 190-217.

Hansen, K.E., Jorgensen, P., Larsen, P.S., 1981, "Experimental and Theoretical Study of Surge in a Small Centrifugal Compressor", ASME Paper 81-FE-1.

Jin, D., Hasemann, H., Haupt, U., Rautenberg, M., 1994, "The Flow Rotation Characteristics of a Centrifugal Compressor during Surge", ASME Paper 94-GT-154.

Lin, F., 1996, "On the Practical Stability and its Application to Nonlinear Control of Surge", ASME paper 96-GT-512.

Ribi, B., Gyarmathy, G., 1993, "Impeller Stall as a Trigger for the Transition from Mild to Deep Surge in a Subsonic Centrifugal Compressor", ASME Paper 93-GT-234.

Ribi, B., Gyarmathy, G., 1995, "The Behaviour of a Centrifugal Compressor Stage during Mild Surge", VDI Berichte 1186 - *Turbomachinery Fluid Dynamic and Thermodynamic Aspects Conference*, Erlangen (D), March, pp.341-356.

Toyama, K., Runstadler, P.W. Jr., Dean, R.C. Jr., 1977, "An Experimental Study of Surge in Centrifugal Compressors", *ASME Journal of Fluids Engineering*, vol.99, March, pp. 115-131.