Military Technical College Kobry Elkobbah, Cairo, Egypt



8<sup>th</sup> International Conference on Aerospace Sciences & Aviation Technology

New Aerodynamic Design Aspect For Active Stabilization of Centrifugal Compressor Surge

### \* K. SAAD ELDIN

#### ABSTRACT

Recent Work has shown that centrifugal compression systems can be actively stabilized against the instability known as surge, thereby realizing a significant gain in system mass flow range . In this context, the paper presents the influence of a new impeller design aspects on increasing the range of stabilized compressor performance; the impeller is made of two coaxial parts working as full bladed impeller at stable regime. It could be transferred into tandem bladed impeller with zero overlap when the flow decreases below the minimum value, so the surge will be delayed or suppressed, the stabilization range will be increased. Herein the effect of tandem and shift angle on the stabilization range is studied. It has been found that range is increased in the form of decreasing the surge the stabilization flow rate limit by about 56%, when the two parts of the impeller are mass shifted by half the pitch angle at high speeds.

### NOMENCLATURE

A	Coefficient.
B	Coefficient.
С	Coefficient .
Ср	Specific heat at constant pressure, J/kg K.
L	Impeller total axial width, m.
m	Mass flow rate, kg/s.
Mo	Mass flow rate at maximum efficiency, kg/s.
Ms	Minimum stable mass flow kg/s.
Nc	Impeller speed of rotation, rpm.

\* Br. Gn. Dr., Tech. Research Center.

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Ncd	Impeller design speed, rpm.
Nr	Compressor speed ratio (Nc/Ncd).
р	Static pressure, N / m <sup>2</sup> .
Do	Total pressure , N / m <sup>2</sup> .
PR	Pressure ratio = (po2c / po1c).
PRo	Pressure ratio at maximum efficiency.
R	Gas constant, J/kg K.
Т	Static temperature, K.
То	Total temperature, K.
η	Compressor Efficiency
С	Compressor .
d	Design .
0	Total, or condition at maximum efficiency.
1	Compressor entry.
2	Compressor exit.
S	stable

# INTRODUCTION

The centrifugal compressors are today playing an essential role in gas applications . In small gas turbines, single stage radial compressor is turbine the multistage axial compressor. usually replacing The compressor characteristics is usually given by the pressure rise versus through flow rate at different impeller speed of rotation . The characteristic curve has a maximum, near the point corresponding to incipient stall conditions where is becoming detached from the surface. The flow incidence flow angle is reduced by increasing the flow rate, so that the right hand side of the characteristic curve marks a region where the flow tends to be stable where the slope of the pressure - flow line is negative.

#### Flow Mechanism at off Design regime

Impeller flow investigation attracted many investigators to study and especially near and at surge, in order to get a suitable active measure . method of stabilization . The stable operating regime of a compressor installed in a jet engine or any compression system is limited by the large amplitude fluctuations in mass flow rate and pressure onset of surge . It effects the performance and thrust as well as rise known as thermal loads . Surge is an possible accurance of mechanical and important factor in compressors design and operation.

Surge is known as the lowest order natural oscillatory mode of the compressor system, (Grietzer 1980). Experiments show that small amplitude, linear disturbances can quickly grow into large amplitude fully developed surge. Haupt (1988) and Fink (1992) presented an experimental study of the influence of impeller blade vibration and unsteady flow behavior. Analysis of the measurements showed that the impeller blades are very sensitive to different periodic excitations and interactions between flow and so the analysis of measurements showed typical broad vibrating blade. band characteristics of unsteady pressure field and also of the blade Results of flow angle investigations of the impeller inlet vibration behavior. the analysis of flow pattern show broad - band pressure together with fluctuations and blade excitation. This can be attributed to a strong reverse suction side of the radial blade in the shroud zone. This flow near the its source downstream of the impeller and is extending has reverse flow back up to a location ahead of the impeller inlet, this reverse flow causes high levels of blade excitation.

Flow mechanism measurements Mizuki (1992) and Bammart (1976) at full span stall showed that the flow entering the impeller near the hub changing its flow direction in the inducer part and leaving the then impeller in the reversed direction near the blade tip region. The high temperature values of reverse flow in the impeller inlet zone, was considered indication that the origin of the reverse flow would be further as an downstream in the compression system . This phenomenon of the flow

occur at off design operating regimes especially near mechanism which surge mode is the heart of the present work in which the concept of the impeller blades arrangement are based .

# Surge Control

Centrifugal compressor surge attracted many investigators to get some active control methods for decreasing the unstable regime margin, and increasing the stable regime of the compressor which corresponds to SO. lower mass flow rates.

Ffowcs williams and Huang (1989) presented an experimental and theoretical study for the stabilization of compressor surge by introducing an active element to counter any tendency of instability in the plenum. The control law governing the active stabilizer is determined from linear theory. The an experimentation of a compression system whose theory is verified in plenum volume is controlled. Suppression of the flow instability was achieved by switching on a loudspeaker pressure waves, which is placed on the wall of the plenum as a controller, and the compressor was set to operate stable a part of its characteristics beyond the natural stall line. The controller on was able to alter the system damping and the resonance frequency. Their the compression system can be actively stabilized by results show that the controller before or even after surge occurs. The decrease switching the mass flow rate, which corresponds to controlled surge is ranged in between 40 and 10 percent for low and high speeds respectively. The last method treats the periodic flow through the plenum and didn't telling us what about the pressure fluctuation inside the impeller flow channels and how much it is reduced and eliminated by the loudspeaker pressure waves. Ffowcs williams et al. (1993) extended the last work and applied a feed back control to the compressor of a 45 kW auxiliary power unit. They found that although surge inception could not be prevented, surge could be

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suppressed within a single period of the limit cycle. They achieved surge control by suppressing a nonaxisymmetric flow phenomenon in the diffuser of the centrifugal compressor. Control was affected by modulating a small extra air flow into the impeller, by this method the engine was able to deliver more than 10 percent extra shaft power before surge occurs.

Paduano (1993) presented an active feed back control by using a circumferential array of hot wires to sense propagated waves of the axial velocity upstream of the compressor during surge, so by wiggling inlet guide vanes simple proportional control law implemented for each harmonic. Control of first spatial harmonic yielded an 11% decrease in the stalling mass flow, while control of the first, second and third harmonics together reduced the stalling mass flow by 23%. Simon (1993) declared that proper choice of sensor as well as actuator crucially affects the ability to stabilize compressor performance.

# New Design Aspect

The present work presents a proposed method for increasing the stabilization of a centrifugal compressor performance, the method bases on splitting the flow passage throughout the impeller blades. The impeller blade geometry has been switched into tandem bladed with zero overlap by a certain shift angle, when the flow rate is reduced below the minimum stable value. Thereby, the surge could be suppressed or delayed to another lower flow rate values. The impeller design aspect treats the surge flow so that the abrupt periodic flow is changed into smooth circulatory flow without pressure fluctuations. The back flow region is splitted by the shifted blades.

The impeller consists of two parts as shown in Fig. 1 the two parts could be shifted relative to each other in theta direction, thereby the

impeller may be operated at stable regime as full bladed impeller or as tandem bladed with zero overlap at unstable regime. During stable operation the two parts of the impeller, coincides to each other, and the blade continuation is achieved . By shifting the twice impeller parts, multi - shaped tandem bladed impeller with different shift angle in theta direction, has been generated.

The shift angle between the impeller frontal and rear parts breaks the continuation of the blades surface so that the air will be splitted and sucked easily from the blade pressure side to the blade suction side. This flow large amplitude fluctuations, and circulates continuously, eliminates the balances the flow around the impeller frontal part.

The compressor characteristics parameters (pressure ratio, efficiency, and mass flow rate) have been measured and calculated for tandem shift angles of 1°, 2°, 3°, 5°, 10° and 15° such that the front part leads the rear part with respect to the direction of rotation. The analysis of these curves the minimum stable mass flow decrease by about 56 % as indicates that compared with the zero shift, at which the pressure and efficiency drops to limited values. This work presents suitable data to design an active control device for increasing the stable margin by selecting the impeller geometry, according to the location of the operating point, especially before surge starts.

# EXPERIMENTAL INVESTIGATION

The objective of the experimental work is to investigate the effect of tandem shift angle of the impeller blades on the stability of the compression system, so the impeller shift angle will be varied between zero and 15° deg.

### Impeller Geometry

procedure the centrifugal compressor of the proposed In the type (HOLSET-4), of 100 mm outlet diameter, is chosen as an turbocharger

experimental running model, as shown in Fig.2. The full bladed impeller geometry sample consists of two parts assembled together adjustable shift angle, as shown in the view picture, Fig. 3. The stable regimes of the compressor characteristics were studied under the following conditions: 1- Changing the impeller blade shift angle by 1, 3, 5, 10 and 15, deg.

- 2- Changing the impeller speed of rotation (4 characteristic speeds of 10000, 15000, 20000, 25000 rpm-)-
- 3- Changing the mass flow rate for each speed from maximum to minimum stable values.
- 4- Measuring the flow parameters (temperatures, pressures) before and after the test section at each operating point.

# **Experimental Setup:**

experimental setup used herein is described in details by Eldin The Fig. 4 illustrates the layout of the experimental setup. The (1993) . compressor sample is placed in the test section (21) of Fig. 4 and is driven by the air turbine (30). The compressor air flow is drafted through the inlet pipe (20) and delivered to the outlet pipe (24). The air flow rate is regulated the manual control gate valve (26). The root compressor (7), of radial by used to compress the air into the turbine (30). It is driven by type , is variable speed electric motor (9) of type "schraga" through a high speed mechanical transmission (8). The diffuser (12) and the air box (13) are for stabilizing the air before the turbine. The bypass gate valve working (17) is controlled to get the air turbine and the root compressor matched together, and the root compressor surging could be avoided especially at turbine low speeds .

# **EXPERIMENTAL RESULTS AND EVALUATION :**

The compressor characteristics at tandem shift angles 1,2,3,5,10 and 15 degrees are determined and plotted in the form of reduced pressure ratio and

efficiency versus reduced through flow rate. Fig. 5 shows the full bladed impeller compressor characteristic map, this will be used as a reference to compare with the other tandem bladed impeller characteristics.

Figs. 6a to 6c shows samples of the compressor maps at leading tandem shift angles . 1,3,10 and 15 degrees . The compressor characteristic are refined in order to obtain the effect of tandem shift angle on the minimum stable mass flow rate (stable margin), the maximum pressure ratio and efficiency at different speeds. Fig. 7 shows the effect of leading shift angle on the minimum stable mass flow rate. At speed 10000 rpm the minimum stable flow increases with increasing the tandem shift angle to a value of 72.7 % and then decreases to a value similar to the original one at shift angle of At speed 15000 rpm the minimum stable flow decreases with increasing 15°. the shift angle and there is a sudden decrease at shift angle of 3° by about 48% and the curve return back to continue in decreasing rate to a minimum value of 47% at shift angle of 15°. At speed 20000 rpm the minimum stable decreases exponentially with increasing the tandem shift angle, it flow decreases to about 52.17% at shift angle of 10° and increases again to about 45% at shift angle of 15°. At speed 25000 rpm the maximum decrease in the minimum stable flow was found about 56% when the shift angle is half the pitch (15°). Fig. 8 shows the relation between the optimum tandem shift angle and speed of rotation, it was found a polynomial of third order, as follows:

$$\theta = A_1 + B_1 N_c + C_1 N_c^2 + D_1 N_c^3$$
(1)

where :

 $A_1 = 30$   $B_1 = 0.0066$  $C_1 = 4.4 \times 10^{-7}$   $D_1 = 8 \times 10^{-12}$  ٢

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s.

The minimum stable flow is also correlated with respect to the speed of rotation as shown in Fig. 9 by the following relation :

$$m_{s} = A_{2} + B_{2} N_{c} + C_{2} N_{c}^{2} + D_{2} N_{c}^{3}$$
(2)

where :

A <sub>2</sub>	11	-0.214		B <sub>2</sub>	11	4.007	Х	10°
C <sub>2</sub>	=	2.27 X	10 <sup>-9</sup>	D <sub>2</sub>	8	4. 33	x	10 <sup>-14</sup>

Nc from 10000 to 25000 rpm in equations (1) and (2)

The previous results can be used in a smart control system . As shown in Fig. 10, the minimum stable flow which is corresponding to the speed of rotation (Nc) is calculated by the formula (2) and then compared by the measured actual flow rate (m). If the measured mass flow is equal to or less than the minimum stable flow (m<sub>s</sub>), the tandem shift angle (θ) will be calculated by the formula (1). The controller will receive the value of the tandem shift angle (0) and transferred it to angular displacement . Sample of the modified characteristic map is illustrated in Fig. 11, which indicates that at high speed 250000 rpm the characteristic curve will be changes from the original one to the modified curve after the minimum stable mass flow the impeller front part will be shifted by decreased by a certain amount, Also the figure shows that the drop in the maximum about 15° deg . pressure ratio and efficiency are about 4% and 40 % respectively from the full bladed one.

#### CONCLUSION

The effect of tandem shift angle on the centrifugal compressor unstable characteristics was found to be significant. The results obtained herein indicated that the stable margin of characteristics could be increase, and the operation safety insured. The decrease in the minimum stable flow was found to be about 65 % at half pitch angle for high compressor speeds. The relation between tandem shift angle and speed of rotation has been found and correlated.

The obtained data may be used in further works to establish a smart controller, used in active control system, and the suggested scheme of this control is presented. The modified characteristics curves indicated that the maximum pressure ratio and maximum efficiency are decreased by 4 % and 40 % respectively at high speeds. The work could be extended to study:

1- The effect of tandem shift angle when the front part is lagging with respect to the rear part.

2- The location of separated plane between front and rear parts.

3- The shape of separated plane in the orthogonal and axial direction.

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Fig. 1 Full bladed impeller sample consists of two parts.



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Fig. 2 The turbocharger model HOLSET - 4.

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Fig. (7) Effect of leading shift angle on the minimum stable flow rate.



Fig. (8) The relation between tandem leading shift angle and speed of rotation.







Fig. (10) The flow chart of suggested control operation.





Fig. (11) The modified controlled characteristics map.

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