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Performance analysis of centrifugal compressor with variable diffuser

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Abstract:

This paper includes design and analysis of centrifugal compressors with variable diffuser. The main agenda of this work is to investigate the effects of centrifugal compressors with variable diffuser on mass flow rate, pressure ratio and performance of a compressor, and to optimize the design to get better performances. A standard centrifugal compressor was designed as per the design specifications of NASA's low speed centrifugal compressor with diffusers having different vanes angles from -6 to +6 degrees. After the performance analysis, we came to know that when a compressor is getting inadequate amount of mass flow rate that can cause surging of the compressor, by increase of diffuser vane angle we can shift the surge point to the lower values so that the mass flow which a compressor is getting could come under operating condition**.** The negative diffuser angles are benefitting when a compressor gets adequate amount of mass flow as they increase the efficiencies.

I. INTRODUCTION

Engine is the main power supplier to any aircraft. Aircraft engines working procedure follows the brayton cycle. In turbo engines, air is drawn through inlet and compressed to increase its pressure. Further, energy of the air is increased by combustion process and expanded in the nozzle to obtain a propulsive force. Centrifugal compressors produce high pressure ratio per stage compared to axial compressors. Centrifugal compressors have been used in smaller aircrafts and auxiliary power unit (APU) in all the aircrafts. Air is drawn axially by highspeed impellers which increase the kinetic energy of air by directing it to flow radially outwards. Main Function of the diffuser is to decelerate the flow and increase the pressure.

II. METHODOLOGY

Design of impeller is carried out using Solid-works Software. NASA low speed centrifugal compressor has been taken for the design and analysis [7].

Various diffuser configurations having vane angle deviations from -6 to +6 degrees with respect -to standard diffuser vane angle were designed.

Analysis of effects of centrifugal compressors with variable area diffuser at different operating speeds on mass flow rate, pressure ratio and performance of impeller was carried out in ANSYS Software.

Compressor maps/curves are analysed and design optimization will be done to get the better performances. Design specifications of diffuser and impeller are tabulated in table 2.1 and 2.2. Figure 2.1 illustrates isometric view of centrifugal compressor designed.

Table 2.1 Diffuser design specifications

Number of Vanes	23
Leading edge circle radius in cm	0.409
Pressure surface angle in degrees	72
Suction surface angle in degrees	80
Rotor exit to vane inlet radius ratio	1.08
Channel Divergence in degrees	7.65
Area ratio	2.9

Table 2.2 Design specification of NASA-LSCC impeller

Rotor Speed in rpm	1920
Inlet hub-tip radius in cm	0.5
Inlet radius in cm	42.939
Exit blade height in cm	13.740
Exit radius in cm	76.200
Axial clearance	0.5725
Number of blades	20
Blade angle at impeller tip in degrees	56.3
Blade angle at exit in degrees	55
Rotor exit tip speed in degrees	153.2

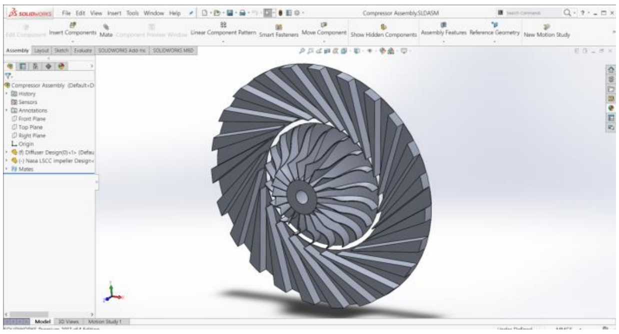

Fig.2.1 Isometric View of Centrifugal Compressor (standard configuration) Assembly

III. ANALYSIS

To determine the effects of variable diffuser on performance of the centrifugal compressor, CFD (Computational Fluid Dynamics) analysis has been carried out using the ANSYS software. The software version used in this analysis was ANSYS 18.1. With this software it became easy to carryout CFD calculations.

A. Fluid flow analysis

Fluid flow analysis involves numerical simulation with fluid to fluid interaction between the interfaces. The material to be meshed and analysed here is of fluid type. In our case, air was taken as fluid. In this analysis, a flow domain has to be created through which air flows. After extracting the flow path from the centrifugal compressor, meshing, rotating and stationary domain creation, applying boundary conditions, and solution to the problem was carried out. Major steps involved in analysis are explained as follows.

1) ***Domain creation:*** A Computational Fluid Dynamics (CFD) domain is the portion of space where the solution of the CFD simulation is calculated. In internal flows, the computational domain is defined by the confines of the geometry itself and the space inside the geometry is discretized into a computational grid.

2) ***Meshing:*** A Mesh is a network that is formed of cells and points. Meshing is a part of the engineering simulation where complex geometries and models are divided into simple elements that can be used as discrete local approximations of the larger domain. Meshing is carried out in ANSYS using auto meshing where unstructured tetrahedral mesh elements are created. Figures 3.1, 3.2, 3.3 illustrates front, rear and cross sectional view of mesh of the flow domain respectively. Tetrahedral mesh elements are preferred in complex geometries.

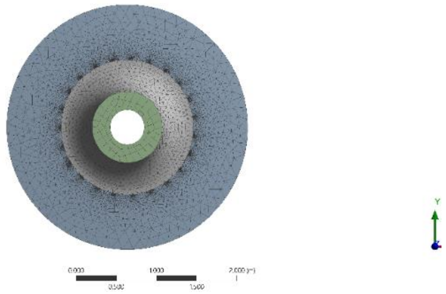

Fig. 3.1 Front view of flow-domain meshing

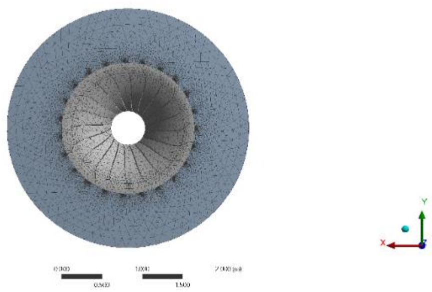

Fig. 3.2 Rear view of flow-domain meshing

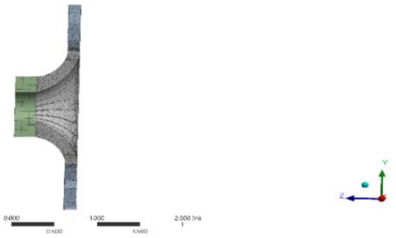

Fig.3.3 Sectional view of flow-domain meshing

*3) **Grid independent study:*** Grid independent study has been carried out for centrifugal compressor configurations such as standard, +2, +4, +6, -2, -4 and -6 configurations. The plots for variation for standard configuration of pressure ratio and becoming independent of no. of grid elements are as shown in fig. 3.4. Similarly grid independent study was carried out for all other configuration mentioned above. All of these compressor configurations have almost similar geometry except deviation in their diffuser angles which slightly makes no. of grid elements different form each other. Hence, grid independent study was done for all the compressor configurations.

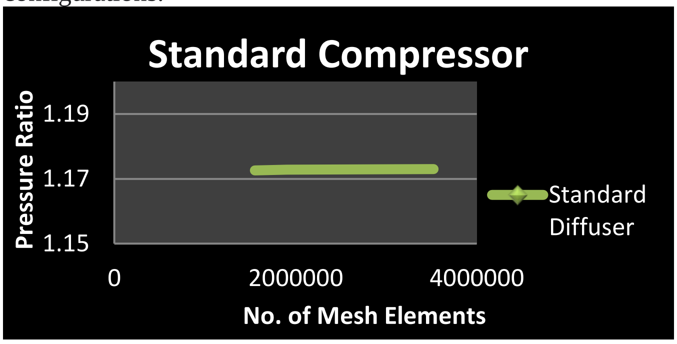

Fig. 3.4 Standard compressor

***B. CFX post processing**:* After the mesh was created, it is then transferred to CFX post process. In the post process, selecting the type of domain (rotating and stationary), applying boundary conditions and equations to solve the problem, writing monitor control, and inserting expressions for CFD calculations were carried out. The steps followed in CFX post processing are explained in detail.

***Domain selection**:*

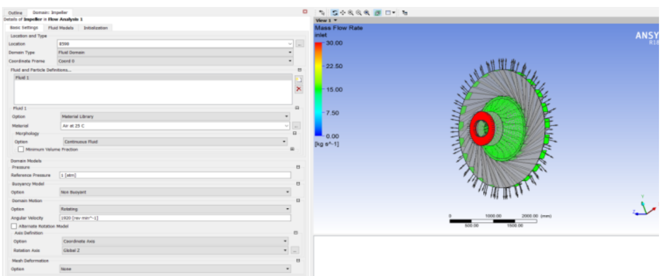

Fig. 3.5 Rotating domain (Impeller) in CFX with rotating speed of 1920rpm

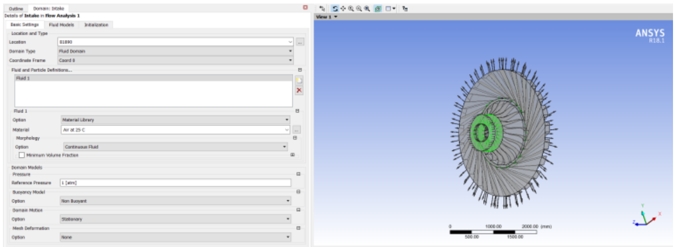

Fig. 3.6 Stationary domain (Intake) in CFX

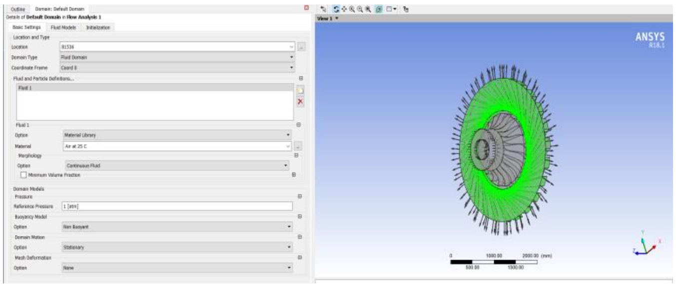

Fig. 3.7 Stationary domain (diffuser) in CFX

- Once after importing the mesh into CFX, stationary and rotating domains were created.
- Rotating domain (Impeller) with rotational speed of 1920 rpm and Intake and Diffuser as stationary domains were created as shown in Fig. 3.5, Fig. 3.6 and Fig. 3.7 respectively.

***Boundary conditions**:* Boundary conditions were applied at intake and diffuser outlet. At the inlet face of the intake domain, Inlet boundary condition was given in terms of mass flow rate, which is equals to 30Kg/s with flow normal to the boundary conditions as shown in Fig. 3.8.

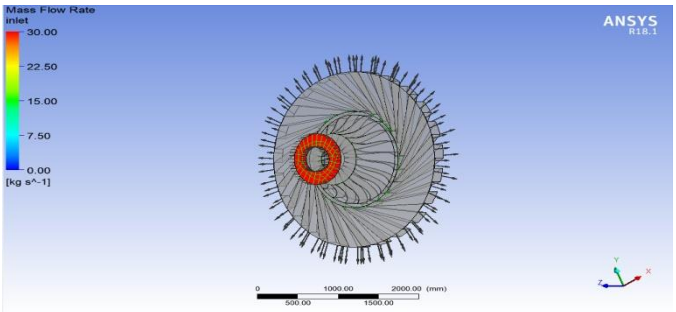

Fig. 3.8 Inlet boundary condition (Mass Flow rate= 30Kg/s).

Exit boundary conditions are applied at the outlet face of diffuser domain as shown in Fig. 3.9. Relative pressure of 1atm (Atmosphere) was applied to guide the flow.

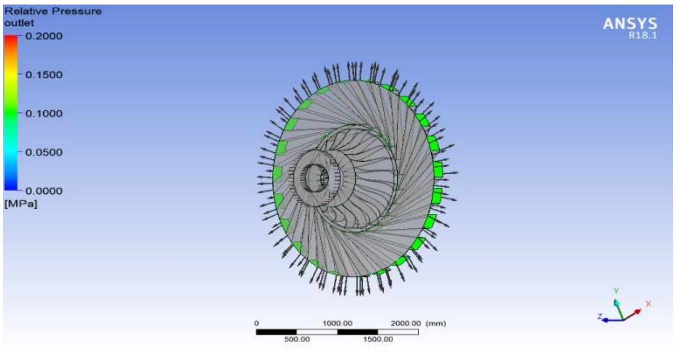

Fig. 3.9 Outlet Boundary Condition (Relative Static Pressure =1 atm.)

Interface: Fluid-Fluid interfaces between impeller and diffuser domain and between impeller and intake are created and are illustrated in Fig. 3.10 and Fig. 3.11 respectively.

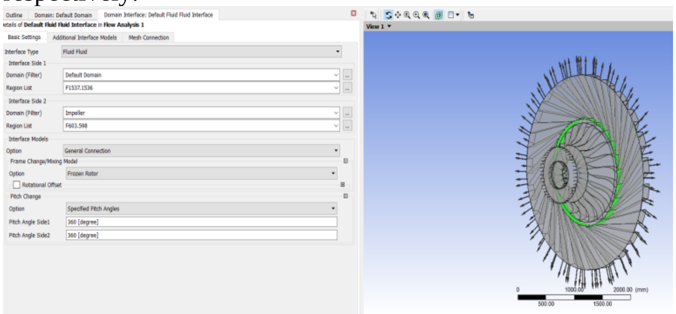

Fig. 3.10 Fluid-Fluid interface between impeller and diffuser domains

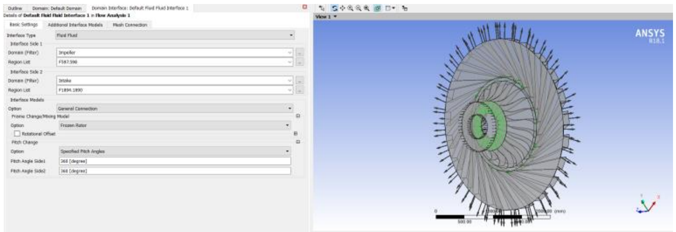

Fig. 3.11 Fluid-Fluid interface between impeller and intake domains

In both the cases, frame change model was kept as Frozen Rotor to get accurate results with less computational time.

The pressure contour shown in fig. 3.12 shows that the pressure kept increasing from inlet to exit of the centrifugal compressor. At the impeller, along with the kinetic energy pressure was also increased and at the diffuser, pressure was increased furthermore due to the decrease in kinetic energy which in turn caused by increased cross sectional area of the diffuser.

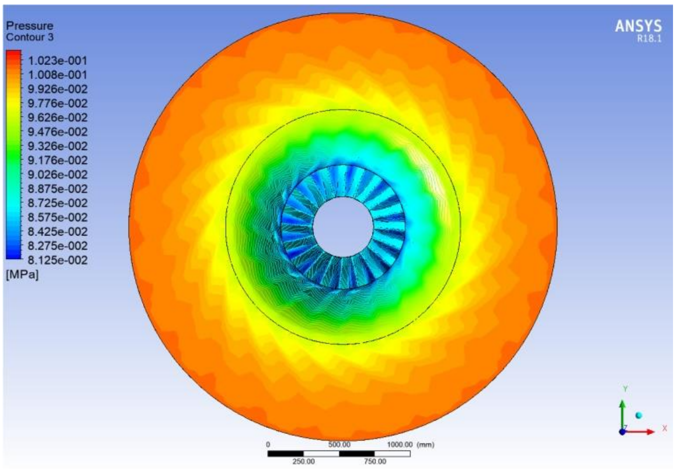

Fig. 3.12 Pressure Contour at Mass Flow rate of 30Kg/s and 1920RPM.

IV. RESULTS AND DISCUSSIONS

Results obtained upon carrying out flow and performance analysis for different diffuser configurations and mass flow rates are discussed.

After doing the analysis for a range of mass flow rates, the graphs between pressure and Mass flow rate for different diffuser configuration were plotted.

1) ***Comparison of Different Configuration with the Standard Compressor:*** The graphs between pressure ratio and mass flow rate for various compressor configurations are compared with standard centrifugal compressor to see the variation and improvement in their surge point and working range of mass flow rates.

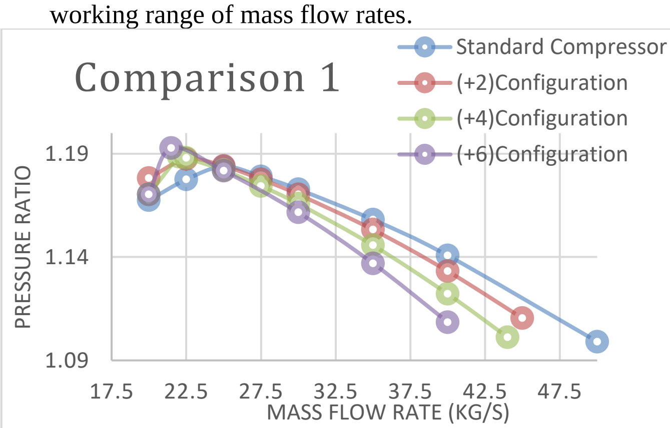

Fig. 4.1 Comparison plot between std. compressor Configuration and Positive Compressor Configurations

Here, positive compressor configurations such as +2, +4, and +6 are compared with standard compressor data as shown in Fig. 4.1.

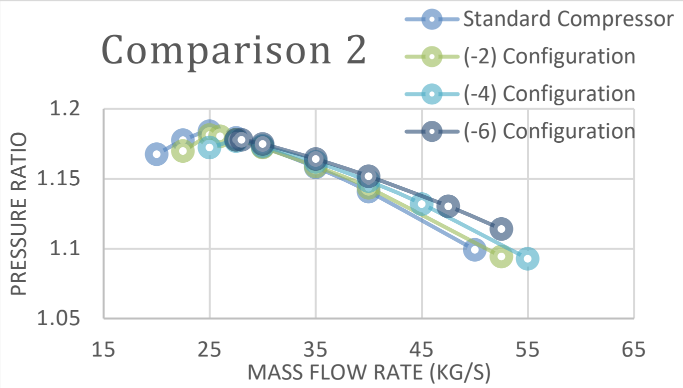

Fig. 4.2 Comparison plot between std. compressor Configuration and Negative Compressor Configurations

Similarly, standard centrifugal compressor's post process data for various mass flow rates was compared with those of compressors having negative diffuser angle deviations. The plots for this comparison are as shown in Fig. 4.2.

2) ***Discussion on fluid flow analysis results:***

This was the simple analysis carried out to predict the variation of pressure ratios, surge point for a wide range of mass flow rates and for different centrifugal compressor configurations. Surge point is a point at which the compressor reaches its maximum pressure ratio at constant impeller speed. Having mass flow rate corresponds to surge point less as much as possible is essential in conditions where density of air is less. This makes the compressor work even in such conditions without affected by reverse flow. Few main observations were made after analysing the plots.

- From the results it was evident that, as the diffuser vane angle was increased from standard angle surge point in of the centrifugal compressor was shifted towards left.
- This shift in surge point towards left in the compressor map is helpful during inadequate mass-flow in higher altitude as the mass-flow decreases with decrease in density.
- This improvement in terms of surge point can be seen in fig.4.1.
- As the diffuser vane angles were decreased from standard diffuser angles, there was negative effect on surge point, but the maximum mass flow rates range was increased.
- From the overall plots, it was noted that the range of mass flow rates a compressor can operate with the help of varying diffuser angles was increased.

3) ***Performance Results:*** Performance analysis was carried out for different compressor configurations at designed point of standard NASA' low speed centrifugal compressor. The performance results are tabulated in table 4.1.

Table 4.1 Performance data of Compressor configurations at mass flow of 30kg/s and impeller speed of 1920 rpm.

Compressor Configurations (based on diffuser angle deviation in degrees)	**Temperature Ratio (TR)**	**Pressure Ratio (PR)**	**Isentropic Efficiency (%)**	**Polytrophic Efficiency (%)**	**Loading factor (Ψ_t)**
0	1.05336	1.1547	78.6504	79.4791	0.6626
2	1.05289	1.1252	64.8389	65.7546	0.6564
4	1.05336	1.1459	74.3817	75.2489	0.6625
6	1.05359	1.1399	71.1407	72.0323	0.6652
-2	1.05304	1.1547	79.1003	79.9223	0.6586
-4	1.05302	1.1555	79.5484	80.3655	0.6583
-6	1.05297	1.1576	80.6341	81.3655	0.6577

4) ***Comparison of performance of std. Compressor with compressor configurations:*** In this comparison study, efficiencies and loading factor variations are plotted against various compressor configurations at designed mass flow rate of NASA LSCC (30Kg/s) and the variation is illustrated in Fig. 4.3

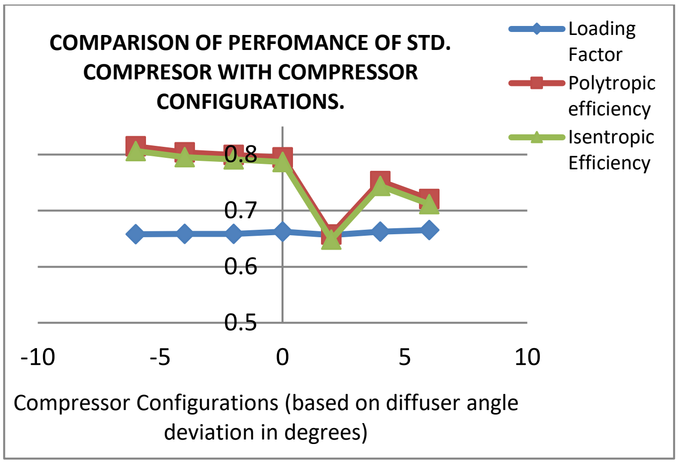

Fig. 4.3 Plot of variation of performance parameters with compressor configurations

5) ***Experimental error analysis:*** The data consisting performance parameters obtained during CFD post process is compared with calculated data for NASA LSCC at designed mass flow rate (30 Kg/s) and impeller speed (1920rpm) and is tabulated in table 4.2.

Table 4.2 Experimental Error Analysis

Parameters	**Calculated Value (CV)**	**Measured Value (MV)**	**Error In (%)** $\frac{(CV-MV)}{CV}\times 100$
Temperature Ratio	1.0449	1.05336	0.8
Pressure Ratio	1.149	1.15476	0.5
Isentropic Efficiency	90	78.6504	12.61
Polytrophic Efficiency	90.35	79.47919	12.031
Loading Factor	0.5536	0.66264	19.69

6) ***Discussion on performance results:*** Performance analysis was carried out in which isentropic efficiency, polytrophic efficiency and loading factors are calculated in CFD post process for different configurations of compressor. The comparison was made between performance parameters and the compressor configurations such as standard, +2, -2, +4, -4, +6 and -6. From the comparison plot as shown in Fig 16, few major points are noted and are as follows:

- As the diffuser vanes angle was increased from 0 to +6 degrees, isentropic and polytrophic efficiencies were dropped by 17.57% and 17.26% respectively at +2 degrees. At diffuser deviations of +4 and +6, isentropic efficiency was dropped by 5.42 % and 9.54% respectively. The variation in polytrophic efficiency was also similar to isentropic efficiency.
- The fall in efficiencies at +4 and +6 diffuser angles were found to be within 5-10 %. These efficiencies drop is acceptable as these configurations (+4 & +6) provide improvement in surge points at less expense of efficiencies.
- Design optimization was needed to eliminate a large drop in efficiency (17.57%) at +2 degree deviation.
- After observing the performance with negative diffuser deviations, the efficiencies were found to be increased by approx. 1 to 2.5%.

7) ***Design optimization**:* After analysing the results obtained, we noted that the design optimization was needed to eliminate the large drop in efficiencies during initial deviations of diffuser vane angles, particularly at +2 degrees. The way we found to avoid the fall in performance was to find the diffuser angle near the +2 deviation so that the drop lies within 5-10 % and can provide improvement in surge point compared to standard compressor configuration. Hence, we designed diffuser with +3 degrees of deviation in diffuser vane angle. The flow analysis and performance analysis were done. The plot for pressure ratio vs. Mass flow rate is as shown in fig 4.4.

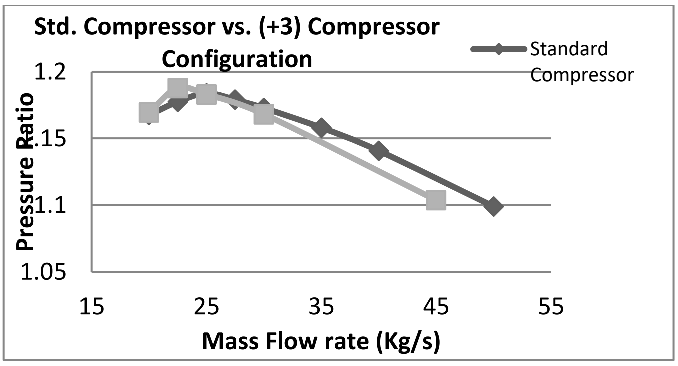

Fig. 4.4 Plot of Std. Compressor (NASA LSCC) vs. (+3) Compressor configuration

Table 4.3 Performance data of the standard, +2 and +3 configurations

Compressor Configurations (based on diffuser angle deviation in degrees)	**Temperature Ratio (TR)**	**Pressure Ratio (PR)**	**Isentropic Efficiency (%)**	**Polytrophic Efficiency (%)**	**Loading factor (Ψ_t)**
0	1.05336	1.15476	78.6504	79.47919	0.66264
2	1.05289	1.12527	64.8389	65.75463	0.656478
3	1.05336	1.14595	74.3817	75.24899	0.662554

The data tabulated in table 4.3 provides the evidence that the (+3) configuration is best suited compared to (+2) configuration. The fall in isentropic and polytrophic efficiencies for (+3) compressor were 5.42% and 5.32% respectively, whereas (+2) configuration gave a drop in efficiency around 17-18 % as seen earlier.

V. CONCLUSIONS

After analysing the results obtained, few major conclusions were been made and are as follows:

- As we increase the diffuser vane angle, we have observed a decline in the mass flow rate corresponds to the surge point. From this it is clear that the surge point has improved. We can make a note that when a compressor is getting inadequate amount of mass flow rate that can cause surging of the compressor, with the increase of diffuser vane angle we can shift the surge point to the lower values so that the mass flow which a compressor is getting could come under operating condition.
- While improving the surging of the compressor by increasing the diffuser vane angle, the compressor experienced decrease in its efficiencies which may limit the changing the diffuser angle. When surging occurs, a compromise has to be made in terms of performances to overcome the reverse flow.
- The results for decreasing diffuser angles have shown that the efficiencies of the compressor were slightly increased but there was no improvement in the surge point. Hence decreasing the diffuser angle is not convenient at mass flow rates near surge point. It is only benefitting when compressor gets adequate amount of mass flow.
- As we look at the overall process and results, we can make a compressor to work for a range of mass flow rates that it couldn't have experienced at its standard diffuser angle.

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