DESIGN POINT EFFICIENCY OPTIMIZATION OF A MULTI-STAGE AXIAL-FLOW COMPRESSOR FOR AERO APPLICATION APPLYING A SPECIALLY DEVELOPED COMPUTER CODE

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Abstract. *This paper presents a methodology for optimizing the design point efficiency of a multistage axial-flow compressor for aero application. Hub-tip ratio and space-chord ratio are geometric parameters of special concern. The stage-stacking method and mean-line technique is used to design and to estimate the compressor performance. Calculations are carried out at the mean blade height streamline for both rotor and stator. Empirical correlations are used to introduce the 3-D real flow effects. For the design point efficiency optimization the Sequential Quadratic Programming (SQP) is incorporated to the method. A specially developed computer program, called "AFCC Optimization Program" and developed by the authors, has been validated with data from an aeronautic eight-stage axial-flow compressor. The results have demonstrated that SQP Method is efficient in searching an optimum, especially when it is isolated. This is demonstrated with the application aiming at improvements to the eight-stage axial-flow compressor efficiency, pressure ratio, geometric and aerodynamic parameters at the design point.*

Keywords*: axial-flow compressor, geometry, design point, optimization, stage-stacking, efficiency.*

1. Introduction

Multistage axial-flow compressors with wide operating flow range and high efficiency are required for aircraft applications. The operation of such compressor is limited by instabilities which may cause a full breakdown of the flow known as surge. These instabilities, which may be caused by high incidence and subsequent stalling of stages, occur due to different phenomena at part- and full-speed operation. The problem at part-speed operation is that the front stages are often heavily stalled and the rear stages choked, whereas at high speeds, the front stages are operating close to choke and the rear stages tend to be stalled and causing surge (Sun and Elder, 1998).

The main design requirements of the multi-stage axial compressor for use in a turbojet engine involve an acceptable level of thermodynamic efficiency, adequate surge margin, and a reduced weight. However, the weight reduction inevitably causes loss of efficiency and the higher efficiency will lead to decrease of the surge margin. As a result, the efficiency and surge-margin improvement, and the weight reduction must be compromised according to certain design criteria.

This paper presents a numerical methodology for optimizing the efficiency at the design point of an eight-stage axial flow compressor with inlet guide vane (IGV) and variable stators (VG) for use in a turbojet engine. Efficiency, surge margin and weight, as already mentioned, are of major concern for the designer.

A specially developed computer program is employed. A Sequential Quadratic Programming (SQP), (Gradient-Based Method) is implemented. Although the global search methods are more robust than the local ones like SQP, its application has shown fast convergence, this being credited to the readiness of the initial values.

Along with the SQP method, the program employs the Mean-Line Technique and the Stage-Stacking Method. For turbomachinery, experts in many areas of machine design have long recognized that Mean-Line Technique (also known as one-dimensional [1D] modeling) is at the heart of good stage performance predictions and design optimization. With the development of modern computer techniques, Mean-Line model calculation can be made exceedingly fast, with considerable accuracy and with integrity over a very wide range of conditions (Japikse et al., 2004).

It will be demonstrated that starting from a newly, called here original, designed axial flow compressor, a new optimized version is obtained with improved the design point performance (efficiency). Among many possibilities, it will be shown that the optimized compressor presents a reduced weight and has a better off-design performance, demonstrated by the upward displacement of its surge line, when compared to the original one. The behavior of the main geometric and aerodynamic parameters will be discussed, which is very important at the outset design.

This paper was divided in two parts: first, the design point efficiency optimization, where the original and optimized compressor parameters will be compared and second, the off-design point analysis, which will be carry out through the performance curves (maps) of both, original and optimized compressors.

2. The design point efficiency optimization

2.1. The sequential quadratic programming (SQP) method

Since the search for the optimum efficiency at the design point is a non-linear problem, a successive quadratic programming method (SQP) was chosen as the optimization algorithm to be employed. The SQP method is a popular and a successful technique for solving nonlinearly constrained problem (Nash and Sofer, 1996).

The problem is stated as follows:

Minimize the function

$$f(x) \tag{1}$$

Subject to

$$g_1(x) = 0 \tag{2}$$

$$g_2(x) \le 0 \tag{3}$$

where $"x"$ is a multi-dimensional variable, $g_1(x)$ and $g_2(x)$ are vectors of constraint functions.

The method, based on the iterative formulation and solution of quadratic programming sub-problems, obtains sub-problems by using a quadratic approximation of the Lagrangean and by linearizing the constraints, that is,

Minimize

$$\frac{1}{2} p^T B_k p + \nabla f(x_k)^T p \tag{4}$$

Subject to

$$\nabla g_1(x_k)^T p + g_1(x_k) = 0 \tag{5}$$

$$\nabla g_2(x_k)^T p + g_1(x_k) \ge 0 \tag{6}$$

and

$$x_l - x_k \le p \le x_u - x_k \tag{7}$$

where B_k is a positive-definite approximation of the Hessian, x_k is the current iterate and x_l and x_u are vectors containing the lower and upper bounds on variables.

Let p_k be the solution of the sub-problem. A line search is used to find a new point x_{k+1},

$$x_{k+1} = x_k + \alpha_k p_k \qquad \alpha \in (0,1) \tag{8}$$

such that a “merit function” will have a lower function value at the new point. Here, the augmented Lagrange function is used as the merit function. When optimality is not achieved, B_k is updated according to the modified BFGS formula (Nash and Sofer, 1996). The finite-difference method is used to estimate the gradient of the functions.

2.2. The AFCC optimized program

The AFCC (Axial Flow Compressor Code) Program, written in FORTRAN, and developed by Tomita and Barbosa (2003) was modified in order to contemplate the optimization subroutine. The new program generated, called AFCC Optimized Program, maximize the efficiency at the design point according to some constraints defined by the designer. After the geometry is calculated by the AFCC Program, the AFCC Optimized Program re-calculates the geometry during the efficiency maximization at the design point.

The structure developed for the AFCC Optimized Program may be summarizing as shown in Fig. 1.

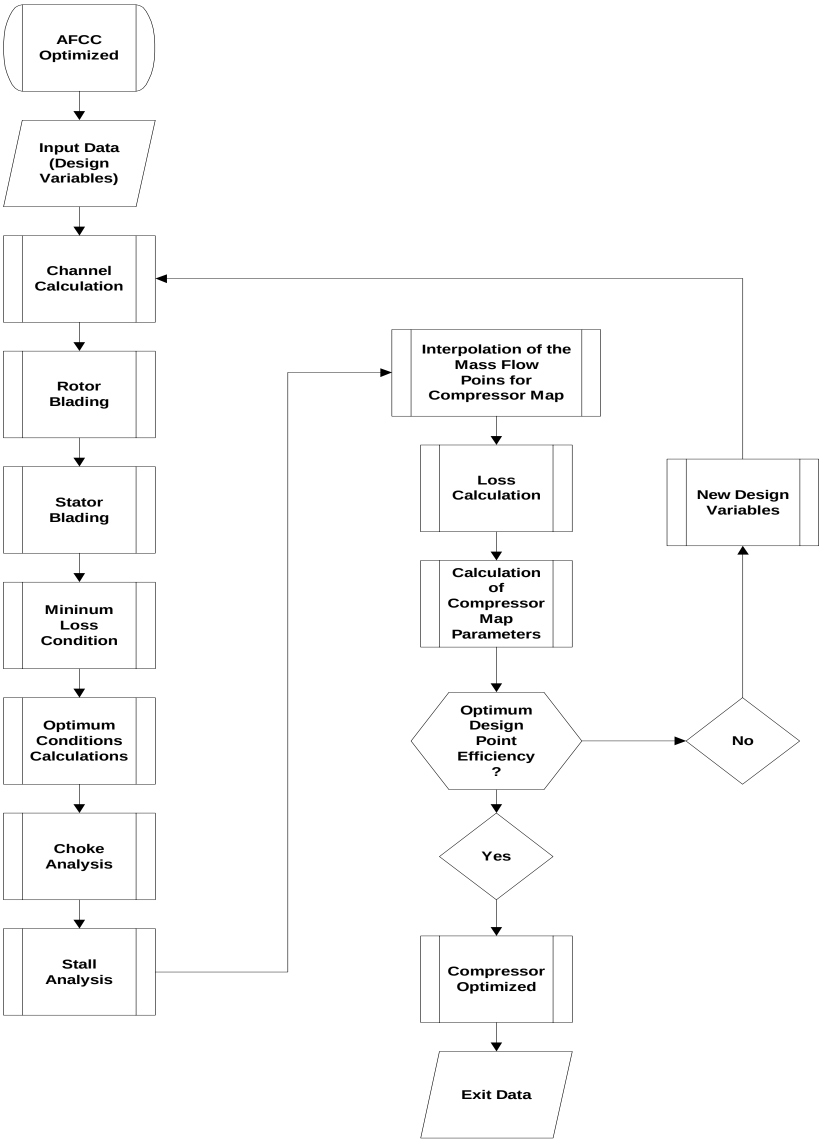


Figure 1. Flowchart of the AFCC Optimized Program

2.3. Formulation of optimum design problem

Any optimization problem involves the identification of design variables, objective function and design constraints.

The design variables are pre-selected variables, which can assume independent values in the design process. The other data of the problem are either given at the beginning of the design process or can be expressed in terms of the design variables.

To optimize the eight-stage axial-flow compressor, the following geometric parameters are taken as the design variables: the inlet hub-tip ratio and the rotors and stators space-chord ratio. These variables have been chosen in order to easily identify performance improvements at the initial design phase. They are non-dimensional and represent the basic size of the machine. Their initial values came from the AFCC Program.

The objective function in a general optimization problem represents a basis for the choice among various equally acceptable designs. In the case of axial-flow compressor used in aero applications, the weight, efficiency and stall margin are among the most important criteria that must be considered (Massardo and Satta, 1990). In the present work, the chosen objective function was to maximize the efficiency at a specific design point.

In order to keep the same dimensions of the original compressor channel, it was created a factor that controls the distribution of the stagnation temperature that load or unload the critical stages, that is, the front and the rear ones. Changing this stagnation temperature factor it is possible to alter the channel size. Although this changing could be done just keeping the inlet hub-tip ratio fixed, the possibility to load and unload the critical stages checking the influence of the stagnation temperature and the hub-tip ratio along the compressor seemed to be more relevant. Yet the stall margin will be increased through the IGV and VG new setting, as will be demonstrated by its off-design performance curves (maps) in the off-design analysis item.

The diffusion factor, blade height, and the axial dimension were chosen as design constraints in order to keep the compressor stable and able to be manufactured, Saravanamuttoo *et al.*(2001) and Walsh and Fletcher (1998).

2.4. Axial flow compressor data

The design parameters for the eight-stage axial-flow compressor that will equip the turbojet are:

Inlet stagnation pressure: 101.3 kPA
Inlet stagnation temperature: 288 K
Inlet Mach number: 0.59
Outlet Mach number: 0.25
Pressure ratio: 11.53
Polytropic efficiency: 0.89
Number of stages: 8
Tip speed: 340 m/s
Inicial IGV setting: 5.6º
Axial channel: Constant Outer Diameter (COD)

Table 1 presents the lateral limits for the design variables. Table 2 presents design constraints values.

Table 1. Lateral limits for the design variables.

Hub-tip ratio	$0.45 \le htr \le 0.75$
Space-chord ratio	$0.50 \le scr \le 0.95$

Table 2. Design constraints values.

diffusion factor	$DF \le 0.55$
minimum blade height$^{(1)}$	$h \ge 10.0$
maximum axial length$^{(1)}$	$dz \le 272.029$

$^{(1)}$: unit in mm

A stagnation temperature factor of 0.2 was chosen to keep the same inlet hub/tip ratio (0.615 mm) for both original and optimized compressor. This value means that 80% of the compressor load is concentrated at the inner part of it remaining 10% for the front and 10% for the last stages.

Other assumptions were made based on literature data Saravanamuttoo *et al.* (2001) and Walsh and Fletcher (1998).

The loss model is based on the model used by Barbosa, (1987).

2.5. Design point analysis

Table (3) summarizes the results generated by the AFCC Optimized Program at the design point. It can be seen an increase of 3.21% in the efficiency, 2.60% in the pressure ratio and a reduction of 7.40% in the corrected mass flow. The reduction in the corrected mass flow was expected, once the compressor geometry was altered to contemplate the increase in the efficiency.

Table 3. Results generated by the AFCC Optimized Program at the design point.

Compressor	*Nominal Speed (rpm)*	*Corrected Mass Flow*	*Pressure Ratio*	*Efficiency (%)*
Original	19600	1.6748	10.3144	80.48
Optimized	19600	1.5593	10.5824	83.69

The optimized axial compressor channel and the stagnation temperature distribution for each stage are presented in Fig. 1 and Fig. 2.

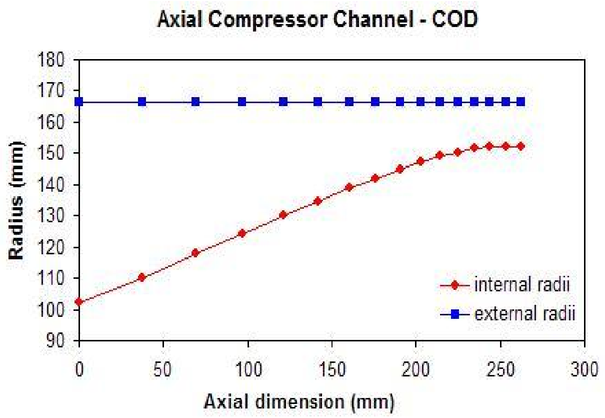


Figure 1. Axial compressor channel.

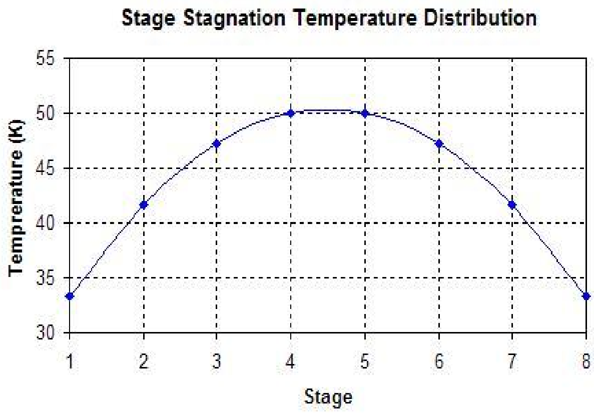


Figure 2. Stage temperature distribution.

Figure 2 presents the stagnation temperature increase for each stage of around 43 K. Due to the stagnation temperature factor, it can be seen that the front and rear stages are less loaded compared to the intermediate stages.

Figure 3 and Fig. 4 present the space-chord (s/c) ratio comparison between the original compressor and the optimized one.

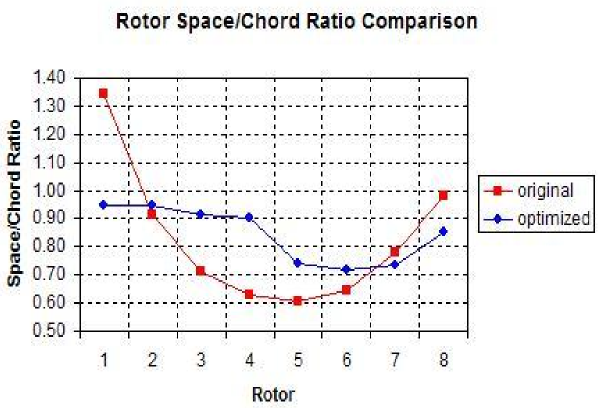


Figure 3. Rotor s/c ratio comparison.

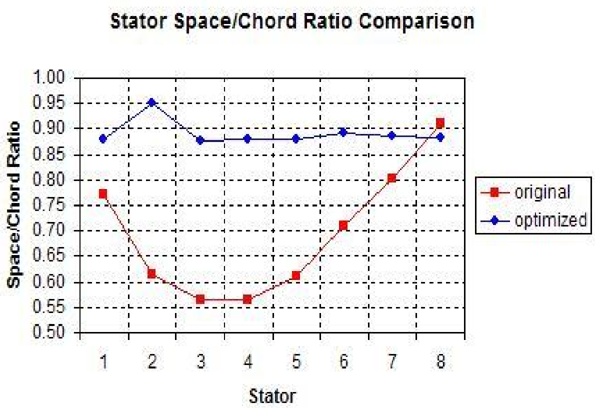
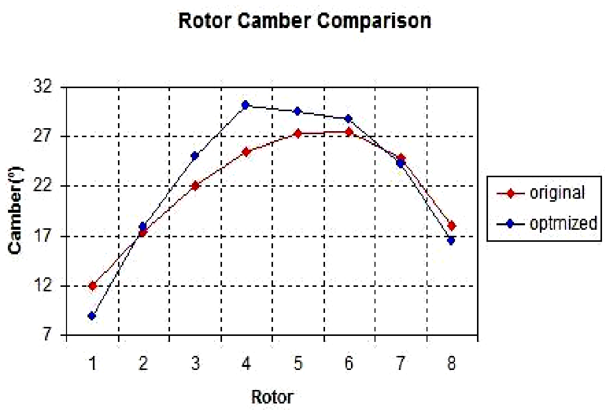


Figure 4. Stator s/c ratio comparison.

Figure 3 shows a great reduction on the first rotor s/c ratio value due to the lateral limits, see Table 1. The rotor s/c ratio shows increased values at the inner part of it while in the stators the values are all higher than the original one.

Figure 5 and Fig. 6 present the rotor and stator camber comparison.



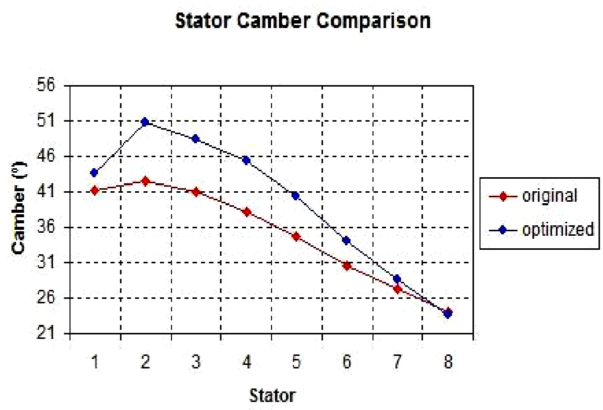
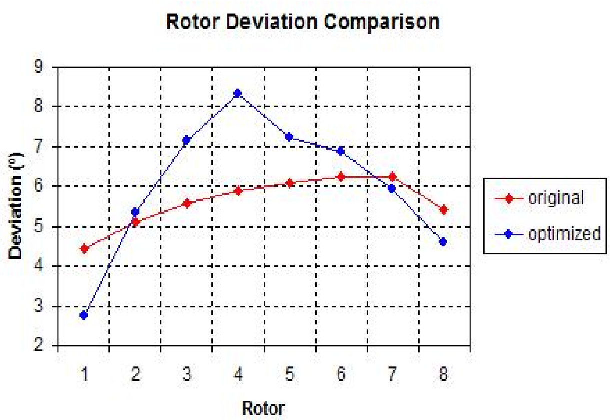


Figure 5. Rotor camber comparison.

Figure 6. Stator camber comparison.

Figure 5 presents camber values less than the original compressor at the front and rear rotors due to the stagnation temperature factor (front and rear stages less loaded), but at the inner part the camber was greater than the original one.

Figure 7 and Fig. 8 present the rotor and stator deviation comparison.



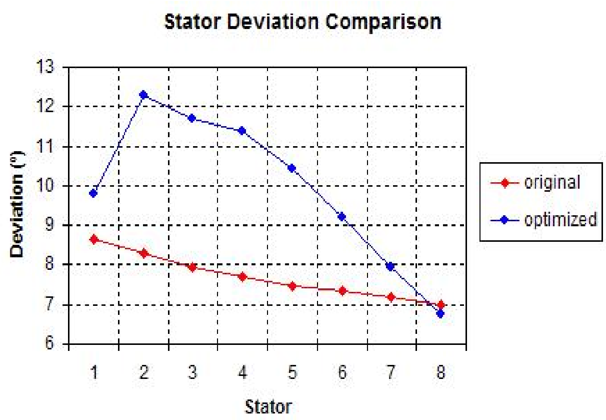
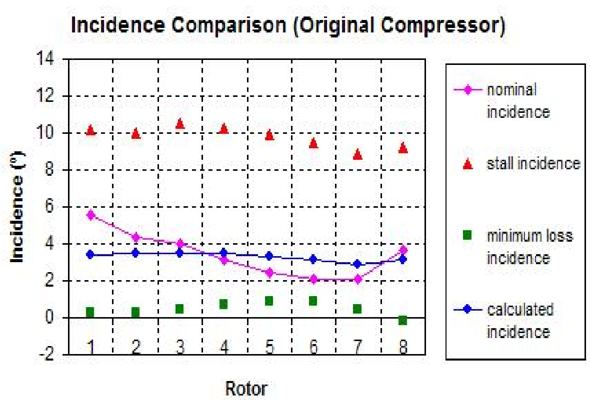


Figure 7. Rotor deviation comparison.

Figure 8. Stator deviation comparison.

As the deviation is function of s/c ratio and camber angles, and both have increased, it can be seen that their values stayed above at the original compressor. The exception is only at the front and rear rotors rows deviations due to weigh factor.

Figure 9 and Fig. 10 present the incidence comparison between original and optimized compressors. The nominal, stall and minimum loss incidence angles were calculated according to Tomita and Barbosa (2004).



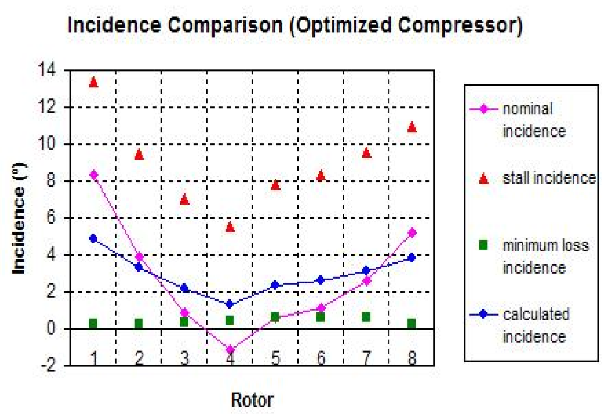
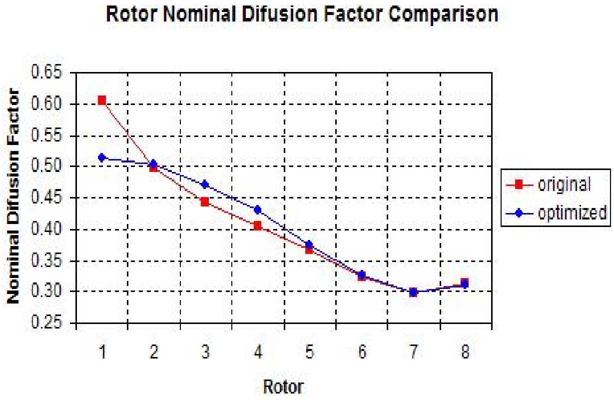


Figure 9. Original compressor incidences

Figure 10. Optimized compressor incidences.

It can be seen from Fig. 9 and Fig. 10 that the compressors cascades aren't stalled because the calculated incidence is below the stall one.

Figure 11 and Fig. 12 present the diffusion factor comparison.



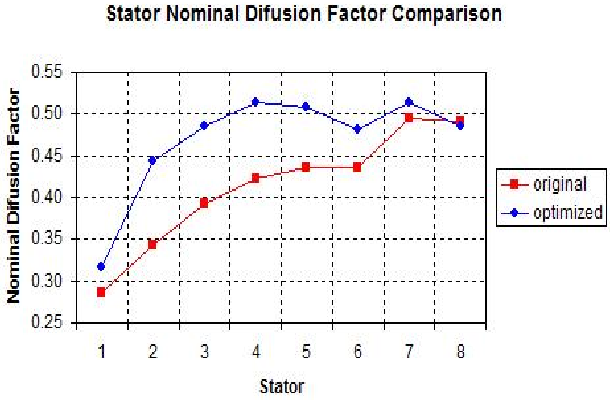


Figure 11. Original compressor diffusion factor. Figure 12. Optimized compressor diffusion factor.

From Fig. 11 it can be seen that the first rotor has a diffusion factor below than the original one due to the stagnation temperature factor and starting from the second stage its behavior is quite similar to the original one. Figure 12 shows that the great gain in the static pressure is due to the stators, mainly the inner ones, letting the first and the rear ones less loaded.

Even with the increase in the efficiency and pressure ratio, keeping the first and rear stages unloaded, Fig. 13 shows that there is almost no increase in the total pressure loss coefficient indicating that there was enough space to optimized the DCA profile used in the eight-stage axial compressor.

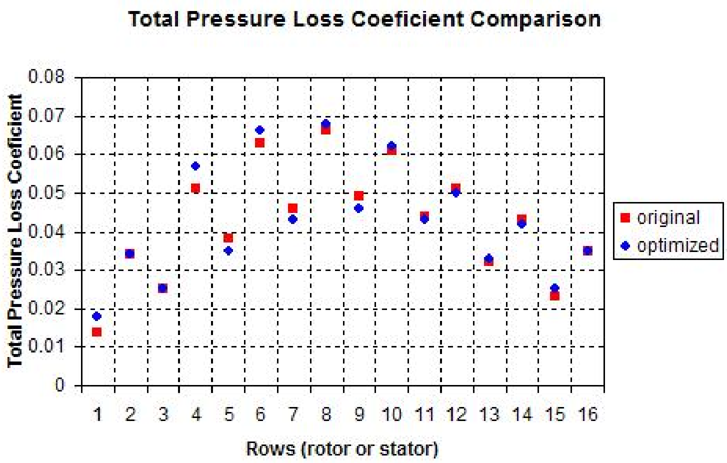


Figure 13. Total pressure loss coefficient comparison.

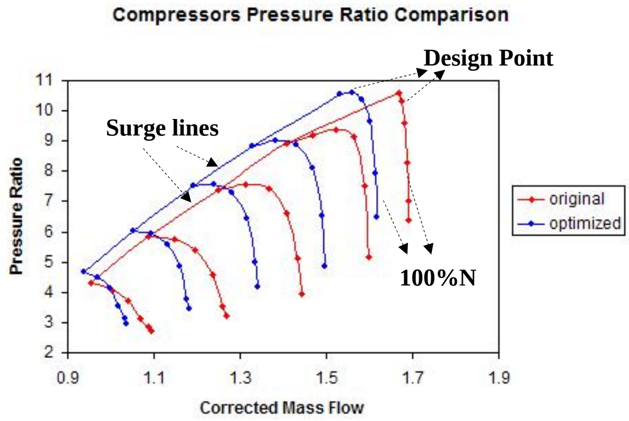
Another important point observed was the number of blades which influence the compressor weight. A reduction of 16% were obtained indicating that, even keeping the same dimensions of the channel, a notable reduction in the compressor weight is feasible which has a great influence in the fuel specific consumption.

3. Off-Design performance analysis

Once the optimized compressor has been designed, its performance was calculated at different speeds, that is, at part load operation. The design rotational speed is 19600 rpm (100%N).

At low speeds stall and choke limits are close. To increase the surge line (stability) and to increase the operational ranges, inlet guide vane (IGV) and variable geometry (VG) were used (Tomita, 2003).

Figure 14 and Fig. 15 show the original and optimized compressor comparison through their performance curves (map).



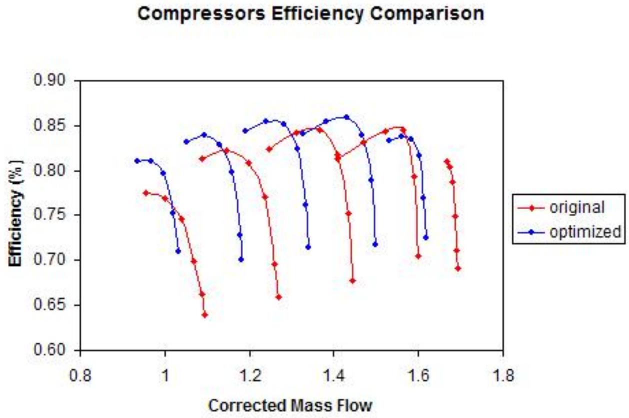


Figure 14. Compressor pressure ratio with IGV and VG

Figure 15. Compressor efficiency with IGV and VG

Figure 14 shows the displacement of the surge line upward (red line to blue line), indicating that the optimized compressor has a better performance than the original one. The upward displacement of the surge line was possible due to use of the inlet guide vane (IGV) and variable stators (VG) although the stator equipped with variable geometry are more expensive and heavier due to the complexity of the additional systems (Tomita and Barbosa, 2003). Figure 15 shows the improvement of the optimized compressor efficiency.

4. Conclusion

The use of local optimization method (Sequential Quadratic Programming) to improve the efficiency at the design point of a known geometric eight-stage axial flow compressor was performed. The main geometric and aerodynamic parameters from both compressors, original and optimized, were calculated. The increase in the efficiency and the weight reduction of the optimized compressor along with its surge line improvement (off-design performance) without compromising the critical geometric and aerodynamic parameters have shown the advantage of the optimization method.

Although the global search methods are more robust than the local one used in this work, the SQP method has shown fast convergence, this being credited to the readiness of the initial values.

The use of the streamline curvature program (Barbosa, 1987) is a natural next step to check the flow at other streamlines, in addition to the mid-height.

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