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ONE-DIMENSIONAL PRELIMINARY CENTRIFUGAL COMPRESSOR DESIGN TOOL FOR UNDERGRADUATE COURSES

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Abstract. *The design of centrifugal compressors is a vast subject in several industrial applications that can be approached in different ways and from the perspective of different design methodologies. Naturally, because of this multitude of different approaches, it makes the design teaching task of these machines labourious, and students may have many doubts in choosing the best option. As an attempt to assist the students in the design process, this paper proposes a methodology that was implemented in a tool capable of assisting in the design teaching of these mechanisms. A one-dimensional computational code using Python language was written, with the addition of appropriate loss modeling to enhance the machine prediction at design-point. The developed computational code is based on examples from open literature, in order to lead to a step-by-step design procedure. Also, the interface has several graphic interfaces that help the designer understanding in each of the design stages, availability for the user to work with different slip factor models and other considerations on the losses associated with these machines. Moreover, it is flexible to implement new subroutines to improve its capabilities. To enable the student himself to self-assess whether the data being entered is compatible with good design practices, several alerts have been programmed to guide the user about possible problems related to its sizing, as an example. It is possible to vary one of the design parameters, so that it is possible to analyze its influence on the performance characteristics of the machine. Using the developed tool, the students will be able to better understand the design process of a centrifugal compressor, developing it for a required condition.*

Keywords: *Centrifugal Compressors, One-Dimensional Toll, Preliminary Design*

1. NOMENCLATURE

C_f	Flow Coefficient
D	Diameter
D_e	Eye Diameter at the Impeller Inlet
D_f	Hub Diameter at the Impeller Inlet
D_h	Hydraulic Diameter
F	Shape Factor
H	Constant Factor between 1.18 and 1.30
L_b	Impeller Flow Length
P	Pressure
Re	Reynolds Number
S	Gap between Blades and Shroud
T	Temperature
U	Tangencial Velocity
V	Absolute Velocity
W	Relative Velocity
Z	Number of Blades
b	Blade Height
c_p	Constant Pressure Heat
f_{in}	Incidence Factor
r	Radius
q	Representation of the Head as a Heat
s	Pitch at the Blade Exit
t	Thickness of the Blade
α	Flow Angle
β	Relative Flow Angle
$d\beta/dm$	Blade Turning Rate
γ	Specific Heat Ratio
δ	Ratio between S and b
θ	Momentum Thickness
ε_{lim}	Impeller Radius Ratio Limit
μ	Slip Factor
μ_{radial}	Slip Factor decrement due to the rotation effect
μ_{turn}	Slip Factor decrement due to the blade turning
σ	Solidity Factor

Subscripts

avg	Average
b	Reference to the Blade

cl	Reference to Clearance Loss	*th*	Reference to the Theoretical Head Available
db	Reference to Diffusion Blade Loss	*vd*	Reference to Vaned Diffuser Loss
df	Reference to Disc Friction Loss	*vs*	Reference to Vaneless Diffuser Loss
ex	Reference to Exit Loss	θ	Tangencial Component
in	Reference to Incidence Loss	*0*	Reference to Total Conditions
m	Refecence to Meridional Line	*1*	Impeller Inlet
r	Radial Component	*2*	Impeller Outlet
rw	Reference to Recirculation and Wake Mixing Loss	*3*	Diffuser Inlet
		4	Diffuse Outlet
sf	Reference to Skin Friction Loss	∞	Reference to Non Slip Condition

2. INTRODUCTION

The design of a high performance aeronautical gas turbine necessarily depends on the degree of performance of the compressor, the pressure ratio supplied and the efficiency of the cycle, with low specific fuel consumption (Saravanamuttoo et al., 2001). However, the design of modern compressors is characterized by a complex process, developed through several steps and with help of advanced computing techniques (Mikhailov et al., 2016).

Some design techniques and methods for optimizing compressor performance are provided by the literature, such as the design method cited by Cheng et al. (2019), in which the design process of an axial compressor follows a sequential analysis of one dimension, then two dimensions and, finally, three dimensions. The authors also state that compressor optimization methods have been developed since 1980.

In a centrifugal compressor project, for example, the first stage uses one-dimensional numerical methods, called midline models. These methods solve the governing equations in the medium flow line, making possible a preliminary analysis of compressor performance in a relatively short time, in relation to computational fluid dynamics techniques (Meroni et al., 2008). Harley et al. (2012), states that the most suitable method for the preliminary design process of centrifugal compressors is the single zone 1D.

The performance of a centrifugal compressor is subject to phenomena described by Aungier (2000) and Boyce (2003), such as: angle of incidence, diffusion, shock waves, friction, top clearance, critical Mach number, among others, responsible for reduce compressor efficiency and, consequently, increase operating costs.

Zheng et al. (2017) studied, through experiments, special and artificial flow modes and their effect on the volute of a centrifugal compressor at different speeds of rotation. In addition, Tamaki (2017) investigated the influence of the angle of incidence of the diffuser blade on the flow and its effects on the performance of the turbomachine. Studies have also shown the effects of hysteresis on a multistage compressor and its characteristics during the surge (Munari et al, 2017).

According to Mikhailov et al. (2016), CFD (Computational Fluid Dynamics) techniques have been widely used by designers, which makes the compressor design process increasingly dependent on high-capacity data processing computers. According to the same authors, one of the most famous analytical and empirical methods for evaluating the performance of axial compressors is the method called stage-stacking, developed by R. Howell et al. 1978. The streamline curvature (SLC) method, presented by Novack (1967) in the late 1960s, has also become well known and used by designers.

Many studies are also carried out with the purpose of preventing and reducing the effects of non-uniform flow at the compressor inlet, as in the Moore-Greitzer model, cited by Lin et al. (2017).

This work aims to present a didactic program for the preliminary design of centrifugal compressors, addressing the main parameters and design methodology, in a way that can be used mainly as an academic teaching tool.

3. CENTRIFUGAL COMPRESSORS

Compressors are mechanisms used to raise the pressure of a working fluid. There are different types of compressors available on the market, for example, the compressor of a refrigerator. In a refrigerator, generally, a crank connecting system is used so that the working fluid is compressed. This compressor is of the positive displacement type, and is not part of our object of study, however, it helps us to understand the general function of the compressors.

Compressors can be divided into two large groups: positive displacement or continuous flow compressors. As for the continuous flow compressors, these can still be divided into axial or centrifugal, and the distinction between each of these basically lies in the path of the working fluid. In the case of axial compressors, the working fluid follows a path in the axial direction of the turbomachinery. For centrifugal compressors, the fluid enters the turbomachine in the axial direction, but its output is in the radial direction of the turbomachinery, and these compressors are of interest in this work (Boyce 2003).

Analyzing the working principle of centrifugal compressors, these machines are composed of a static housing, which contains an impeller inside. This impeller contains several divergent passages, bounded by structural blades. In these defined regions, the fluid will undergo a first rise in static pressure, and there will be an increase in its absolute speed. At the exit of the impeller, the fluid will find a region with fixed diverging passages, and in this region the fluid will be decelerated and there will be a second increase in static pressure (Saravanamuttoo et al., 2001). Figure 1 shows a representation of the centrifugal compressors and the increase in static pressure throughout the stage (Boyce, 2003).

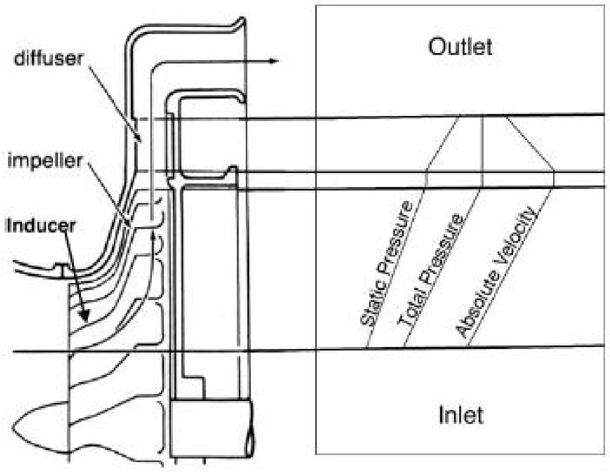

Figure 1. Behavior of a Centrifugal Compressor (Boyce, 2003).

4. CALCULATION PROCEDURE

The programmed model follows the steps presented in the example in Chapter 4 of the book Gas Turbine Theory by Saravanamutto et al. (2001). In this example, based on some input parameters defined by the author, the characteristics of the centrifugal compressor are calculated, and then the author uses some more parameters to determine the dimensions of the compressor diffuser.

In order to complement the code, 5 slip models were added to the code (in addition to keep the slip factor constant). These slip models were obtained from Boyce (2003) and Qiu et al. (2007).

The loss models were implemented through two references, Boyce (2003), and Velásquez (2017). This decision was made because many of the equations presented by Boyce (2003) are not clear or some of the variables are not defined. Based on Velásquez (2017), losses due to the formation of shock waves were disregarded.

As for the code calculation structure, the following steps are taken: determining the properties of the impeller; calculation of the diffuser dimensions; estimate of adiabatic losses. At each of the stages, the user will be able, through buttons present in the interface, to know more about each of the necessary parameters for the calculations. The user will also be alerted if any calculated parameters exceed any limits established by good design practices. Figure 2 shows a flow chart with the code calculation procedure.

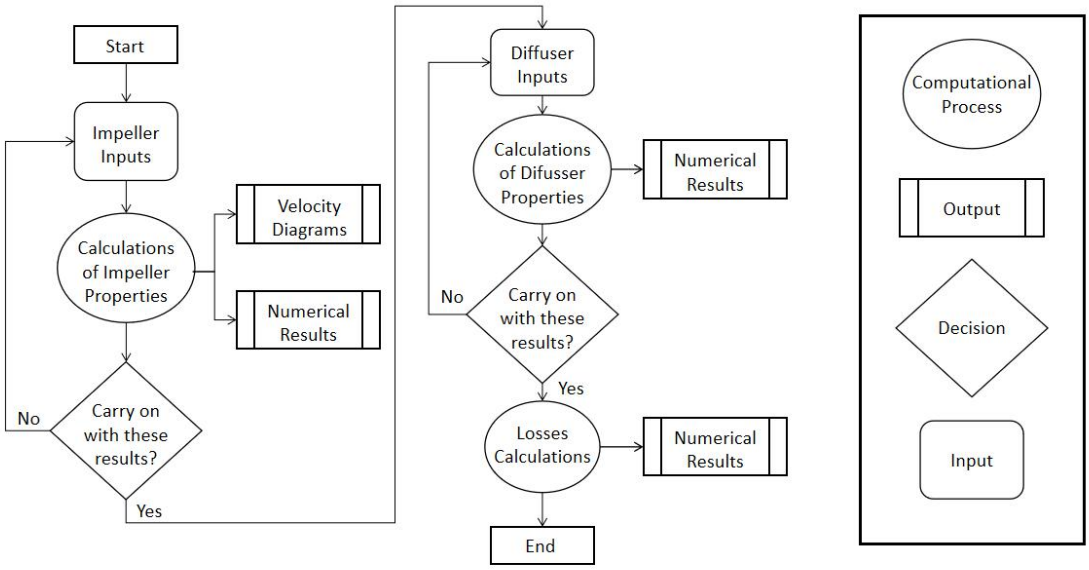

Figure 2. Calculation procedure of the design tool.

5. SLIP FACTOR MODELS

The fluid that passes through the impeller is not directed exactly with the physical angle of the blades of this component, so that the tangential speed of the absolute speed that the working fluid will actually experience will be less than the tangential speed if it follows the physical angle of the shovel. The slip factor is then defined as the ratio between these two speeds, as shown in Eq. (1).

$$\mu = \frac{V_{\theta 2}}{V_{\theta 2\infty}} \tag{1}$$

As this is a factor that directly affects the performance of centrifugal compressors, it has become the object of study in different works, thus creating different correlations that can be used to try to predict the behavior of this parameter.

In the code developed for this work, in addition to the possibility of considering the slip factor as a fixed value, the program user will be able to choose between 5 different models for determining this property, these being the models of: Stodola, Wiesner, Stanitz, Balje and NREC. It must be kept in mind that each of these correlations was developed for specific cases. It is important before choosing which correlation, to consider the experimental conditions in which they were obtained.

Stodola's formulation considers that the working fluid has an angular velocity relative to the movement of the impeller, and that it is this relative velocity that causes the appearance of the slip (Boyce, 2003). Equation (2) shows Stodola's correlation.

$$\mu = 1 - \frac{\pi}{Z} \cdot \left[1 - \frac{\sin \beta_2}{\frac{V_{m2} \cdot \cot \beta_2}{U_2}} \right] \tag{2}$$

Stanitz developed his correlation based on experimental tests using eight different impellers. Through the obtained results, it was noticed a relation between the value of the Slip Factor with the peripheral speed of the impeller, number of blades and angle of exit of the rotor blade (Boyce, 2003). The proposed correlation is presented in Eq. (3).

$$\mu = 1 - \frac{0.63 \cdot \pi}{Z} \cdot \left[1 - \frac{1}{\frac{W_{m2} \cdot \cot \beta_2}{U_2}} \right] \tag{3}$$

The correlation proposed by Balje was developed using impellers with axial inlet and radial outlet (Boyce, 2003), that is, the influence of the blade outlet angle is not considered in this correlation. Equation (4) shows this correlation.

$$\mu = \frac{1}{1 + \frac{6.2}{Z \cdot \left(\frac{D_2}{D_{1,m}} \right)^{\frac{2}{3}}}} \tag{4}$$

Wiesner proposed a correlation for the calculation of the Slip Factor based on different methods previously used. In his conclusions, the author classifies what would be the best method, among those analyzed in his work, for determining this parameter. In the proposed correlation, a new parameter, called the impeller radius ratio, is introduced, so that the way the slip factor is calculated depends on this parameter.

For $r_1/r_2 \leq \varepsilon_{lim}$, the Wiesner correlation will be given by Eq. (5).

$$\mu = 1 - (\sqrt{\cos \beta_{2b}} / Z^{0.7}) \tag{5}$$

For $r_1/r_2 > \varepsilon_{lim}$, Wieser proposes Eq. (6).

$$\mu = 1 - \left(\frac{\sqrt{\cos \beta_{2b}}}{Z^{0.7}} \right) \cdot \left[1 - \left(\frac{\frac{r_1}{r_2} - \varepsilon_{\lim}}{1 - \varepsilon_{\lim}} \right)^3 \right] \tag{6}$$

ε_{lim} is calculated using Eq. (7).

$$\varepsilon_{\lim} = \frac{1}{\ln^{-1}(8.16 \cdot \sin \beta_{2b} / Z)} \tag{7}$$

The last model used in the code was published by Qui et al. (2007), representatives of Concepts NREC. The proposed model was developed to be able to be applied to predict the slip factor for axial, radial and mixed flow rotors. The model proposed by the authors is based on the loading of the paddle, and the difference in speed between the pressure and suction sides of the paddles. Equations (8) to (12) show how the Slip Factor is determined through the proposed correlation.

$$F = 1 - \sin \frac{\pi}{Z} \cdot \left(\sin \frac{\pi}{Z} + \beta_{2b} \right) \cdot \cos \beta_{2b} \cdot \sin \alpha_2 - \frac{t_2}{s \cdot \cos \beta_{2b}} \tag{8}$$

$$s = \frac{2 \cdot \pi \cdot r_2}{Z} \tag{9}$$

$$\mu_{radial} = \frac{F \cdot \pi \cdot \cos \beta_{2b} \cdot \sin \alpha_2}{Z_2} \tag{10}$$

$$\mu_{turn} = \frac{F \cdot s \cdot \frac{V_{r2}}{U_2}}{4 \cdot \cos \beta_{2b}} \cdot \left(\frac{d\beta}{dm} \right)_2 \tag{11}$$

$$\mu = 1 - \mu_{radial} - \mu_{turn} \tag{12}$$

6. CENTRIFUGAL COMPRESSOR LOSSES MODELS

The energy losses present in a centrifugal compressor must be known so that it is possible to determine the performance parameters of the turbomachine. This task is as important as assessing the blade-loading parameters, since the energy losses are directly related to the adiabatic efficiency of the compressor.

These losses are generally divided into external, in the impeller and in the diffuser.

6.1 External Losses

These losses are generally divided into two categories: Clearance Loss and Disc Friction Loss.

As for Clearance Loss, these are basically related to the flow that crosses the region that comprises the top clearance, located between the rotor and the centrifugal compressor housing. The way used to estimate these losses in the developed code, is presented in Eq. (13) (Boyce, 2003).

$$\Delta q_{cl} = 0.17 \cdot q_{th}^{\delta} \tag{13}$$

The parameter δ is estimated by Eq. (14).

$$\delta = S / b \tag{14}$$

Regarding Disc Friction Loss, these losses are caused by flow in the rotor backface, where torque is generated. Losses in seals, bearings and gearbox can also be considered in this category. These losses are calculated using Eq. (15) (Boyce, 2003).

$$\Delta q_{df} = \frac{C_f \cdot \left(1 + \frac{P_2}{P_1}\right)}{2 \cdot \left(\frac{W_{r1}}{U_2}\right) \cdot q_{th} \cdot \left(\frac{D_2}{D_e}\right)^2 \cdot \left(1 - \left(\frac{D_h}{D_2}\right)^2\right)} \tag{15}$$

Where the coefficient C_f is determined through Eq. (16) in case the flow is turbulent, and by Eq. (17) in case the flow is laminar.

$$C_f = 0.0622 \cdot \mathrm{Re}^{-0.2} \tag{16}$$

$$C_f = 2.67 \cdot \mathrm{Re}^{-0.5} \tag{17}$$

6.2 Impeller Losses

Impeller losses are classified into four categories, namely: Shock losses, Incidence Loss, Diffusion Blade Loss and Skin Friction Loss.

Regarding the Shock in rotor losses, these losses occur at the entrance of the rotor, and are not considered in the developed code.

The Incidence Loss is caused by an operation outside the design point. A positive incidence causes a reduction in flow, due to the change in speed. Although the work does not consider the calculation in conditions outside the Design Point, these losses are still considered, and their contribution is calculated through Eq. (18) (Velásquez, 2017).

$$\Delta q_{in} = f_{in} \cdot \frac{W_1^2}{2 \cdot U_2^2} \tag{18}$$

Since f_{in} is a factor that can vary between 0.5 and 0.7. For this work it was used as being 0.6.

Regarding the Diffusion Blade Loss, it develops due to the negative speed gradient in the boundary layer and recirculation of the flow in the impeller. The flow deceleration increases the thickness of the boundary layer and can cause the flow to separate. The adverse pressure gradient in the compressor impeller is unfavorable to the flow, which increases the chance of separation occurrence and consequent increase in losses. This loss is calculated using Eq. (19) (Boyce, 2003).

$$\Delta q_{db} = \left(\frac{2 \cdot \theta \cdot \sigma}{\sin \beta_1}\right) \cdot \left(1 + \frac{\theta \cdot \sigma \cdot H^2}{2 \cdot \sin \beta_1}\right) \tag{19}$$

Finally, the Skin Friction Loss, is the loss generated due to the shear forces acting on the rotor wall. This loss is determined through Eq. (20) (Velásquez, 2017).

$$\Delta q_{sf} = 5.6 \cdot C_f \cdot \frac{L_b \cdot V_2^2}{D_h \cdot U_2^2} \tag{20}$$

6.3 Diffuser Losses

As with impeller losses, losses in the diffuser region are divided into four categories: Recirculation and Wake Mixing Loss, Vaneless Diffuser Loss, Vaned Diffuser Loss and Exit Loss.

Recirculation and Wake Mixing Loss originate at the impeller outlet, where a recirculation mat is formed when the flow begins to enter the diffuser region. They are estimated through Eq. (21) (Boyce, 2003).

$$\Delta q_{rw} = 0.02 \cdot D_f^2 \cdot (\tan \alpha_2)^{0.5} \tag{21}$$

Regarding the Vaneless Diffuser Loss and the Vaned Diffuser Loss, these are estimated, generally, through empirical relationships obtained from experimental trials. For the determination of Vaneless Diffuser Loss, Eq. (22) (Boyce, 2003) is used. For Vaned Diffuser Loss, Eq. (23) (Velásquez, 2017) is used.

$$\Delta q_{vs} = \frac{4 \cdot C_f}{12} \cdot \frac{D_2}{b_1} \cdot \frac{1}{\cos \alpha_{avg}} \cdot \left[1 - \left(\frac{D_2}{D_3} \right)^{1.5} \right] \tag{22}$$

$$\Delta q_{vd} = \frac{c_p \cdot T_{02}}{U_2^2} \cdot \left[\left(\frac{P_4}{P_{04}} \right)^{\frac{\gamma-1}{\gamma}} \cdot \left(\frac{P_4}{P_{03}} \right)^{\frac{\gamma-1}{\gamma}} \right] \tag{23}$$

Finally, in relation to Exit Loss, this is estimated to be half the kinetic energy at the outlet of the diffuser. An approximation for this loss is given by Eq. (24) (Boyce, 2003).

$$\Delta q_{ex} = 0.25 \cdot V_4 / U_2^2 \tag{24}$$

7. THE GRAPHICAL USER INTERFACE (GUI) DEVELOPED

The computational code was developed using the programming language Python. The choice of this language to carry out the necessary tasks is due to the fact that it is an open source language, with a large community of users, and which is currently in constant use both in the academic community and outside.

The developed GUI allows the user to perform two different tasks. The first one is intended to show some aspects of the design of centrifugal compressors, and the second allows the user to perform a sensitivity analysis, thus being able to verify the behavior of some performance parameters of the turbomachine with the variation of an input parameter.

7.1 Centrifugal compressor sizing module

As already mentioned in item 4 of this document, the user will have a program that will assist him in the design of centrifugal compressors, based on a simple step by step. In certain situations, the program will indicate to the user if he is exceeding any limit established by good design practices, so that he/she makes a decision as to whether to continue with the process, or modify any input parameter in order to respect these limits.

Figure 3 shows an overview of the first interface window for the design of the centrifugal compressor. In this first window, the user must enter some input parameters for the design of the impeller of the centrifugal compressor. Next to each of the spaces for the input parameters, there are buttons for the user to activate if he/she has any kind of doubt about that parameter. An example of an explanation of one parameter is found in Fig. 4, showing the explanation of the Slip Factor entry.

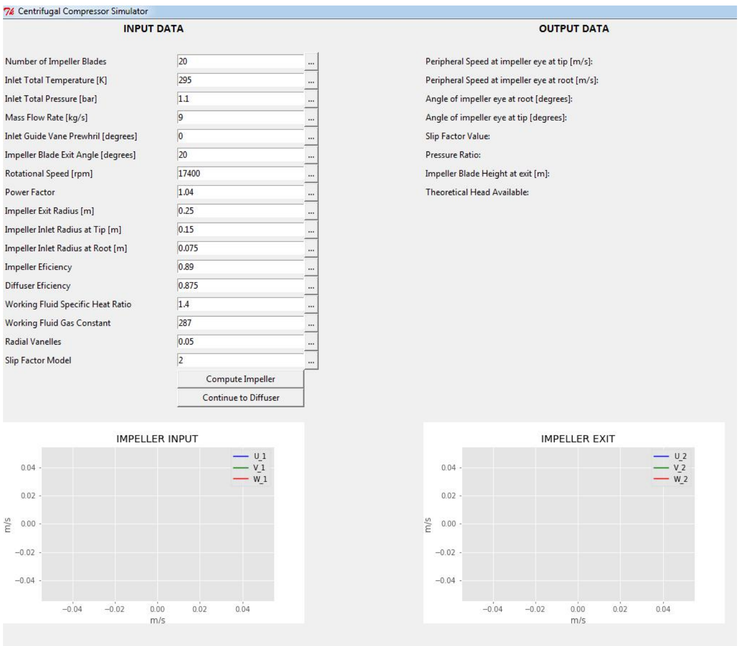

Figure 3. First window of the centrifugal compressor design tool.

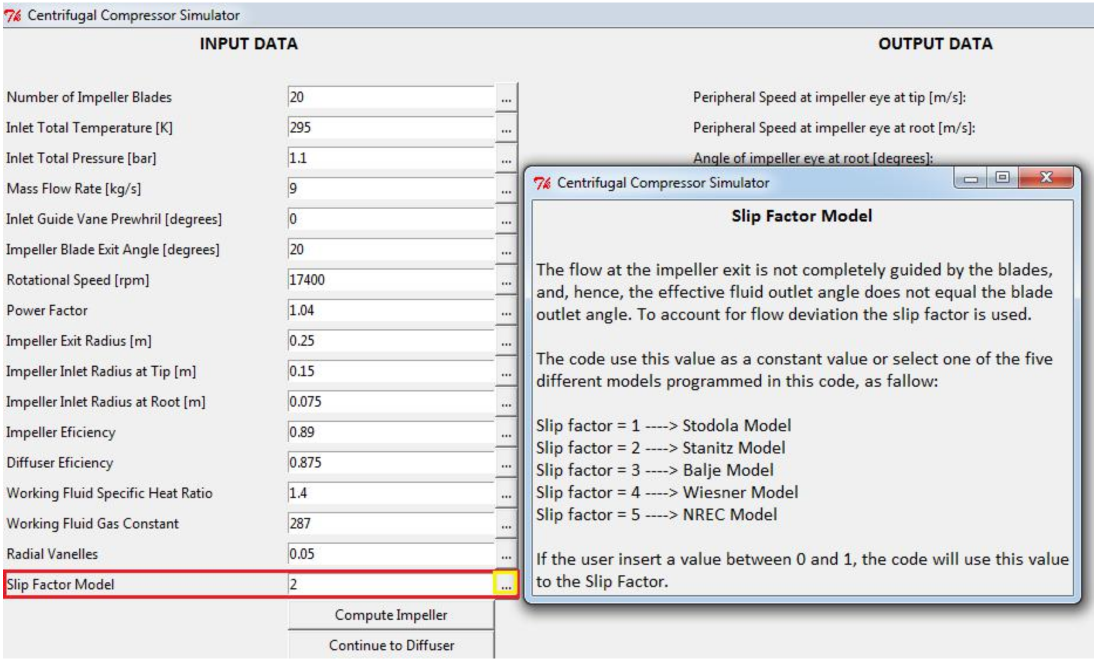

Figure 4. Example of an explanation of one input parameter.

After defining these parameters, the user must calculate the impeller properties, and as a result he/she will obtain a list of output parameters related to the flow, in addition to two graphs representing the speed triangles of the impeller inlet and outlet. These output results are shown in Fig. 5.

Figure 5. Impeller output data.

At this moment, if any output parameter is outside the limits established in the code development, it will be highlighted with a red fill, and a button will be positioned beside it. This button can be activated by the user, opening a page with a brief description of the recommended limit of this parameter, and what would be an adequate working range. This characteristic is shown in Fig. 6, showing a calculation where the pressure ratio has exceeded the limits of good design practices.

In the situation shown in Fig. 6, the user must decide whether to modify the inputs and recalculate the impeller, or whether to proceed to the diffuser calculation step, and then to the adiabatic loss calculation step. Both of these steps, calculating the diffuser and adiabatic losses, have procedures similar to those already shown for calculating the impeller. An overview of these last two windows is shown in Fig. 7.

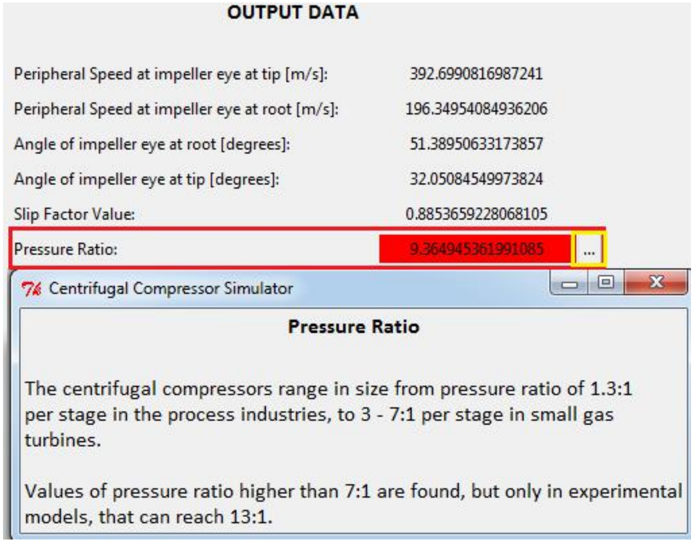

Figure 6. Example of an output parameter that exceeded the limits of good design practices.

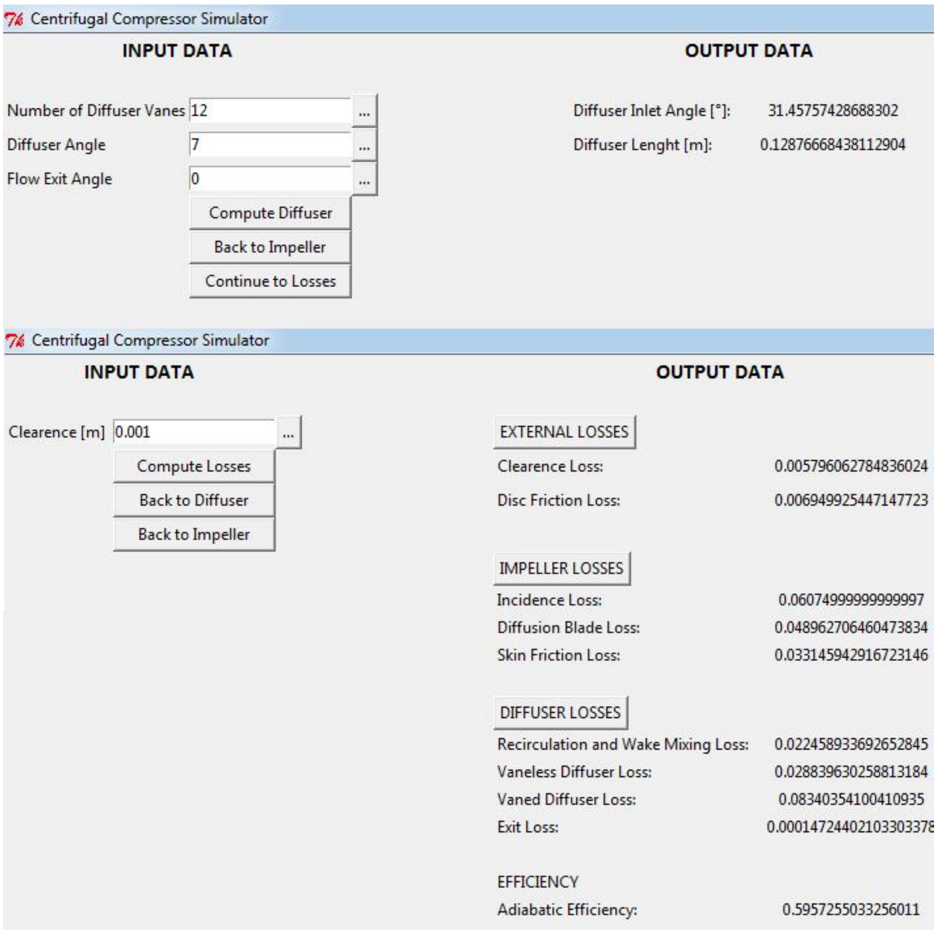

Figure 7. Overview of the last two windows of the software.

7.2 Sensitivity analysis module

This second analysis that can be done by the user of the GUI is intended to analyze the sensitivity of the input parameters, in the behavior of the turbomachine. With this module, the user will be able to choose between one of the input parameters, and vary it within a specified range, having to also provide the number of points to be calculated within this range. This program module was created so that the student can identify the effect of the variation of each of the input parameters on the performance of the studied compressor. This analysis, however, does not represent an evaluation in conditions outside the design point, but a study of the sensitivity of the input parameters. Figure 8 presents an overview of this GUI module showing also an analysis of the compressor pressure ratio when the rotational speed is varied.

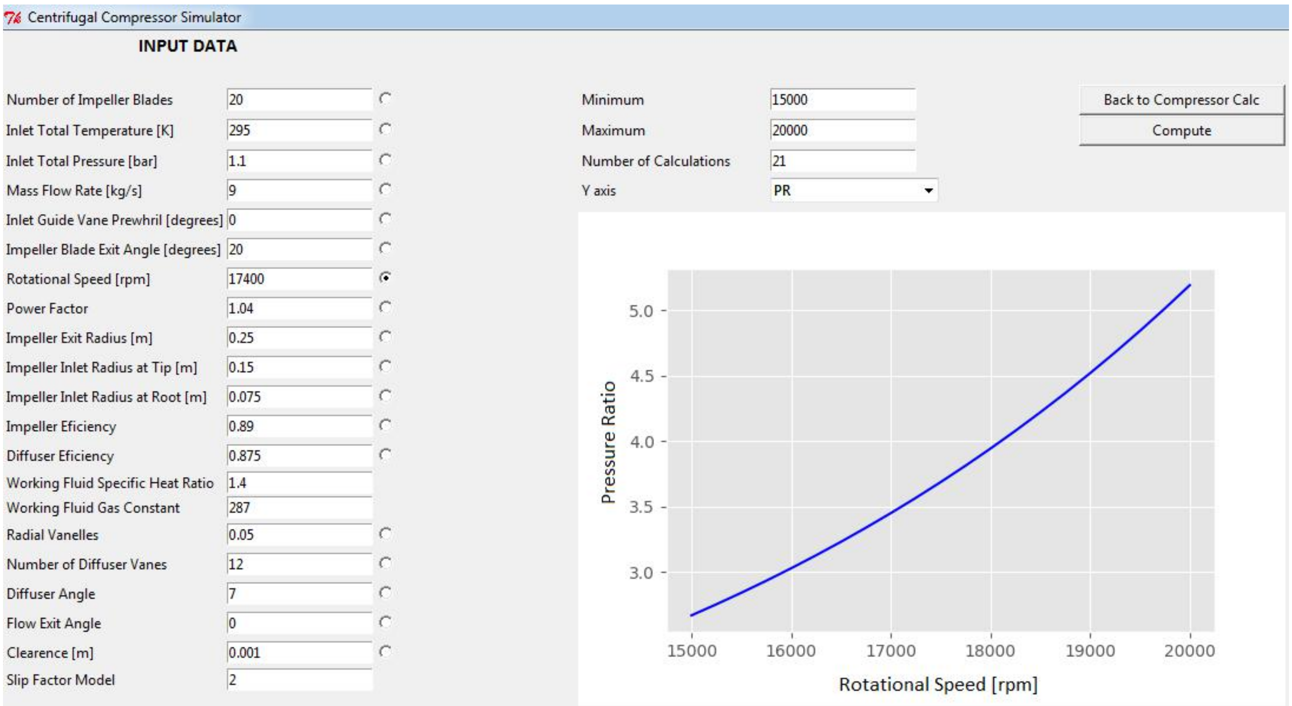

Figure 8. Overview of the Sensitivity Analysis Module.

8. CONCLUSIONS

The work was developed in order to create a digital tool capable of assisting in the design of centrifugal compressors. The code was developed considering examples available in the literature, in addition to considering in its calculation procedure different models of Slip Factor and adiabatic losses.

It is possible to conclude that, even though it is a tool that is based on a simple methodology, it would be able to assist in the learning of undergraduate students, through some resources present within the developed GUI, so that the user can understand and make decisions when designing these turbomachines.

The authors of the work believe that through a code similar to this it would be possible to make the teaching task more dynamic, proposing, for example, that students use the program to design a compressor that must meet certain requirements. In addition, it is believed that it is possible to reduce the number of hours needed to understand the content.

Despite presenting a good solution, mainly for didactic purposes, the developed GUI is still very rudimentary, and needs to be improved to make it more friendly to the user, and perhaps more intuitive.

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11. RESPONSIBILITY NOTICE

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