Numerical representation of three stages axial compressor performance map used for small turbojet engine.

Khaled I. Azzabi[1]; Ahmed M. Algadal[2]; Aiman Elmahmodi[3]; Mohamed T. Swei[4]
[3]Aeraonutical department, Faculty of engineering- University of Tripoli
[1,2,4]College of engineering technology-Janzour
azzabikh@cetj.edu.l
aljedo2000@yahoo.com

Abstract.
A numerical representation of three stage axial flow compressor performance map used for a small turbojet engine was developed and verified experimentally. It well known that, the compressor map in the standard format cannot be used directly in a performance calculation program; accordingly in the present work a successful development of non-linear model of a turbojet engine depends largely on the establishing of a consistent compressor performance map that will guaranteeing acceptable accuracy. In the present work the transformation functions of the three stages axial flow compressor performance map for the small turbojet engine are developed to model the non-linear thermodynamic relations obtained from test results.
Keywords: Numerical representation, Axial flow compressor, Compressor performance map, Compressor computational module, Turbojet engine, compressor characteristic map, Working point

Nomenclature

n	Rotational Shaft Speed [rpm]
h	enthalpy [J/kg]
P	Pressure [Pa]
T	Temperature [K]
m	Mass Flow rate [kg/s]
s	entropy[J/kg]
Suffixes	
in	Compressor Entry
ref	Reference Condition
d	Design point
r	referred value

Introduction

Gas turbine simulation software utilizing object oriented programming has been developed with extended capabilities [1, 2], and thus the engine performance simulation that can be used directly in control systems development has to be fast and accurate [3].
Once the component geometry has been fixed at the design point then the map may be generated to define its performance under all off design conditions. Compressor map in the standard format cannot be used directly in a performance calculation program. Numerical representation of compressor map with acceptable accuracy is integrated with the TMM25 turbojet engine mathematical model [4].The map presents the characteristics of the compressor for the entire working regime. Values of the map must accurately represent the component and to be easily introduced to the simulation computer program [6]. The compressor map usually, shows the performance curves of a given compressor, in which pressure ratio, and isentropic efficiency, are plotted against referred mass flow for fixed values of non-dimensional speed [7].

The Compressor

In the axial compressor the fluid passes through a succession of expanding passages with consequent reduction in velocity. The flow is decelerating or diffusing and the pressure rises as it progresses through the compressor blades, the performance of the compressor will largely depend on its design and operating regime. As shown in the figure 1, the compressor increases both the pressure and temperature of the air (2-3). The work input to the compressor is used to raise the pressure ratio, and the temperature rise depends on the loss during compression process. Compressor isentropic efficiency is expressed in terms of the ratio of ideal to actual work , and the compressor pressure ratio is defined as:

$$\pi_c = \frac{P_3}{P_2} \tag{1}$$

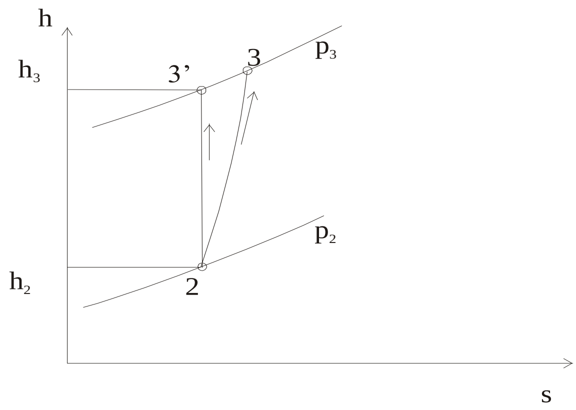

Figure 1. h-s diagram for axial compressor

Compressor performance map

The characteristic performance maps of the compressor, being plotted on non-dimensional basis, pressure ratio, and isentropic efficiency, against the referred non-dimensional mass flow for fixed values of the non-dimensional speed. Once the compressor geometry has been fixed at the design point then the compressor Map generated to define performance under all off design conditions.

Parameter groups

Non-dimensional corrected parameter group directly proportional to quasi-dimensional groups. The compressor test data is referred to standard condition with non-dimensional group for inlet temperature and pressure as:

$$\theta = \frac{T_{in}}{T_{ref}} \tag{2}$$

$$\delta = \frac{P_{in}}{P_{ref}} \tag{3}$$

Hence, referred mass flow is defined as:

$$m_r = m\sqrt{\theta}/\delta \tag{4}$$

And referred speed is defined as:

$$\bar{n}_r = \frac{n/\sqrt{\theta}}{n_d} \tag{5}$$

The use of these parameter groups allows describing numerically component performance throughout the operational range. This is needed where the change of ambient conditions, linear scales of the compressor and change of working fluid are considered.

Compressor isentropic efficiency is expressed in terms of the ratio of ideal to actual work. The following relation gives the isentropic efficiency:

$$\eta_c = \frac{\pi_c^{\frac{\gamma-1}{\gamma}} - 1}{T_3/T_2 - 1} \tag{6}$$

The general form of equation for compressor map is given by:

$$\pi_c = f\left(\bar{n}_r, \bar{m}_r\right) \tag{7}$$

$$\eta_c = f\left(\bar{n}_r, \bar{m}_r\right) \tag{8}$$

Compressor Testing Facility

This is illustrated in figure 2, and consisting of electric motor to drive the compressor, a set of gearboxes, and a stand supporting the compressor. The compressor test driving motor has a maximum power of 50 kW at 2700 rpm. Turbine utilizing compressed air is connected to the compressor driving shaft, this is to cover as much as possible of the compressor map. The combustion chamber was not positioned in its volume to avoid unnecessary pressure drop. In order to extract the compressor characteristics; pressures and temperatures were measured at compressor inlet (atmospheric conditions P_a, T_a) and outlet (P_2, T_2), figure 3 shows the measurement points.

Testing procedure:

Throttling valves controlling the air mass flow rate are set at full open.

The operator sets the desired testing speed; the electric motor is fixed at the desired speed by the control unit.

The valves are slowly closed until surge occurs; the valves then slowly opened until fully opened. Meanwhile the electric motor control unit was maintained at the desired speed.

The procedure was then repeated for different rpm.

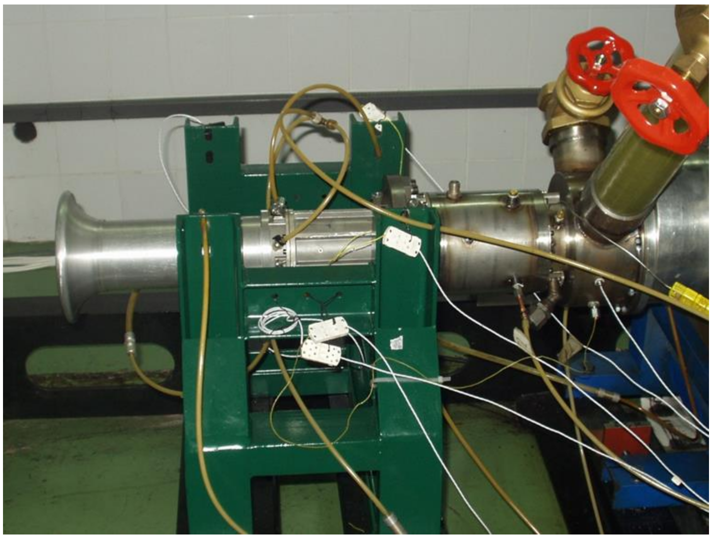

Figure 2. Compressor at testing installation

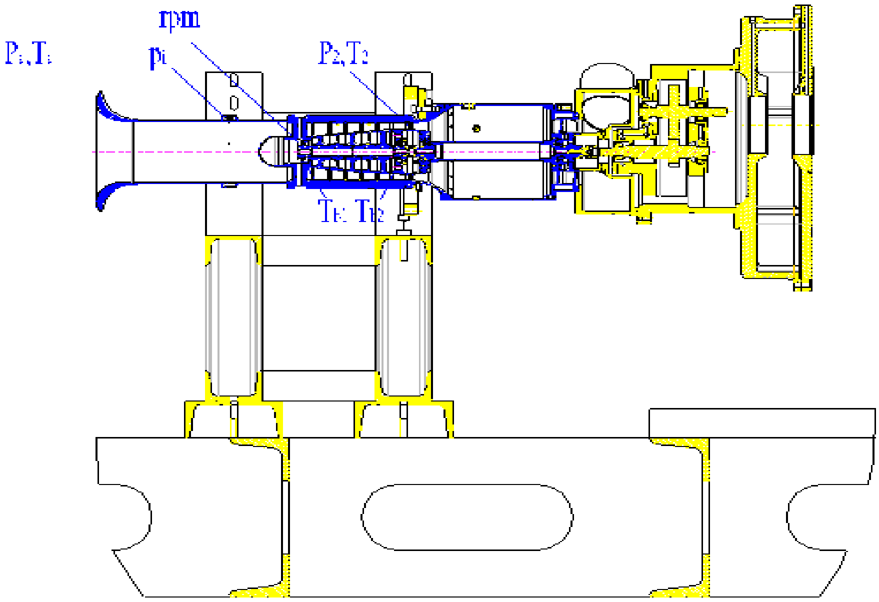

Figure 3. Compressor test measuring points

Compressor map numerical representation

The compressor map could be approximated as set of polynomials for referred mass flow and referred speed vs. both pressure ratio and efficiency. The shape of polynomials is predicted according to that obtained with Al Gammal's work [5]. The representation of the compressor map is considered as per for real time simulation of gas turbines. It was found that the transformation of coordinates of the shape with the function Y=f(X/Y), can be well approximated with second order polynomial. This polynomial was used to predict the characteristic lines of the performance map that obtained with the tests and to compute the rest of the lines that to be consistent this map. The transferred coordinate representing either pressure ratio or efficiency as appropriate is given by [4]:

$$Y = a_0 + a_1 \frac{X}{Y} + a_2 \left(\frac{X}{Y}\right)^2 \tag{9}$$

The transformation suggested for pressure ratio and referred mass flow is given by:

$$X = \frac{m_r + b_{11}}{m_{rS} + b_{12}} \quad and \quad Y = \frac{\pi_c + b_{21}}{\pi_{cS} + b_{22}} \tag{10}$$

Where $b_{11} = -0.5, b_{12} = -1.2, b_{21} = 3.5, b_{22} = 4.8$

For pressure ratio and referred mass flow the values of the coefficients of the polynomial of equation (9) are given as a function of referred speed as:

$$\begin{aligned} a_0 &= 0.52 + 1.3\bar{n}_r - 1.98\bar{n}_r^2 + 0.96\bar{n}_r^3 \\ a_1 &= -0.36 + 3.16\bar{n}_r - 6.19\bar{n}_r^2 + 3.27\bar{n}_r^3 \\ a_2 &= -0.79 + 3.45\bar{n}_r - 5.67\bar{n}_r^2 + 2.94\bar{n}_r^3 \end{aligned} \tag{11}$$

The transformation suggested for efficiency and referred mass flow is given by

$$X = \frac{m_r + c_{11}}{m_{rS} + c_{12}} \quad and \quad Y = \frac{\eta_c + c_{21}}{\eta_{cS} + c_{22}} \tag{12}$$

Where $c_{11} = 5, c_{12} = 10, c_{21} = 18.5, c_{22} = 100$

For efficiency and referred mass flow, values of the coefficients of the polynomial of equation (9) are given as a function of referred speed as;

$$\begin{aligned} a_0 &= -8.132 + 33.02\bar{n}_r - 46.41\bar{n}_r^2 + 19.35\bar{n}_r^3 \\ a_1 &= 6.19 - 24.38\bar{n}_r + 34.14\bar{n}_r^2 - 14.26\bar{n}_r^3 \\ a_2 &= -1.15 + 4.50\bar{n}_r - 6.27\bar{n}_r^2 + 2.62\bar{n}_r^3 \end{aligned} \tag{12}$$

Predicted characteristics lines are presented with measured results in figures 4 and 5. The figures show the compressor map in standard form, the pressure ratio and efficiency are plotted against relative reduced air mass flow, for constant relative reduced speed. Test converge the performance map up to 0.7 relative reduced speeds that is due to the limitation of the available power of the electric motor used through the tests.

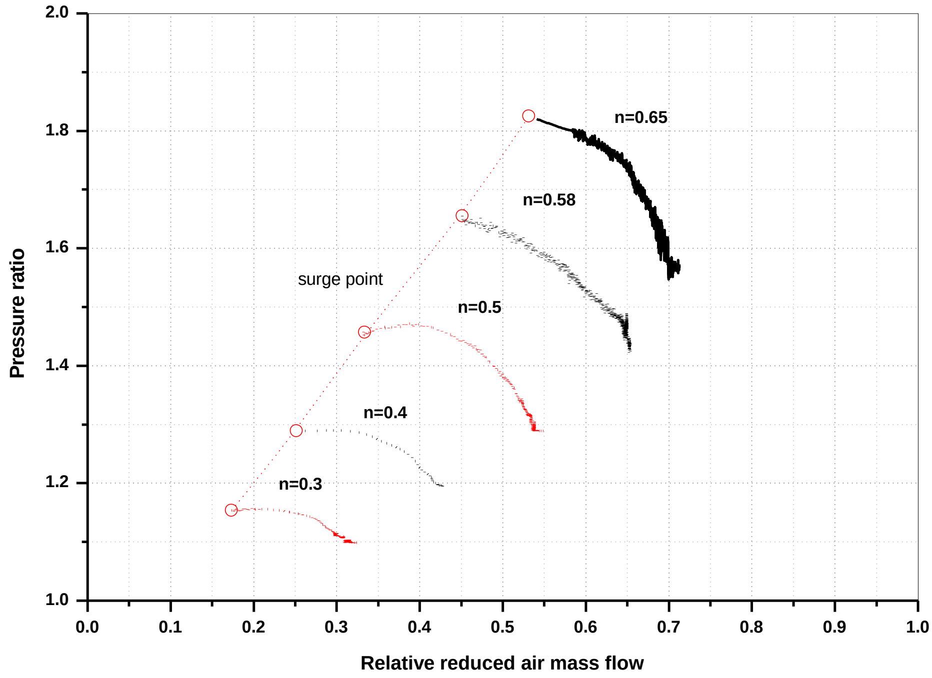

Figure 4. Measured results of compressor pressure ratio vs. relative reduced air mass flow

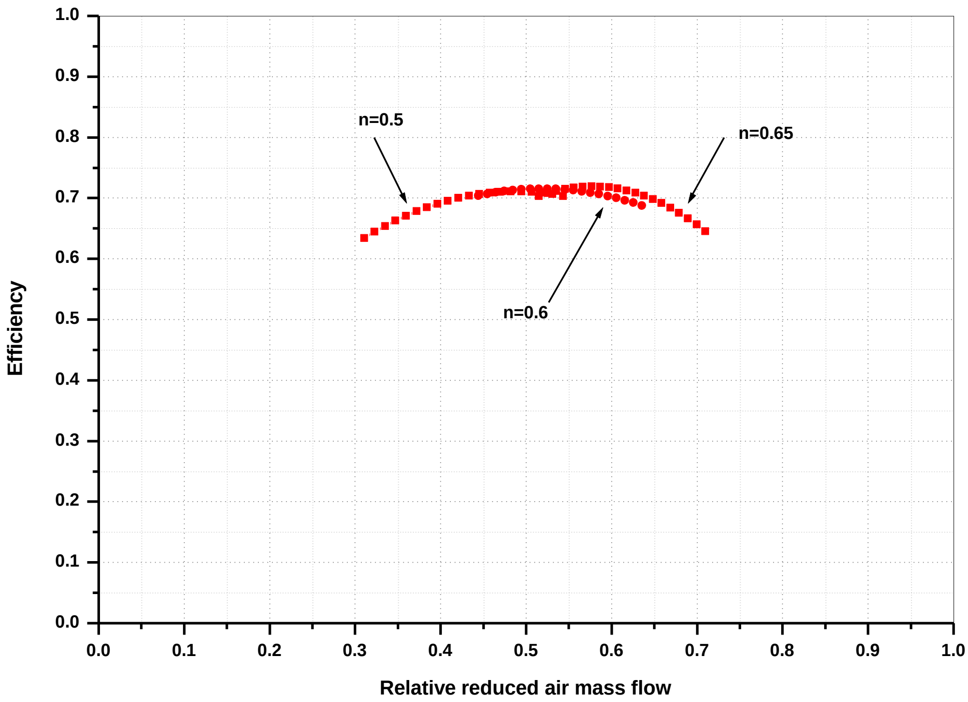

Figure 5. Measured results of compressor efficiency vs. relative reduced air mass flow

Compressor computational module

Figure 6 illustrates the computational module of compressor, the input properties of the air that calculated by the intake module will passing through the compressor, temperature and pressure will increase depending on the present work numerical represented compressor performance map, and the working point.

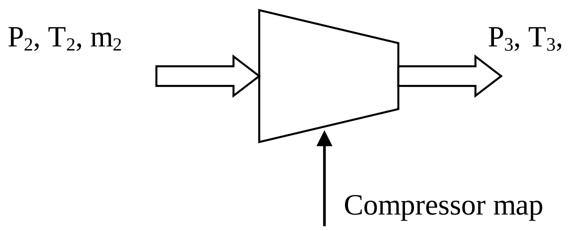

Figure 6. Computational module for compressor

Conclusions

A numerical representation of the three stage axial flow compressor performance map that will be used for the small turbojet engine was presented. Most of the compressor performance map has been obtained by the test, up to 0.7 relative reduced speeds, which is actually covers engine starting working area. Finely, the compressor map is integrated with the small turbojet engine mathematical model [4].The performance map presents the characteristics of the compressor for the entire engine working regime. Thus the computational module of the compressor will control the process, as the air passing through the compressor the temperature and pressure will increase depending on the characteristic map of compressor and the working point.

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