

Implementing Turbomachinery Physics into Data Map-Based Turbocharger Models

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ABSTRACT

A convenient way of modelling turbochargers is based on data maps. These models are easy to put into place, require low CPU charge and are control-oriented. Data relative to compressor and turbine are read from tables: pressure ratio and efficiency are determined as functions of mass flow rate and rotary speed on two distinct data maps. Nevertheless, this type of model has drawbacks:

- Usually, only higher turbocharger speed data are mapped (> 90000 rpm) although the low rpm zone is the most useful zone for normalized driving cycles simulations. Moreover, maps are poorly discretized, leading to the use of specific extra-interpolation methods (many are identified in [5]).
- These methods are purely mathematical, which gives inaccurate results in extrapolation zones. Relation between pressure ratio and efficiency is then broken (i.e., if one implements a pumping model for the compressor, the pressure ratio will be affected, but not the efficiency).

The present paper develops a new extra/interpolation model incorporating physical laws. An analysis of turbomachinery equations is performed. A new approach for extra-interpolating performance maps, which satisfies the physical laws stated in turbomachinery equations, is derived from this work. Results from this new model are compared with standard methods.

The major conclusions drawn from this study are: 1 - The new model improves the simulation accuracy while keeping the same easiness of use and robustness. 2 - Extrapolation in the low rpm zone is derived from physical equations. 3 - This method is applicable to both compressor and turbine. 4 - The pressure ratio and efficiency maps are now linked.

INTRODUCTION

In recent years, drastic strengthening of the pollutant emissions standards (EURO - Europe, TIER - USA) combined to the raising of raw materials prices have led to the reduction of vehicles fuel consumption. In the same time, engines power has increased (see as an example: Figure1), leading engine designers to use innovative techniques in the area of combustion, injection, and, of course, turbocharging. The latter is now widely used in diesel and gasoline engines to increase power output for a given engine displacement. The modelling of turbocharger is therefore a cornerstone regarding modern engines simulation. In literature, two distinct ways of modelling turbochargers can be found: the first way is essentially based on physical equations, while experimental measurements are used only to correlate the model ([2], [3], [4]). The second one uses the opposite philosophy: a standard approach appears to be to store turbocharger performance data into lookup tables and to use mathematical algorithm to interpolate - extrapolate data ([5], [6], [7], [8]). This paper will only focus on the second approach, for reasons that will be developed below. The main issue encountered using this control-oriented method is that the overall model validity is linked to the accuracy of experimental data and on the algorithm. Even though some authors state physical law of turbomachinery into their papers, in the end it appears

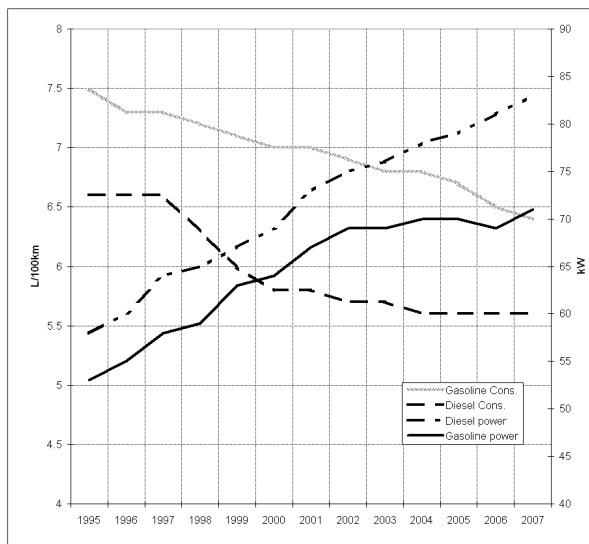


Figure 1: Engines power & fuel consumption evolution over the 12 past years in France. Source: [1]

that the turbocharger mathematical formulation is never controlled by physical laws, which leads to low accuracy and low robustness in the case of extrapolation. In this paper the basis of the physics of turbomachinery will be formulated. The goal is to incorporate these very basic physical formulations into more conventional 0d mathematical turbocharger models. Results from this new model will be compared to conventional methods and to engine bench experimental data.

SETTING THE PROBLEM OF 0D TURBOCOMPRESSOR MODEL

THE CHOICE OF 0D MODELLING As mentioned in the introduction, there are various approaches for modelling a turbocompressor. Therefore, it is necessary to take into account the way the model will be used. In this paper, we have chosen to focus on those two applications:

- A tool for the design and development of engines and its associated systems. This is a 'model-based' approach, which will allow, thanks to simulation, to predict performances and behavior of the engine. In this aim, it is necessary to get a predictive model, while keeping calculation time as low as possible. Indeed, it is often necessary to undertake multiple calculations to validate a given engine architecture.
- A tool for developing control laws of the engine. In this case, the formulation of the model must be compatible with a control laws development process (a control-oriented model).

Regarding these expectations, it seems that a 0D model is a good candidate to fulfill the needs expressed above. Indeed, it requires low calculation time, it is easy to implement and can give, if well defined, good accuracy. However, there are many parameters that will affect the validity and the accuracy of these results.

LIMITATIONS OF 0D TURBOCHARGER MODELS In [5], these issues are well defined and they will be summarized here. The first limitation comes from experimental mapping. Data maps provided by suppliers do not cover the entire operating range. This is mainly due to the fact that for very low mass flow rates, the sensors accuracy is not sufficient to provide good measurement. A second limitation is that no measurement can be made in the compressor surge zone, as this is a transient and highly unstable zone. And finally, turbocharger flow bench are steady in temperature and mass flow rate, so there will be a gap between flowbench measurements and the real performance of the turbocharger once mounted on the engine. Indeed, aerodynamics wave advection is a non-linear phenomenon, which explains why the mean-value averaging of mass flow rate and pressure won't give a correct result. Benson [12] shows that the error induced by neglecting pulses of the flow is about 10% on the turbine mass flow rate and power calculation.

To conclude this section, we can say that even though 0D turbocharger models appear to have the good structure to answer the problematic, one must keep in mind that the overall model accuracy will depend on the inter-extrapolation process and on a data map that does not take into account real engine conditions. Consequently, in the case where pure mathematical intra-extrapolation methods are employed directly on supplier data maps, the validity of the model results can be questioned. This is why a new 0D model, which incorporates physical laws, has been developed, as well as a correlation tool.

TURBOCHARGER BACKGROUND

The maximum power delivered by an engine for a given displacement is directly linked to the amount of fuel efficiently burned at each cycle inside the cylinder. This amount is limited by the quantity of air which can be introduced into the cylinders. Therefore, in order to increase the maximum power output of a given engine, the inlet air must be compressed to higher density than ambient conditions before entering each cylinder. This is shown ([9]) by the following equations (eq.1, eq.2) :

$$P = \frac{\eta_f \cdot \eta_v \cdot N \cdot V_d \cdot Q_{HV} \cdot \rho_{inl} \cdot F_{f/a}}{2} \quad (1)$$

$$T = \frac{\eta_f \cdot \eta_v \cdot V_d \cdot Q_{HV} \cdot \rho_{inl} \cdot F_{f/a}}{4 \cdot \pi} \quad (2)$$

where P and T are the power and torque output of the engine, η_v and η_f the volumetric and fuel conversion efficiencies, Q_{HV} is the lower fuel heating value, ρ_a is the inlet air density and $F_{f/a}$ is the fuel/air ratio. One can see that for the same engine architecture, turbocharging will increase the maximum power and torque by increasing ρ_a , the other parameters remaining constant.

A turbocharger is composed of a turbine, located at the exhaust, which uses exhaust gases energy to drive the compressor which compresses inlet gases. In our study we will only consider radial compressors and turbines since they are the only ones used in automotive applications.

COMPRESSOR A centrifugal compressor is composed of a static inlet, a mobile rotor with blades, a diffuser and a collector, of which the role is to bring compressed air to the engine air inlet system (fig.2). The role of the rotor is to transfer energy between the turbocharger shaft and the fluid. The flow is deflected by the blades, this means that a reaction force is applied by the fluid to the rotor. For a better understanding of the physical processes taking place in the rotor area, a h-S (enthalpy / entropy) diagram is drawn (fig.3).

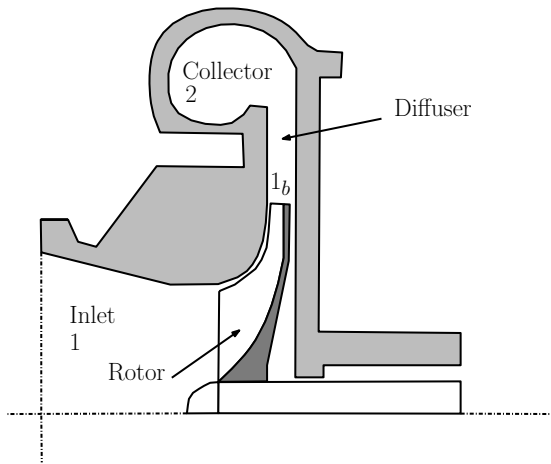


Figure 2: Side view of centrifugal compressor

Global thermodynamics Inlet air is inducted at pressure p_1 and speed C_1 . It is accelerated through the rotor up to speed C_{1b} . One can notice that static pressure p_{1b} has almost not increased (compared to p_1), but kinetic energy has increased. This means that the gain of enthalpy through the rotor is due to the acceleration of the air. After this step, the fluid needs to increase its density, i.e. its pressure. That's the role of the diffuser, where an energy transfer occurs from kinetic to potential energy: the flow speed decreases along the diffuser to reach the value of C_2 whereas the static pressure increases to p_2 . At the final state, p_2 is almost equal to the final total pressure p_{02} . Entropy is created during the compression, which means that there are losses. Consequently a centrifugal compressor can be characterized by two parameters: the amount of energy transferred from the rotor shaft to the fluid (which is directly linked to the pressure drop across the compressor) and the efficiency of this process. The first law of thermodynamics put this behavior into a mathematical equation:

$$-W_c = \frac{Q_m \cdot C_p \cdot T_{01}}{\eta_c} \left(\left(\frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \quad \eta_c = \frac{\left(\left(\frac{P_{02}}{P_{01}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right)}{\frac{T_{02}}{T_{01}} - 1} \quad (3)$$

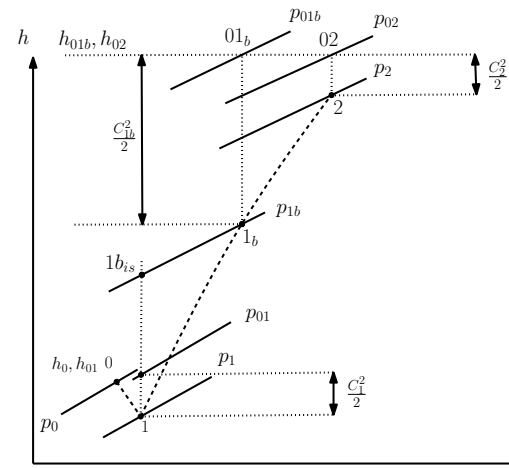


Figure 3: h-S diagram of a centrifugal compressor

Turbomachinery thermodynamics In order to formulate the basics of turbomachinery thermodynamics, some notations need to be introduced (fig.4). U is the tangential speed of blades, V is the absolute speed of the fluid exiting the rotor, W is the speed of the fluid in the system of coordinates linked to the rotor (ie. it rotates). We can define between these parameters what is called the two triangles of the characteristic speeds for a centrifugal compressor: subscript '2' is related to the values at the exit of the rotor, while subscript '1' is related to the entering ones. The angles β_1 and β_2 are the geometric angles of the blades at the inlet and the exit of the rotor, so they are fixed for a given compressor. We can also distinguish the ideal case from those with losses, by the slightly modified exit angle of the flow. Considering the parameters introduced just above, we can now develop Euler's equation applied to turbomachines (ref [10]). First of all, we can express the torque applied on the blades by the flow, related to the mass flow rate:

$$T = \dot{Q}_m \cdot (r_2 V_{\theta 2} - r_1 V_{\theta 1}) \quad (4)$$

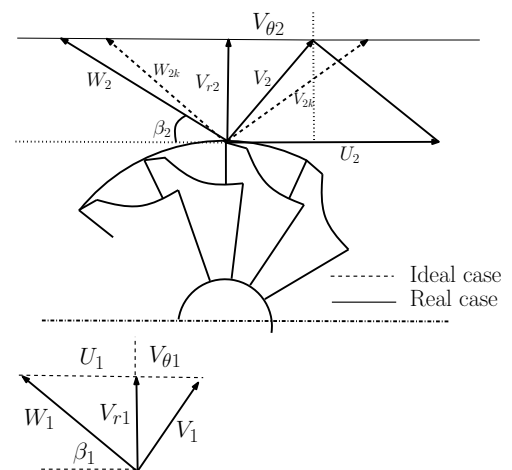


Figure 4: Speed characteristics triangle of a centrifugal compressor

Then, we can deduce from Euler's equation:

$$\Delta h_{is} = U_2 V_{\theta 2} - U_1 V_{\theta 1} \quad (5)$$

If we consider, that the flow is irrotationnal when entering the rotor, which is almost the case in real applications ($V_{\theta 1} = 0$), we get:

$$\Delta h_{is} = U_2^2 - U_2 V_{r2} \cotan(\beta_2) \quad (6)$$

In addition, one can notice that:

$$V_{r2} = \frac{Q_m}{\rho \cdot S_{rot}} = \frac{Q_m}{\rho \cdot S_{rot}} \quad (7)$$

Considering that the flow is 'observed' after its exit of the rotor blades, it can be stated that the fluid has been accelerated but almost not compressed, as it has not entered the diffuser yet. Therefore, we can consider that $\rho = \text{constant} = \rho_{inl}$. In the end, we get a linear evolution between the specific isentropic enthalpy gain Δh_{isM} and the mass flow rate Q_m :

$$\Delta h_{isM} = U_2^2 - \frac{U_2 \cotan(\beta_2)}{\rho_{inl} \cdot S_{rot}} Q_m \quad (8)$$

This relationship is to be noticed, as it means that we can link, in theory and at a given rotational speed, the evolution of the specific isentropic enthalpy gain to the mass flow rate by a linear mathematical expression. Based on experimental data, the specific isentropic enthalpy change is calculated. The result is plotted versus mass flow rate at various rotational speeds (fig. 5).

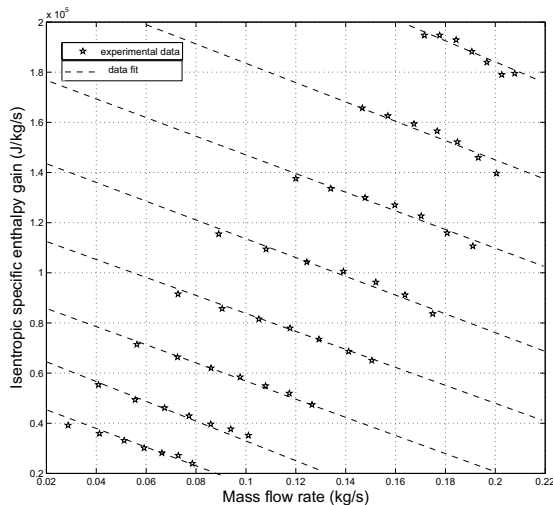


Figure 5: Isentropic enthalpy gain per unit of mass vs. mass flow rate at various rotational speeds

Qualitative study of compressor losses In this section, we are going to analyze the losses which occur in a centrifugal compressor. The aim is to put in light the fact that quadratic function can be fitted on compressor max efficiency as a function of rotational speed (fig.6).

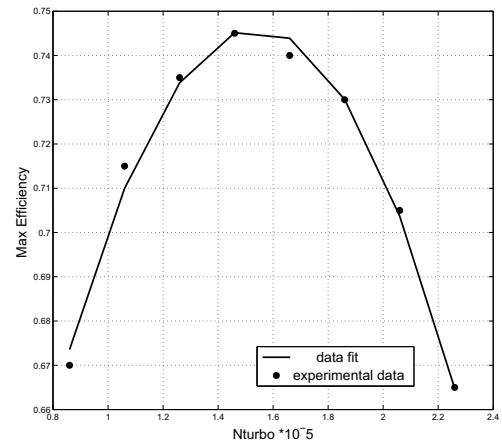


Figure 6: Centrifugal compressor maximum efficiency as a function of rotational speed (experimental data + fit)

In first approximation, we will only consider the following kinds of energy losses for a centrifugal compressor: aerodynamics losses (friction losses and blade incidence losses), mechanical losses (bearing) and flow leaks. Aerodynamic friction losses are due to the fluid viscosity, which coupled to rotational movement, will create shear stress in the boundary layer and thus dissipate energy. Blade incidence losses take into account the flow separation caused by off-design inlet gas angle at the impeller eye. Consequently, these losses are nil at the nominal speed and increase with speed changes. Leakage always occur, as clearance must exist between moving part (rotor) and static part (stator). Friction losses are mass flow rate dependant, while other parameters are rotational speed dependant (fig.7). By studying these plots, one can say that in order to reach maximum efficiency, a compromise must be reached between these losses: aerodynamic friction losses are low for low mass flow rates, but low mass flow rates are reached for low rotational speed (if we exclude the surge zone), where blade incidence losses are high. In fact, this compromise is found for intermediate rotational speeds and mass flow rates, as one can notice on every centrifugal compressor data map. By increasing or decreasing rotational speed, the maximum efficiency at iso rotational speed will decrease, as plotted in fig.6. What is noticeable is that this curve is almost a perfect quadratic function. This shape has been noticed for every centrifugal compressor in our database.

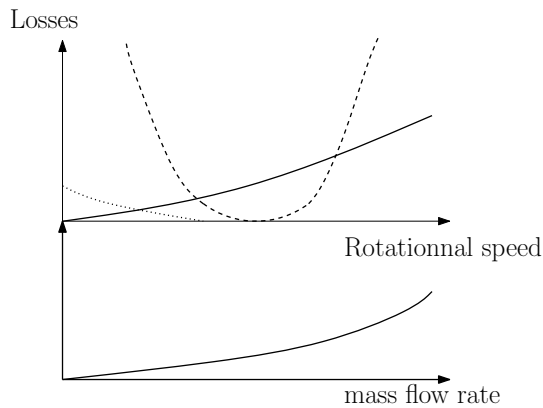


Figure 7: Qualitative power losses evolution. Above: bearing losses (plain), leaks losses (dotted), blade incidence losses (dashed). Below: Aerodynamic losses

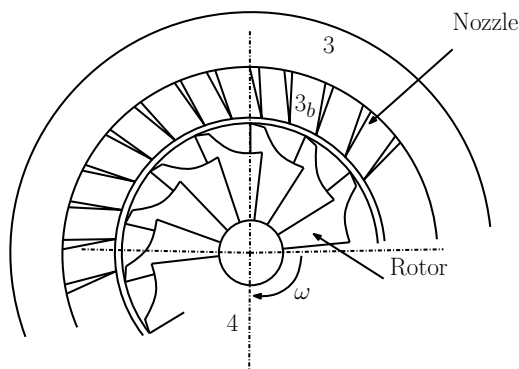


Figure 8: Radial flow turbine

TURBINE Turbocompressor turbine is driven thanks to the energy of the exhaust gases. This energy is generated by the gases released by the exhaust valve and expanding toward the ambient pressure. As mentioned before, we will consider the case of radial flow turbines. Such devices are composed of a casing which collects exhaust gases, a nozzle with stator or moving blades, and a rotor (fig.8)

Global thermodynamics As for the compressor, we plot the h-S diagram for the thermodynamics processes through the turbine ((fig.9).

The nozzle accelerates the flow to speed C_{3b} , which compensates the pressure loss ($p_{03b} \simeq p_{03}$). In other words, it means that the total enthalpy h_{03b} is almost equal to h_{03} . Only the form of energy changes (pressure to kinetic energy) in the stator. The transfer of work occurs through the rotor, which is designed for minimum kinetic energy at the exit. Consequently, it's through the rotor that the total enthalpy exchange occurs ($h_{04} \ll h_{03b} \simeq h_{03}$). VGT (Variable Geometry Turbine) are characterized by their nozzle

having moving blades. So regarding rotary speed, mass flow rate and desired energy provided to compressor, one can adjust the nozzle section and incidence. For a full open nozzle, the area is maximal, so the flow is almost not accelerated. When the nozzle is almost closed, the area is minimum so maximum kinetic energy will be transferred through the rotor.

As noticed for the compressor, the process is not isentropic for the turbine. Consequently a radial flow turbine can be characterized by two parameters: the amount of energy transferred from the fluid to the rotor shaft to the fluid and the efficiency of this process. We can formulate this thanks to the first law of thermodynamics:

$$W_t = Q_m \cdot C_p \cdot T_{03} \cdot \eta_t \cdot \left(1 - \left(\frac{P_4}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}}\right) \quad \eta_t = \frac{1 - \left(\frac{T_{04}}{T_{03}}\right)^{\frac{\gamma-1}{\gamma}}}{1 - \frac{P_{04}}{P_{03}}} \quad (9)$$

Thanks to compressor and turbine thermodynamic formulations (3, 9), we have the basis of a 0D turbocharger model: the values of P_x , T_x , η_x , Q_m are read in data maps. Once the amount of power delivered by the turbine is calculated, as well as the power consumed by the compressor, the last thing to evaluate is the turbocharger speed. It is done by verifying the dynamics fundamental principle in order to achieve the power balance between the turbine and the compressor:

$$\omega_t = \frac{1}{I_{turbo}} \int T_{turb} - T_{comp} \quad (10)$$

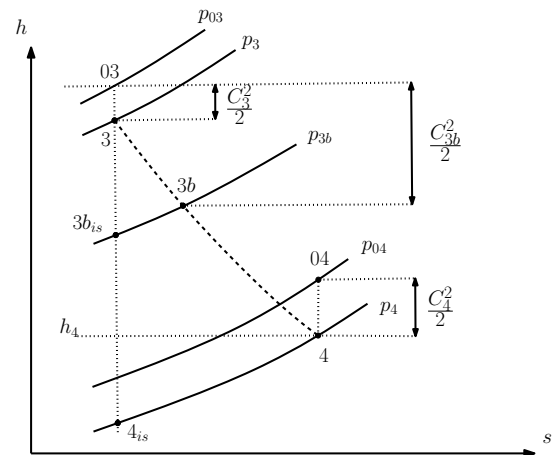


Figure 9: h-S diagram for centrifugal turbine

Turbomachinery thermodynamics We keep the same notations as introduced for the compressor. Subscript '3' is related to entry values, while subscript '4' is for exiting values. We can write Euler's turbomachines equation in the case of a turbine:

$$\Delta h_{is} = U_4 V_{\theta 4} - U_3 V_{\theta 3} \quad (11)$$

By developing as done above and considering that the enthalpy change is negative for the turbine we get:

$$\Delta h_{isM} = U_4^2 - \frac{U_4 \cot(\beta_4)}{\rho_{inl} \cdot S_{rot}} Q_m \quad (12)$$

This relationship means that we can link as for the compressor, at a given rotational speed, the evolution of the specific isentropic enthalpy drop through the turbine to the mass flow rate by a linear evolution. From measurements, we calculate the specific isentropic enthalpy drop through the turbine and plot it versus mass flow rate at various rotational speeds (fig. 10).

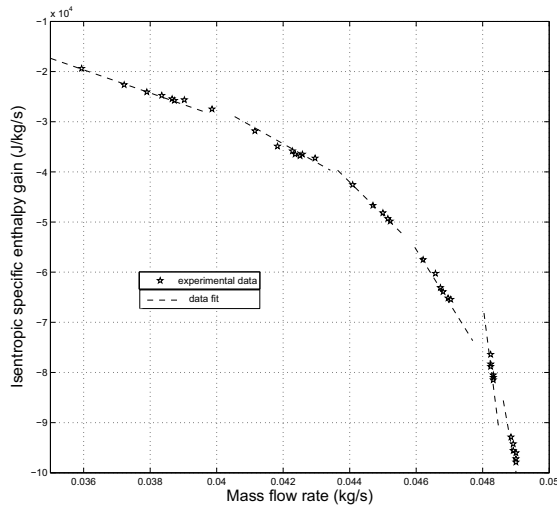


Figure 10: Specific isentropic enthalpy drop vs. mass flow rate a turbine

The curve is the evolution of the specific enthalpy versus the mass flow rate. This curve is composed of local straight lines, which are iso rotational speeds. As for the compressor, the linear evolution of the specific enthalpy has been shown for a given rotational speed.

STANDARD TURBOCHARGER 0D MODEL

We propose to test intra-extrapolation methods which appear (according to author's advice) to be the 'standard' method. The model developed by Jensen, Kristensen et al.[11] has been chosen, as it is one of the most stable ones, both in intra-extrapolation, while giving quite accurate results. This model is also used by Pr. Guzzella in [7]. The goal of this study is to understand better the pros of data map models, while showing the cons of pure mathematical algorithm for extra-interpolations.

JENSEN KRISTENSEN COMPRESSOR MODEL
Jensen and Kristensen have developed a model in which the compressor work is expressed in terms of a dimensionless head parameter ψ :

$$\psi = \frac{C_p \cdot T_{amb} \cdot \left[\left(\frac{P_{01}}{P_{amb}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right]}{0.5 \cdot U_c^2} \quad \text{with} \quad U_c = \frac{\pi}{60} \cdot d_c \cdot \omega_t \quad (13)$$

A dimensionless mass flow rate parameter is also introduced, in order to express compressor mass flow rate and Mach number at the inlet of the compressor:

$$\phi = \frac{Q_m}{\rho_{amb} \cdot \frac{\pi}{4} \cdot D_c^2 \cdot U_c} \quad M_{inl} = \frac{U_c}{\sqrt{\gamma R T_{amb}}} \quad (14)$$

Head parameter ψ , as well as the efficiency η_C are then expressed in terms of ϕ and M_{inl} :

$$\psi = \frac{k_1 + k_2 \cdot M_{inl} + k_3 \cdot \phi + k_4 \cdot M_{inl} \cdot \phi}{k_5 + k_6 \cdot M_{inl} - \phi} \quad (15)$$

$$\eta_C = \frac{a_1 + a_2 \cdot M_{inl}}{a_3 - M_{inl}} \cdot \phi^2 + \frac{a_4 + a_5 \cdot M_{inl}}{a_6 - M_{inl}} \cdot \phi + \frac{a_7 + a_8 \cdot M_{inl}}{a_9 - M_{inl}} \quad (16)$$

Parameters k_i et a_i are determined via least square method based on supplier data map.

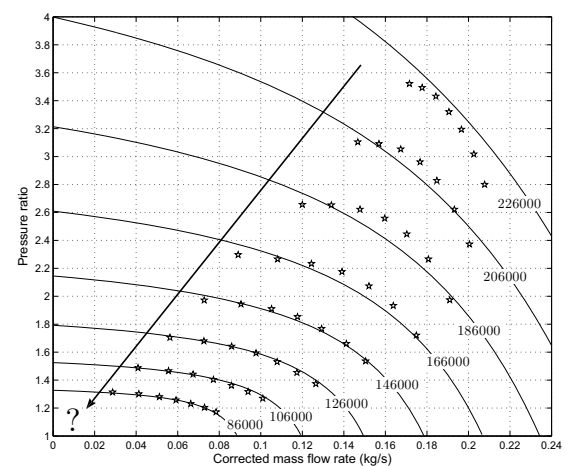


Figure 11: Comparison between pressure ratio from supplier data (stars) and the Jensen extrapolation (solid lines). Average error = 1.5%, Max error = 5.5%