Tool for Sizing Suction Pumps for Hybrid Laminar Flow Control Concepts

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***Abstract*—This paper describes a tool developed for estimating performance of suction pumps for Hybrid Laminar Flow Control (HLFC) with constraints related to real aircraft application.**

The suction pump has three components: a compressor, a generator and a motor. The last two must fit the overall aircraft electric system, which is unknown at this stage. Therefore, general linear approximations are used. The compressor is independent from other systems and is optimized specifically for HLFC. The models have been validated with data from previous HLFC concept studies.

The tool, developed in Python, provides sufficient details to ensure the constraints are satisfied, while still requiring minimal computational efforts and allowing the user to choose freely any optimization variables, constraints and objectives functions. This represents the first step toward a system-level optimization of HLFC concepts.

***Index Terms*—HLFC, suction pump, compressor, tool, optimization**

I. INTRODUCTION

For aircraft development, reducing operating cost is one of the main goals, along with having a clean and ecological system. A good solution to satisfy both these objectives is to reduce the fuel consumption. One way to do that is to maintain the flow laminar over the wet surface of the airplane to avoid friction drag which causes energy dissipation and consequently increases fuel consumption.

Laminar Flow Technology has been studied for both wings and empennage. Several solutions have already been proposed which can be divided into two major categories: passive or active systems. The passive solution is called Natural Laminar Flow (NLF) and has the advantage of not requiring any active system, the flow control relying solely on the shape of the airfoil. There is a drawback: Reynolds number and leading edge sweep are limited. Therefore, NLF is not optimal for medium range commercial aircraft. The active solution, Laminar Flow Control (LFC), also aims to maintain the flow laminar but with the support of an active system to suck air from the surface to stabilize the flow in the boundary layer. The first idea was, for an application over the wing, to suck air from both lower and upper surface of the wing over a major extent of the chord. Doing so, the flow is kept laminar for at least 75% of the chord [1] but the power required for the suction is high and diminishes the overall performance benefits. A hybrid solution, Hybrid Laminar Flow Control (HLFC), has then been developed, which uses the natural laminar flow principle to stabilize one type of instability in the laminar flow (Tollmien-Schlichting) while employing active control in the leading edge region to manage the cross-flow instabilities which are associated with swept wings on high-speed aircraft. With HLFC, a suction system is required to suck the air from the leading edge and maintain laminarity over a large extent of the chord.

To balance the drag reduction and the performance penalties associated with an active system, a specific suction distribution is required over the wing. This distribution is obtained through a system of chambers under the skin, which are connected to a compressor. This component is the major driver of the active part of HLFC. The compressor may be either axial or centrifugal, and must be optimized for the pressure and mass flow required. The HLFC concept may be adapted to different flight conditions for different aircraft, and even for different parts of the same aircraft. A suitable compressor for one concept may be far from ideal for another concept: therefore, there is a wide range of possibly interesting compressors for this application.

A tool has been developed to size and optimize suction pumps with a focus on the ranges required for HLFC, where for example pressure ratios are lower than three. The final goal for this tool is to be extended to cover the complete HLFC system allowing for easy modifications to system architecture, including constraints related to real aircraft application. It must also be compatible with the aerodynamic simulation tool to be able to optimize HLFC systems at the same time as the aerodynamic performance.

The main goal of this part of the tool is to determine the performance of a suction pump for giving inlet and outlet conditions. In the context of preliminary design for HLFC systems, it is important to be able to optimize the pump while respecting different constraints. Since the goals and constraints may vary with different HLFC concept, the choice of variables must be completely flexible, but also simple. A high level of fidelity is not required and one dimension models are favored.

There are many small tools to determine efficiency at design or off-design conditions for compressors, but publicly available solutions usually require specific inputs for specific outputs, or are focused on a given type of compressor. These restrictions are inconvenient for an application to HLFC suction pump. Amid commercial solutions, SoftInWay Inc. proposes a software suite that allows for the complete design of any type of compressor with many design variables and constraints. This would be a good solution if the compressor design only was of interest. But here it is important to consider other aspects of the tool:

- updatability for continuous development and inclusion of new models by the user if required,
- adaptability for multiple types of components met in HLFC,
- interfaceability with other tools used in HLFC concept development (in particular aerodynamic tools).

To achieve these, a new tool has been developed using Python 2.7.

II. Compressor Models

The HLFC system must provide the correct mass flow suction to maintain laminar flow. This limits the options to dynamic, with continuous flow, compressors: axial, centrifugal, and mixed-flow compressors. This tool will be used for preliminary design only and thus one dimensional flow model is used, using only the average values of the flow. And two models have been defined: the first one is extremely simple and mainly gives the power required for the suction, whereas the second provides stages' information, including dimensions, for either axial or centrifugal stages.

A. Overall Model and Variables

The overall compressor model is valid for any type of compressor and does not consider the stages design.

The key parameter is the specific energy transfer Y. It represents the energy transferred from the compressor to the flow. Its basic expression is simply the difference of total enthalpy h_o, but it may also be expressed with the total temperature at the outlet $T_{o,out}$ and at the inlet $T_{o,in}$ [2]:

$$Y = \Delta h_o = c_p\left(T_{o,out} - T_{o,in}\right). \quad (1)$$

The isentropic condition assumes that no energy (heat) is transferred to or from the gas during the compression, and all supplied work is added to the internal energy of the gas, resulting in increases of temperature and pressure [3]. This represents our "ideal" case and with that assumption, the isentropic energy transfer Y_i is obtained with the isentropic outlet temperature, $T_{o,out,i}$:

$$Y_i = c_p\left(T_{o,out,i} - T_{o,in}\right). \quad (2)$$

This expression is transformed with usual isentropic flow relations, using the total pressure at the outlet, $p_{o,out}$, and at the inlet, $p_{o,in}$:

$$Y_i = c_p T_{in}\left(\left(\frac{p_{o,out}}{p_{o,in}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right). \quad (3)$$

For the compressor description, three efficiencies are defined [4]:

- mechanical efficiency η_{mech},
- isentropic efficiency η_i,
- compressor efficiency η_{Cmp}.

This compressor is bound to a motor and a generator with their own efficiency to form the suction pump. The relation between the efficiencies and the powers are indicated in Figure 1.

B. Stages Model

To get a more detailed compressor design, one must consider the stage design. Keeping in mind that this tool is not aimed at a complete definition of the compressor with 3D flow simulation, the level of fidelity remains relatively low and only the average values for the different stages are considered. It is also of interest to use the same process for axial and centrifugal compressors to be able to switch the type of compressors for a HLFC concept without any additional work for the user.

T. M. Schobeiri develops a stage model in [5] that can be used equally for axial and centrifugal compressors, as well as turbines. This model relies on three sections within each stage. The first section is at the stator inlet, the second is between the stator and rotor, and the third after the rotor.

At each of these sections, it is possible to establish a velocity diagram to show the blade velocity u as well as the flow velocity, either relative to the blades (w) or absolute (v) as seen in Figure 2. With this model, the different velocities within the compressor are expressed with (vectors in bold):

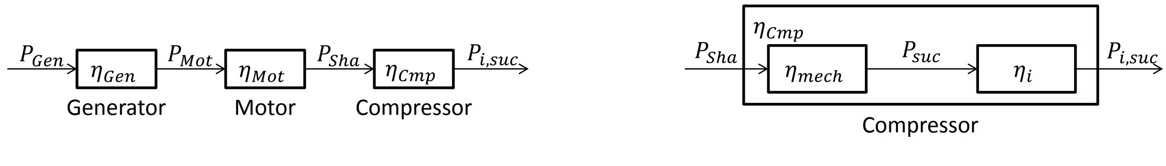


Figure 1. (a) Efficiencies and powers for the suction pump: the generator receives the power P_{Gen} from the aircraft system. It is redistributed to the motor with an efficiency of η_{Gen}.The behaviour is similar for motor and compressor. (b) Efficiencies and powers used for the compressor: P_{Sha} is the shaft power, power received by the compressor. The power $P_{i,suc}$ and P_{suc} are the powers required for isentropic and real compression respectively. Their ratio gives the isentropic efficiency η_i. The mechanical efficiency η_{mech} is the ratio of P_{Sha} and P_{suc}, and η_{Cmp} is the total compressor efficiency.

$$
\begin{aligned}
\boldsymbol{u_1} &= \mathbf{0}\\
\boldsymbol{v_1} &= v_{u_1}\boldsymbol{e_1} + v_{m_1}\boldsymbol{e_2}\\
\boldsymbol{w_1} &= \boldsymbol{v_1}\\
\boldsymbol{u_2} &= u_2\boldsymbol{e_1}\\
\boldsymbol{v_2} &= v_{u_2}\boldsymbol{e_1} + v_{m_2}\boldsymbol{e_2}\\
\boldsymbol{w_2} &= (v_{u_2} - u_2)\boldsymbol{e_1} + v_{m_2}\boldsymbol{e_2}\\
\boldsymbol{u_3} &= u_3\boldsymbol{e_1}\\
\boldsymbol{v_3} &= -v_{u_3}\boldsymbol{e_1} + v_{m_3}\boldsymbol{e_2}\\
\boldsymbol{w_3} &= -(v_{u_3} + u_3)\boldsymbol{e_1} + v_{m_3}\boldsymbol{e_2}.
\end{aligned}
\tag{4}
$$

The subscripts 2 and 3 represent respectively the inlet and outlet of a rotor in a compressor's stage. The subscript u indicates the component of the flow velocity tangential to the blade velocity u and m indicates the component orthogonal to u, which is called the meridional flow. This choice of convention for the velocities and angles gives a simple expression for the "Euler turbine equation", which expresses the specific energy transfer, Y_{Euler}, with the blade linear velocity, u, and the component of the flow velocity tangential to u, v_u:

$$
Y_{Euler} = u_2 v_{u_2} + u_3 v_{u_3} \tag{5}
$$

The "Euler turbine equation" is a well-known relation to obtain the energy transferred to the fluid by a turbine or a compressor [6]. As opposed to the previous equations for the specific energy transfer, the Euler equation is based on a mechanical consideration: the conservation of angular momentum. The principal difference from the thermodynamic equations is the absence of efficiency. Hence the specific energy calculated corresponds directly to the specific energy effectively transmitted to the fluid.

This equation, obtained through the derivation of the conservation law of angular momentum, is valid for all types of compressors.

In addition to this frame of reference, Schobeiri suggests using dimensionless variables for the third section. In the tool, these variables have been extended to each section to unify each stage and ease the development. In the following definitions, i is the section number:

- meridional velocity ratio: $\mu_i = v_{m,i-1}/v_{m,i}$,
- circumferential velocity ratio: $\nu_i = u_{i-1}/u_i$,
- stage flow coefficient: $\varphi_i = v_{m,i}/u_i$,
- stage load coefficient: $\lambda_i = Y_{Stg}/u_i^2$,
- stage degree of reaction: $r = \frac{\Delta h''}{(\Delta h' + \Delta h'')}$.

The relations between these variables are as follow:

$$
\begin{aligned}
1 &= \varphi_3(\cot\alpha_3 - \cot\beta_3),\\
\lambda_3 &= \varphi_3(\mu_3\nu_3\cot\alpha_2 - \cot\beta_3) - 1,\\
\nu_3 &= \mu_3\varphi_3(\cot\alpha_2 - \cot\beta_2),\\
r\left(2\lambda_3 + \frac{v_3^2 - v_1^2}{u_3^2}\right) &= \mu_3^2\varphi_3^2(\nu_3^2 - 1)\cot^2\alpha_2\\
&\quad -2\mu_3\nu_3\varphi_3\lambda_3\cot\alpha_2 + \lambda_3^2\\
&\quad +2\lambda_3 - \varphi_3^2(\mu_3^2 - 1).
\end{aligned}
\tag{6}
$$

With these equations, it is possible to determine the geometry of each stage and deduce the mass by assuming a certain thickness for the different parts of the compressor.

Each stage is considered as a compressor on its own, and the overall model described previously is applied to compute the energy transfer for a single stage. The whole compressor is simply modelled with a succession of stages. The overall model is applied with the inlet flow of the first stage and the outlet flow of the last stage to obtain the performance of the complete compressor.

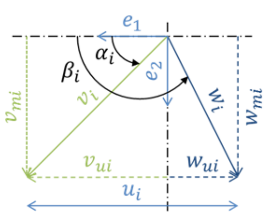
III. Efficiency Approximation

The compressor efficiency is a key parameter in designing a compressor: it gives the relation between the isentropic power, related to thermodynamic equations, and the power obtained with the Euler equation, related to mechanical equations.

A. *Wright Simplification of the Cordier-Diagram*

The Cordier-diagram was presented in 1953 by Otto Cordier [7] and regroups extensive empirical data for compressor efficiency, providing an overview of the best achievable efficiency over a wide range of specific diameters D_s and specific speeds N_s [6].

The first model relies only on an approximation of the Cordier-Diagram to give the maximum efficiency for a



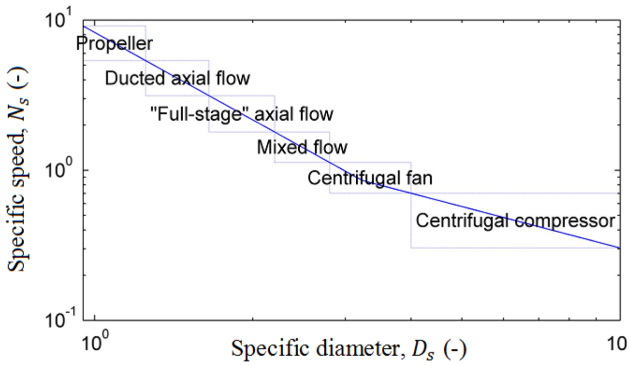


Figure 2. (a) Angles and velocities convention used for the stages. The velocity u is the blades' velocity, v is the flow absolute velocity, w is the flow relative to the blades velocity, e_2 is the compressor rotation axis, e_1 is positioned to form an orthogonal frame of reference. (b) Cordier-diagram as described in [6] with the associated type of compressors, it shows the optimal specific speed N_s as a function of optimal diameter D_s.

given specific speed N_s or specific diameter D_s. The Cordier-diagram is approximated by a simple equation giving the best possible specific speed N_s for a given specific diameter D_s as shown in Figure 2. The best achievable efficiency η_{Cmp} is then given by a second equations, thus determining all three parameters D_s, N_s and η_{Cmp}.

The major advantage of this model is that it requires very limited knowledge of the compressor and is thus applicable to any design, as rudimentary as it could be. The drawback is a very limited design space, which gives no room for optimization. And it is of course impossible to estimate off-design performance with this equation.

B. *Complete Cordier diagram*

The complete Cordier-diagram will show the best efficiency for a given pair (N_s, D_s), but contrary to the previous equation, this pair doesn't have to be optimal. Thus, this model may be applied to various compressors, and it is possible to deviate from the optimum design to accommodate different constraints.

Inside the tool, for optimization purposes, it is necessary to have continuous functions. Therefore, the values from the literature, [8] and [9] for axial and centrifugal respectively, have been interpolated with polynomials equations of the fifth order with two variables ($\log N_s$ and $\log D_s$). The results of these interpolations are shown in Figure 3. The highest difference between the values computed with the new equations and the original table are below 4%, which is acceptable given the uncertainty inherent to the one-dimensional model.

This model is used to compute the efficiency of a compressor under constraints but it is not suitable for off-design performance. The diagram gives indeed the best achievable efficiency for a compressor designed for such condition. Logically, off-design performance will be lower than what is found in the diagram.

C. *Models for off-design efficiency*

1) *Axial Compressor*

For an axial stage, it is possible to estimate the performance for off-design conditions based on a reference compressor (here the design conditions) using an empirical relation shown in Figure 4. This relation gives the ratio of efficiencies between design and off-design points based on ratios of flow coefficients and loads coefficients [10]. These dimensionless ratios are expressed as follows:

$$x = \frac{\lambda_3}{\lambda_{3,ref}} \frac{\varphi_{3,ref}}{\varphi_3} \quad y = \frac{\eta}{\eta_{ref}} \tag{7}$$

This model requires a reference compressor which may be obtained with the Cordier-diagram previously mentioned.

2) *Centrifugal Compressor*

For a centrifugal stage, such simple relations were not found but there are simple models to compute the off-design performance. H. W. Oh, in [11], gathers different models and shows the results for different combinations. These models estimate the impact of the:

- incidence loss,
- blade loading loss,
- skin friction loss,
- clearance loss,
- mixing loss,
- vaneless diffuser loss,
- disc friction loss,
- recirculation loss,
- leakage loss.

The Schobeiri compressor model, with each stage designed, only gives mean values of the flow and this is not enough to compute all these losses as the blade design is not detailed enough. Nevertheless, it is possible to use usual values for certain parameters (blade solidity for example) and determine others to reach the efficiency obtained with the Cordier diagram for the design conditions. With these parameters, the losses for off-design conditions may then be estimated and the efficiency computed.

IV. Tool Architecture

As a main feature, the tool must be able to design and optimize a compressor for any kind of input. This means that the user may provide any data and the tool will then compute the optimum outputs, with a user-defined objective function. This is particularly useful for changing the HLFC architecture, study case, or to perform a sensitivity study and analyze off-design

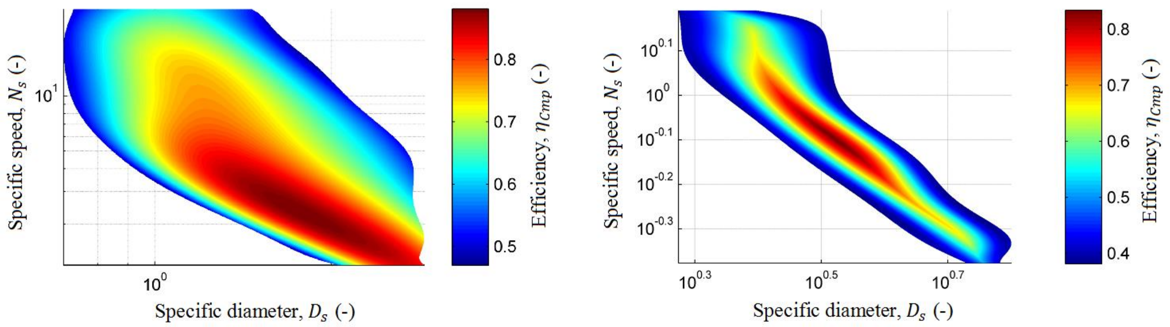


Figure 3. Cordier diagram drawn based on polynomial interpolation for axial compressor, 2D view with colour map for the efficiency. (a) Diagram for axial compressor. (b) Diagram for centrifugal compressor.

performance. In the future development of the tool, this flexibility will allow different modules to be combined without worrying about the inputs.

The tool should be used for preliminary design and first estimation of the performance of different HLFC architectures. This means that aerodynamic design and overall aircraft performance will also be at a simple preliminary study level. Therefore, a high level of detail is not required for the compressor design and the tool does not include any Computational Fluid Dynamic (CFD) model.

A. Equations Solver

Flexibility in solving the equations is the key feature of the solver, because it must work for any set of inputs. An immediate solution would be to have different set of equations to use depending on the set of inputs provided. But the amount of possible input sets grows exponentially with the number of variables, and so this solution was put aside. In this tool, this flexibility is achieved by defining all the equations for a given object and then going through each of these equations in a loop, until no more equations can be solved.

For example, a circle is defined by five different parameters:

- radius R,
- diameter D,
- hydraulic diameter D_h,
- perimeter P,
- area A.

These parameters are coupled with eight equations:

1. $A = f(R)$,
2. $D = f(D_h)$,
3. $D = f(R)$,
4. $D_h = f(D)$,
5. $P = f(R)$,
6. $R = f(D)$,
7. $R = f(P)$,
8. $R = f(A)$.

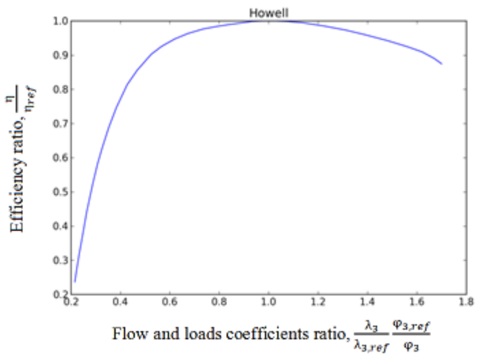
When the solver is called for the first time, it will simply go through the list of equations and try to solve each of them. If one of the equation is successfully solved (i.e. provided a coherent numerical results), the solver will go through the list of equations again as shown in Figure 4, excluding the equations successfully solved.

In the case of a circle, if D were used as input, the solver will solve the equation 4 and 6 on the first run, extending the known parameters to D_h and R. On the second run, equations 4 and 6 are excluded, and the solver can now solve 1, 2, 3, 5, 7 and 8. The seventh and eighth equations are also solvable because A and P were found with the first and fifth equations respectively. On the third run, all equations have been used and the solver will stop.

This example also shows that several equations may have the same output. Here the equations 6, 7, and 8 all give R as output. As 6 will be used first, it will define the value of R. The results of the equations 7 and 8 will be compared to this value of R and if the difference is reasonably small, the equations are considered successfully solved. If the difference is too high, a warning is issued to the user. This could happen if the user provided incoherent inputs (e.g. $D = 1$ and $A = 1$). In other words, the solver checks itself if all equations are coherent and warns the user if any results do not fall in line.

The advantage of this method is that the user may use any parameters as inputs, without having to change anything in the code, and every possible output will be computed. And this principle is extended to all geometries encountered in HLFC, flow and even compressor stages. This means that it is possible to use any of the stages parameters as inputs and simply use the most suitable set of inputs, either as simple parameter or as optimization variable, for the current HLFC architecture. Of course, it comes with a computational cost. But this cost was judge acceptable in regards to the small impact it will have in the end, when aerodynamic is also computed.

Some equations are not linear (see (6) of the compressor stage model) and cannot converge under certain conditions; other equations are valid only for a certain domain. With this solver, it is possible to include conditions in each equation to ensure the equations are solved correctly, respecting the domain of validity of each equation.



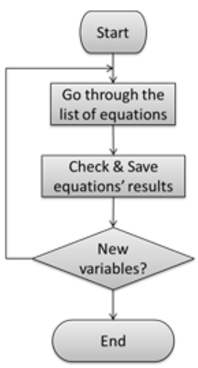


Figure 4. (a) Relation between stage load and flow coefficients and efficiency. This curve only gives the efficiency as a percentage of a reference compressor. It assumes that the reference, the design point, is the optimal configuration. (b) Equations solving process: an extensive list of equation is provided; the inputs are not specified giving the user complete freedom over the choice of variables

B. Components

Different modules are combined to form an "assembly". This represents the complete system for which the performance is evaluated. Modules are available to compute the flow and key parameters for electric motors, generators, and compressors.

The flow module's main function is to compute all parameters of the fluid, with the usual thermodynamic parameters (pressure, temperature, density, etc.), as well as dimensionless quantities (Mach, Prandtl, Reynolds numbers for example). As of now, only dry air as an ideal gas is implemented. In addition to this, it is possible to get the free stream flow characteristics for a given altitude, and the corrected mass flow, an interesting parameter to compare flow for different HLFC concept, which may operate at different altitudes.

The electric motor and generator modules are extremely simplified. It seems indeed unnecessary to put much effort in the design of such components as they depend on the overall aircraft design and electric system. These are not known for the HLFC preliminary design and the study of different concepts. Therefore, the modules rely on linear approximation to compute the mass of the different components based on the power required, with usual power-to-weight ratios.

The major component included at this time in the tool is the compressor. The compressor module includes the models described in this paper. The user may switch between these models depending on the constraints and the level of details required.

V. Optimization

In terms of HLFC concept, the performance is estimated with an equivalent drag count for the system. This equivalent drag depends on the pump power, the flow velocity at the exhaust to the free stream, as well as the mass of the complete system.

The pump drag is expressed with the pump power P_{pump}, the free stream velocity U_∞, the dynamic pressure q_∞, the thrust T obtained with the HLFC system, the surface of reference S_{ref} and the ratio of aircraft power to calorific fuel burn $\eta_{a/c}$:

$$D_{pump} = \frac{1}{q_\infty S_{ref}}\left(\eta_{a/c}\frac{P_{pump}}{U_\infty} - \mathrm{T}\right) \qquad (8)$$

The thrust is expressed with the flow velocity and pressure at the exhaust (U_x and p_x) and in free stream (U_∞ and p_∞), the sucked mass flow $\dot{m}$ and the exhaust area A_x:

$$T = \dot{m}\cdot(U_x - U_\infty) + A_x(p_x - p_\infty) \qquad (9)$$

The pump drag coefficient is obtained with:

$$C_{Dpump} = \frac{D_{pump}}{q_\infty S_{ref}} \qquad (10)$$

Using the expression of the lift induced drag C_{Dlift}, it is possible to estimate the impact of the added mass Δm on the performance with the lift coefficient C_L and the aspect ratio of the wing AR:

$$C_{Dlift} = \frac{C_L^2}{\pi AR} = \frac{1}{\pi AR}\left(\frac{(m+\Delta m)g_0}{q_\infty S_{ref}}\right)^2 \qquad (11)$$

The coefficient ΔC_{Dlift} is the change in drag due to the additional mass. This new mass is supposed very small compared to the mass of the aircraft which gives a simplified equation:

$$\Delta C_{Dlift} \approx \frac{2m}{\pi AR}\left(\frac{g_0}{q_\infty S_{ref}}\right)^2 \Delta m \qquad (12)$$

A possible objective function to be minimized is:

$$C_{Dnet} = C_{Dpump} + \Delta C_{Dlift} \qquad (13)$$

This function is used to compare the different suction pumps for HLFC but it not enough to evaluate the complete HLFC architecture: the aerodynamic performance is not considered here and the other components are not designed, in particular the piping. It is nevertheless a first step toward the combined optimization.

It is also possible to use any other parameters for the optimization. It is for example often required to minimize the size of a component due to space restriction in an aircraft.

The optimizer used is the function "fmin_slsqp" from the package "scipy" of Python 2.7. The sequential least squares programming is adapted because all functions used are continuous. Furthermore, it is well suited to nonlinear constraints.

VI. Tool Validation

Numerous studies on HLFC have included compressors, either roughly designed or taken off-the shelf and therefore not optimized for this HLFC use. Before using this tool for new HLFC design in the future, it is compared with data from previous studies. HLFC was vastly studied in the 80s' with [12], [1], [13] and [14] for example. The manufacturing capacities at that time pushed the people to put HLFC aside. But a new interest arose at the beginning of the century with [15] and [4] among others. The data about the compressors available in these studies was used as input in the tool presented here as a validation.

In some cases, as for [13], the data available is not enough to define a compressor as there are still many variables unknown which can be chosen freely. In this situation, usual values have been used and a realistic compressor was design with a coherent result. In some other cases, the data available is more than sufficient to define the compressor which allows comparing inputs and outputs. This is true for the studies from T.M. Young [15] and Prof. Atkin [4].

The compressor model without stages design was tested against the data from Prof. Atkin. The values for the inlet of the compressor were conserved, as well as the assumption for the exhaust pressure and Mach number. The pump efficiency was also taken as input as shown in the table I.

TABLE I. PARAMETERS USED FOR THE DESIGN OF COMPRESSORS WITH THE DATA AVAILABLE IN [4].

Variable	*Description*
$\dot{m}$	Mass flow
$p_{o,in}$	Total pressure at inlet
$T_{o,in}$	Total temperature at inlet
M_{in}	Mach number at inlet
p_{out}	Pressure at outlet
M_{out}	Mach number at outlet
η	Compressor efficiency

The compressor inlet and outlet were fully computed, giving results very close to the original data, with differences below 0.2% (see table II), which may be explained by the rounding of numerical values. The results for all cases presented in [4] are similar.

TABLE II. COMPARISON OF THE VALUES FOR THE INLET AND OUTLET FLOWS COMPUTED WITH THE TOOL AND THE VALUES AVAILABLE IN [4]. THE SMALL DIFFERENCES ARE IMPUTABLE TO THE ROUNDING OF THE VALUES PROVIDED IN THE REFERENCE DOCUMENT.

Variable	*Description*	*Difference*
p_{in}	Pressure at inlet	0.000%
T_{in}	Temperature at inlet	0.000%
$p_{o,out}$	Total pressure at outlet	0.000%
T_{out}	Temperature at outlet	0.008%
$T_{o,out}$	Total temperature at outlet	0.009%
v_{out}	Exhaust velocity	-0.003%
P_{Sha}	Shaft power	0.052%
T	Net thrust	0.034%
CD_{pump}	Pump drag coefficient	-0.184%

The tool also provided additional data for the flows (area, density, volumetric flow, and diverse dimensionless numbers, etc.) and for the compressor (isentropic and polytropic efficiency for the simple compressor model). The user could also extend the model by assuming specific diameter and specific speed. This completes the simple model with the rotation speed and the compressor diameter. If the efficiency is estimated with the Cordier diagram, the user could set any of these parameters as optimization variables to reach the best achievable efficiency.

Young uses a 1-stage centrifugal compressor and gives details about the flow at the inlet and outlet of the suction pump and provides the major parameters for the compressor in [15]. Again, some values (but not all) are taken from the source to test the tool. These chosen parameters are indicated in the table III below.

TABLE III. PARAMETERS USED FOR THE DESIGN OF COMPRESSORS WITH THE DATA AVAILABLE IN [15].

Variable	*Description*
$\dot{m}$	Mass flow
$p_{o,in}$	Total pressure at inlet
$T_{o,in}$	Total temperature at inlet
M_{in}	Mach number at inlet
$p_{o,out}$	Pressure at outlet
M_{out}	Mach number at outlet
η	Compressor efficiency

Here, the centrifugal stage will also be designed. Therefore, the value of a few other parameters must be assumed (they are not provided in T.M. Young's thesis) and usual values from the literature will be used. These parameters are the flow angles at the inlet, the meridional velocities ratios, the diameters ratios, the specific speed and specific diameter. It is then possible to design the compressor and compare the results with the data provided in [15], as shown in the table IV.

TABLE IV. COMPARISON OF THE VALUES FOR THE FLOW AT THE INLET, OUTLET AND COMPRESSOR COMPUTED WITH THE TOOL AND THE VALUES AVAILABLE IN [15]. THE HIGH DIFFERENCES ARE EXPLAINED WITH THE ASSUMPTION USED IN THE REFERENCE DOCUMENT.

Variable	*Description*	*Difference*
p_{in}	Pressure at inlet	-0.43%
p_{out}	Exhaust pressure	0.03%
$T_{o,out}$	Exhaust total temperature	-17.28%
v_{out}	Exhaust velocity	-8.42%
ρ_{out}	Exhaust density	15.02%
m	Compressor mass	17.80%

The values are here quite different but this is easily explainable: in the reference document, it is assumed that the stagnation temperature is unchanged during the compression. This is unrealistic and the tool computes an increase of temperature corresponding to the 17% indicated. It is possible to force the exhaust temperature to the value indicated by Young, and in this case, the results obtained with the tool are similar to the ones showed in [15]. For the mass, the order of magnitude is the same, which is satisfactory given the low precision of the model for mass estimation.

The stage model also provides additional parameters, for example the Mach number directly at the exit of the centrifugal stage. It is much higher than the one assumed at the pump exhaust and this shows the need for a diffuser. The tool includes, for now, only a perfect diffuser with no pressure losses in order to reduce the Mach number. The objective function with the pump drag is also computed, allowing the pump to be optimized with respect to any of the variables if need be.

VII. Summary and Outlook

The tool presented here provides a solution to quickly estimate the performance of suction pump used in HLFC with a level of details sufficient to ensure all constraints for real aircraft integration are satisfied. The tool is particularly adapted to preliminary design and concept evaluation with a great flexibility regarding the inputs.

Nevertheless, an HLFC system is not limited to the suction pump. The ducting from the suction area to the compressor and from the compressor to the exhaust will also have an influence on the overall performance of the system. Therefore, it is necessary to extend this tool in the future to allow modelling the other systems, such as the ducts, valves and other fittings.

In addition to the system performance, a closer look must be taken at the aerodynamic performance of HLFC. For now, aerodynamic and system studies are separated, missing any potential synergy between these two aspects of HLFC. The next step is to combine aerodynamic and system tools in order to optimize suction distribution and system architecture simultaneously to increase overall performance.

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