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Preliminary Design and Sizing of the Thermal and Energy Management Subsystem for LAPCAT MR2

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Abstract

This paper deals with the preliminary design and sizing of the Thermal and Energy Management Subsystem elaborated by ESA for the LAPCAT MR2 high-speed vehicle. The working principle of this subsystem is based on the exploitation of liquid hydrogen boil-off. The friction heat loads penetrating the aeroshell, all along the point-to-point hypersonic mission, generates boil-off within the cryogenic tanks that can be exploited for cooling and power generation purposes. This work aims at analyzing the various components which the TEMS consists of, suggesting proper Estimation Relationships (ERs) for mass, volume and power budgets. Eventually, a short discussion about examples of sensitivity analysis aimed at deriving the impact of operating parameters on subsystem budgets is proposed as introduction to future works of the research.

Keywords: *Thermal and Energy Management Subsystem, Sizing Parametric Models, LAPCAT MR2.*

1. Introduction

One of the major challenges to cope with in the field of the hypersonic speed transportation is the definition of adequate thermal management strategies to be adopted to withstand very high temperatures, heat fluxes and heat loads all along the mission. Many research activities on this topic envisaged subsystems and technologies optimized to perform thermal protection, control and management issues. Conversely, the current trends in hypersonic speed transportation design are going towards the development of multi-functional subsystems highly integrated in the vehicle, whose working points are the results of multidisciplinary optimization processes. In this context, within the research activities carried out in the framework of LAPCAT-II [1], ATLLAS-II [2] and HIKARI [5] projects, ESA developed the concept of TEMS, the Thermal and Energy Management Subsystem. This is a clear example of multifunctional subsystem combining together the management of thermal loads and the electric power generation and supply, making benefits of liquid hydrogen boil-off, that is usually considered a drawback. The complexity of the TEMS and of the constituent components, the maturation levels of the technologies as well as the integration on-board the reference vehicle needs to be furtherly investigated in order to confirm the technical feasibility of the concept. Moreover, some components are supposed to work out of the usual performance boundaries, preventing the exploitation of statistical laws to derive high-level sizing suggestions. Thus, this paper aims at in-depth analysing each of the constituent components, evaluating the results of a preliminary sizing obtained through dedicated semi-empirical parametric models.

After a short introduction to the reference vehicle, and related mission profile, Section 2 provides a description of the reference case study, briefly describing the architecture of the TEMS, together with the envisaged operating points. While Section 3 presents an overview of the approach that has been followed to derive the estimation parametric models, Section 4 critically analyses the results in terms of mass, volume and power budget estimation for a specific preliminary selected design point. In addition, Section 5 provides an overview on the sensitivity analyses aimed at demonstrating the impact of the different operating points on the sizing to be further investigated. Eventually Section 6 collects the conclusions and proposes future works of the research.

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2. Case Study: LAPCAT MR2

2.1. LAPCAT II project

LAPCAT II (Long-Term Advanced Propulsion Concepts and Technologies) [1] was a follow-up of the previous EC co-funded project LAPCAT. The main goal was the development of different vehicle concepts enabling potential reduction of antipodal flight times to about 4 hours along with the critical technologies and know-how to realize this ambitious goal. In particular, this paper will use the Mach 8 configuration as reference: a wedge-shaped wave-rider powered by a combination of different propulsion systems, i.e. the ATRs (turbojet engines based on an air-turbo ramjet cycle) for Mach below 4 and the DMR (Dual-Mode Ramjet/Scram-jet engine) which operates between Mach 4 and Mach 8, able to carry 300 passengers over long-haul routes.

2.2. Reference vehicle and mission scenario

The selected vehicle, known as LAPCAT MR2, is equipped with six ATRs and one DMR in order to cover the propulsive power need along the mission profile. The typical mission is in fact characterized by a first subsonic cruise, a supersonic ascent followed by the ATR switch-off, the ignition of the DMR and a subsequent hypersonic climb and cruise at 33 km of altitude. The descent is then considered unpowered gliding. A reference long haul route is the Brussel – Sidney direct flight, which has a great circle distance of 16700 km and a flight time of about 2h45min [4]. The estimated Maximum Take-Off Weight (MTOW) is about 400 tons, with an Operating Empty Weight (OEW) which is almost half of the MTOW (200 tons), leaving the remaining fraction for the liquid hydrogen (LH2) that is used as propellant for the powerplant.

2.3. The Thermal and Energy Management Subsystem (TEMS)

Although the cruise altitude is almost three times the flight level with respect to conventional aircraft, flying at this high speed causes high heat fluxes on the vehicle shell and, consequently, higher temperature of the skin layers. For the selected range of Mach numbers, the temperature of the pressure side of the aero-shell may reach 1000 – 1200 K in the worst condition [2]. This leads to the adoption of proper materials for the TPS, belonging particularly to ceramic family rather than metallic one, especially for Mach higher than 4 . However, flying at higher speed may also be an advantage not only considering the overall mission time but also looking at the heat load accumulated by the vehicle (thermal paradox) [2]. Indeed, heat fluxes grow with increasing speed, but the heat has less time to penetrate the structure while the wall temperature doesn't increase proportionally with the total temperature.

Apart from the external balance, other strategies shall be considered to control the amount of heat flowing through the shell into the internal compartment in order to guarantee proper survivability levels of both systems and payload. To face this problem, LAPCAT MR2 is equipped with the innovative TEMS, whose scheme is reported in Fig. 1, which is conceived to use the boil-off vapours coming from the evaporation of the LH2 within the tanks as main cooling means for the cabin, the powerplant and the air-pack of the Environmental Control System (ECS), being finally injected in the combustor of the engines. Moreover, the high-pressure liquid hydrogen is used to drive a dedicated turbine providing mechanical power to the devices of the TEMS itself, producing enough power margin for the other on-board subsystems. Tanks and cabin are in fact highly integrated in a bubble-structure architecture, which characterize the overall internal configuration of the wave-rider (Fig. 1).

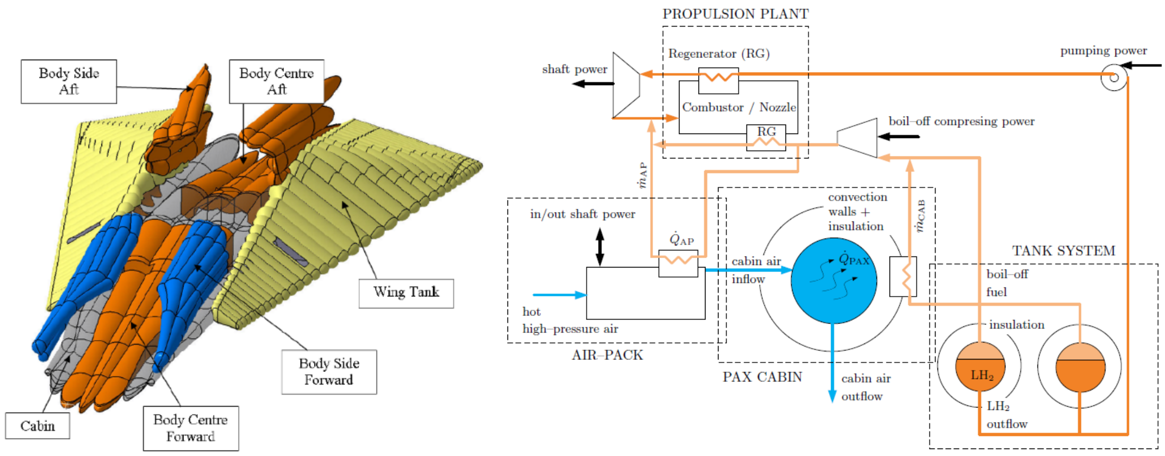

Fig. 1. Tanks and cabin layout of LAPCAT MR2 (left) and scheme of the TEMS (right) [5]

The computation of heat loads [2] due to both external and internal phenomena led to the derivation of the main operating conditions of the TEMS, which are listed in Table 1.

Table 1. Operating ranges for the main components of the TEMS [5]

Component	Temperature [K]		Pressure [bar]		Mass flow	Power
	In	Out	In	Out	[kg/s]	[MW]
Fuel Pump	20	[26, 28]	1	[60, 80]	[0, 100] [0, 35]	[1, 15] [0.5, 5]
Boil-off compressor	[20, 270] [50, 270]	[150, 950] [240, 950]	1	60	[0, 8] [0, 6]	[0, 15] [0, 7]
Expander	1300	[1200, 1300] [1250, 1300]	[60, 80]	60	[10, 100] [6, 32]	[0, 120] [0, 40]

These data are used as starting point of the preliminary sizing of the TEMS components, reported in the following Sections.

3. Methodology overview

In order to properly size the different components, when statistical data is not available or applicable, an integrated methodology based on estimation relationships able to define physical characteristics as function of a set of operational parameters and performance is suggested. The approach here presented can be applied to a wide range of active and passive elements since it is made up of several modular steps which start from the identification of the governing equation of the selected component (Fig. 2) and end with the parametric budget computation.

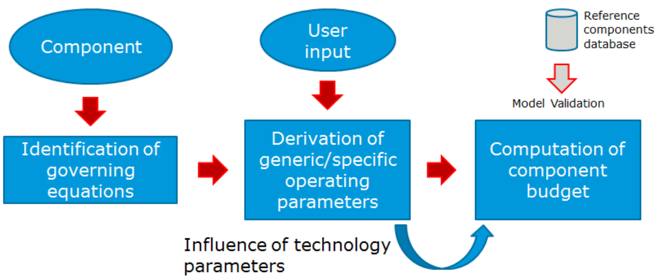

Fig. 2. Components budgets derivation process [6]

The first step is in fact related to the identification of equations describing the physical phenomena under which a component behaves (this is the reason why they are called governing equations). This can be done by referring to the applicable scientific literature describing the behaviour of specific devices, by constituting relationships based upon the main operating parameters (Eq. 1). These parameters are the core of the methodology. Thus, a proper classification of these parameters is performed within the second phase of the method, with the objective of analysing their impact on the governing equations.

$$G = g(GOP, SOP) \tag{1}$$

Where

- GOP are the General Operating Parameters, representing the main performance of a component (e.g. rotating speed of the machine, mass flow, pressure rise, efficiency, etc.). They are referred to as "generic" since different kinds of components can be described by means of them, as they describe the basic physical occurring phenomena.
- SOP are the Specific Operating Parameters, representing peculiar of a specific kind of machine (or at least of a family of machines). They can often be expressed in non-dimensional way to allow comparisons between similar components. They may assume also the aspect of simple coefficients conceived to represent the main characteristic of a machine with single value or through couples of variables (e.g. specific speed and specific diameter).

Secondary Parameters (SP) can be also defined to describe some very detailed aspects of the considered components, being generally function of SOP as indicated in Eq. 2.

$$SP = h(SOP) \tag{2}$$

The identification of the relationships occurring among parameters and between variables and parameters, allows expressing each physical and geometrical characteristic (e.g. mass, diameter, etc.…) as a function of this set of parameters. In the third phase, through the exploitation of results coming from empirical estimations, simulations or literature, it is possible to find out the proper mathematical formulations to represent those relationships occurring between each variable and its subset of dependent parameters. A generic estimation relationship F can be formalized in the form expressed in Eq. 3.

$$F = f[g(GOP, SOP), h(SOP), TP] \tag{3}$$

Where

- TP are the Technology Parameters reflecting the technology maturation level of a component. They may be expressed through coefficients and corrective factors, allowing a higher flexibility of the model (since they may be adopted to enlarge the limits of the correlations) even if causing a sort of discontinuity with the aim of the "exact formulation" of the estimations.

The implementation of the methodology used to derive the parametric semi-empirical models for mass, volume and power estimations of the different components is presented in detail in [6].

4. Derivation and implementation of Estimation Relationships (ERs) for main TEMS components

4.1. Introduction

Following this approach, it is possible to derive parametric models for each component of the TEMS, with the final goal of obtaining preliminary results for their sizing. Generally, mechanical components can be divided in active and passive, looking at their capability of generating or consuming power (e.g. pumps and turbomachinery are active components whilst pipes and tanks are example of passive components). The main elements (Fig. 1) are then selected for the sizing procedure and, notably, turbopump, compressor, turbine, tanks, pipes and heat exchangers. Sections 4.2 to 4.6 briefly presents the parametric models built per each component and the first results obtained considering the hypotheses on the case study for a specific design point. Section 4.7 collects the results for the whole TEMS subsystem.

4.2. Turbopumps

Turbopumps are widely used devices for different aerospace applications, from engine feed (typically for rockets), to thermodynamics cycles that require moderate to high power. Different models for performance estimations are present in literature, starting from theoretical derivation of the

operational parameters [8]. A typical and useful characterization of these machines can be performed through the computation of two main parameters, the specific speed N_s and the specific diameter D_s [9]. Through these numbers, different machines can be compared directly on a single diagram, known as the Balje diagram, that represents a sort of map upon which many other coefficients can be plotted (as function of these two main parameters). The different types of turbopump architectures can be identified on the diagram, depending on the ranges of N_s and D_s. From speed and diameter numbers other variables and coefficients can be derived, as the head ψ and flow φ coefficients.
In analogy with the definition of N_s and D_s, similar constituting parameters, namely the speed σ and diameter δ numbers, can be computed, generally referring to a simpler diagram known as Cordier diagram.
It has to be noticed that the reference diameter plays a fundamental role in the turbopump model. Indeed, it can be used to determine other features like length, volume and mass (Fig. 3) [7]. Thus, the main objective of this turbopump model is to determine their diameter as first step for the overall characterization. Considering the governing equations, the final budget can be generically formulated as in Eq. 4:

$$\mathcal{G} = g(GOP, SOP) = \ g(\sigma, \delta) = g(N, Q, H, D) \tag{4}$$

Considering that the volumetric flowrate Q, rotational speed Ω and head rise H (or pressure rise Δp or specific head Y) are supposed to be known by the user, since a first evaluation of the performance should be available at this stage of design, the computation of specific speed Ω_s and speed number σ is straightforward. As final step, it is possible to use the definition of optimal efficiency to relate σ and δ so that the diameter can be derived [6].

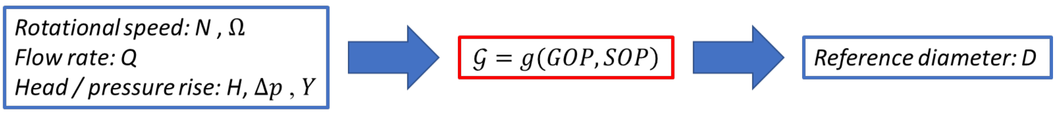

Fig. 3. Variables flowchart for input/output of governing equation of turbopumps

The impeller diameter is directly related to the specific diameter by definition. From the analysis of the various types of existing pumps, the inverse formulation is only able to differentiate between machines with different operating fluids but it does not contain any information related to the impact of number of stages of the impeller, as well as of the impeller constructional type. In case of multi-stages impellers, it shall be considered that they are mounted on the same shaft and, thus, they run at the same rotational speed. Moreover, since they are acting in series, they are supposed to handle the same fluid flow as an ideal single stage, with the same total developed head. As a consequence, the impeller diameter of a multistage configuration should be smaller. Thus, the previous equation can be corrected making benefit of the affinity laws stating that:

- the pump volume flow rate varies directly with the rotational speed
- the pump-developed head varies directly as the square of speed

Assuming that the head developed by each stage of the impeller is equal to the head developed by the ideal single stage machine divided for the number of stages, the following equation (Eq. 5) is suggested.

$$D = \frac{D_s Q^{\frac{1}{2}}}{Y^{\frac{1}{4}}} \sqrt{\frac{1}{(n_{stage})_{imp}}} \tag{5}$$

Considering the possible different configurations for each of these elements, a model able to express all the missing contributions as proper percentages with respect to the impeller diameter has been developed, giving suggestions on how to tune these percentages depending on the characteristics of the element that is considered. It is worth noticing that the identification of proper factors representing the impact of the different components on the pump diameter has been based on an in-depth investigation of several components. The analyses demonstrate that the estimation of diameter is affected by the presence/absence of a diffuser, the inlet and the outlet geometry, the presence of other elements (such as inducer, connectors, etc..).

A similar approach can be used to evaluate the overall length of the turbopump on the basis of its constituent components. The main contribution in this case is due to the impeller. The length of the impeller can be usually estimated on the basis of the mass flow that the pump should handle and remembering that the length of the impeller is in proportion with the inlet pump diameter. In addition, the type of impeller as well as the number of stages shall be considered. The other elements contributions are usually estimated as percentage with respect to the impeller length, apart from bearings and turbines, for which equations coming from some best practices may be used. Then, a high-level estimation of the volume can be executed by simply multiplying the diameter of the machine and its length, while the masses of pump can be evaluated taking into account a proper integration coefficient is used to make the estimation sensible to different range of speed numbers and mass flow rates. Eventually, power budget for the overall turbopump can be evaluated with a simple formulation, where the power required is the product of volumetric flow rate and delivery pressure with an additional efficiency.

This approach has been exploited to size the turbopump of the LAPCAT MR2 vehicle hypothesizing a high rotational speed ($N = 35{,}000\, rpm$), the highest mass flow ($\dot{m} = 100 \frac{kg}{s}$) rate and head rise ($\Delta P = 80\, bar$) and considering a single stage impeller and a two-stages turbine in a direct driven configuration. Moreover, additional assumptions related to more constructional details should be done in order to properly estimate diameter, length and eventually, mass budget. As far as the fuel pump is concerned, LAPCAT MR2 hypothesized turbopump, a baseline configuration with a single stage pump, with frontal inlet and with an inducer has been envisaged. Moreover, the diffuser will be a volute diffuser, trying to maximize the developed pressure head
These assumptions are collected in Table 2, together with the results.

Table 2. LAPCAT MR2 – outputs of turbopump sizing

	Hypotheses			**Items**	**LAPCAT MR2 estimated value**
	Input	**Value**	**Comment**		
Diameter	$k_{D/diff}$	*1*	*Volute type diffuser*	D_{TP}	0.549 m
	$k_{D/inlet}$	*0*	*Frontal inlet*		
	$k_{D/outlet}$	*0*	*Not applicable*		
	$k_{D/misc}$	*0.01*	*Compact design*		
Length	$k_{L/ind}$	*2.5*	*Inducer*	L_{TP}	0.873 m
	$k_{L/pb}$	*0*			
	$k_{L/out}$	*1*	*Mean of the case-studies*		
	$k_{L/other}$	*2*	*Compact design*		
Mass	I_N	0.6	*Considering N=35000 rpm*	M_{TP}	456 kg
	$I_{\dot{m}}$	1	*Considering a flow rate of 100 kg/s*		

4.3. Compressors and turbines

Gas turbines are one of the most used category of turbomachines through which it is possible to transfer energy from a driving fluid to a rotor or a shaft in order to produce mechanical and/or electrical power. In aeronautics, gas generators based on Joule-Brayton cycle are the core of airbreathing engines, like turbojets, turbofans and turboprops, which usually adopt them as internal stage constituted by a compressor, a combustor and a turbine. However, gas generators cycles are also selected for specific on board subsystems to provide a dedicated source of power in operation, to feed some other devices to which they are mechanically connected or to exploit particular types of thermodynamic cycles like the Rankine or vapour cycles for thermal management systems.

All of these machines have in common the working principle, which is based on a set of operating variables determining some families of input and output parameters. Looking at the gas generator it is possible to represent a general scheme of the machine and of its working parameters as in Fig. 4.

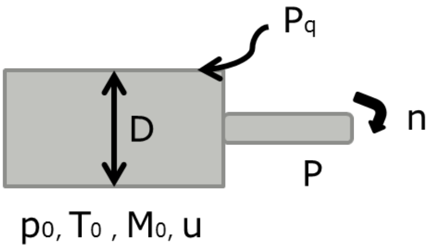

Fig. 4. General schematic of a gas generator

During the conceptual design, it is a common procedure to refer to specific performance, in order to be independent from the dimensions of the machine. However, when looking at off-design conditions it is better to refer to non-dimensional variables and, in case of a fixed fluid and machine size, to the so-called corrected performance. Actually, for the purposes of this study, specific and corrected performance are not so suitable to identify the physical characteristics of the assembly since they do not include them in the formulation. On the other hand, dimensionless variables can represent in a better way the phenomena affecting the behavior of the component as well as its physical parameters.

For this reason, in order to identify a proper "shape" of the governing equation, dimensional analysis has been performed. The classical approach, also described in literature [10], makes benefit of the Buckingham π Theorem, stating that if there is a physically meaningful equation involving a certain number n of physical variables, then the original equation can be rewritten in terms of a set of $\pi = n - m$ dimensionless parameters, where m is the number of physical dimensions involved. In this case it is possible to say that the governing equation of the phenomenon can be written as Eq. 6.

$$\mathcal{G} = g\left(P^{\alpha}, D^{\beta}, p_0^{\gamma}, (RT_0)^{\delta}, u^{\varepsilon}, N^{\eta}, P_q^{\vartheta}\right) \tag{6}$$

Where

R is the individual gas constant in $\frac{J}{kgK}$

p_0, T_0 are the initial conditions of the fluid in terms of pressure and temperature $\left[\frac{N}{m^2}\right]$ and $[K]$

N is the rotational speed [rpm]

u is the speed of the fluid $\left[\frac{m}{s}\right]$

$P_q = \dot{m}_b H_i$ for a general arrangement where the combustor is included, having

$\dot{m}_b$ fuel mass flow in $\frac{kg}{s}$

H_i lower heat of combustion of the selected fuel in $\frac{J}{kg}$

Main operating parameters are then the rotational speed and the power of the machine as well as the initial conditions of the flow and the thermal power provided. The input summary necessary to obtain the main geometrical value through the governing equation is clarified in Fig. 5.

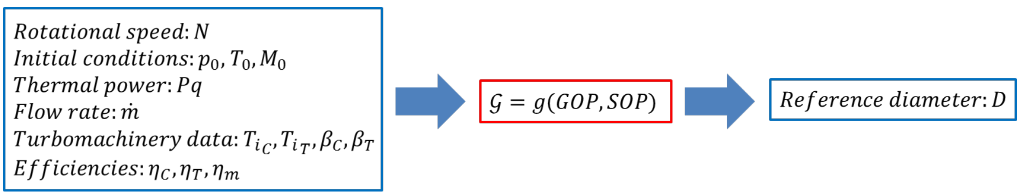

Fig. 5. Variables flowchart for input / output of governing equation of turbomachinery

Once that the reference diameter of the machine is derived, the identification of the dimensions and of the mass is possible. Particularly, [11] includes some semi-empirical models for the estimation of gas generator components of turbojets and turboshafts for aeronautical applications. As far as the performance envelope remains within the ranges prescribed by [11] it is reasonable to apply the methodology also for simple gas generators (i.e. not necessarily implemented within an engine cycle) by neglecting the terms related to inlet, fan stage and combustor (when this is not required).

The mass of the compressor is computed in [11] as function of the reference diameter D, of the number of stages $\mathcal{N}$ and on the length to inlet mean diameter ratio L_C/D. Particularly, the number of stages can be computed looking at the compression ratio of each single stage. Then, the mass of the compressor can be directly derived exploiting existing statistical correlations. Similarly, the length and mass of the turbine can be expressed as function of reference diameter, number of stages and rotating speed, as for the compressor. Additional contributions to the overall mass of the machine coming from controls, accessories and, notably, support structure can be added to have a clearer view over the real mass of the component. For the purpose of this study, a reference value in the range of 10 - 20% is suggested for controls and accessories. Moreover, this kind of approach can be adopted to compute the mass of structural components. The work performed in [11] suggest to use a reference additional percentage to the total mass which is around 10 – 18 %

Considering LAPCAT MR2 TEMS architecture (Fig. 1), the compressor receives the hydrogen boil-off coming directly from the tanks as well as a dedicated boil-off flow which is used to cool down the cabin. The two flows are mixed together before entering the compressor. On the other hand, the expander receives the liquid hydrogen which is used as fuel for the powerplant. The LH2 is also adopted as cooling fluid for the engines within a regenerator before entering the turbomachinery. Compressor pressure ratio and turbine expansion ratio can be derived looking at inlet/outlet pressure reported in Table 1. Moreover, even if the cooling system of the propulsion plant is not yet fully characterized, it is possible to hypothesize that the thermal power exchanged through convection is represented by Eq. 7.

$$Pq = hS\left(T_{powerplant} - T_{LH2}\right) \tag{7}$$

Where
h is the convective heat exchange coefficient of the fluid $[W/m^2K]$
S is the interface surface m^2
$T_{powerplant}$ is the temperature at powerplant interface $[K]$
T_{LH2} is the temperature of the fluid $[K]$

For the purpose of this study, the reference value of 1300 K is selected, as advised by [5]. Considering the aforementioned analysis and the general input coming from [5], a possible implementation for the boil-off compressor and the expander of MR2 TEMS system is here proposed. The application of the described methodology leads to the results collected in Table 3.

Table 3. LAPCAT MR2 – output of compressor and turbine sizing

	Items	LAPCAT MR2 estimated value
Diameter	$D_{Compressor}$	0.70 m
	$D_{Expander}$	0.50 m
Length	$L_{Compressor}$	0.47 m
	$L_{Expander}$	0.17 m
	L_{Total}	**0.64 m**
Mass	$M_{Compressor}$	168 kg
	$M_{Expander}$	97 kg
	$M_{Accessories\,\&\,Structure}$	80 kg
	M_{Total}	**345 kg**
Volume	$V_{Compressor}$	0.33 m^3
	$V_{Expander}$	0.08 m^3
	V_{Total}	**0.41 m^3**

It is particularly interesting to see that, with the proposed input and formulation, the resulting number of stages computed is about 5 (Table 4).

Table 4. Additional operating parameters for the case study

	Parameter	***Value***
Boil-off compressor	Rotational speed [rpm]	20000
	Number of stages	5
Expander	Rotational speed [rpm]	20000

4.4. Tanks

Tanks are one of the most common element of a hydraulic system, having the main function of collecting the fluid necessary for the operations and of maintaining the environmental conditions necessary to guarantee its characteristics within the specific ranges of application. The design of tanks and the selection of their type depend largely on the kind of fluid which they host and on vehicle integration aspects. For the purpose of this study, in order to evaluate the mass of the components, non-integral tanks are considered. This allows estimating the main characteristics of the vessels with a general approach, without restrictions related to specific shape or volume requirements. The proposed methodology is focused on the evaluation of the main structural aspects which determine the final mass and volume of the tank. Thus, the focus is mainly concentrated on the identification of pressure levels and induced stresses on the materials. Moreover, the analysis applied to LAPCAT MR2 (which adopts integral tanks) is mainly related to the selection of equivalent rigid tanks capable of hosting the same amount of LH2. The expected outcome is a fictional value of mass allocated on rigid tanks that can be used to understand the mass saving produced by the adoption of integral tanks. Independently from the type of the tank, the design shall start from the analysis of the required fluid volume. A general expression for the propellant tank design volume [12] is reported in Eq. 8.

$$V_t = V + T + B + U \tag{8}$$

Where
V_t is the tank design volume $[m^3]$
V is the actual volume of fluid required by the system $[m^3]$
T is the volume of fluid trapped within the system (not usable) $[m^3]$
B is the volume of fluid subjected to boil-off (only in case of cryogenic fluids) $[m^3]$
U is the ullage volume $[m^3]$

Together with volume, the assumptions related to wall thickness, material and shape are fundamental to complete the evaluation of dimensions and mass. The selection of material and thickness are actually related to the working loads, whilst the shape can be traded depending on the configuration.

Since in this case it may be difficult to represent separately governing equation $\mathcal{G}$ and estimation $\mathcal{F}$, the latter is directly included for clarity. The choice of the material is directly related to the maximum working tank pressure, determined by the operating conditions, which determines also the thickness of the wall. Moreover, the required fluid volume is function of working pressure as well for a given fluid. Thus the estimation relationship for volume and mass can be proposed as in Eq. 9.

$$\mathcal{F}' = f(GOP, SOP) = f(p_t, fluid, shape, material) \tag{9}$$

It is clear that, for fixed mechanical characteristics of the material, a lighter one guarantees a lower mass per unit volume.
The summary of the variable flowchart is shown in Fig. 6.

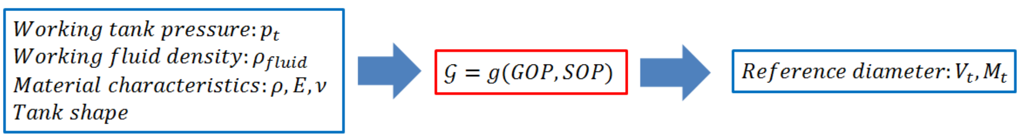

Fig. 6. Variables flowchart for input / output of estimation relationships for tanks

In particular, two kinds of rigid tanks are considered, as in [12]:

- Spherical tanks
- Cilindrical tanks (with both ellipsoidal and spherical tank ends)

LAPCAT MR2 has integral tanks. They have been grouped in 5 families [5] and, notably:

- Body Center Forward Tank (BCFT)
- Body Side Forward Tank (BSFT)
- Body Center Aft Tank (BCAT)
- Body Side Aft Tank (BSAT)
- Wing Tank (WT)

Since it is not useful to evaluate original tanks masses, being already included within structural mass of the airframe, a parallel approach is here proposed in order to evaluate the hypothetical mass of equivalent rigid tanks having cylindrical shape with both spherical and ellipsoidal tank ends. The reference volumes are reported in Table 5, whilst an Aluminium alloy based material has been selected as reference (characteristics in Table 6).

Table 5. LAPCAT MR2 LH2 tanks volume [5]

Tank location	Volume [m^3]
Wing	1356.64
Body Centre Aft	152.99
Body Centre Fwd	568.35
Body Side Aft	420.09
Body Side Fwd	233.42
Total Tanks	2731.49

Table 6. Characteristics of reference tanks material

Parameter	***Value***
Ultimate tensile strength [MPa]	290
Young Modulus [GPa]	70
Density [kg/m3]	2700
Poisson ratio	0.33

Results of the estimations are collected in Table 7. Working conditions have been kept constant and consistent with the study performed in [5].

Table 7. Mass estimation for LAPCAT MR2 case study in case of adoption of non-integral tanks

Tank	***Result considering spherical ends [kg]***	***Result considering ellipsoidal ends [kg]***
BCFT	3628	1898
BSFT (both sides)	1494	770
BCAT	846	465
BSAT (both sides)	2578	1380
WT (both wings)	8204	4186
TOTAL	16750	8699

4.5. Pipes

Together with tanks, pipes are another kind of simple passive components of a typical hydraulic system. Their main function consists in guaranteeing a continuous fluid flow within a prescribed path inside the system and among the different utilities. They are typically used to feed active components with the aim of reducing the pressure losses due to hydraulic friction and discontinuities like valves, junctions and turns. As for the tanks, the determination of the physical characteristics of the pipes is mainly related to the evaluation of the operating pressure. However, conversely to what happens for the tank, the volume occupied by a pipe is not so important since, apart from the determination of its diameter, it depends essentially to the distance that the duct shall cover and from the path it shall follow. For the purpose of this analysis, since the distances between the different elements of the circuits are difficult to be determined, a specific mass in kg/m for a simple straight path is considered.

The friction factor λ is function of Reynolds number and of material roughness. A rougher material will produce higher distributed pressure losses, thus requiring a higher inlet pressure to reach a fixed outlet pressure. For this reason, the operating pipe pressure p_p can be highly influenced by the selection of the material. As for tanks, it is not easy to distinguish between governing equations and estimation relationships, thus the general shape of an estimating function is directly presented in Eq. 10 and sketched in Fig. 7.

$$\mathcal{F} = f(GOP, SOP) = f\left(p_p, \dot{m},\ fluid, pipe\ material\right) \tag{10}$$

The pressure level is influenced both by the pressure required by the components fed at the outlet of the pipe (i.e. by the pressure level which shall be maintained at outlet) and by the pressure drop produced by the fluid flowing at a certain speed in a pipe made by a specific material. Moreover, mass flow influences both the pressure levels and the drops.

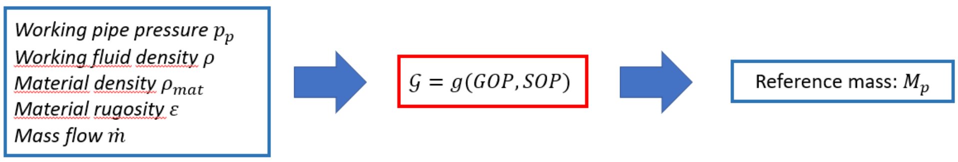

Fig. 7. Variables flowchart for input / output of estimation relationships for pipes

The results for the different parts of the circuit can be in any case determined following the input provided in [5] concerning pressure levels and diameters. Pressure levels indicated in Table 8 are referred to the pressure required at the end of the segment. The difference between the inlet and outer pressure is then the pressure drop due to pipes friction.

Table 8. Mass breakdown of the pipes for LAPCAT MR2 case study

Circuit segment	Pressure levels [MPa]	Mass flow [kg/s]	Diameter [m]	Specific Mass [kg/m]
From LH2 tank to pump	0.1	100	0.10	0.85
From pump to engine	8	100	0.10	1.8
From engine to expander	8	100	0.92	152.7
From LH2 tank to compressor	0.1	8	0.46	3.9
From compressor to expander	6	4.4	0.13	2.3
From compressor to air pack	6	3.6	0.13	2.3
From air pack to expander	0.075	3.6	0.13	1.1

All results are obtained considering aluminum alloy pipes, as already indicated for tanks in Table 6. The specific mass has been computed considering a minimum pipe wall thickness of 1 mm.

4.6. Heat exchangers

Heat exchangers are simple and effective solutions to control and manage thermal energy and related transfer in different situations. Being usually based upon conductive and convective heat exchange they are adopted to balance temperature of compartments through direct contact and/or by means of driving fluids. The sizing of such components is mainly related to the selection of the architecture and to the identification of the thermal power that they shall manage in operation. Estimation of mass and volumes of the exchanger is then performed looking at these main aspects, by using semi-empirical formulations usable in preliminary design, as proposed in [13]. The main strategy adopted for the proposed study is then to evaluate firstly the thermal power managed by the components, identifying proper governing equations, subsequently using the output of this first step to determine mass and volume as consequence. Considering two fluids flowing within an exchanger at a certain temperature, it is possible to say that the thermal power balance is expressed by Eq. 11.

$$\dot{Q} = \dot{m}_c c_{p_c}(T_{co} - T_{ci}) = \dot{m}_h c_{p_h}(T_{hi} - T_{ho}) \tag{11}$$

Where
$\dot{m}_c$ and $\dot{m}_h$ are the mass flows of cold and hoot fluids respectively in $[kg/s]$
c_{p_c} and c_{p_h} are the specific heat at constant pressure of both fluids in $[J/kgK]$
T_{co} and T_{ho} are temperature of cold and hot fluids at exchanger outlet in $[K]$
T_{ci} and T_{hi} are temperatures of cold and hot fluids at exchanger inlet in $[K]$

The governing equation of a general exchanger has the shape of Eq. 12 and is graphically represented as in Fig. 8.

$$\mathcal{G} = g(GOP) = g\left(\Delta T, \dot{m}, c_p\right) = g(\Delta T, h, S,) \tag{12}$$

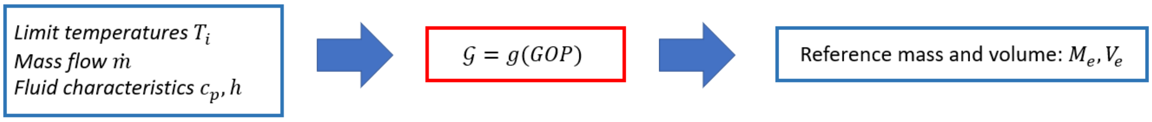

Fig. 8. Variables flowchart for input / output of estimation relationships for heat exchangers

The derivation of estimation relationships is then easy when considering semi-empirical models for the determination of mass and volume, since both estimations will be expressed as Eq. 13.

$$\mathcal{F} = f(GOP) = f\left(\dot{Q}\right) \tag{13}$$

Other dimensions can be computed basing on the exchange surface and volumes previously determined. The main operating parameter is then the thermal power, which can be computed by means of other variables.

The TEMS circuit indicated in Fig. 1 has four main exchangers for cabin, powerplant and air pack cooling. The exchanger for cabin cooling uses LH2 boil-off to cool down an air flow which is maintained between the LH2 tanks and the cabin walls in proper channels as indicated in Fig. 9 [5].

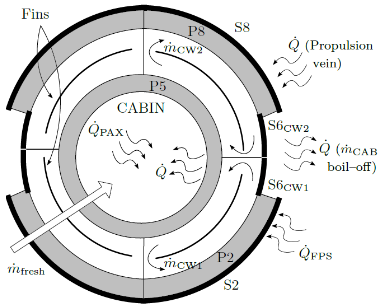

Fig. 9. Cabin cooling architecture [5]

Boil-off is also used as main fluid for cooling the high-pressure and high-temperature air flow coming from the air pack, which is used for conditioning and pressurization of the vehicle. Moreover, the cooling of the powerplant is performed through two regenerators using boil-off coming from the compressor and, on the other hand, by the liquid hydrogen pumped by the powerplant feeding system.

In order to perform the estimations, the assumptions listed in Table 9 are considered.

Table 9. Assumptions for MR2 heat exchangers characterization

Heat exchanger	Inlet temperatures [K]		Outlet temperatures [K]		Mass flow (cooling fluid) [kg/s]
	Hot fluid	Cold fluid	Hot fluid	Cold fluid	
Cabin exchanger	308	50	301	270	20
Air pack exchanger	3240	950	300	1200	3.6
Engine LH2 regenerator	1400	28	1000	1300	50
Engine boil-off regenerator	1400	950	1000	1200	4

The results included in Table 10 have been obtained for the considered exchangers.

Table 10. Results for TEMS heat exchangers estimation

Heat exchanger	Mass [kg]	Volume [m^3]	Exchange surface [m^2]
Cabin exchanger	493	2.3	14.4
Air pack exchanger	2267	10.8	37
Engine LH2 regenerator	159017	763	1659
Engine boil-off regenerator	2517	12	46.2
Total*	**5277**	**25.1**	**97.6**

* Excluding LH2 regenerator

As it can be seen, the results are very high, especially in terms of mass. The proposed relationships, in fact, use the physical breakdowns of dedicated exchangers which are not integrated within structural assemblies, being characterized by a conventional architecture (fluid heat exchangers with parallel or counter flow configurations). It is then clear how the integration of such components within structural elements is fundamental to reduce the required mass.

4.7. Overall subsystem

Considering the assumptions and results of Sections 4.2 to 4.6, the summary of Table 11 can be collected concerning TEMS mass breakdown. For pipes mass it is recommended to look at Table 8.

Table 11. Mass budget of TEMS

Component	Mass [kg]
LH2 Turbopump	456
Boil-off compressor	168
Boil-off expander	97 (177 considering accessories)
Heat Exchangers	5277
Total*	**6078**

* Excluding LH2 regenerator

Mass of gas generator accessories and structure has been allocated 1/3 on turbine and 2/3 on compressor. For the main parts of TEMS circuit pipes, the average value of 2 kg/m can be considered (this value is obtained neglecting the engine – expander path contribution). Moreover, it would be necessary to consider an additional mass in case of utilization of non-integral tanks of about 8700 *kg*. The mass allocated to heat exchangers can be also reduced depending on the level of integration of the devices with the airframe structure.

As far as LAPCAT MR2 TEMS volume is concerned, Table 12 provides an overview of the contributions of the different items to the final breakdown.

Table 12. Volume budget of TEMS

Component	Volume [m^3]
LH2 Turbopump	0.48
Boil-off compressor	0.33
Boil-off expander	0.08
Heat Exchangers	25.1
Total	**25.99**

Tanks total volume is about 2731 m^3 as specified in Table 5 whilst pipes volume depends on the routings and mutual position of the different components.

The evaluation of power consumed and produced by active components of TEMS allows determining the overall budget presented in Table 13.

Table 13. Power budget of TEMS

Component	Power [kW]
LH2 Turbopump	- 11300
Boil-off compressor	- 6000
Boil-off expander	+ 20000
Total	**2700**

Power has been computed looking at fluid volumetric flow rate and pressure levels (Table 1).

5. Example of sensitivity analysis of budgets for the turbopump component

The approach described in Section 3 and 4 allows performing a sensitivity analysis of components budgets directly evaluating the impact of operating parameters on results. This is valid both for active and for passive components since the methodology can be applied in a general purpose way. As example, this Section reports the some budgets trends for turbomachinery.
For these kind of components it is common to refer to map views. As example, the maps for mass and volume trends for the assembly compressor – turbine of the TEMS are shown in Fig. 10.

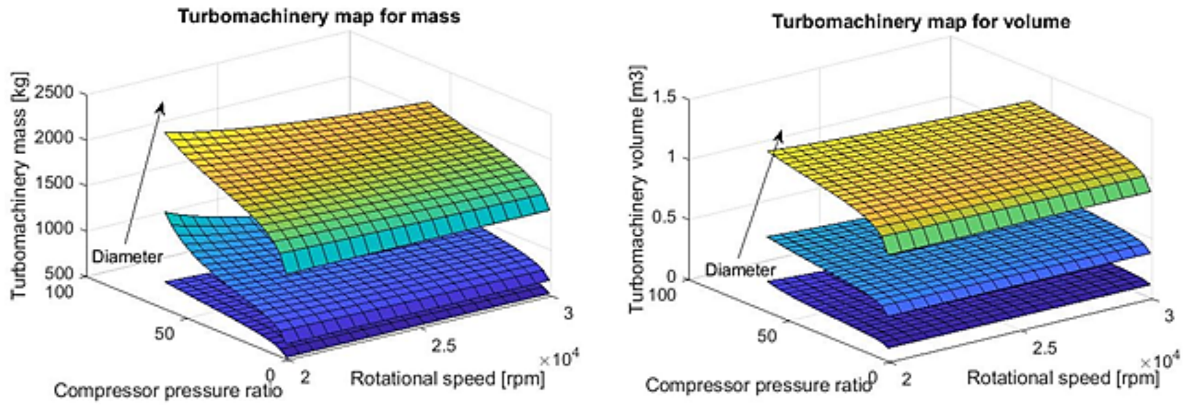

Fig. 10. Turbomachinery maps as function of operating parameters

The diameter is here a clear representation of machine dimensions and, thus, a higher value brings a higher mass. Moreover, the reduction of the pressure ratio also produces a considerable lightening of the machine. The contribution of rotational speed is less remarkable. For low diameters, high pressure ratio and low rotational speed a higher effect on mass increase can be noticed.
Additional views and representations can include 2D versions of the aforementioned maps, typically relating mass with rotational speed and diameters (Fig. 15).

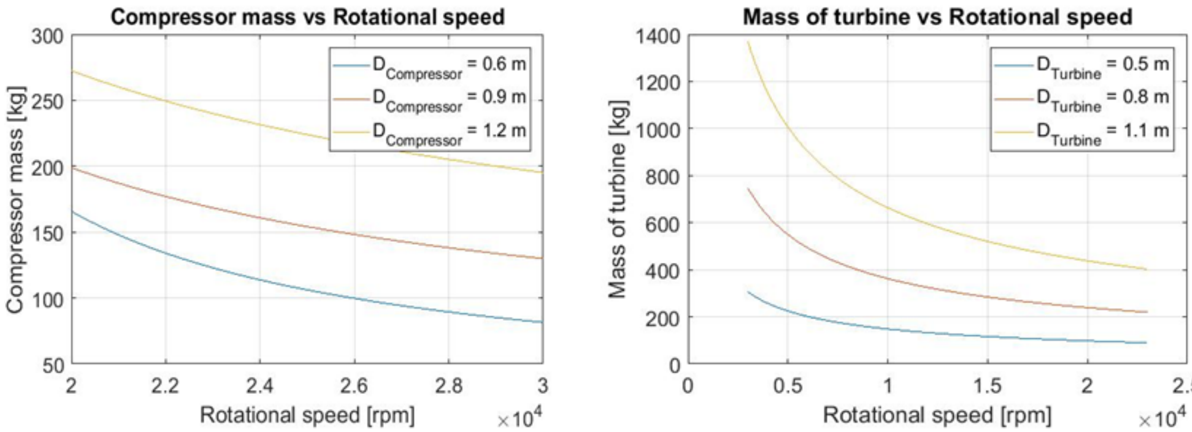

Fig. 11. Compressor and turbine mass as function of rotational speed and diameter

The increasing rotational speed produces a reduction of mass for both compressor and turbine. This reduction appears to have an asymptotic trend for turbine, whilst the theoretical mass of compressor does not reach a steady value. However, in both cases, it is reasonable to hypothesize that a minimum value, under which it is not possible to go for constructional reasons, shall be reached at a certain rotational speed.

6. Conclusions and future works

The proposed study allows determining a general approach for the estimation of main physical characteristics of the components of on-board subsystems for aerospace applications starting from the evaluation of the operating variables. The methodology proposed in this paper has been applied to the Thermal and Energy Management Subystem (TEMS) of LAPCAT MR2 as main case study. The components for which estimation is proposed are then turbopumps, turbine and compressors as well as tanks, pipes and heat exchangers. The approach appears to be feasible for a wide range of components and working conditions and provides an alternative solution to the pure empirical models present in literature. However, it represents only a strategy for evaluation of system budgets at preliminary design level, thus requiring detailed analysis for the evaluation of specific low-level results.

The application of the proposed methodology to the estimation of mass, volume and power of TEMS allows determining a mass of 801 kg for active components, 8700 kg for tanks, and 5277 kg for heat exchangers (in case of conventional devices) excluding LH2 regenerator. The total mass of the subsystem is then about 6078 kg plus the contribution of tanks (if applicable) and pipes. Moreover, the total volume of the system has been estimated around 2757 m^3 excluding pipes (but including tanks). The total mass of TEMS is about 2 % of the MTOW of LAPCAT MR2.

Power budget of active components has been evaluated and a total value of 20000 kW for produced power and of 17300 for consumed power have been obtained. Future improvements of the estimation relationships described in this study will mainly deal with enhanced and more detailed equations for the computation of the breakdowns of the different components and, notably, compressor, turbine and passive elements, which stands at a lower level of detail if compared to turbopumps. This is more critical when considering those components actually highly integrated with structural elements for which it is not possible to refer to pure literature models as basis for the estimation (e.g. heat exchangers etc…). Moreover, since most of the relationships should be applicable independently from the fluid used within the circuits, a dedicated analysis on the impact of the kind of fluid (either gas or liquid) shall be carried out as well as a complete sensitivity analysis on the different operating parameters.

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