Automatic and Detailed Analysis of Customer's Requirements for Rapid Decision Making in Actuation Design

Professor Dr.-Ing. Jean-Charles Maré

Université de Toulouse; INSA, UPS; Institut Clément Ader; 135, avenue de Rangueil, F-31077 Toulouse, France, E-mail: jean-charles.mare@insa-toulouse.fr

Abstract

The analysis of customer's requirements is addressed in the case of actuation design. Once introduced the need for early decisioin making in the today's very challenging times, the focus is put on mission profiles that can provide efficient indicators for performance assessment and design comparisons. It is proposed to firstly condition the mission profile and then to extract quantities that are representative of the energy and power needs, the thermal and mechanical fatigue and the control performance in the frequency domain. A particular attention is paid to the consistency of power and control performance requirements then preliminary calculations are proposed to facilitate the early sizing evaluation of control demand for electrohydraulic or power by wire actuators.

KEYWORDS: actuator, mission profile, preliminary design, requirements, sizing

1. Introduction

1.1. Industrial and economical context

In a highly competitive market that can no more put aside the need for a more sustainable development, the mechanical power transmission industry is currently facing challenging issues. This particularly deals with global performance optimization, energy saving, robustness improvement and time to market shortening. In the field of actuation, and more especially for embedded applications, the model-based design has recently made a step forward with the increased offer of advanced computer aided tools that can enhance and speed up not only modeling and simulation but also design exploration, robustness evaluation or even more global optimization. In parallel, electrical drives, especially high performance brushless DC (BLDC) motors combined with motor controllers and power electronics have opened new opportunities to the fluid power industry. By a judicious combination, the electric and hydraulic technologies can

merge their advantages to better meet all customers' requirements, not only for the main power transmission function but also the often neglected secondary functions:

- For the electrical technology, easiness of control, efficient power modulation, light and efficient power distribution, easiness of power supply re-configuration and energy recovering under aiding loads.
- For the hydraulic technology, high power density of hydromechanical power transformers, low speed / high effort mechanical output directly driving the load, easy bypass (null force), easy cushioning at stop ends, easy overload protection and reduced reflected inertia.

Even if it is often pushed as the ultimate solution, electromechanical actuation is still neither attractive nor mature for a lot of applications. This typically comes from the acquisition and maintenance costs (especially for rolling elements) and the complexity of design solutions for overload protection, heat rejection, tolerance / resistance to jamming and cushioning at stop-ends.

1.2. Customer's requirement document

At the very beginning of projects, assessing the functional and conceptual architectures with respect to the customer's requirements is a critical task that early engages the final performance. Unfortunately, this step is often shortened to meet the project planning: the main constraints are set up backwards from the delivery date, considering the almost incompressible activities of testing, integration, machining and procurement.

For the preliminary design, it is of particular importance to facilitate early decision making while a very few data are available. Therefore, producing a complete requirement document is a key work that must be carefully accomplished by a close collaoration between the customer and the supplier. For architecture selection and sizing, this document should typically address:

- modes of operation,
- power capability,
- performance of the control of power (accuracy, stability, rapidity),
- reliability,
- environment (temperature, vibration).

1.3. Mission profile

As both fatigue, service life and reliability are closely linked, the requirements should provide to the designer relevant data to consider these topics in parallel with the power and control requirements for the preliminary sizing.

The power requirement is most of the time defined simply by the stall force F_0 or torque T_0 (no velocity), the no-load linear velocity V_0 or angular velocity Ω_0 and the nominal force F_n or torque T_n at nominal velocity V_n or Ω_n . It can also be given as a velocity vs. effort plot that represents the power need plot in the load power plane. It is not a generalized practice to go deeper with mission profiles giving the time history of the power variables (effort and position / velocity) at the actuator/load interface. Another solution when the power need is given as an effort / velocity plot in the power plane can be applied adding marks equally time spaced to graphically display the transient and the steady operation areas.

For the very common valve-controlled hydraulic cylinders that basically waste pressure or flow to meter the power delivered to the load, there is no thermal issue at the actuator level: in addition to its main function of power conveyor, the hydraulic fluid is used as a heat conveyor to evacuate heat from the actuator. Moreover, the actuator reflects most of the time a negligible inertia to the load. For these reasons, the actuator sizing for power capability can be performed using only the stall effort, the no load velocity and the nominal effort at nominal velocity or, if available the power need locus. In both cases, there is no time information. The actuator can be designed without any particular consideration to thermal fatigue or effort need to accelerate its own inertia. On its side, the mechanical fatigue can be addressed using a generic duty cycle (constant or sine effort and velocity) that enables accelerated tests through a simplified reproduction of the product life.

This situation changes significantly with power-on-demand architectures involving:

- hydraulic motor displacement control (e.g. in aerospace for Airbus 380 flaps and slats power control units),
- power-by-wire actuators using electrically-controlled pump velocity (e.g. Airbus 380 and 350 electro-hydrostatic actuators for backup primary flight controls),
- or even full electromechanical actuators (e.g. electromechanical actuators for the Boeing 787 electromechanical backup spoiler).

This time, the thermal effects cannot be neglected as they affect a lot the limits of operation and the service life though their influence on fluid properties, seals life, coils and windings temperatures.

Moreover, even if the mission profile was not mandatory for sizing in the past, it also becomes a major input data, as soon as energy saving is addressed. Unfortunately, defining a representative mission profile is a critical task for many reasons:

- it fully drives the sizing of power-by-wire actuators,
- it must be representative of the effective use, whatever the customer, the environment, the operating and maintenance practices,
- for position actuators, the velocity/position time history is easy to record while the effort time history is rarely known. The effort delivered to the load highly depends on the instantaneous load and its ageing (e.g. increased friction or backlash). It cannot be easily recorded without installing dedicated effort sensors that require significant modifications in the power transmission path to the load.

2. Extracting indicators from the mission profile

To enable early and efficient decision making, the power need has to be defined at the actuator / load interface using the extreme and nominal power variables (F_0 ,/ T_0 , V_0 / Ω_0 and F_n / T_n at V_n / Ω_n) in combination with a typical mission profile. For position control, it is also a common practice to mention in addition the load inertia.

2.1. Mission profile conditioning

The mission profile is generally defined by top level designers according to the different phases of operation. However, it has to be conditioned to efficiently enable architecting and preliminary sizing, especially considering the followings:

- the mission profile, being often generated by assembling the different phases of operation, is not continuous or derivable; velocity and acceleration that are got by derivation of the load position can display huge discontinuities or infinite values.
- in long missions where some phases are rather dynamic and others are rather static, a non constant the time sampling is often used to significantly reduce the amount of stored data to make the mission profile.

When attempting to condition the mission profile, different issues must be fixed. First, filtering the mission profile must conserve the consistency between the position and

effort data. Second, the filter must be sufficiently dynamic to not remove the transients that contribute to the power demand. Third, the filter must enable extracting first and second derivatives of the position signal to get velocity and acceleration.

As a first approach, it is proposed to pass all data in a centered and weighted moving average filter, e.g from Savitzky and Golay /1/. As a major advantage, this type of filter does not introduce any phase lag while it outputs in addition the filtered first and second derivatives of the original input signal. Its major limitation comes from the need for a constant time sampling.

As an alternative, it is suggested to implement a non-linear second-order state observer of an equivalent actuator having the specified dynamics and the specified extreme power capabilities, **Figure 1**:

- the filter is presented in a block diagram view, as its natural decomposition,
- the filter dynamics (natural frequency ω_n) and stability (dimensionless damping factor z) are set according to the closed loop performance requirements,
- the linear filter is modified to bound speed and acceleration, according to the values mentioned in the requirements while ensuring consistency between position, velocity and acceleration (simply inserting saturations on position, velocity and acceleration in the upper path would not ensure consistency!),
- if required, a position limitation can be implemented in the same manner as the velocity one. In this case, the deadzone represents an equivalent stopend stiffness that must be combined with an additional viscous damping. In order to ensure acceleration continuity at stop end, the damping must start to establish when the stop-end is reached and is fully established for a given deformation.

The gain applied to the velocity limit is typically set to 1000. Getting a rigorous consistency between position and effort remains a real issue if the effort required by the load depends on the filtered and limited position or velocity (to be solved with the issuer of the mission profile).

This approach focuses only on the pursuit function (following the demand) but not on the disturbance rejection function (rejecting the influence of transmitted effort on the controlled output). The same filtering approach can be developed if the rejection requirements are more detailed than simply specifying a static error under permanent load, as usually done.



Figure 1: Conditioning the position mission profile

2.2. Mission profile analysis

Once conditioned, the mission profile is ready for detailed analysis. Considering the time histories of effort, position, velocity, acceleration and instant power, the following quantities can be expressed:

a) Extreme values

Extreme values provide the designer with the peak values during transient operation. When the peak values are specified, the consistency with the mission profile can be checked.

b) Mean values

Mean

Simple mean values are often of little interest as they do not consider the major non linear effects driving the sizing. However, they can serve as first indicators for architecture and technology selection.

Root mean square

In power by wire actuation, keeping the temperature of windings, coils and power electronics at an admissible level is often a major sizing driver. The heat generated by copper or conduction losses is proportional to the squared current that is itself proportional to the produced electromagnetic effort (force or torque) generated by the electrical machine. The mean heat is consequently calculated as the root mean square (RMS) of the demanded effort that takes the form of (1) in case of rotary motion:

$$T_{rms} = \sqrt{\frac{1}{t_f} \int_0^{t_f} T^2(t) dt}$$
(1)

with *t* the time variable, t_f the total duration of the mission, *T* the instant torque to load, T_{rms} the equivalent root mean square torque at load interface.

Root mean cube

Mechanical components like rolling elements (e.g. bearings) do not display any endurance limit above which they can endure an unlimited number of load cycles. Their sizing is mainly driven by the mechanical fatigue that makes an exponential dependency of load to the number of cycles before failure. In the case of variable loads, an equivalent root mean cube (RMC) load is calculated from the Fischer formula (2) as:

$$T_{rmc} = \sqrt[3]{\frac{\int_0^{t_f} \omega(t) T^3(t) dt}{\int_0^{t_f} \omega(t) dt}}$$
(2)

with $\boldsymbol{\omega}$ the load angular velocity.

c) Integral of absolute

Calculating the integral of absolute data is of particular interest. When applied to velocity using (3), it generates the cumulated travel θ_c of the load during the mission. For hydraulic actuators, this quantity is a good indicator of the functional volume of fluid spent to perform the mission and of the seals wear. When the actuator is fixed displacement and supplied at a constant pressure, multiplying the total travel by the stall effort gives an estimate of the hydraulic energy to be supplied, including the load demand and the energy loss at the valve orifices.

$$\theta_c = \int_0^{t_f} \left| \omega(t) \right| dt \tag{3}$$

When applied to the power delivered to the load (4), it provides an indication of the total energy \mathcal{E} to be delivered to the load if the power transmission systems is non regenerative (e.g. valve control).

$$\mathcal{E} = \int_0^{t_f} \left| \omega(t) T(t) \right| dt \tag{4}$$

The interest of energy recovery can be easily assessed calculating the integral of positive (or negative) power that represents the energy \mathcal{E}_+ delivered to the load (or the energy \mathcal{E}_- potentially recovered from the load).

3. Dynamics

Architecting and preliminary sizing must not dissociate the power capability and the performance of the control of power, each one being very dependent on the other. However, the designer often misses approaches to link the power and control performance requirements, especially considering dynamics.

3.1. Thermal time constant

In power-by-wire applications, the main definition parameters of the electrical motor are the peak and the continuous torque. Calculating the RMS torque from (1) to specify the equivalent continuous torque is only relevant if the motor time constant is long in comparison with the torque demand dynamics. For long missions, it becomes interesting to calculate an indicator of the torque demand dynamics from the thermal point of view. This can be performed filtering the torque demand by a first order function of time constant τ as (5). The first order can be seen as a simplified representation of the motor windings temperature / torque dynamics.

$$T_f(s) = \frac{1}{1+\tau s} T(s)$$
(5)

with T_f the filtered torque demand.

Two types of indicators are therefore suggested:

- the filter time constant τ_x that makes the peak filtered torque over the mission x times lower (e.g. 2) than the non filtered peak one.
- or the factor k_y of reduction of the peak torque when filtered with a time constant y (e.g. 30 mn).

When related to the range of the power need (by an appropriate selection of x an y), these indicators provide a clear view on the mission dynamics with respect to thermal transients.

3.2. Making the dynamic requirement

When the actuator is not itself a component of an outer loop belonging to a larger system, there is no particular need to manage the phase lag it introduces. Therefore, the control performance is often specified in the time domain in terms of overshoot and tracking error. In this case, the mission profile can be used to identify the equivalent required dynamics, assimilating the system under design to a non linear filter as given

on Figure 1. The damping is set according to the specified overshoot while the natural frequency is set to meet the tracking requirement.

Another approach can be found in applying a fast Fourier transform (FFT) to the mission septpoint (e.g. position demand) and disturbance (e.g. load effort). Plotting magnitude vs. frequency provides a useful view into the frequency spectrum of these signals that links the time domain of the mission to the frequency domain. However, this approach requires the mission profile to be sufficiently continuous and equally time sampled to not generate alias frequencies.

3.3. Consistency between power and control performance requirements

It is quite common that the power and the dynamic requirements are set independently without particular check of consistency. It is therefore of particular importance to determine which requirement is the most demanding in terms of power. This is illustrated with the two following examples.

a) Example 1

The first example comes from the design of an electrohydraulic actuator for the thrust vector control a first stage of a commercial launcher. The actuator is required to steer the nozzle at a no load velocity of 15°/s, a stall force of 450 kNm. The response time for a $\theta_0 = 1^\circ$ step is required to be $t_r \le 0.08$ s. The moment of inertia of the nozzle to be steered is 8000 km².

Assuming the actuator behaves at least as a second order system with a critical damping gives, for the required response time, a natural frequency ω_0 of

$$\omega_0 = 3/t_r = 35.7 \ rd/s \tag{6}$$

The peak velocity Ω_p during the step response of a critically damped second order system is

$$\Omega_p = 0.4596\theta_0 \,\omega_0 = 17.2\,^{\circ}/\text{s} \tag{7}$$

and the peak acceleration $(d\Omega / dt)_p$ is

$$\left(d\Omega/dt\right)_{p} = \theta_{0}\omega_{0}^{2} = 1400\,^{\circ}/\mathrm{s} \tag{8}$$

It produces an inertial torque of 197 kNm.

This fast analysis shows that the dynamic requirement is more demanding in terms of speed than the power requirement. Oppositely, the specified stall effort seems consistent with the inertial effort required to produce an acceleration meeting the required response time.

b) Example 2

This example comes from the design of a landing gear steering electromechanical actuator that is required to develop 8000 Nm stall torque, 20° /s no load velocity, +/- 40° travel with a nominal power of 1 kW at 10° /s. The specified accuracy is 0.1° and the bandwidth at -3 dB is 5 Hz. According to the level of power, the electric motor inertia is estimated to produce globally and equivalent steered inertia of 1000 kgm^2 .

The achievable steering angle magnitude for testing the frequency response is easily calculated assuming a sine load position and a pure inertial load. **Figure 2** obviously shows that the frequency response measurement is strongly limited by the acceleration and speed limits that are specified as power requirements.



Figure 2: Limit magnitude of the frequency response

3.4. Demanding dynamic requirements

For early decision making, including also budget and planning, the designers need to quantify how demanding the dynamic requirements are with respect to the amount of power and the load characteristics. According to the pre-selected technology, this question can be roughly answered even with a very few input data.

a) Position controlled electrohydraulic actuators

The dynamics of electrohydraulic actuators is driven by the combination of two second order dynamics (natural frequency ω_{sv} of the servovalve plus, hydromechanical natural frequency ω_{b}) where

$$\omega_h = \sqrt{\frac{2BF_0}{\alpha_v \eta P_s z_M M}} \tag{9}$$

with *M* load mass, P_s net supply pressure, z_M maximum travel from centered position are specified while *B* effective fluid Bulk modulus, α_v dead volume and η mechanical efficiency are to be estimated from usual values.

Once the hydraulic mode has been damped by an appropriate action, the practical closed loop stability under simple proportional control is achieved /2/3/ when the loop gain k_l of the actuator position control is typically set to

$$k_{l} = \frac{0.4}{1/\omega_{sv} + 1/\omega_{h}}$$
(10)

This value is directly linked to the closed loop response time $(3/k_l)$, tracking lag $(1/k_l)$ and bandwidth (k_l) . If a higher performance is required, a more advanced controller has to be designed or/and the architecting and power sizing choices are to be modified.

b) Position controlled power-by-wire actuators

The dynamics of electromechanical actuators is highly related to the amount of inertia that is reflected at load level by the actuator moving parts (motor rotor and reducer). Therefore, the power sizing of the actuator turns into an algebraic problem as the torque needed during the acceleration phases depends highly on the actuator inertia itself. In order to avoid multiple loops, the torque of the mission profile can be augmented by an estimated actuator inertia torque as explained below.

On basis former works using scaling laws /4/, it can be shown that the actuator reflected inertia J_a at load level rotary motion assumed) is

$$J_a = k_1 C_0^{5/3} + N^{2/3.5} k_2 C_0^{5/3.5}$$
(11)

with C_0 the rated torque at load, *N* the reducer ratio, k_1 the constant characterising the reducer (giving typically 6 10^{-5} kgm² for a 100 Nm output torque reducer), k_2 the constant characterising the motor (giving typically 4 10^{-4} kgm² for a 10 Nm BLDC motors).

The total mass of the actuator increases with the rated motor torque (i.e. reduces when the reducer ratio is increased). When the actuator mass is an important criteria, one can consider as a first approach that the reducer ratio N is determined to run the motor as rapidly as possible according to the level of required power P

$$\omega_0 = k_3 C_0^{-1/3.5} \tag{12}$$

The k_3 constant characterises the motor (giving typically 600 rd/s for a 10 Nm BLDC) and makes finally the actuator inertia reflected at load level as

$$J_a = k_1 C_l^{5/3} + \frac{k_2}{\left(k_3 \mathcal{P}^{1.9}\right)^{2/3.5}} C_l^2$$
(13)

4. Automating the customer's requirements analysis

All the above mentioned proposals of customer's requirements analysis can be easily implemented as a worksheet in order to drive the architecting and preliminary sizing activities. This has been widely employed and is still under extension in the authors research team with the objective to supply designers with preliminary design processes and their associated computer aided tools.

5. Conclusion

The analysis of customer's requirement is an important activity that drives decision making in the very early phases of a project. Accordingly, it has been proposed to extract different types of indicators in order to address globally and as soon as possible the power needs, the energy saving, the thermal issues, the natural dynamics and the control demand. It has been shown that the mission profile, although being rarely provided by the customer, is a key component of the requirements from which rich information can be extracted. Getting information on dynamics and checking consistency of the power and control performance requirement have been addressed to complete the preliminary analysis. All the suggested approaches can be easily implemented, extended and capitalized as an automatically calculated requirement analysis sheet.

6. References

- /1/ Savitzky A., Golay M.J.E. 1964. Smoothing and Differentiation of Data by Simplified Least Squares Procedures, Anal. Chem., 1964, 36 (8), pp 1627– 1639
- Maré J-C. 2002. Actionneurs hydrauliques Commande, Article S7531, série
 Mesure et Régulation, Encyclopédie des Techniques de l'Ingénieur, Septembre 2002
- Yu L., Maré J-C. 2002. Preliminary design rules for electrohydraulic position control actuation systems with structure compliance, Proceedings of the 2nd Fluid Power Net Phd symposium, July 3-6, 2002, Modena, Italy
- /4/ Liscouët J. 2010. Conception préliminaire des actionneurs électromécaniques
 Approche hybride, directe/inverse. Institut National des Sciences Appliquées de Toulouse, 2010