SELECTION OF SERVO MOTORS AND DRIVES

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Abstract: *The choice of motor and drive as well as mechanical transducer is a very important step in servo system design, because non-optimal selection leads to poor system performance and increased installation and maintenance costs. It is still an iterative process with skill in choice dependant on obtaining correct specifications and safety margins. Fundamental principles and procedures in selecting proper size motors and amplifiers are explained, and an example illustrating load analysis and motor/drive selection is included. It also describes methods of selecting other associated components such as power supplies, transformers etc.*

I. LOAD ANALYSIS

In order to select a motor of an appropriate size, speed and torque requirements have to be known first. Servo motors should have just enough speed, peak torque and rms torque capabilities, along with optimal gearing arrangement, to meet the load requirement as well as the cost objective. Equally important is selecting the type and size of the drive and power supply to meet the system requirements. Unfortunately, there is no simple straightforward procedure in servo system component selection due to complexity and variety of motor types, power transmission devices and other peripheral components available in the market. Nevertheless, a basic principle of analyzing the load and selecting a motor is presented here. It is assumed that the type of motor (brushed DC, PM synchronous motor, etc.) is already chosen. Refer to discussions on various motor types and their relative merits and demerits in previous discussion. To simplify equations, SI metric units are used throughout the article.

The load in motion control systems may be either rotational or linear. The load requirement must be converted to the equivalent load of the motor shaft in order to calculate the total torque requirement for the motor. If you have freedom in choosing gearing ratio, one desirable choice would be to convert the maximum load speed into the maximum motor speed. Since the maximum speed of the motor can only be known after the final selection, and for most motors available torque at maximum speed is smaller than the stall torque, an iterative process has to be taken. One may start with motor data selected from a rough estimation, verify the result, and then repeat the procedure with another motor. As a rule of thumb, one may start with a gear ratio that converts the maximum load speed into to half of the maximum motor speed. With this selection, nearly continuous stall torque is available at the maximum speed and the motor operates reliably at this speed without requiring frequent maintenance or bearing life. Some other mechanical constraint may govern the gearing ratio. It is very important that the shaft resonance frequency [1,2] has to be considered during early part of the design to eliminate unexpected servo performance. Once your gear ratio is determined, the next step is to determine peak and rms load torque seen from the motor shaft. For load analysis, it is convenient to convert all torque or inertia into the motor shaft. For rotational motion, the gearing ratio N is defined as

$$N = \omega_m / \omega_o, \tag{1.1}$$

where ω_m and ω_o are speed of the motor and load, respectively. Now, reflected torque and inertia to the motor can be calculated from load inertia Jo' and torque requirement at the load To' by the following equations,

$$J_o = J_o' / N^2 \tag{1.2}$$

$$T_o = T_o' / (N \eta) \tag{1.3}$$

Here, the efficiency of the mechanical converter is denoted by η. When a linear motion is required at the load, Eqs. 1.1-1.3 can be modified as

$$N = \theta m / Xo \tag{1.4}$$

$$J_o = m_o / N^2 \tag{1.5}$$

$$T_o = F_o / (N \eta) \tag{1.6}$$

where θm is the angular move of the motor shaft while the load moves the distance Xo, m_o is the mass of the load, and Fo is required force at the load. In Eq. 1.3 and Eq. 1.6, efficiency η is not negligible (for instance, η = 0.8 - 0.9 for Ball screws) and has to be included for accurate calculation.

The load torque of a motion system has the following components, which are reflected to the motor shaft as:

$$T_L = T_I + T_F + T_D \tag{1.7}$$

The inertial torque T_I is the torque to accelerate and decelerate the load, while the friction torque T_F includes load thrust torque and static frictions of the mechanical system. T_D is the viscous damping torque, which is proportional to the speed. In most cases, the inertial torque demands the peak torque, while other components are negligible during acceleration. When total inertia is calculated, inertia of the motor and the transducer must be included in addition to the reflected load inertia of Eq. 1.2 or Eq. 1.5. In other words, total inertia reflected on the motor shaft is

$$J = Jm + Jo, \tag{1.8}$$

and the inertial torque is

$$T_I = J\,\alpha, \tag{1.9}$$

where α is angular acceleration, which is the amount of speed change per desired time in case of trapezoidal velocity movement. Since the total inertia is unknown until motor is selected, one may use data from an assumed motor inertia and verify the validity after the selection process is done. When the application is to move an end effector vertically, counter-balancing weight may be added. In this case, total inertia must include the balancing weight, too. The advantage of counter-balancing is to have zero steady-state torque when the load is stopped. This advantage is obtained at the expense of increased inertia.

The load motion profiles, usually expressed as the load velocity vs. time, are used to calculate peak and rms torque. These profiles can be broken into several simple segments such as acceleration, high-speed run, deceleration, low-speed run (typically a machining segment), dwell, etc. Fig. 1.1 shows an example of a motion profile.

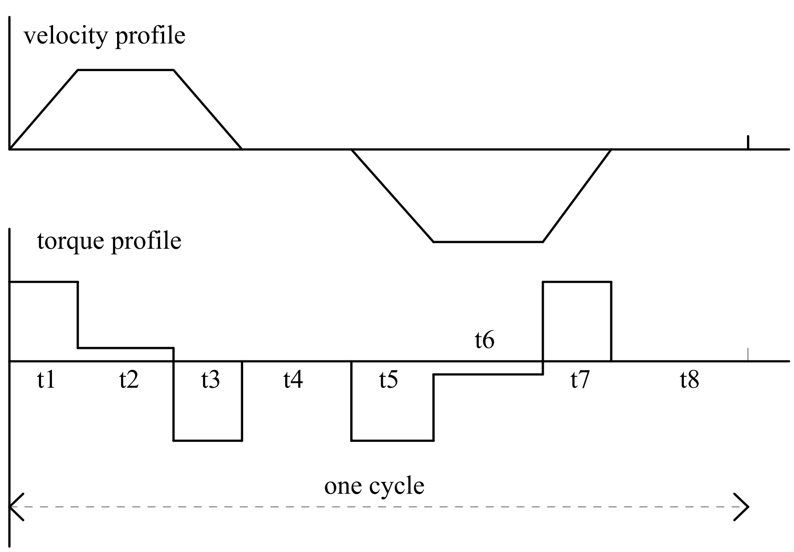

Fig. 1.1 Example of a Motion Profile

Peak load torque is simply the highest (motoring) torque demand found in one or more motion segments. The motor to be selected must be able to supply this peak torque at the desired speed for specified amount of time. The rms value of the load torque through the entire operation cycle can be calculated as

$$Trms = \sqrt{\frac{T_1^2 t_1 + T_2^2 t_2 + ... + T_n^2 t_n}{t_1 + t_2 + ... + t_n}}, \qquad (1.10)$$

where T_i is the load torque in i-th segment and t_i is duration of the i-th segment. A more detailed discussion on load analysis can be found in [3,4].

II. SELECTION OF A MOTOR AND DRIVE

Now, on the torque-speed curve of the selected motor, one can map calculated peak and rms torques can determine if the motor is suitable. If not, one can select a different size motor or the same motor with different winding for repeated procedure. Since motor performance is affected by the drive, it is best to work with system torque-speed curve. If system curve is not available, one may use motor curve to select the motor first, and then select a drive. When selecting a drive, it should be able to supply enough current and voltage to the motor to meet both peak and rms torque requirements. As a minimum, drive should supply peak and rms currents desired for the system calculated by

$$I = T / Kt, \qquad (2.1)$$

where Kt is the torque constant of the motor. Furthermore, the drive should be able to supply the voltage of

$$V = R I + Ke\, \omega_m, \qquad (2.2)$$

where Ke is the electrical (back emf) constant of the motor, R is the motor resistance, I is the required motor peak current obtained from Eq. 2.1, and ωm is the maximum speed of the system. Due to the variation of the load and possible overcurrent trip, often it is desirable use maximum motor speed and peak and rms torques of the motor rating instead of calculated requirement of the system.

Servo motors often require modifications necessary to meet the complex requirements of a machine. Standard options for the various models of servo motors include but are not limited to: fail-safe brakes; environmental sealing; feedback devices including tachometers, resolvers, and encoders; and connectors. Special options include modifications to the shaft and mounting surfaces, special windings and fan cooling provisions to mention a few. See specific product literature or contact manufacturer for availabilities. Operation in ambient environment other than that for which the motor is rated may require special consideration. Operation in ambient environment less than the rated temperature will result in a greater continuous torque rating but excessively cold ambient environment may require a different bearing lubrication. Operation in ambient environment above the rated temperature results in a derating of the motor continuous torque. The derating will be a factor of the motor design and its ultimate temperature rating. Some manufacturers offer gearheads as standard options on many motor models. They provide the convenience of having motor and gearhead sourced and tested from a single supplier. Gearmotors offer the following advantages to a servo application :

- The ability to operate the motor over its optimum speed range
- Minimize motor size by multiplying torque
- Minimize reflected inertia for maximum acceleration
- Provide maximum torsional stiffness

When selecting a servo drive, its interface with your controller must be checked. Traditionally, analog velocity command has been very popular. Recently, other forms of interface are available. Pulse type interface (either A-B pulse format or Pulse & direction format) is now popular. Some drives features position control (either programmable or non-programmable) function integrated into the amplifier. Considerations for type and resolution of the feedback device, I/O and other product features must be made.

Another important consideration in the selection of brushless motor amplifiers is the decision for analog or digital technology. Analog technology has been a standard for brush and brushless motor amplifiers. With analog amplifiers a voltage reference command results in speed or torque, depending on the mode of operation, proportional to the command. As well, compensation (tuning) of the amplifier is done by analog circuitry requiring potentiometer/switch and, or compensation "comp" board changes for different system configurations (mode of operation and motor & load variations). Digital electronics on the other hand, enhance drive features and minimize set-up concerns. The command reference for digital drives can be done by either analog or digital inputs. Digital input options are typically in a pulse format or through an RS 232 or 485 interface. Typically, better regulation of the system can be obtained by the digital command format. Digital drives also have the feature of tuning through a digital interface. This greatly benefits the issue of load compensation relative to inertia, friction and compliant factors which are often difficult and time consuming to simulate under lab conditions. Another benefit of digital technology is the diagnostics enhancements it offers. Monitoring of internal functions of the drive such as following error or current can be easily done. Fault diagnostics are typically more detailed than those with analog drives and fault histories are maintained through power loss.

III. SELECTION OF OTHER SYSTEM COMPONENTS

The power supply in a servo system fundamentally delivers DC power to a servo amplifier. This is done by the rectification and filtering of AC power delivered from a 1∅ or 3∅ main. Depending on the specific servo amplifier chosen, the power supply may be internal to the drive, for which no selection process is required, or external, thus requiring selection decisions. In addition to the basic power sizing requirement, typical DC supplies for servo amplifiers include shunt regeneration circuitry, which dissipates regenerative energy from the load when the motor is decelerated rapidly. The following information gives general guidelines for power supply and regeneration selection.

The power output rating of a power supply must exceed or equal to the combined average power of all servo drives operating simultaneously. The average power of an individual servo drive is based on the power calculation of rms torque and speed. Taking into account motor and drive losses, as a general rule, for permanent magnet servos (DC brush or brushless), 1 kW of input power is needed for every 750 watts (approximately 1 HP) of output power. In addition, power supplies often supply the logic power for the servo amplifier. The voltage ratings and ampacity of this power supply must also be appropriate for the amplifiers being supplied.

Consideration should be made for the motor thermal time constant if the on time at a peak torque value is for any substantial period. Typical complementary amplifiers, having a 2 times peak current rating for 2 to 8 seconds, do not present a problem even for small motors having a short thermal time constant. Problems may occur however if an amplifier is "oversized" for the motor meaning that the drive continuous current is greater than the motor continuous rating. In these cases, other evaluation methods can be used taking into account this motor time constant.

Regeneration is the process whereby kinetic energy of a motor and load is converted into electrical energy during deceleration by a four quadrant servo controller. Regeneration also occurs if motor rotation is opposite to that of the commanded torque as in an unwind web application. Dissipation of this electrical energy can be done by one of two methods. These are line regeneration and shunt (resistive) regeneration. Line regeneration is a more costly approach, which involves the conversion of DC power into AC power, synchronous with the AC source. (In battery operated systems, regeneration into the battery (recharging) will occur without the need for additional electronics.) Shunt regeneration is accomplished by the shunting (short circuiting) of the DC bus through a power resistor. This technique is more common and can handle higher transient loads than line regeneration. Shunt regeneration is a feature of most low power servo systems as part of the power supply. For decelerating loads, the amount of regeneration required is a function of the sum of simultaneously decelerating inertial loads where multiple amplifiers are being operated from a common supply. The loads need to be defined in terms of the system inertia (load plus motor inertia), maximum speed, and the deceleration time from maximum speed. These parameters define the kinetic energy that must be either stored or dissipated. The power supply has some capacity to store some of this energy, and some energy will be dissipated in the motor windings and amplifier circuitry. Any remaining energy will need to be dissipated in an appropriately sized regeneration system.

Transformers serve two basic purposes in a servo system: voltage matching and isolation from the line. Voltage matching is fundamental and required for many systems where standard AC line voltages like 115, 1∅ and

230 or 208, 3∅ are not appropriate. Line isolation is required for some products or can benefit other applications by the reduction of PWM noise for sensitive environments.

IV. EXAMPLE

An automatic component storage/retrieval system uses at least 3 axes of motion control. The elevator axis servo system as shown in Fig. 4.1 is to be designed. In order to have zero net torque when stopped, a counter-balance weight has to be added. It has the following requirements:

Total load mass = 64.2 lbs = 29.121 kg

(End-effector 31.0 lbs, Balance 31.0 lbs, Belt 2.2 lbs)

Maximum load speed = 90.18 in/sec = 2.2906 m/sec.

Friction at constant speed run = 10.0 lbf = 4.536 kgf

Motion profile

- total cycle period = 2.0 sec.
- accel time (0 to max. speed) = 0.2 sec
- maximum constant speed run = 0.4 sec.
- decel time (max. speed to stop) = 0.2 sec.
- pause 1.2 sec.

Pulley: 14 teeth, belt span = 0.5 in = 0.0127 m

[Selection Procedure]

(1) An initial estimated selection indicates that maximum motor speed is about 3000 rpm. Therefore, 1500 rpm is chosen to find the gear ratio. If gear ratio is selected as 30/16, then

$$N = (30/16)\, 2\,\pi / (0.0127 \times 14) = 66.2597 \text{ rad/m}$$

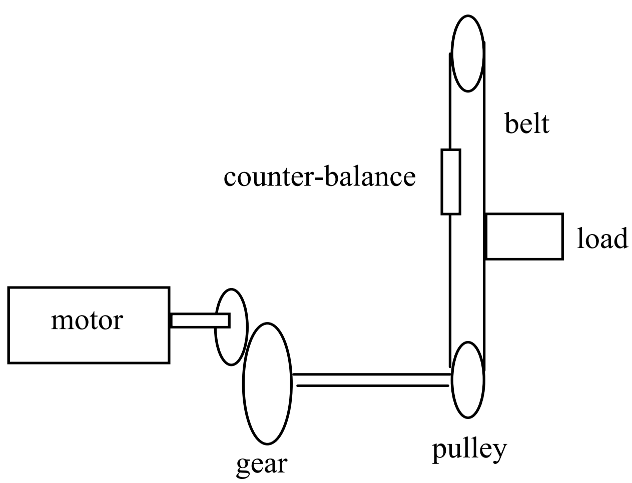

Fig. 4.1 An Elevator Servo System

which converts the maximum load speed into 151.774 rad/sec or 1449.3 rpm.

(2) Load inertia reflected to the motor shaft can be calculated (including power train inertia of 0.00066 kg m^2) as:

$$Jo = 29.121 / N^2 + 0.00066 = 0.007293 \text{ kg m}^2$$

Assuming that the motor inertia Jm is 0.00150 kg m^2, total inertia J is,

$$J = Jo + Jm = 0.008793 \text{ kg m}^2$$

(3) Friction torque, neglecting the efficiency, reflected to the motor shaft,

$$To = 4.536 / N = 0.06846 \text{ kgf m} = 0.67078 \text{ Nm}$$

(4) Peak angular acceleration α and peak torque T_{peak} are

$$\alpha = 151.774 / 0.2 = 758.872 \text{ rad/sec}^2$$

$$T_{peak} = J\alpha + To = 7.3435 \text{ Nm.}$$

(5) Rms torque can be calculated as

$$Trms = \sqrt{(2 \times 7.3435^2 \times 0.2 + 0.67078^2 \times 0.4) / 2.0} = 3.2978 \text{ Nm}$$

(6) After reviewing torque-speed curves of several motors, one ferrite DC motor is selected. Both peak and rms torque points are marked on the speed-torque curve of the selected motor. Its motor inertia is found to be 0.00167 kg m^2. After re-calculation, the selection is confirmed.

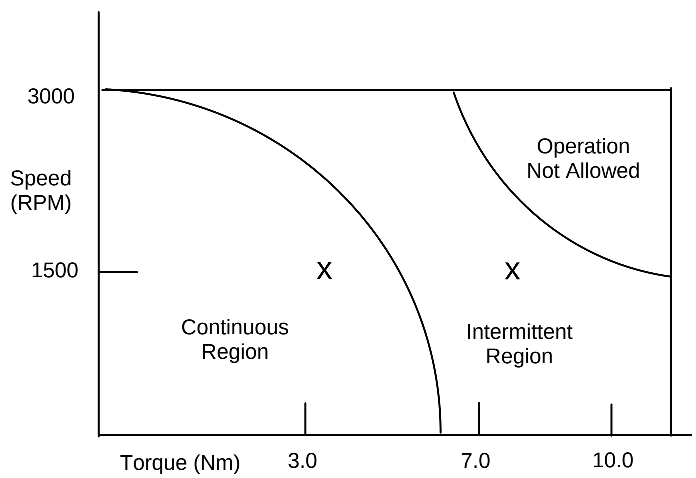

Fig.4.2 Speed-Torque Curve of a DC Motor

APPENDIX A. CALCULATION AND MEASUREMENT OF INERTIA

For simple geometric structure, it is possible to calculate from its density (ρ) and shape. For example, a disc shown in Fig. A.1A with its diameter of d, and thickness of t can be calculated by

$$J = (\pi / 32)\, \rho\, t\, d^4 \qquad [\text{kg m}^2] \qquad \text{(A.1)}$$

If the above disc has a concentric hollow of diameter d_2 as in Fig. A.1B,

$$J = (\pi / 32)\, \rho\, t\, (d^4 - d_2^4) \qquad [\text{kg m}^2] \qquad \text{(A.2)}$$

For steel, $\rho = 7.85 \times 10^3$ kg /m^3.

When the object is composed of combination of simple geometrical shapes, the total moment of inertia is the arithmetic sum of all moments of inertia of the components.

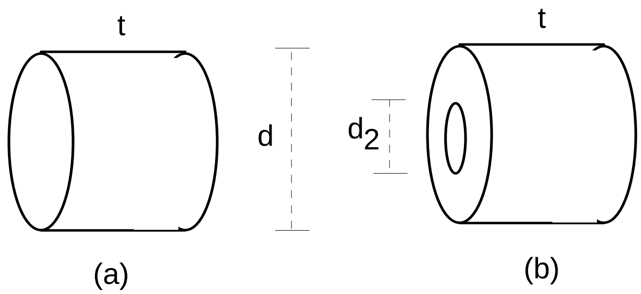

Fig. A.1

When the geometry is complex, using the calculation may not be a simple task. An alternative method is to use a one wire hanging method. As in Fig. A.2, an object is suspended on a piece of steel wire. First, an object of known inertia (Jo) is hung and the period of oscillation (to) is measured. For accuracy in measurement, one may use a period of 20 rotational vibrations and divide it by 20. Then, the object to be measured (Ju) is hung and its period (tu) is measured. The recommended torsional angle may be about 45 degrees. Now the unknown inertia Ju is

$$Ju = Jo\,(tu / to)^2 \qquad (A.3)$$

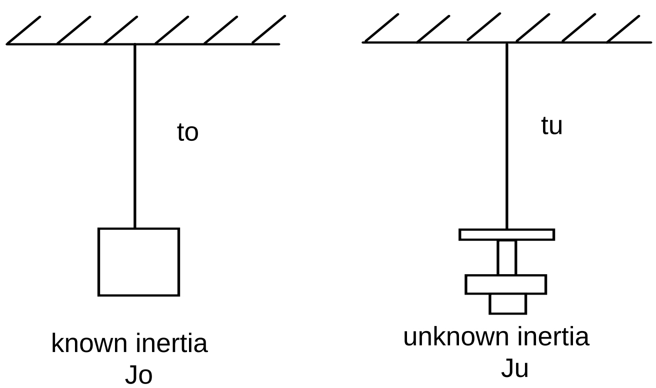

Fig. A.2 One Wire Hanging Method

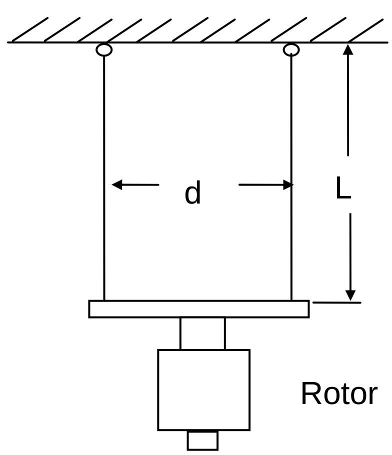

Fig. A.3. Two Wire Hanging Method

There is another similar method, which does not require the reference inertia. Suspend the armature with the shaft oriented vertically using two parallel wires as indicated in Fig. A.3. The wires should be attached diametrically, equally spaced from the center line of the shaft. The ratio L/d should be approximately 10. Rotate the armature by a small amount from the equilibrium position and release it. Measure the angular oscillation. The moment of inertia J [kg m^2] is determined from the following formula:

$$J = \frac{c\ m\ d^2}{L\ f^2} \qquad \text{(A.4)}$$

where m: armature mass (kg)

c: a constant 6.2×10^{-2}

Refer to [5] for detailed procedures.

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