Servo Motor Selection Criterion for Mechatronic Applications

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***Abstract*—Modern mechatronic applications often involve complex motions, resulting in highly dynamic motor loads. The selection of an appropriate motor is based on the characteristics of the load, besides other technical, as well as economic, considerations. However, motor characteristics, such as rotor inertia, affect the motor load, which complicates the analysis. The selection criterion presented in this paper separates the motor characteristics from the load characteristics and its graphical representation facilitates the feasibility check of a certain drive and the comparison between different systems. In addition, it yields the range of possible transmission ratios. The method is illustrated with an industrial case study.**

***Index Terms*—Drives, mechatronics, servo motors.**

I. Introduction

THE heart of many mechatronic systems is the servo drive generating nonmonotonic motions. Therefore, its load is much more dynamic than that of a classical drive. In general, such a load cannot be reduced to either a purely inertial load, requiring torques proportional to the acceleration, or an external load driven at constant speed; it is a mixture of both. This complicates the motor and reducer selection, and existing criteria turn out to be inadequate. The motor selection procedure can be subdivided into two parts. First, find all motors that are capable of driving the given load, and second, from this subset of feasible motors, choose the motor that satisfies best an additional criterion (price, weight, volume, etc.). This paper addresses, in general, the first question of feasibility. It develops a criterion as a practical tool to eliminate all motors that cannot drive the given load, whatever the transmission ratio of the reducer is. It is useless to consider such motors in a further economical study. Consequently, this criterion will speed up the whole drive selection process.

A new motor selection criterion is also needed because many existing criteria are only designed for classical dc servo motors. Synchronous servo motors, however, gradually supersede classical dc servo motors in high-performance drives [1]. Advanced control circuits and power electronics improve their quality and lower the price. The main advantages of ac-servo motors are their robustness due to the absence of brushes, their higher torque and speed bandwidths, and lower maintenance [2]. Furthermore, there are far less problems regarding electromagnetic interference (EMI). Another basic difference between a synchronous servo motor and a dc servo motor is the torque–speed characteristic. In a dc motor, the torque at high speed decreases because of the commutation limit (Fig. 1). The torque of a synchronous servo motor, however, is almost constant at all speed values, as flux weakening is almost impossible in permanent magnet ac machines when no special rotor designs are used [3]. The result is an almost rectangular T–ω curve. Two important properties of such a motor are the maximum torque $T_{\max,\text{motor}}$ and the maximum speed $\omega_{\max,\text{motor}}$, i.e., the coordinates of the upper right corner of the T–ω rectangle. A third important motor characteristic is its rotor inertia J_{motor}.

Section II sets the problem and indicates the shortcomings of existing selection methods. Section III derives the new selection criterion and its graphical representation. Section IV gives a short procedure for applying the method in practice. The use of the criterion and its advantage over conventional methods is illustrated in the industrial case study of Section V.

II. Classical Selection Criteria

One formulation of the problem is the following. *Determine which motors cannot perform a given dynamic task, whatever the transmission ratio of the reducer between the motor and the load and, for those that can, find the range of feasible transmission ratios.* This test considers only the dynamics of the problem; accuracy and other aspects need another test, not discussed in this paper. The dynamic task is a motion of a given load or mechanism and is specified by the speed $v(t)$, acceleration $a(t)$, and required force $F(t)$, for a linear motion or the angular velocity $\omega(t)$, angular acceleration $\alpha(t)$, and required torque $T(t)$ for a rotational motion. These functions of time are obtained by dynamic simulation of the system. There are no further conditions on these functions. The derivation of the criterion in Section III

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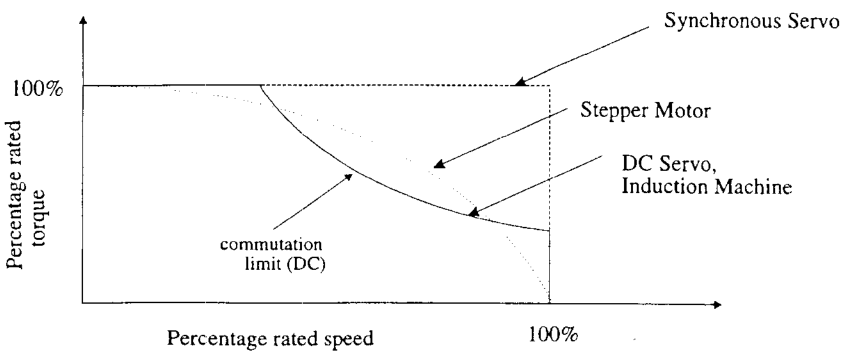

Fig. 1. T–ω curves for different motor types.

linear motion, as well. The maximum absolute values or peak values of these functions are indicated as follows: ω_{peak}, $T_{\text{peak}} \cdots$. The transmission ratio of the reducer is defined as $n = \omega_{\text{motor}}/\omega_{\text{load}}$ and is, in most cases, a reduction ($n \geq 1$).

Obviously, the motor's maximal power $P_{\max}$ should be larger than the peak power of the task, but this is not a sufficient requirement. Physical power always consists of two components, torque and speed, and both of them are limited in real actuators. Therefore, a dynamic feasibility test has at least two parts (and sometimes more) if several failure modes exist. The maximal motor speed causes an upper bound n_u on the feasible reduction of the reducer: $n_u = \omega_{\max,\text{motor}}/\omega_{\text{peak}}$. Similarly, the maximal motor torque causes a lower bound on the reduction: $n_l = T_{\text{peak}}/T_{\max,\text{motor}}$. If $n_l > n_u$, the motor is unsuitable. However, if $n_l < n_u$, extensive simulations for each motor are normally needed to decide whether there is really a range of feasible transmission ratios between these bounds.

Some studies suggested a drive selection criterion to avoid these simulations, but they are not applicable to general dynamic tasks and are often cumbersome. Most of them assume a purely inertial load [4]–[6]. For that special case, Pasch and Seering [4] derive the optimal transmission ratio, $n_{\text{opt}} = \omega_{\text{motor}}/\omega_{\text{load}} = \sqrt{J_{\text{load}}/J_{\text{motor}}}$, and also optimize the duration of the motion between two points for velocity-limited systems. The tradeoff between precision and dynamic load is demonstrated in [5]. Chen [6] and Vukobratović *et al.* [7] specifically investigate the selection of a dc motor. They show that peak power alone is not an adequate selection criterion. Their criterion combines the required physical power with the concept of acceleration power. The suggested calculation of the appropriate transmission ratio, however, is cumbersome. The recursive procedure for the choice of robot actuators in [8], on the other hand, is restricted to fixed transmission ratios (e.g., direct drive).

III. New Selection Criterion

The advantages of the criterion derived in this section are as follows: 1) its applicability to all kinds of loads; 2) the separation of load and motor characteristics; and 3) the combination of all motor and load characteristics into one graphical representation, which can be used for all motors. Therefore, extensive simulations are no longer needed to check whether a particular motor can drive the load.

A. Normalization of the Problem

Inserting a transmission between motor and load changes the dynamic task presented to the motor. A large transmission ratio, for example, reduces the influence of external torques on the motor behavior. At the same time, the motor has to rotate at a higher speed and generate higher accelerations for the same output motion. Hence, higher inertial torques are required. A good choice of transmission will balance both opposite effects.

In general, a motor–task combination is feasible if there exists a transmission ratio n such that the following apply:

1) the maximal speed required from the motor is smaller than $\omega_{\max,\text{motor}}$;
2) a certain norm $\|\cdot\|_p$ of the motor torque is smaller than a corresponding motor specific limit M_p;

or

$$\omega_{\text{peak}} n \leq \omega_{\max,\text{motor}}$$
$$\left\| \frac{T(t)}{n} + J_{\text{motor}} n \alpha(t) \right\|_p \leq M_p. \tag{1}$$

The subscripts "$_p$" indicate the norm used. The second requirement is motor dependent because of the rotor inertia J_{motor}. To normalize these inequalities, multiply the first one by $\sqrt{J_{\text{motor}}}$ and divide the second one by $\sqrt{J_{\text{motor}}}$:

$$\omega_{\text{peak}} \sqrt{J_{\text{motor}}} n \leq \omega_{\max,\text{motor}} \sqrt{J_{\text{motor}}},$$
$$\left\| \frac{T(t)}{\sqrt{J_{\text{motor}}} n} + \sqrt{J_{\text{motor}}} n \alpha(t) \right\|_p \leq \frac{M_p}{\sqrt{J_{\text{motor}}}}. \tag{2}$$

Then, define a normalized transmission ratio

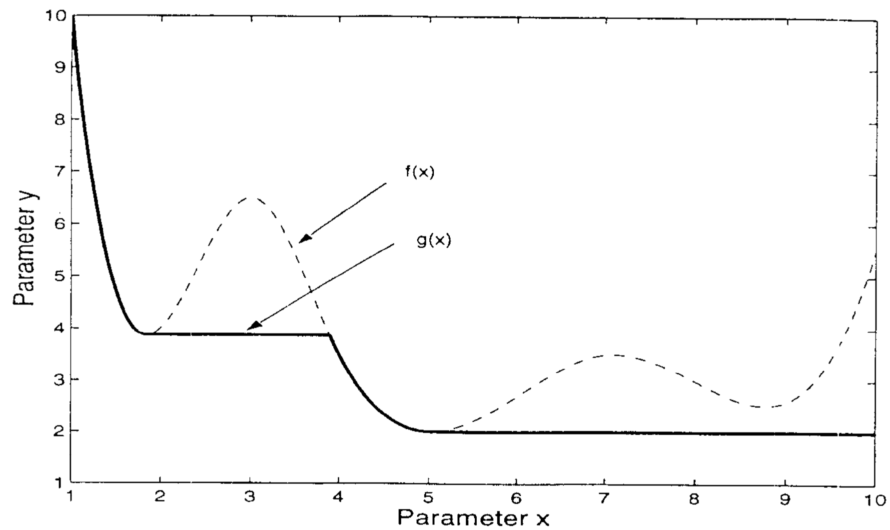

Fig. 2. Function $f(x)$ (dotted line) and its MLB $g(x)$ (solid line).

and substitute it into inequalities (2):

$$\begin{aligned} \omega_{\text{peak}} n^* &\leq \omega_{\text{max.motor}} \sqrt{J_{\text{motor}}} \\ \left\| \frac{T(t)}{n^*} + n^* \alpha(t) \right\|_p &\leq \frac{M_p}{\sqrt{J_{\text{motor}}}}. \end{aligned} \tag{4}$$

Because of the one-to-one relationship between n and n^*, the necessary condition for dynamic feasibility is equivalent to finding a normalized transmission n^* such that inequalities (4) hold. The normalized required speed and torque are defined as

$$\begin{aligned} \omega^*(n^*) &\equiv \omega_{\text{peak}} n^* \\ T_p^*(n^*) &\equiv \left\| \frac{T(t)}{n^*} + n^* \alpha(t) \right\|_p. \end{aligned} \tag{5}$$

Notice that these functions are independent from the motor parameters. So, this normalization separates the load from the motor quantities, and inequalities (4) become

$$\begin{aligned} \omega^*(n^*) &\leq \omega_{\text{max.motor}} \sqrt{J_{\text{motor}}} \\ T_p^*(n^*) &\leq \frac{M_p}{\sqrt{J_{\text{motor}}}}. \end{aligned} \tag{6}$$

The right-hand sides are defined as normalized motor limits

$$\begin{aligned} \omega^*_{\text{max.motor}} &\equiv \omega_{\text{max.motor}} \sqrt{J_{\text{motor}}} \\ M_p^* &\equiv \frac{M_p}{\sqrt{J_{\text{motor}}}}. \end{aligned} \tag{7}$$

In the $T^* - \omega^*$ plane, the combinations $(\omega^*(n^*), T_p^*(n^*))$ describe the so-called load curve, parameterized in n^*. The SI units of the normalized quantities are T_p^* in $\text{N}/\sqrt{\text{kg}}$; ω^* in $\text{rad} \cdot \text{m}\sqrt{\text{kg}}/\text{s}$ and n^* in $\text{m}\sqrt{\text{kg}}$ when the output motion is rotational or ω^* in $\text{m}\sqrt{\text{kg}}/\text{s}$ and n^* in $\sqrt{\text{kg}}$ when the output motion is translational.

B. The Graphical Representation

a function. For a function $f(x)$, the MLB $f_{\text{MLB}}(x)$ is a monotonically decreasing function, for all values x pointwise lower bound to $f(x)$, defined by

$$f_{\text{MLB}}(x) = \inf_{\xi \leq x} f(\xi). \tag{8}$$

Fig. 2 shows a function $f(x)$ and its MLB $g(x)$.

Theorem 1 For a function $f(x)$ with parameter description $(x_f(n), y_f(n))$:

$$\exists n: \begin{cases} x_f(n) \leq a \\ y_f(n) \leq b \end{cases} \Leftrightarrow \begin{array}{l} \text{point (a,b) lies above} \\ g(x) = f_{\text{MLB}}(x). \end{array}$$

Proof:

$(\Rightarrow)$ Suppose point (a,b) lies under $g(x)$

$\Rightarrow$ All points with $x_f < a$ have $y_f > b$

$$\Rightarrow \nexists n: \begin{cases} x_f(n) \leq a \\ y_f(n) \leq b \end{cases}$$

$\Rightarrow$ contradiction

$(\Leftarrow)$ point (a,b) lies above $g(x)$

$\Rightarrow f_{\text{MLB}}(a) = c < b$

$\Rightarrow \inf_{\xi \leq a}(f(\xi)) = c$

$$\Rightarrow \exists n: \begin{cases} x_f(n) \leq a \\ y_f(n) \leq c < b \end{cases} \square$$

The MLB concept gives a single line in the $\omega^* - T_p^*$ graph, such that the region below contains the representation of all motors that cannot drive the given application under any circumstances.

Based on this theorem, the requirements of inequalities (6) are reformulated as follows. *A motor with parameters $\omega^*_{\text{max,motor}}$ and M_p^* can drive the given load if, in the $\omega^* - T_p^*$ plane, the point with coordinates $(\omega^*_{\text{max.motor}}, M_p^*)$ lies above the MLB of the corresponding load curve.*

Consequently, the same figure can contain the coordinates of several motors, which simplifies the comparison between them. In practice, the load curve has a global minimum and so does its MLB. Clearly, no motor with M_p^* smaller than this minimum can drive the given load.

In conclusion, the graphical feasibility criterion can be stated as follows.

Criterion of Feasibility: A motor can perform a given dynamic task if its representing point in the $\omega^* - T_p^*$ graph is above the MLB of the load curve of the task.

C. Norms Used in Practice

In general, the p norm $\|\cdot\|_p$ of a periodic time signal $f(t)$ with period τ is defined as

$$\|f(t)\|_p = \left(\frac{1}{\tau}\int_0^\tau f^p(t)\,dt\right)^{1/p} \tag{9}$$

The norms that have most physical meaning for motor torques are

$$\|T(t)\|_2 = \sqrt{\frac{1}{\tau}\int_0^\tau T^2(t)\,dt} \tag{10}$$

$$\|T(t)\|_\infty = \max_{0\le t\le\tau} |T(t)|. \tag{11}$$

Since torque is proportional to the motor current, the norm $\|\cdot\|_2$ is a measure for the rms current in the windings, whereas the norm $\|\cdot\|_\infty$ is a measure for the peak current. Both should be limited to avoid thermal destruction of the windings and demagnetization of the permanent magnets, respectively. The limit of the $\|.\|_2$ norm is the rated motor torque $T_{\text{rated,motor}}$ and, for the $\|\cdot\|_\infty$ the maximal motor torque $T_{\text{max,motor}}$. Therefore,

$$M_2^* = \frac{T_{\text{rated,motor}}}{\sqrt{J_{\text{motor}}}} \tag{12}$$

$$M_\infty^* = \frac{T_{\text{max,motor}}}{\sqrt{J_{\text{motor}}}}. \tag{13}$$

D. Range of Feasible Transmission Ratios

Supposing that one motor (with rotor inertia J_{motor}) fulfills the graphical feasibility test, then what is the range of feasible transmission ratios? The first step toward the answer is to find the range of normalized transmission ratios $[n_l^*, n_u^*]$ for which inequalities (6) hold. The graphical representation is helpful to determine this range. To that end, indicate in the figure the parameter value n^* for different points of the MLB of the load curve. Inequalities (6) describe a rectangle in the figure having $(\omega^*_{\text{max,motor}}, M_p^*)$ as the upper right-hand corner. The range $[n_l^*, n_u^*]$ are all normalized transmission ratios with corresponding load curve points within this rectangle. Fig. 3 shows schematically a load curve and its MLB and the representation of one motor. The range of normalized transmission ratios for this motor is 3 m$\sqrt{\text{kg}}$ to 7 m$\sqrt{\text{kg}}$. The second step calculates the real transmission ratios from (3): 3 m$\sqrt{\text{kg}}/\sqrt{J_{\text{motor}}} \le n \le 7$ m$\sqrt{\text{kg}}/\sqrt{J_{\text{motor}}}$.

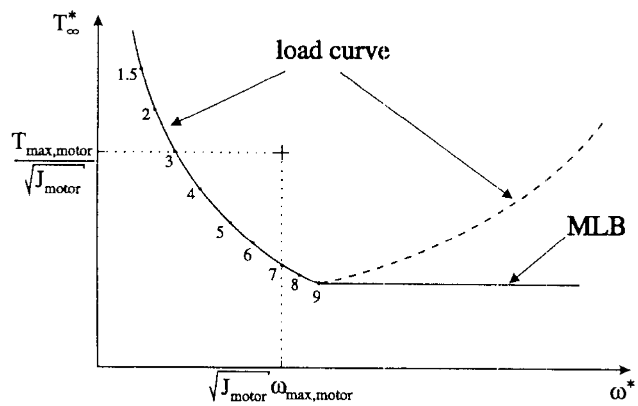

Fig. 3. Schematic $\omega^* - T^*_\infty$ graph.

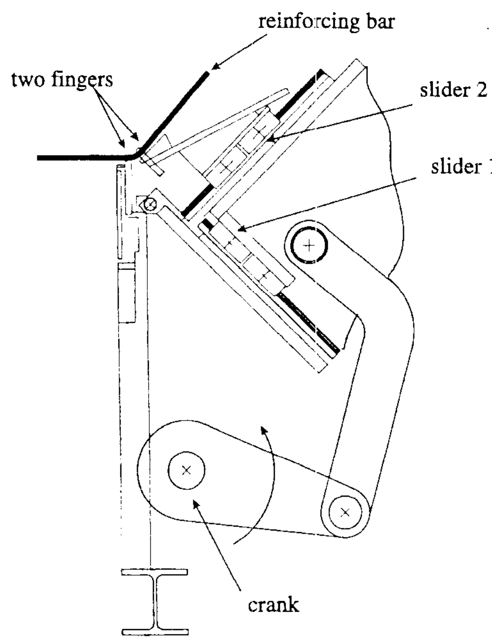

Fig. 4. Industrial bending machine.

load curve itself. In fact, for the other sections, there always exist smaller transmission ratios that require less torque and/or less speed from the motor.

IV. Practical Procedure

In practice, the feasibility criterion requires following steps.

1) Calculate or simulate the speed, acceleration, and torque to drive the load $\rightarrow \omega(t), \alpha(t), T(t)$.
2) Choose the p norms to evaluate the torque (often 2 norm or ∞ norm).
3) Use (5) to plot the load curve in the $\omega^* - T_p^*$ graph, i.e., simulate the motion for a motor with $J_{\text{motor}} = 1$ kg·m², and for different transmission ratios n, and plot each time the normalized required speed and torque.
4) Indicate all candidate motors on the graph.
5) Eliminate those motors, the representation of which is under the MLB of the load curve.
6) For the remaining motors, find the range of feasible

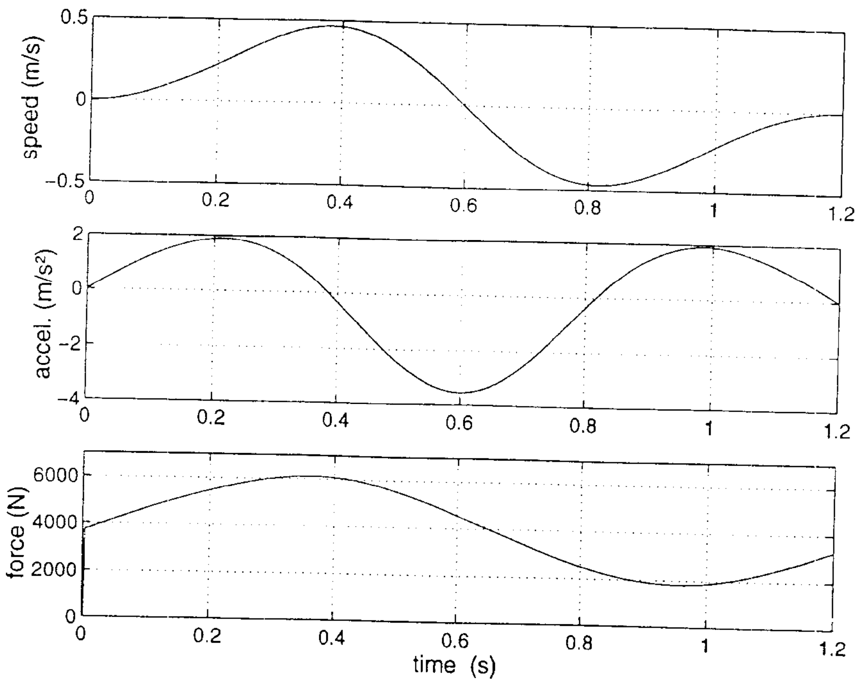

Fig. 5. Speed, acceleration, and required force of slider 1.

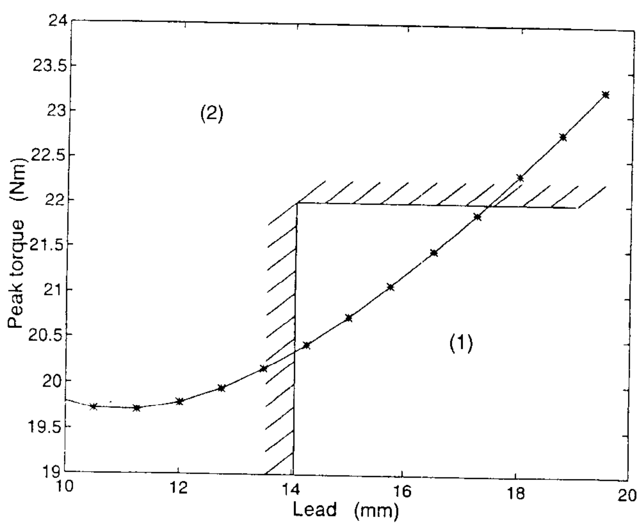

Fig. 6. Peak torque and limits of motor A: (1) feasible region (2) infeasible region.

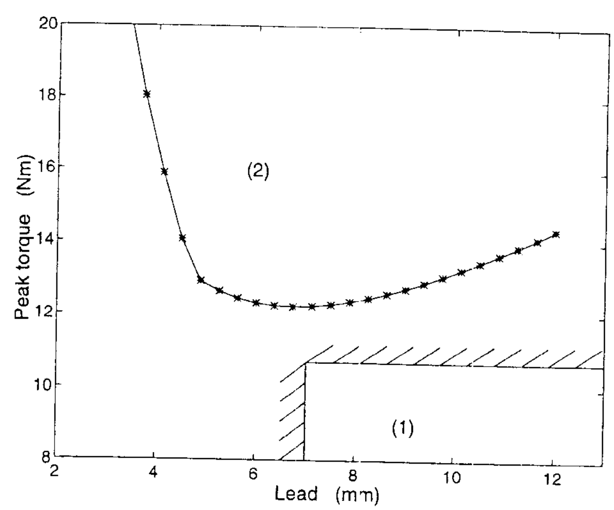

Fig. 7. Peak torque and limits of motor B: (1) feasible region (2) infeasible region.

V. Example

A. Problem Description

The presented criterion is used for the selection of feasible motors in an industrial machine which is used to bend reinforcing bars for the production of meshes. Fig. 4 shows the machine in the position with a 45° bending angle. The machine clamps each bar between two fingers. One finger is fixed, whereas the motion of the other plastically bends the section of the bar between the fingers to the desired angle. The motion of the moving finger is controlled by three servo which rotates the platform on which the finger is mounted. Two additional motors translate the finger to the right position through a ball screw transmission.

The total load of these motors contains the following components. First is the plastic moment T_{plast}, which is constant during the motion and only depends on the cross section of the bar and its material. Second is the torque $T_{\text{mech}}(t)$ required for the motion of the mechanism. It consists of a gravitational term, an inertial term, a coriolis term, and all kinds of friction. All these terms are variable with the configuration of the mechanism.

As an example, consider the selection of the motor driving

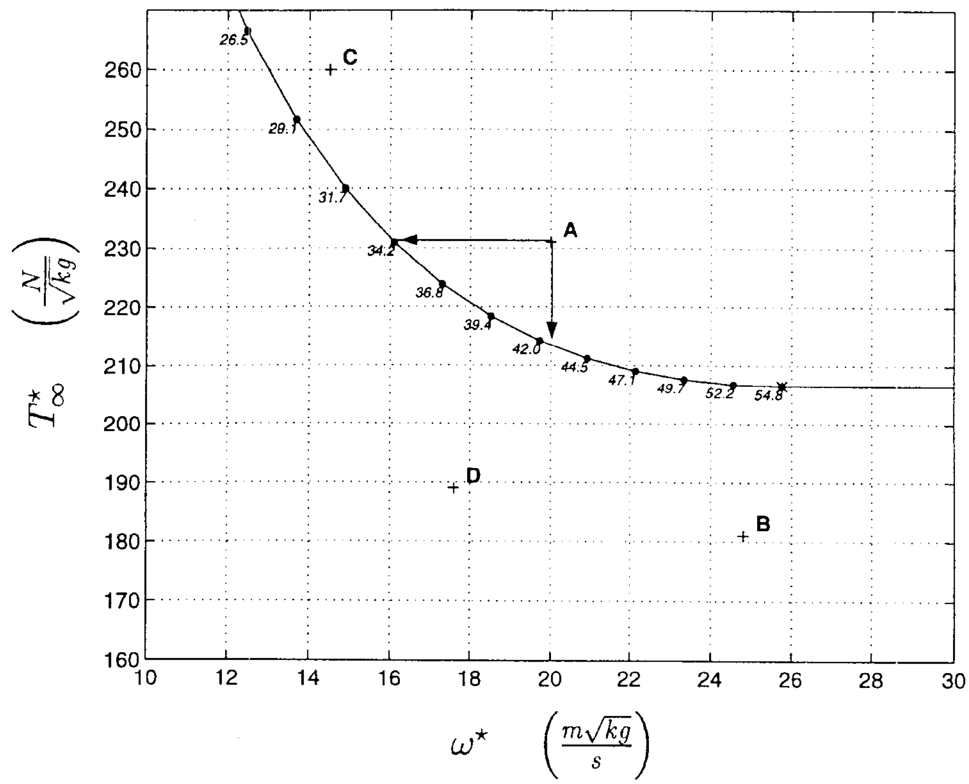

Fig. 8. $\omega^* - T^*_\infty$ graph representing the load and four motors A, B, C and D.

the bending task. Fig. 5 shows these quantities. This load is highly dynamic because of the peak accelerations (2 m/s^2 and -3.6 m/s^2) and force (6120 N).

The selection of an appropriate drive involves technical (power, geometrical size, weight, etc.), as well as economical (price, amplifier size, acquaintance with supplier, etc.) considerations. Prior to all these decisions, however, those motors that cannot dynamically cope with the load should be rejected, since they can never be eligible as a solution. The classical selection criteria are designed for constant speed motions or for purely inertial loads. For general motions, these criteria give merely a vague indication, but are inadequate to reject a motor. Only extensive simulations and iterations can give a decisive answer. The presented criterion, on the other hand, shows in one single plot which motors are unsuitable and the range of possible transmission ratios for the others.

Consider two candidate motors, A and B, with the dynamic characteristics listed in Table I.

B. Application of Classical Criteria

The following classical criteria are often used.

- The maximal power of the motor should be larger than the motion peak power. The required peak power from the dynamic analysis is 2860 W. Therefore, both motors have higher power and pass this test.
- The power criterion of Chen [6] requires motors with at least double peak power (or 5720 W) for dynamic motions of inertial loads. Motors A and B fail this alternative

TABLE I
DYNAMIC CHARACTERISTICS OF MOTORS A, B, C, AND D

motor	power P (kW)	max. speed $\omega_{\max}$ (r/min)	rotor inertia J_{rotor} (kg·cm^2)	max. torque $T_{\max}$ N·m
A	4.6	2000	91	22
B	5.4	4000	35	10.7
C	3.8	3000	43	12
D	3.4	4000	18	8

- One popular quantity to gauge the dynamic capability of a motor is its maximal no-load angular acceleration: $\alpha_{\max} = T_{\max}/J_{\text{rotor}}$. This acceleration is 2400 rad/s^2 for motor A and 3000 rad/s^2 for motor B. Therefore, from this criterion, motor B performs dynamically better than motor A.
- A first choice for the transmission ratio is the minimal reduction to drive the peak force: $n = F_{\text{peak}}/T_{\max}$. For motor A, $n = 6120\text{N}/22$ N·m or lead = 23 mm and, for motor B, lead = 11 mm. Simulations with these transmission ratios, however, show required peak torques 25.8 and 13.7 N·m, respectively, exceeding the limits of both motors.
- The optimal transmission ratio n_{opt} according to the inertia match principle is $n_{\text{opt}} = \sqrt{m_{\text{load}}/J_{\text{rot}}}$, where m_{load} is the mass of the driven load. In this case, $m_{\text{load}} =$ 500 kg. The corresponding lead of the optimal ball screw transmission is 26.8 mm for motor A and 16.6 mm for motor B. The required peak torque for these optimal transmission ratios are 28.7 and 17.8 N·m, respectively, and, therefore, these ratios are not implementable on the

The indications of these criteria are insufficient to make a clear decision. Therefore, for each motor, a series of simulations with different transmission ratios is needed to calculate the required peak torque T_{peak} for each combination. The limits of the motors are the maximal speed and maximal torque. These correspond to a minimal lead for the ball screw and a maximal peak torque of the motion. Figs. 6 and 7 show the peak torque simulation results for motors A and B and the motor limits. The feasible region, indicated by (1), is rectangular and in the lower right-hand corner of the plot. So, motor A can drive the load if ball screw lead is between 14–17.5 mm, whereas motor B cannot drive the load at all. These simulations are time consuming and unique for every motor.

C. Application of the New Criterion

The presented criterion unifies all these calculations in one plot, which has the following advantages.

1) The load curve clearly separates the motors which cannot dynamically drive the load.
2) The range of feasible transmission ratios can easily be derived from the numbers on the load curve.
3) The check for an additional motor merely requires the indication of its generalized coordinates on the plot and not a complete new set of simulations.

Fig. 8 shows this plot for the given selection problem. The load curve is a parametric curve in n^*. The figures in italics on the load curve represent the parameter value for that point. Motors A and B are represented by a little "+" mark with coordinates $(T_{\max,\text{motor}}/\sqrt{J_{\text{rotor}}}, \omega_{\max,\text{motor}}\sqrt{J_{\text{rotor}}})$ or $(20\ \text{m}\sqrt{\text{kg}}/\text{s}, 231\ \text{N}/\sqrt{\text{kg}})$, and $(24.8\ \text{m}\sqrt{\text{kg}}/\text{s}, 181\ \text{N}/\sqrt{\text{kg}})$ respectively. The conclusions are as follows. First, since the representation of motor B is under the load curve, it will fail dynamically to drive the load. Second, the feasible generalized transmission ratios for motor A lie on the load curve between the two arrows, i.e., $34.3\ \sqrt{\text{kg}} \leq n^* \leq 42.8\sqrt{\text{kg}}$. Translated to lead of the ball screw, these limits are $(\sqrt{0.0091\ \text{kg}\cdot\text{m}^2}/42.8\ \sqrt{\text{kg}})\ 2\pi \leq \text{lead} \leq (\sqrt{0.0091\ \text{kg}\cdot\text{m}^2}/34.3\ \sqrt{\text{kg}})\ 2\pi$, or 14 mm $\leq$ lead $\leq$ 17.5 mm, which is, of course, the same as the simulation result. Third, for evaluation of two additional motors, C and D, with characteristics as in Table I, simply indicating the motor coordinates is sufficient to determine that motor D cannot drive the load.

VI. Conclusion

The selection problem of a servo drive aimed at complex motions involves the characteristics of the load, transmission, and motor. Classical approaches considering, for example, the required peak power are shown to be inadequate. The use of normalized torques, velocities, and transmission ratios, however, yields a criterion that completely separates the load from the motor characteristics. By virtue of this normalization, the simulations for one standard motor ([illegible]

graph shows both the dynamic feasibility of all candidate motors for the given load and the range of possible transmission ratios. From this feasible subset of motors, the designer can select the most appropriate one according to another criterion.

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