DESIGN STUDIES ON A ROBOTIC DEVICE FOR ULTRASONIC INSPECTION*

by

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SUMMARY

Design studies in progress on a special purpose robot are described. The robot is to be used for ultrasonic inspection of extruded and forged parts immersed in water in a large tank. The studies focus on establishing the best tradeoff between light weight-with the associated fast movement time for large motions, - and rigidity - as characterized by high structural natural frequency. High natural frequency allows high servo bandwidth and consequently faster response to small disturbances and greater dynamic accuracy. [1,2]

INTRODUCTION

Ultrasonic inspection is frequently used to improve product reliability and to avoid scrapping raw material after expensive machining has uncovered flaws. Manual ultrasonic inspection is slow and labor intensive and thus expensive. When properly performed, however, manual inspection has the advantage that suspected flaws can be investigated more carefully as the specimen is being scanned. Most automated systems scan the specimen in a predetermined pattern without regard for the results of the ultrasonic test. The test results are then graphically displayed to an operator who may have to perform a manual inspection of questionable regions before passing or rejecting the specimen.

The systems under design will position an ultrasonic transducer in five degrees of freedom in a long ultrasonic immersion tank. Motion commands to the scanner are generated by the computer control system in response to a stored search pattern and the current ultrasonic test results. By using a combination of wide beam transducers (with attendant high false alarm rate) and narrow beam transducers (which take more passes to cover an area) an improved inspection strategy is sought. The optimum

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strategy is dependent on the distribution of ultrasonic reflectors (possible flaws) in the specimen.

The robotic inspection system is especially interesting as an example of robot design because of the interaction between sensors and motion control and because of the nondeterministic nature of the task. Of particular interest here is the tradeoff between a light manipulator structure capable of high accelerations and hence faster motions, and a more rigid manipulator structure with its increased accurracy, higher structural natural frequencies and higher bandwidth.

Optimization of the design cannot be performed on the basis of an analytical performance index. However, with the large number of design parameters brute force simulation of behavior is not practical. A two stage process is proposed whereby three specifications: clamped joint natural frequency, minimum movement time, and static stiffness; are used to constrain the choice of physical design parameters. Design parameters are considered "optimum" if they meet these constraints and minimize a static cost function. The values at which the constraints are set are determined by simulation. Thus for the purposes of simulaton it is assumed that all systems with the same constraint values would perform identically.

DESIGN PARAMETERS OF INTEREST

Design of a complex mechanism involves many choices, not all of which can be reduced to solving an equation. In this section we state the parameters to be determined by this study and other design decisions which are assumed given or are beyond the scope of this study.

The arrangement and type of axes assumed is shown in Figure 1. Three linear motions are of primary interest. Two additional motions, rotations at the distal end of the manipulator, have minimal effect on the decisions of interest and are ignored here. Vertical motion of link 1 is powered by motor 1. The size of the motor, the link cross section, and the speed reduction or effective drive radius are to be determined for each of the three motions. Motor 1 and link 1 ride on link 2. Two configurations are under consideration. Shown in Figure 1 is the configuration in which motor 2 rides on link 2. Also being considered is motor 2 mounted on link 1. To be determined are the size of the motor, the link cross section, the speed reduction, and for the second configuration, the cross section of the drive belt. Motor 3 rides on link 3 and both are to be sized together with the speed reduction in the drive. The linear motions lend themselves to operation in the long ultrasonic immersion tank. The tank length (30.5 m) dicates that motor 3 ride with link 3 rather than be mounted in a stationary arrangement.

Selection of a motor specifies motor weight, rotor inertia, and its acceleration and speed characteristics. Within a family of motors only one design decision specifies all three parameters. For the design study permanent magnet direct current motors are considered. The relationship between motor parameters has been determined empirically from manufacturer's data as shown in Figures 2 and 3. Knowing this approximate relationship we specify a motor in terms of its mass, although any of several other parameters could be used. Motor inertia follows from this specification.

Time optimal control of permanent magnet d.c. motors results in movement times dependent on the effective load inertia, the distance traveled and the motor parameters. Time optimal control with voltage limits as solved by Szabados [3,4] was applied to the same motors used in

Figures 2 and 3. An example of the movement times for these motors for one radian of travel and for various load inertias is shown in Figure 4. Possible movement times are bounded by the dashed line in that figure. The best motor to use is not necessarily the motor which minimizes movement time, however. It should also be pointed out that the speed reduction used varies the effective load inertia as it appears in Figure 4.

The link cross sections determine both the rigidity and mass of the link. The links one and three are represented by tubular members with wall thickness a constant fraction of outer radius. For a given material the outer radius specifies the variable link parameters. Line 2 is respresented as a short, stubby beam much the same as for links one and three. This description is not so accurate but suffices for this general design study which seeks to avoid the detail design.

FEASIBLE CONSTRAINTS

The designer's delima is that by increasing rigidity he increases mass and movement time. Rigidity is characterized here by lowest natural frequency ω_c and end point stiffness k_e. End point stiffness constriants are directly related to accuracy of end point location with constant gravitational and/or drag loads. Natural frequency is related in a more complex manner to the dynamic performance. Movement time or its inverse Ω (which has the same units as ω_c) is related to performance in a complex dynamic fashion also. Simulations are under way in which the dynamic performance is studied for different relative values of movement time and natural frequency. The search strategy used and the distribution of flaws are important variables in the outcome of these studies which are incomplete at this time. We discuss here the determination of feasible values for ω_c and Ω, and the least costly way to provide a feasible value.

To determine the limits of feasibility we propose the following procedure. Maximize the performance index L

$$L = \omega_c$$

subject to the equality constraints on $\Omega = 1/(\text{movement time})$ which is required to be the same for each of the three axes. Ω_i for axis i depends on its total effective load inertia, J_{Ti}, which in turn depends on the link masses $m_{\ell i}$ and motor masses m_{mi}, the rotor inertias J_{mi} and the effective drive radii r_i. See Figure 1 for a complete description of the terms in each J_{Ti}. Having empirically related motor mass and inertia we can eliminate J_{mi} and write

$$\Omega_1(r_1, m_{\ell 1}, m_{m1}) - \Omega = f_1(r_1, m_{\ell 1}, m_{m1}) = 0$$

where $m_{\ell i}$ = the mass of link i

m_{mi} = the mass of motor i

r_i = the effective drive radius of axis i.

Similarly constraints on Ω_2 and Ω_3 can be formally written as

$$f_2(r_2, m_{\ell 1}, m_{\ell 2}, m_{m1}, m_{m2}) = 0$$

and

$$f_3(r_3, m_{\ell 1}, m_{\ell 2}, m_{\ell 3}, m_{m1}, m_{m2}, m_{m3}) = 0$$

We adjoin the constraint equations to the performance index by way of Lagrange multipliers λ_i to define

$$H = \omega_c + \lambda_1 f_1 + \lambda_2 f_2 + \lambda_3 f_3 = L(x,u) + \lambda^T f)x,u)$$

We arbitrarily designate six variables of the problem as decision variables and form the decision vector u

$$u = [m_{\ell 1}, m_{\ell 2}, m_{\ell 3}, r_1, r_2, r_3]^T \quad .$$

The three remaining variables are designated state variables and form the state vector x

$$x = [m_{m1}, m_{m2}, m_{m3}] \quad .$$

To solve this extremization problem we propose the first order gradient technique adapted from Bryson and Ho [5] with steps as follows:

(a) Select initial values for u

(b) Determine x from $f(x,y) = 0$

(c) Approximate $\frac{\partial L}{\partial x} \simeq \frac{\Delta L}{\Delta x}$ and $\frac{\partial L}{\partial u} \simeq \frac{\Delta L}{\Delta u}$

(d) Calculate $\frac{\partial f}{\partial u}$ and $\left(\frac{\partial f}{\partial x} \right)^{-1}$ wither by approximations with $\frac{\Delta f}{\Delta u}$ and $\left(\frac{\Delta f}{\Delta x}\right)^{-1}$ or with analytical differentiation of empirical fits to the constraint equations

(e) Compute $\lambda^T = - \left(\frac{\partial L}{\partial x} \right) \left(\frac{\partial f}{\partial x} \right)^{-1}$

(f) Compute $\frac{\partial H}{\partial u} = \frac{\partial L}{\partial u} + \lambda^T \left(\frac{\partial f}{\partial u} \right)$

(g) Vary u by $\Delta u = K\left(\frac{\partial H}{\partial u} \right)^T$

(h) Repeat (b) through (h) until ΔL is very small.

THE COST OF ACHIEVING ω_c and Ω

Given that values of ω_c and Ω are feasible, what is the best way to acieve them? The obvious answer is: "in the way that minimizes cost". Less obvious is the way in which to compute cost. Empirical relationships for cost are not known except for the motors and other relatively minor components. The proposed method of ascribing cost is by mass of the components.

A case of practical interest is when the values of ω_c and Ω on the feasibility boundary are used. This assumes no cost differential associated with achieving the highest performance. The question then remaining is which pair of ω_c, Ω values gives the best performance. This the authors hope to answer this question with digital computer simulations.

CONCLUSIONS

The movement time vs. rigidity trade-off is one faced in many arm designs. The work described here continues and the results should shed light on designs for other applications as well.

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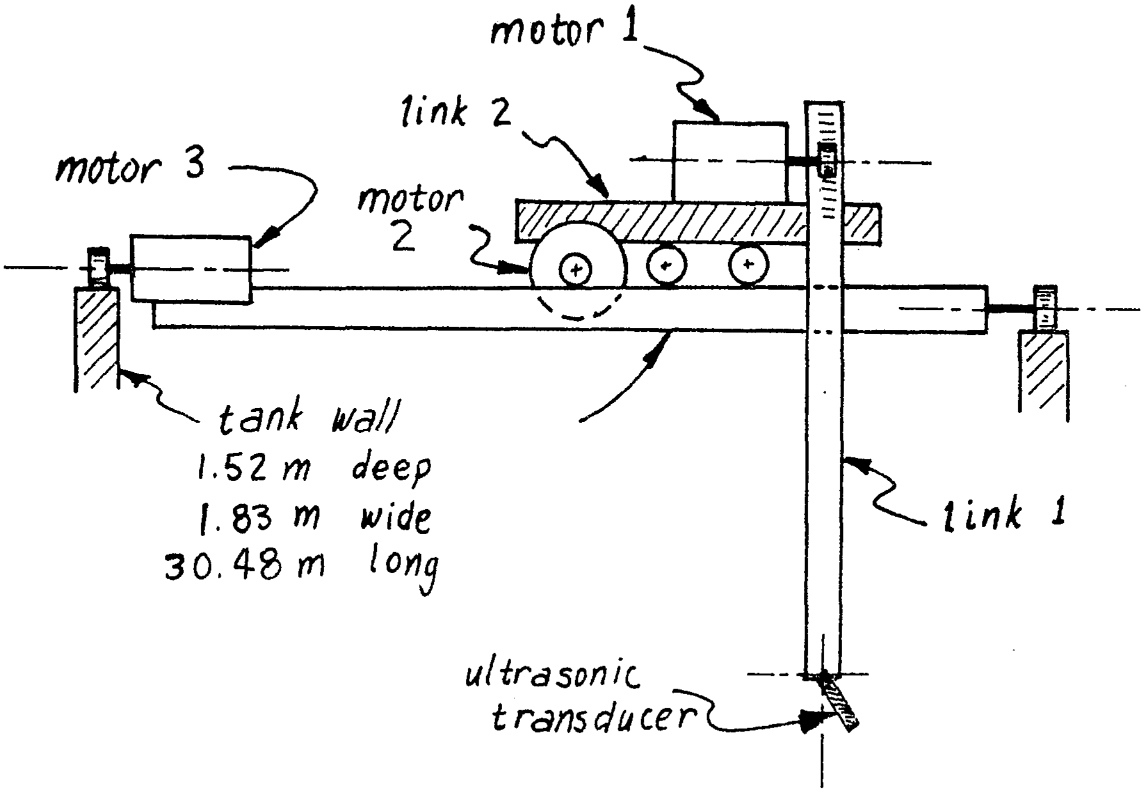

$m_{\ell i}$ = mass of link i

m_{mi} = mass of motor i

J_{mi} = rotor inertia of motor i

r_i = effective radius of the drive of motor i

Design parameters:

$m_{\ell i}$ = specifies stiffness of link with assumed geometry

m_{mi} = specifies motor Torque and J_{mi}

r_i = selected to optimize movement time

Inertia driven by motor 1:

$$J_{T1} = J_{m1} + m_{\ell 1} r_1^2$$

Motor 2:

$$J_{T2} = (m_{\ell 1} + m_{\ell 2} + m_{m1} + m_{m2})r_2^2 + J_{m2}$$

Motor 3:

$$J_{T3} = (m_{\ell 1} + m_{\ell 2} + m_{\ell 3} + m_{m1} + m_{m2} + m_{m3})r_3^2 + J_{m3}$$

Figure 1. Schematic of the robot for ultrasonic testing.

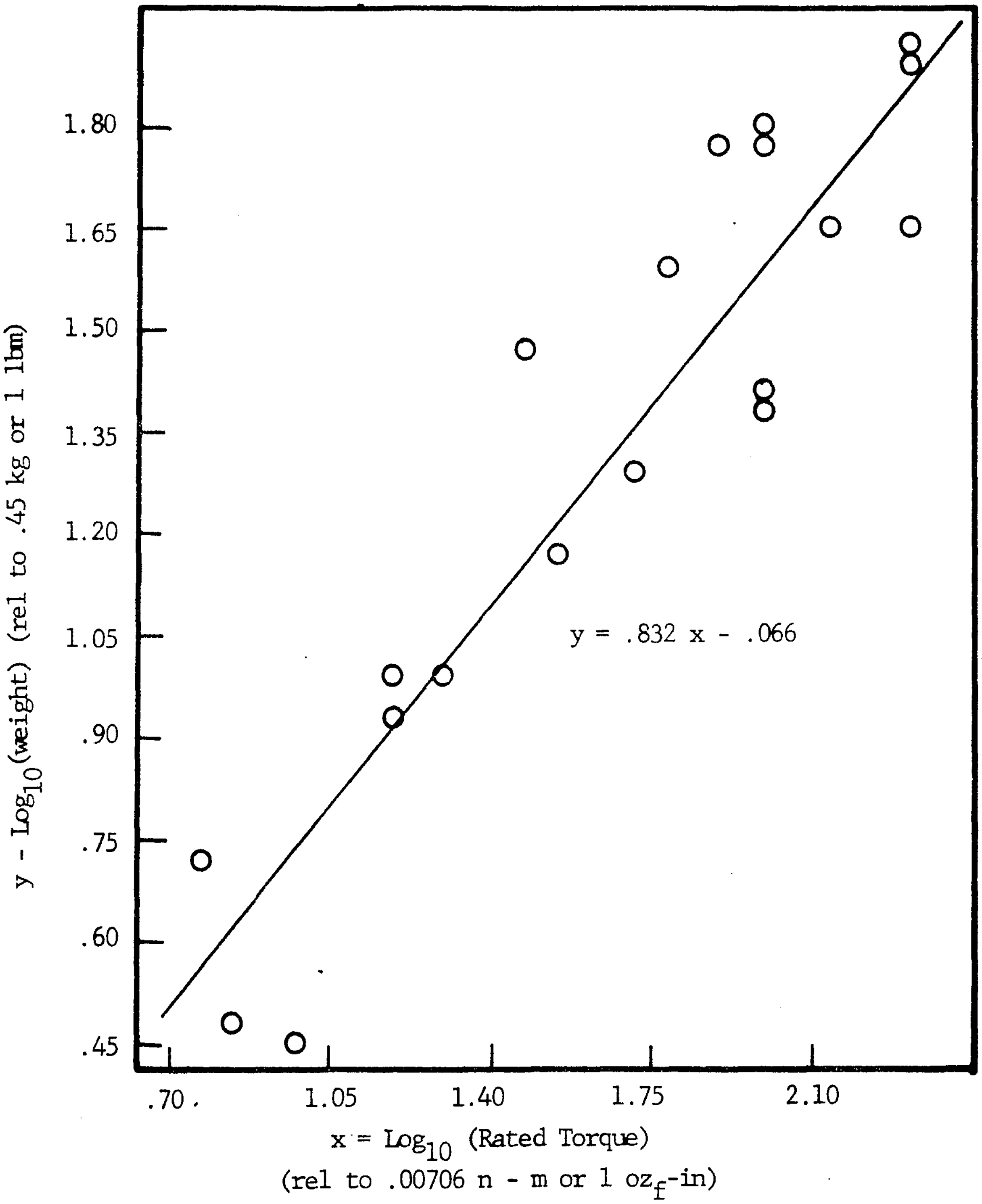

Figure 2. Empirical fit to manufacturer's data.
R = .840 correlation

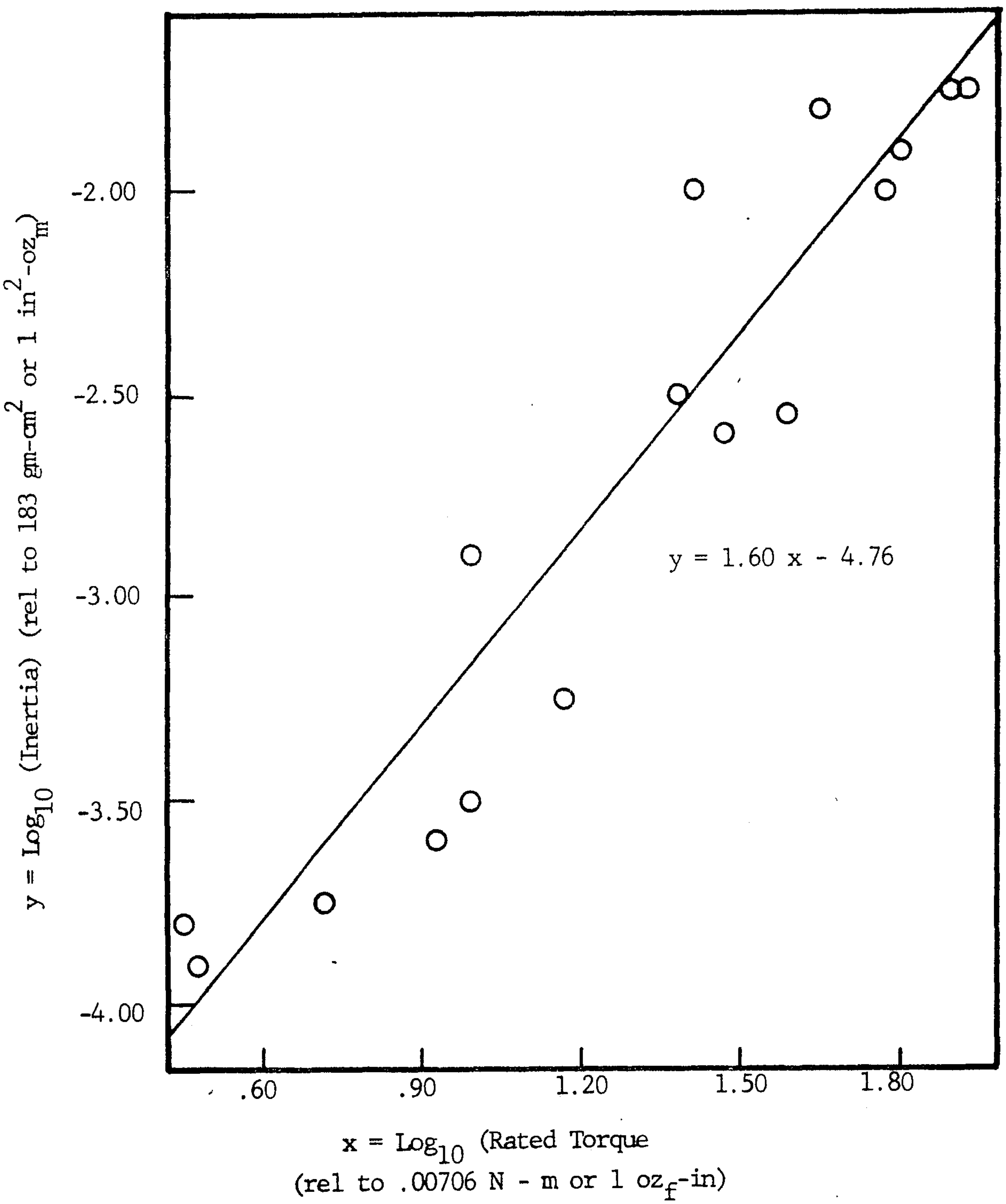

Figure 3. Empirical fit to manufacturers data. R = .900.

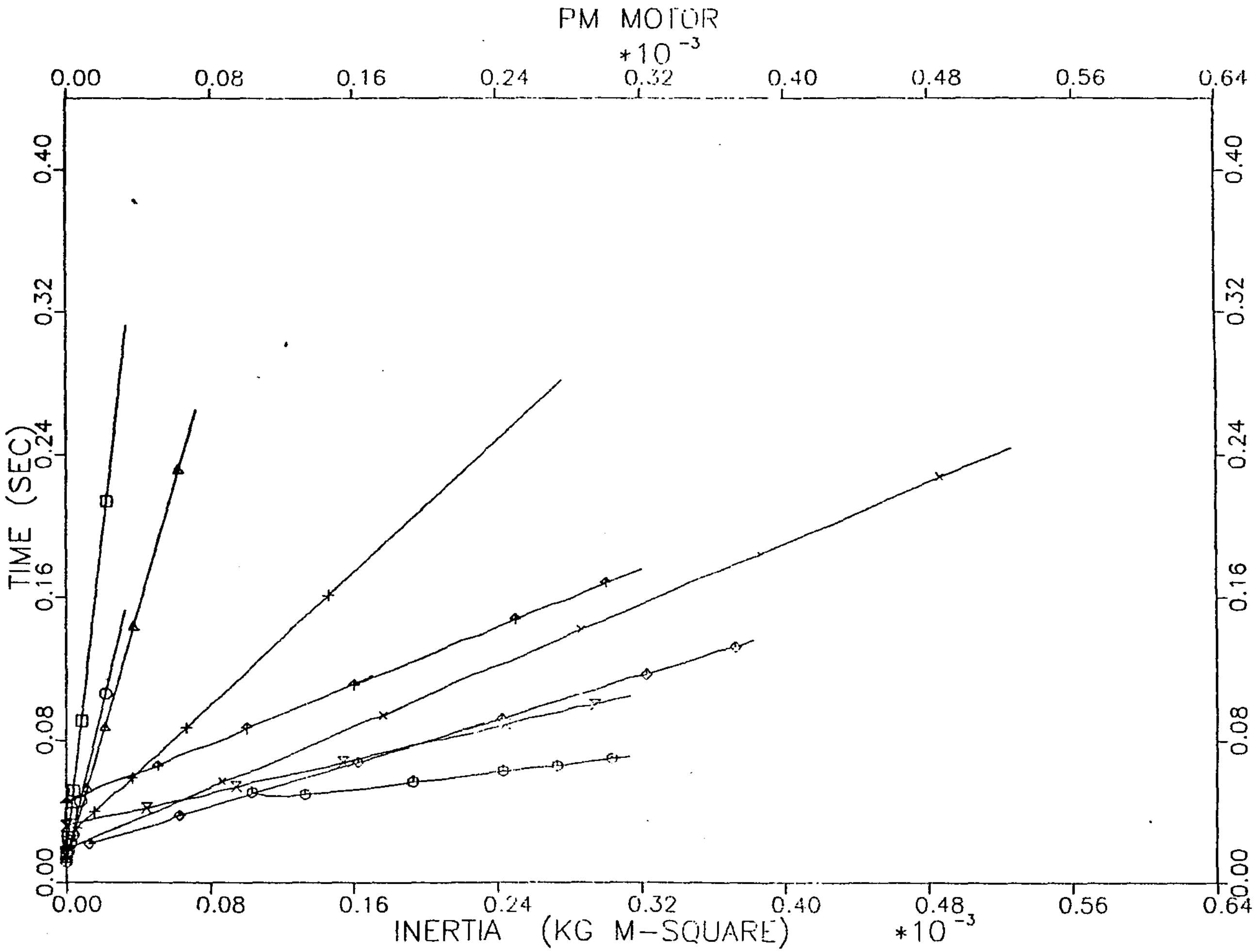

Figure 4. Minimum movement time vs. Load Inertia for a family of d.c. Permanent magenet. Rotation is 1 rad.