DESIGN AND CONTROL CONSIDERATIONS FOR INDUSTRIAL AND SPACE MANIPULATORS

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ABSTRACT

This paper is a progress report summarizing theoretical and practical results concerning integration of design and control aspects of manipulator arms for industrial or space applications. The relationships between task specifications, gross motions, fine motions, actuator type and location, size and strength of structural members, control servos and strategies, and overall design evaluation are briefly discussed, with some technical examples.

MAJOR DESIGN - CONTROL ISSUES

Three questions dominate the design of a mechanical arm:

What is it going to do?
How shall it be built?
How shall it be controlled?

The first concerns task specifications like reach, speed, and payload. The second concerns structure and actuators. The third involves both simple stabilization but also vibration supression, use of feedback sensors from the hand and joints, and general strategy of operation for high efficiency and accuracy with low overshoot and power consumption. For space manipulators, overall weight is a crucial consideration.

A comparison of current industrial robots and the people they augment or replace yields some insights. A typical step in the manual assembly of a washing machine gearcase reads "Obtain pinon and assemble to gearcase". That is, fetch some object and do some thing with it. More concretely, a gross motion (much larger than the pinon itself) followed by some fine motions (usually much smaller than the pinon or whatever). Most industrial robots are incapable of fine motions because they were designed for gross motions and because fine motions require sensory feedback from the task of a kind which no current industrial robots have access to. Work currently under way at MIT and C.S. Draper Labs [1] is addressing, among other things, the issue of giving robot arms fine motion capability.

An important measure for both human and robot arms is the ratio of gross motion time to fine motion time. A high ratio may indicate wasted time in mere parts feeding activities which crowds the time needed for the careful work of assembly. But, for people, the gross motion time is fairly consistently lower-bounded for a given task. Overall task time is usually shortened by strategies which group many gross motions, such as carrying several little parts simultaneously, and take advantage of the human hand's dexterity. One can hope to build a robot arm strong enough to exceed a human's gross motion speed. Some of the problems of doing so are discussed below.

Exceeding a human's fine motion speed, which includes measurement and strategy - invocation time along with mere speed of motion, is much more difficult. The human equipment

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Work supported by NASA contract NAS8-28055 and DOD-ARPA Contract DAH C15 73 CO 278

actually consists of two devices, an arm of 5 degrees of freedom which positions the hand and wrist, plus the hand, itself a fine motion device with several dozen degrees of freedom and many sensors. One can gain some design freedom in a robot fine motion device by separating it from the gross motion device but this still leaves the robot at a disadvantage. Current technology and understanding of the problem indicate that

a) robot gross motion must be very fast to gain time for fine motion to occur, or else strategies like multipart handling must be adopted

b) robot fine motion must be specialized and carefully designed with limited degrees of freedom and other simplifications

c) contradictions could arise in attempting to build an arm which simultaneously is intended to perform both gross and fine motions economically, especially if the arm is physically large

For manipulators aboard spacecraft, the severe weight limitation replaces fine motion control as the major design-control problem. If the arm is to be large, like the Space Shuttle cargo boom, this means that large flexures of the arm structure can be expected, which in turn means oscillations of arm and payload at very low frequencies (perhaps one cycle in two minutes). If structural damping cannot quickly eliminate these oscillations, then the control system must be designed to do so or else long mission times will result. Alternatively, the control strategy can be designed to keep such oscillations small to begin with. This relation between structure and servo recurs in design of industrial arms where unwanted interactions between servo and structural natural frequencies could occur or an attempt to avoid these interactions could result in a structurally overdesigned arm. Some examples below discuss these points.

<u>Questions Related to Gross Motion Patterns</u>

a) how large is the arm to be and what kinematic articulations should it have

b) how fast should it be able to make a gross motion of some meaningful size

c) what range of inertial and gravitational loads must it be able to carry at the above speeds.

Remarks: It is generally true that the more specialized or dedicated an arm is, the fewer degrees of freedom it needs, the minimum being one. Programmable robots presumably must be capable of a variety of tasks, especially if an economical number of them is to be manufactured. Beyond this, choice of task for design purposes is difficult. The major variables seem to be payload weight and distance to be moved. For assembly of a small gasoline engine the distances are all small but a factor of 100 or more separates the weight of the lightest and heaviest parts. For an auto instrument panel distances are large but all the parts are in a small size and weight range. Large and heavy parts pose the biggest challenge, especially because conventional fixed automation machinery cannot handle them.

For tasks of a given class, a kinematic analysis can be performed to minimize the size of a given linkage that will reach a specified set of endpoints. It is necessary that the arm not be fully outstretched at such points so that fine motions can occur.

The time quoted in item b) must include time for the arm to settle down on the target point. Settling oscillations and overshoot can result from the structure or the servo. Settling time can be approximately related to servo bandwidth (for a rigid arm). For example, if stop to stop motion time is to be one second and settling time is budgeted 10% of that time, then for an effective damping ratio of 0.5, the servo with arm and load must be flat to about 13 hz, a very difficult goal to meet with a large arm.

Items a), b), and c) determine to a large degree the actuator torque requirements and the servo bandwidth needed to throw around the arm and load. The capabilities of the actuators need to be balanced (see technical discussion below) so that no joint is over or under designed with respect to the others. A number of industrial robots seem to have weak wrists in relation to their elbows and shoulders.

<u>Actuator Type and Location</u>

d) for the torque requirements from above, what type of actuator should be used

e) what sort of transmission should couple the actuators to the arm

f) how much accuracy should the arm have

g) how much resolution should it have

Remarks: Families of actuators can empirically be described rather accurately, relating their peak torque or rotor inertia to their total weight. For a family of DC torque motors all operating at the same supply voltage, relation is

$$\text{mass in kg} = 2.1 \times (\text{torque in nt-m})^{0.875}$$

while for a family of hydraulic rotary vane actuators, all operating at the same supply pressure, the relation is

$$\text{kg} = 0.235 \times (\text{torque})^{0.55}$$

Comparison of these relations indicates that for these torque motors to compete on a torque to mass basis with these vane actuators, gears of ratio at least 10 or 15 to one will be necessary. Even with vane actuators, the weight of a hydraulically driven arm is mostly actuator weight. One can locate the actuators in the arm's base and transmit power through shafts, cables, tapes or chains, which will save weight but introduce compliance. Gears contribute both compliance and backlash, which decreases accuracy, resolution and servo stability. Hydraulic actuators directly coupled to the joints develop high torque but compliance appears in the fluid, an effect which can be reduced by careful design of the control system. Large hydraulically driven arms with fast gross motion requirements will need large servo valves which in turn have low enough bandwidth to affect settling time and the speed of fine motions.

Thus the issue of actuators, their type and location on the arm is a complex one affecting all aspects of design and control. It is not clear whether there is one clear cut solution suitable for all situations.

Fine Motion Patterns

h) how small must the fine motions be

i) how rapidly must they be performed

j) what and how many arm degrees of freedom must be involved.

Remarks: Resolution of the joint sensors, size of the arm, backlash in gears and friction in the actuators or joints all can limit the fineness. If a rotary actuator far from the hand must contribute to the fine motion, then the radius from the joint to the hand times the joint sensor resolution indicates but does not absolutely limit the fineness. (some types of actuators can be jogged open loop with predictable results).

The rapidity of fine motions is an issue for industrial arms equipped with touch or force feedback. References (1) and (2) describe a force vector measuring system, located in the wrist, capable of resolving three components of force and three of torque about a chosen point. Such a system can be used to assemble objects in much the same way people do, by making some small deliberate collisions occur and judging from the direction of the resulting contact force how to move next. To avoid large contact forces, the appropriate change in the arms's trajectory must be made quickly. A way of accomplishing this is to interpret the force vector as a servo command. However, contact forces build to large values quickly if arm inertia is large and the objects and their supports, including the arm itself, are stiff.

Any type of low pass cutoff will make rapid fine motions difficult. For hydraulics the crucial items are the servo valve and the compliance represented by the fluid within the actuator. Sizing the arm and valves for rapid gross motions and heavy loads will yield large slow valves and large fluid compliances, inconsistent with rapid fine motions. Computation time lags and filtering time associated some types of high accuracy joint sensors also add to this problem.

Structural Members

k) for the given kinematic configuration, how strong or thick should the structural members be

l) should the members be sized for static stiffness (an issue related to accuracy in a gravity environment) or dynamic stiffness in conjunction with the arm's masses (related to structural vibration and its interaction with the servos).

Remarks: The links must not only support their own weight and that of the actuators and payload, but should not create, in concert with these masses, structural natural frequencies close to those of the servo because this will make gross and fine motions difficult to accomplish quickly and could prevent using the servo to damp out structural vibrations. These issues are discussed in some detail below.

Design Evaluation

Some competing criteria are:

m) how closely does the arm meet the speed, reach, strength and accuracy requirements originally posed.

n) how efficiently, in terms of arm weight and power consumption, are these requirements met

Remarks: For a space manipulator, low weight is a severe requirement which must be included directly in item m). For industrial arms, the idea of load factor efficiency criterion for item n), makes sense where load factor means the ratio of dynamic payload, (usually less than mere lifting capacity since an economic time to move the payload is usually enforced) to the weight of the moveable parts of the arm itself. Experience indicates that a load factor of 5% to 10% may be typical and that 20% would be quite an improvement. Substitution of control techniques for structural weight as a vibration supression method could allow increases in load factor.

An allied efficiency criterion is energy consumption. Typical large industrial manipulators use 10 to 30 horsepower. It seems reasonable to compare this to a "payload power" such as (payload) x (reach) / (slew time).

TECHNICAL EXAMPLES OF GROSS MOTION DESIGN CONSIDERATIONS

One of the criteria by which mechanical arm performance can be judged is its payload-speed relation, that is, the tradeoff between payload mass and the speed with which that mass can be moved. If a mechanical arm is limited in the torque it can produce at its actuators, then large payload masses obviously cannot be moved as fast as small payload masses. This is because of the larger inertial forces as well as the larger gravitational forces which must be overcome.

The limiting behavior of the arm as the payload is increased occurs when the arm is able to hold only itself and the payload against gravity with no spare torque left over for acceleration. Hence, the arm can move its maximum payload mass only very slowly. The limiting behavior for zero payload mass occurs at the speed at which the inertia and weight of the arm itself limit the speed of motion. Thus, for a given arm design, if we were to define a particular arm motion and plot maximum possible payload against motion completion time, our curve might look like Figure 1.

In Figure 1 are indicated the absolute payload limit, determined by the arm configuration and gravity, and the minimum task completion time, determined by the arm configuration and inertia. Clearly, the shape of this curve depends on many factors, including the arm size, mass distribution, nature of the task motion, and type of actuators. The same arm would have a different curve for a different task motion. Hence, there is a certain arbitriness about the choice of a standard task for determination of arm performance. For design purposes, a task should be chosen which is representative and which taxes the arm's capabilities.

The idea of a payload-speed curve is best illustrated by example. Therefore, let us consider a one degree of freedom arm and actuator shown in Figure 2, a desired arm motion shown in Figure 3a, and a desired velocity profile shown in Figure 3b. Notice that the trajectory is a 90° down sweep from straight out horizontal to straight down. The triangular velocity profile produces a stepwise constant torque requirement while geometry and gravity superimpose a cosine torque requirement. The result appears as Figure 4. For long task times, acceleration torque is small, so the peak torque occurs at $t=0$. However, for short task times, the acceleration torque is large, so the peak torque occurs at the halfway point.

Now that the peak torque for a given task time can be found, we can obtain a performance curve by assuming a maximum torque capability for the actuator and solving for payload as a function of the total task time. We assume an idealized actuator which can deliver its maximum torque at the maximim required speed. Task time t_f does not include any servo effects like overshoot or settling time.

One Degree of Freedom Numerical Example

For a numerical example of the preceding analysis, we assume the following parameter values, which crudely reperesent a three degree of freedom arm to be discussed below.

Max actuator torque	691 nt-m
Inertia of arm at shoulder (no payload)	13.4 kg-m^2
Radius of arm's c.g.	0.798 m
Length of arm to payload	1.1 m
Mass of arm (no payload)	24.87 kg

After the analysis we obtain the following results:

Gravitational payload limit (arm horizontal)	45.95 kg
Gravitational payload limit (arm at 45°)	72.46 kg
Minimum motion time at zero payload	0.39 sec

Figure 5 is the performance curve for this example, showing payload and load factor versus lower bound task time. Lower bound implies that in a practical system the servo effects would tend to lengthen the task time for a given payload.

Performance of Multiple Degree of Freedom Arms

When a manipulator has more than one degree of freedom, the payload-speed curve becomes more complicated. Consider a planar three degree of freedom arm with actuators located at the shoulder, elbow and wrist. The performance of the actuators must be balanced to their respective inertial and gravitational loads encountered during a typical task. Note that each actuator must be able to accelerate and support all the actuators, structure and load outboard of itself. To investigate this, we assign realistic values to actuator and structural weight and apply the same task as in the above single degree of freedom example. The rotary vane actuators discussed earlier are used and the characteristics of the arm and the task are intentionally similar to the one degree of freedom example. See Figure 6.

Each actuator has its own performance curve corresponding to the maximum payload it can handle at any given task time. The "interior" of the performance curves, represented by the hatching in the figure, is the net performance curve of the arm for this particular task. The actuators represented in this plot are fairly well balanced compared to each other; that is, no one actuator is excessively under - or overdesigned. The large load factors indicated in the figure may not be possible in practice due to the dead weight of such items as pipes, valves, fittings and so on, all difficult to estimate in advance. Thus the curves serve to indicate a performance goal against which a real arm may be judged.

The previous analyses assumed that the required torque profiles were produced open loop. A closed loop torque generating method is preferable for trajectory and endpoint control. one way to produce these torques is to generate a commanded position and velocity profile for the arm and request torques in proportion to position and velocity errors. The proportionality constants (servo position and tach gains) can be chosen by diagonalizing linearized equations of motion for the arm and assigning undamped natural frequency and damping ratio to each joint of the arm.[1] The gains for the above three joint arm model were chosen to yield the same undamped natural frequency ω_s at each joint with unit damping ratio. Simulations show that the settling time is approximately inversely proportional to ω_s. Note that actuator limitations and structural frequencies upper bound the choice of ω_s. Structural aspects are discussed in the next section.

FLEXIBLE DYNAMICS AND FINE MOTION CONTROL

The separation of structural design from control system design can result in incomplete understanding of their interaction and in decreased performance. Problems have been traditionally avoided by requiring the lowest structural frequencies to be much greater than the servo bandwidth. This results in additional inertia requiring higher torques and lower ratios of payload weight to arm weight. In an attempt to better understand this interaction and to allow for its consideration in design, computer programs have been developed which allow one to model an arm as a collection of distributed beams, lumped masses, and joints with control about a given equilibrium position described by a transfer function. These programs allow one to obtain closed loop eigenvalues, frequency responses, and impulse time responses.[3] Special attention has been given to flexural vibration in a plane.

While detailed arm models should be analyzed for a particular arm design, the simplified model of Figure 7 indicates several important points for arm design in general. This example consists of two identical beams joined by a rotary joint. One end of the arm is free and the other end is clamped to ground. The joint is regulated about a straight equilibrium position by feeding back the angular position and velocity of the joint. When the beam segments are essentially rigid, the arm is a simple second order system with undamped natural servo frequency ω_s. Adding negative velocity feedback moves the complex conjugate eigenvalues from a value of $0 \pm j\,\omega_s$ on the imaginary axis of the complex plane in a circular arc to the real axis. (Here $j = \sqrt{-1}$.) Figure 8 displays the eigenvalues in nondimensional form as ω_s and ζ (ζ is the damping ratio for the rigid analysis) are varied. As ω_s approaches the lowest structural frequency ω_c (the cantilevered natural frequency with actuator locked) the second order behavior disappears, and the structure can no longer be called "essentially rigid." When $\omega_s / \omega_c \geqslant 0.38$ the lowest eigenvalues can no longer be brought to the real axis. At $\omega_s / \omega_c \approx 0.5$ an effective damping ratio for the arm structure and servo of 0.7 is the maximum that can be attained. Increasing the velocity feedback past this point results in decreased rather than increased damping on that eigenvalue and at the same time moves a slow real eigenvalue near the origin. Figure 8 thus exhibits the limitations imposed by the arm structure on the speed of the servo system. Increasing ω_c is necessary after $\omega_s / \omega_c \geqslant 0.5$ in order to get a faster well damped response. This usually results in a heavier structure or in slower gross motion response. The tradeoff is thus well characterized as between gross motion response and fine motion response.

Adding a second joint with the same feedback arrangement as the first complicates the analysis and even more so the display of the results The rule that remains is that if the arm and servo display undamped natural frequencies the lowest of which approaches the locked actuator natural structural frequency, then the actual eiganvalues of the arm and servo inevitably display insufficient damping. The frequency above which damping of first mode vibrations of the arm is insufficient again seems to be one-half of the lowest locked actuator structural natural frequency. Improved performance (and a more complex system) results from more complete feedback where angular velocity and position of each joint is fed back to the other joint as well as to itself. Still better performance might result from a regulator configuration which allowed measurement and/or reconstruction of information on higher modes of vibration to be fed back to the joint control.

The locked actuator natural frequency of an arm can be increased by a redistribution of mass as well as by making the entire arm more rigid. This may involve a tapered structure or relocation of actuator masses. This relocation requires power transmission channels such as shafts, cables, or hydraulic lines which add both weight and compliance, at least partially offsetting the improvements.

When it is necessary to increase the structure's cross section size in order to increase ω_c, gross motion times suffer as described below. The value of ω_c increases linearly with r, the radius of the structure's cross section, and the inertia of the arm increases as r^2. The task time t_f to perform a gross motion of given angle θ can be shown to increase as r and thus as ω_c assuming no gravity and a bang-bang control strategy with maximum torque T_m.

One might consider what penalty one must pay for operating conservatively at a smaller ratio of ω_s / ω_c than is necessary. Consider moving from the limiting case of $\omega_s / \omega_c = 0.5$ to the slightly conservative case of $\omega_s / \omega_c = 0.3$, by increasing r, with a damping ratio of .65 in both cases. In the limiting case the nondimensional root is 80% farther from the origin than the conservative case. Thus operating at $\omega_s / \omega_c = .3$ would require an 80% larger value of r at the expense of an 80% increase in gross motion time.

The minimum value of r established by strength considerations may be larger than the minimum required to yield a sufficiently high value of ω_c. For a bending moment T_m applied to a beam, we can show that the minimum value of cross sectional radius r_{min} is proportional to $T_m^{1/3}$ and that

$$(1)\quad \omega_{c,\,min} \propto \frac{r_{min}}{\ell^2} \propto \frac{T_m^{1/3}}{\ell^2} .$$

where ℓ is the total length of the arm. As discussed previously ω_c determines the minimum settling time and thus the fine motion control speed. A comparable gauge of gross motion speed with bang-bang control of the second link inertia is

$$(2)\quad \sqrt{\frac{\theta}{t_f^2}}\Bigg|_{min} \propto \sqrt{\frac{T_m}{\ell^3\, r_{min}^2}} \propto \frac{T_m^{1/6}}{\ell^{1.5}} .$$

It is postulated that there exists a preferred ratio of gross motion speed to fine motion speed. If so, equations (1) and (2) might indicate when the structure cross section is lower bounded by the gross motion speed requirements (via maximum torque) and when it is lower bounded by the fine motion speed requirements (via ω_c). Viewed alternatively it might indicate when we must increase r from r_{min} to achieve an adequate value of ω_c. As T_m increases or ℓ decreases the ratio of $\omega_{c,min}\ /\ \theta/t_f^2$ will increase and the value of r will tend to be determined by minimum strength requirements with the resulting ω_c adequate to achieve fine motion control. When T_m decreases and ℓ increases the tendency will be for the ω_c resulting from r_{min} to be too low requiring r $> r_{min}$ for adequate fine motion control. These simple relations cannot predict where the limiting condition will change, only the tendency or relative change as the arm parameters change.

CONCLUSIONS

This paper has set forth the major design - control issues for mechanical arms. Further work will be reported in subsequent papers.

REFERENCES

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3. W.J. Book, "Modelling, Design and Control of Flexible Manipulator Arms" PhD Thesis, MIT Mechanical Engineering Dept, 1974.

Figure 1. General Payload/Task Time Curve

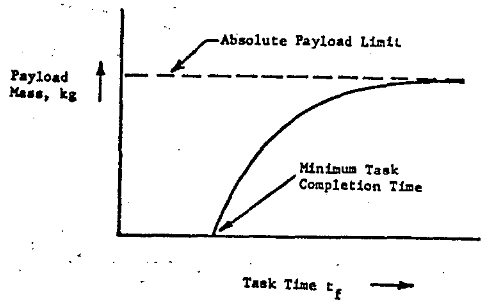

Figure 2. One Degree of Freedom Arm

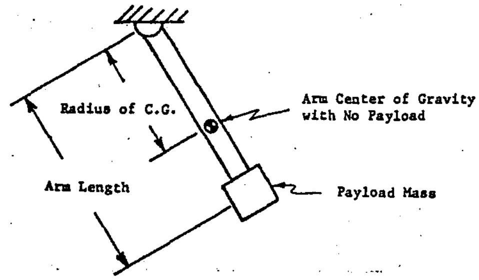

Figure 3. Desired Task Motion

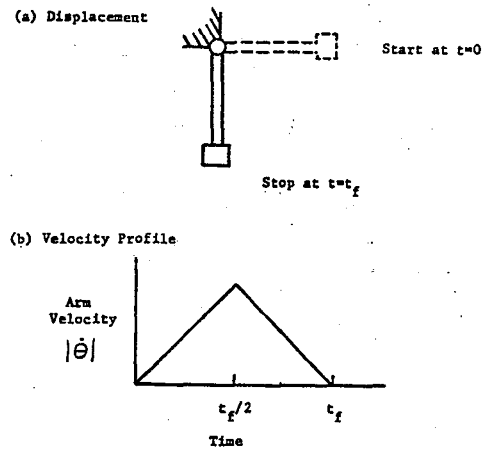

Figure 4. Task Torque History

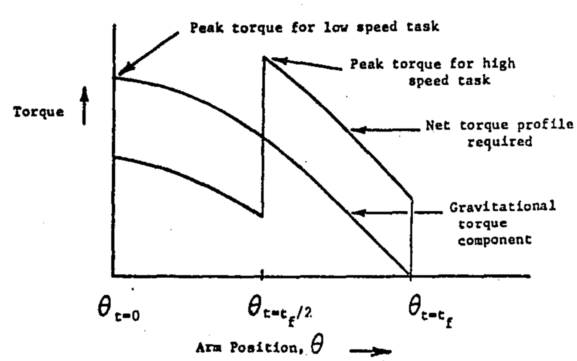

Figure 5. One Degree of Freedom Performance Curve

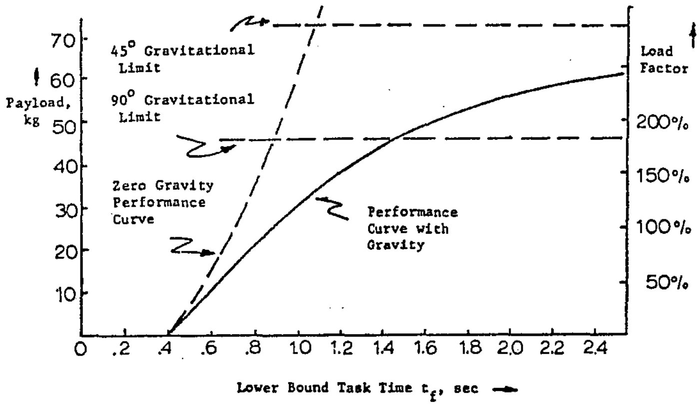

Figure 6. Three Joint Model Performance Curve

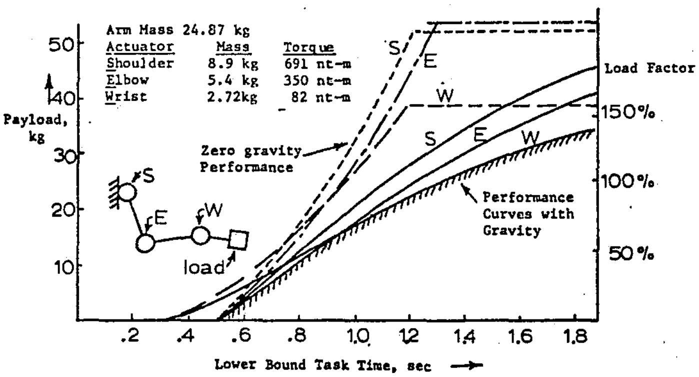

Figure 7. Simple arm example.
First eigenvalues shown in Figure 8.

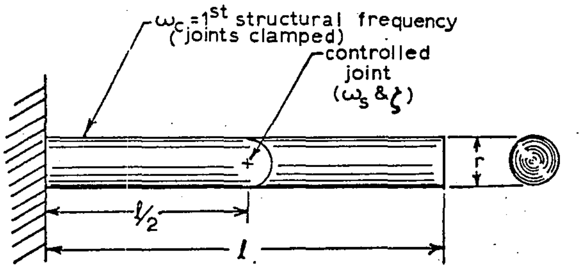

Figure 8. Locus of complex first eigenvalues for variation in ω_s / ω_c and ζ . (Positive root of conjugate pair shown. Locus of real eigenvalues and eigenvalues of higher modes not shown).