Rule-Based Damping Control for Magnetic Bearings

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ABSTRACT

The derivative control of a conventional PID controller is modified using fuzzy logic. The modified magnetic bearing controller produces low gains in the high-frequency range. Therefore, the bearings can be made stiff in the low-frequency range with less audible noise.

INTRODUCTION

Conventional magnetic bearing controllers include PID (proportional, integral and derivative) and phase compensation circuits among others. The derivative or phase compensation controls are the electronic means that provide the necessary bearing damping over a frequency range, typically 10 to 1000 Hz, for suppressing rotor resonances. The frequency response of a PID controller has a gain curve shaped like a bath tub. In other words, the controller has high gain in the high-frequency range (500 to 2000 Hz, typical) due to the derivative controls. Because of the high gain, magnetic bearings are sensitive to noise in the audible frequency range. In fact, music can be played through a magnetic bearing using it as an acoustic amplifier.

It is common to hear broad-band noise when a magnetic bearing is activated, because amplified system noise excites both rotor vibration modes and stator structural modes. One must compromise to achieve an acceptable noise level by limiting the magnetic bearing to low dynamic stiffnesses in the low-frequency range (typically 10 to 500 Hz). This can be a serious limitation for some applications. For example, during start-up of an induction motor, the inrush current of the motor creates a large dynamic side pull. To support the motor with magnetic bearings, the low-frequency stiffnesses must be high, or the rotor may not be able to remain levitated during the transient.

High gain at high frequency is characteristic of linear controller designs. Even using complicated, compensating circuitry, it is difficult to achieve a good compromise. Herein, we propose a rule-based control scheme to reduce the high-frequency gain and thus achieve a quiet but stiff magnetic bearing at lower frequencies. The scheme involves the use of fuzzy logic [1] to manipulate the derivative part of the bearing control.

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RULE-BASED DAMPING CONTROL

Our goal is to minimize high-frequency controller gains. Considering the PID control as three parallel paths, as shown in Figure 1a*, the D-path is modified by using fuzzy logic, as shown in Figure 1b. The modified control is labeled PIFD ("F" for "fuzzy"). No further phase compensation in series with the modified controller is required.

The output of the modified D-path is generated using instinctive rules determined by engineers. These rules are applied to two inputs - measured rotor displacement and the associated velocity. The velocity is a filtered differentiation of the measured displacement. The two inputs and one output are each categorized into two linguistic variables - SMALL and LARGE. Smallness and largeness are quantified using membership functions in fuzzy logic. Table I shows how the intuitively derived rules governing the box labeled "ruled-based damping control" in Figure 1b may be tabulated.

Note that |X| and |dX/dt| are absolute values of displacement and velocity. The sign of the output naturally is "-SIGN(dX/dt)." To facilitate continued discussion, the rules are numbered below.

If	Then use
Rule 1: SMALL |X| and SMALL |dX/dt|	LARGE damping force
Rule 2: SMALL |X| and LARGE |dX/dt|	SMALL damping force
Rule 3: LARGE |X| and SMALL |dX/dt|	LARGE damping force
Rule 4: LARGE |X| and LARGE |dX/dt|	SMALL damping force

Rules 1 and 3 are for the low-frequency range where |dX/dt| is usually small and the bearing needs damping. Rule 2 is intended for reducing the gain in the high-frequency range where |dX/dt| is high and the controller is better off doing little with it. Rule 4 is for a troublesome situation when the D-path is apparently not doing the job and the controller may be better off using less derivative gain. Agreement on these rules, which are based on the designer's control experience, may not be universal.

The membership functions used to quantify smallness and largeness can be defined with straight lines that overlap between SMALL and LARGE to represent the inherent subjectivity. The smaller of the two input memberships (|X| and |dX/dt|) is used for calculating the output. If more than one rule applies, the output force will be an average. A numerical example is provided below to clarify these details.

NUMERICAL EXAMPLE

An analytical simulation was performed to compare the linear PID control with the PIFD control. The simulated system was an electromagnet suspending a weight of 400 lb. The dynamics of the sensor filter and the power amplifier/coil subsystem were ignored. The nonlinear magnetic force was included in the formulation. The natural frequency was set at 400 radians/sec (64 Hz). For the linear controller, the stiffness was about 2×10^5 lb/in. and the damping coefficient was 125 lb-sec/in.

Figure 2 presents three membership functions used in the simulation. Note that the higher bound of SMALL overlaps with the lower bound of LARGE in the three functions. For example, 2 mils may represent a small displacement in some applications, while 1 mil may be viewed as a large displacement in other applications. For a vibration amplitude of

*For ease of readability, all figures and tables are included at the end of this paper.

2 mil at the natural frequency, the damping force of the linear controller would be 100 lb. Therefore, the maximum damping force of the rule-based controller was set at 100 lb. At any on-line control instance, there exists a specific set of displacement and velocity magnitudes (|X|, |dX/dt|). Using the displacement and velocity membership functions, a set of memberships (UX,UDX) is calculated. The corresponding membership for the damping force is taken as "MIN(UX,UDX)", or the smaller of the two. All four rules are checked for their applicability. If more than one rule applies, the output force is the sum of the damping forces divided by the number of the applied rules.

The impulse responses of the system using the conventional PID control and the PIFD control are compared in Figure 3. Apparently, the rule-based control system is overdamped when the vibration amplitude is small and less well damped than the PID control system when the amplitude is large. This is precisely what the four rules were designed to achieve.

Figure 4 compares the frequency responses of the two controllers. For the PIFD controller, two describing functions [2] are plotted - one for a vibration amplitude of 1 mil, the other for 0.5 mil. The PIFD controller is naturally nonlinear and amplitude dependent. The smaller vibration amplitude has higher gain (stiffness) and more phase lead (damping). Compared to the linear PID controller, the PIFD controller has obviously much lower gain in the high-frequency range. It also has more evenly distributed phase lead in the low-frequency range, which is beneficial to the control of rotor criticals.

The harmonic contents of the PIFD-controlled system are presented in Table II for a sinusoidal vibration amplitude of 1 mil. The large numbers shown in the table are the dynamic stiffnesses of the closed-loop system. Only odd harmonics exist because the controller output is an odd function of its velocity input. The fundamental signals are approximately an order higher in amplitude than the third and fifth harmonics.

The PIFD control force for a sinusoidal vibration amplitude of 1 mil at 100 Hz is compared with the corresponding linear control force in Figure 5. The ripples on the nonlinear force are the source of the harmonics.

DISCUSSION AND CONCLUSIONS

The rule-based approach is a time-domain controller design method. Preferably, it should be implemented in digital signal processors (DSP) or other commercial fuzzy logic hardware. With floating-point DSP boards, the sampling rate should be high enough for magnetic bearing applications, because the rules of fuzzy logic are simple. Taking the latest DSP TMS30 from Texas Instruments, for example, the sampling rate was estimated to be 50 kHz for one control axis using the PIFD controller in the above numerical example. Therefore, each DSP is capable of controlling five axes, or two radial bearings and one thrust bearing.

If necessary, the harmonic contents of the rule-based controller output can be partially filtered out prior to input to the power amplifiers. The phase lag of this additional low-pass filter should not be a problem, because significant phase lead is achievable in the low-frequency range as indicated by the preceding example.

The rule-based controller using fuzzy logic can provide damping without causing high-frequency noise. By manipulating the membership functions, one can change the frequency bandwidth with a desirable amount of phase lead. This flexibility also enables an increase in the proportional gain of the controller and achievement of high dynamic stiffness at lower rotor critical frequencies.

ACKNOWLEDGEMENT

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REFERENCES

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2. Cook, P. A. 1986. "Nonlinear Dynamical System." Prentice Hall, Chapter 3.

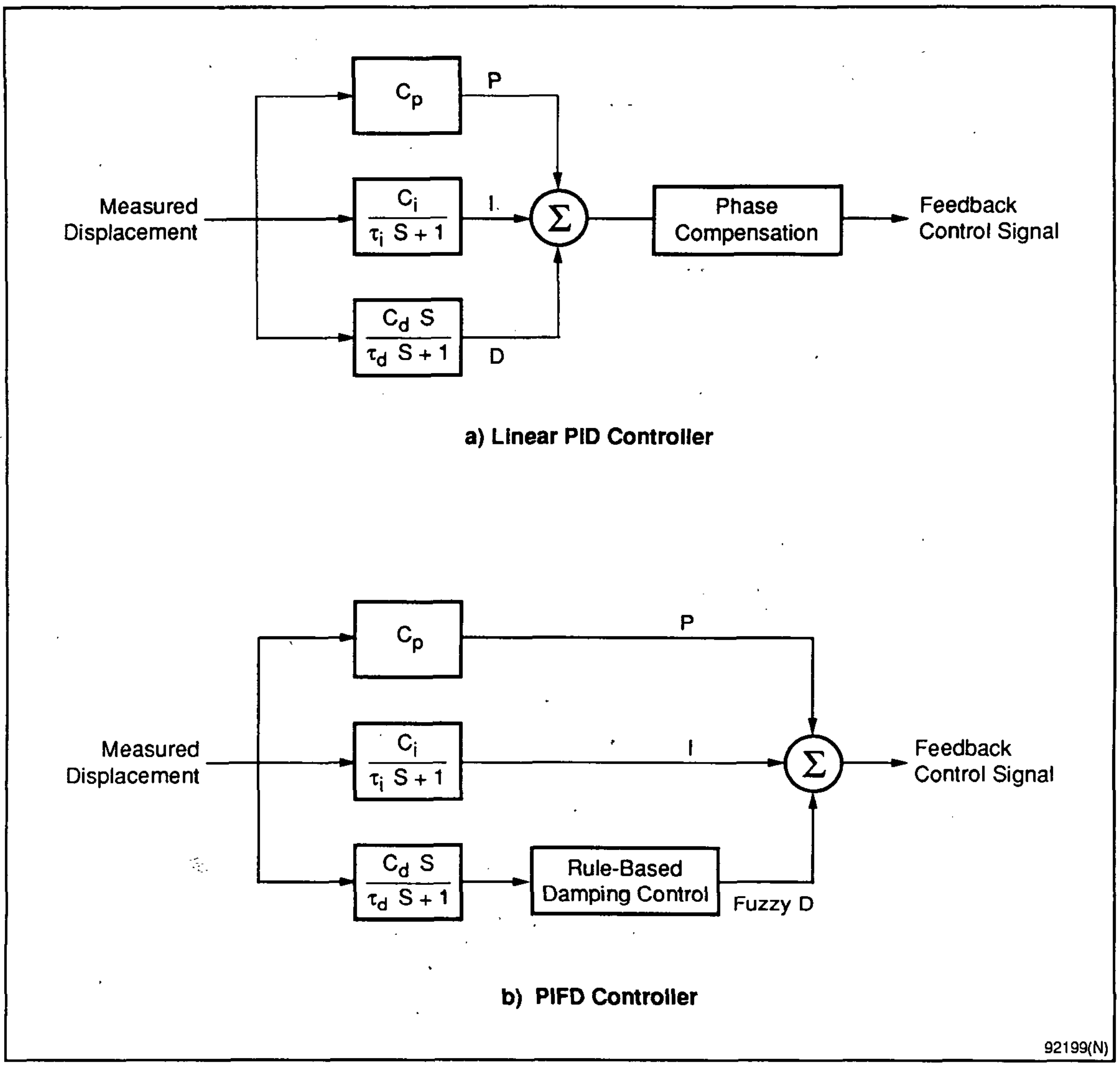

Figure 1. Conventional versus Rule-Based Damping Control

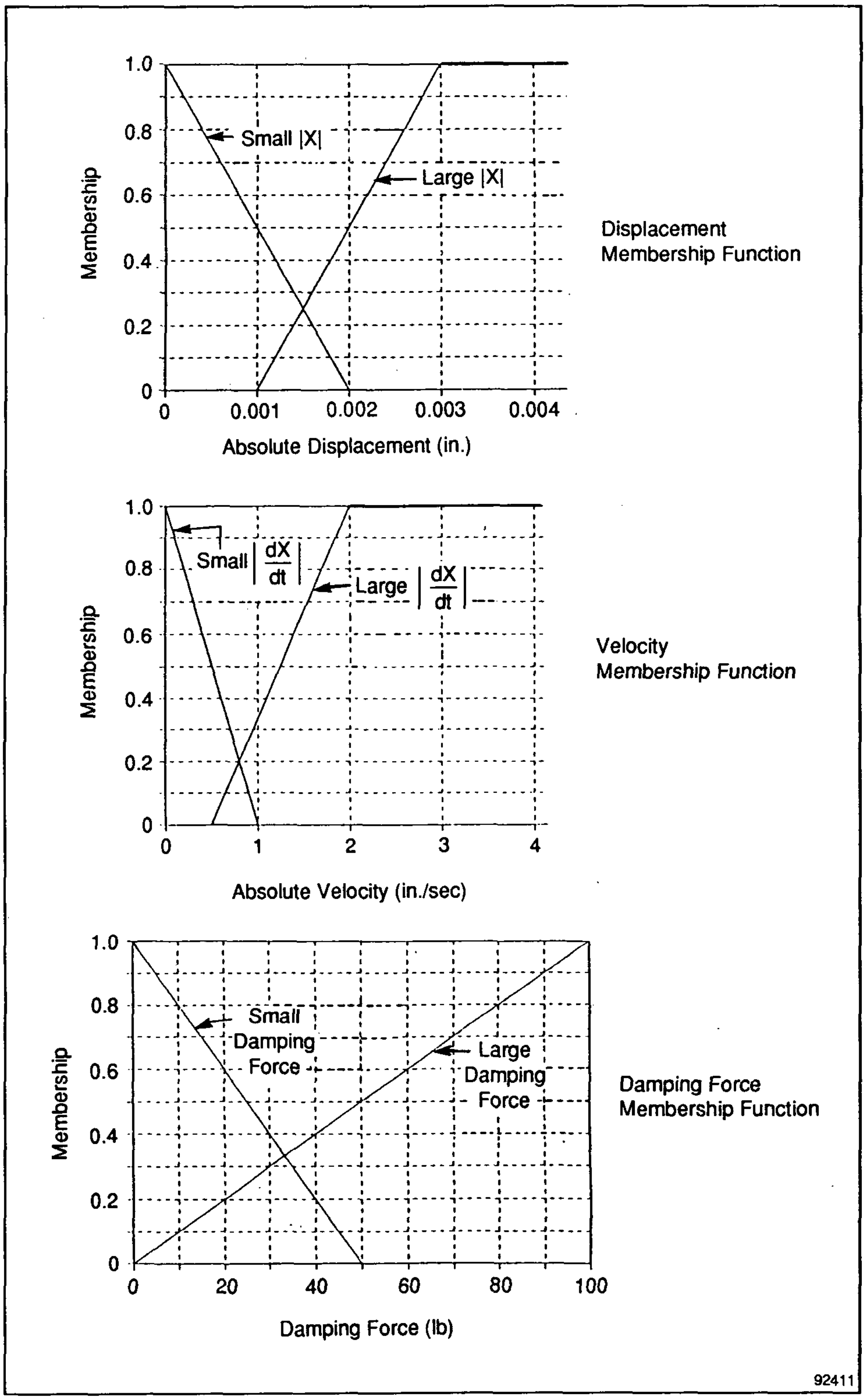

Figure 2. Membership Functions

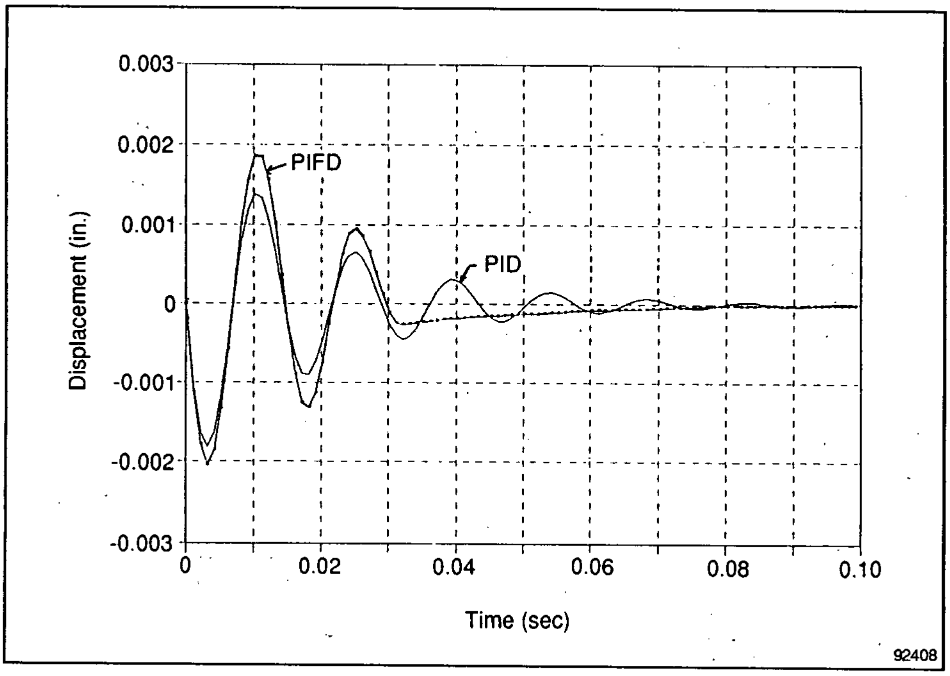

Figure 3. Comparison of Impulse Responses

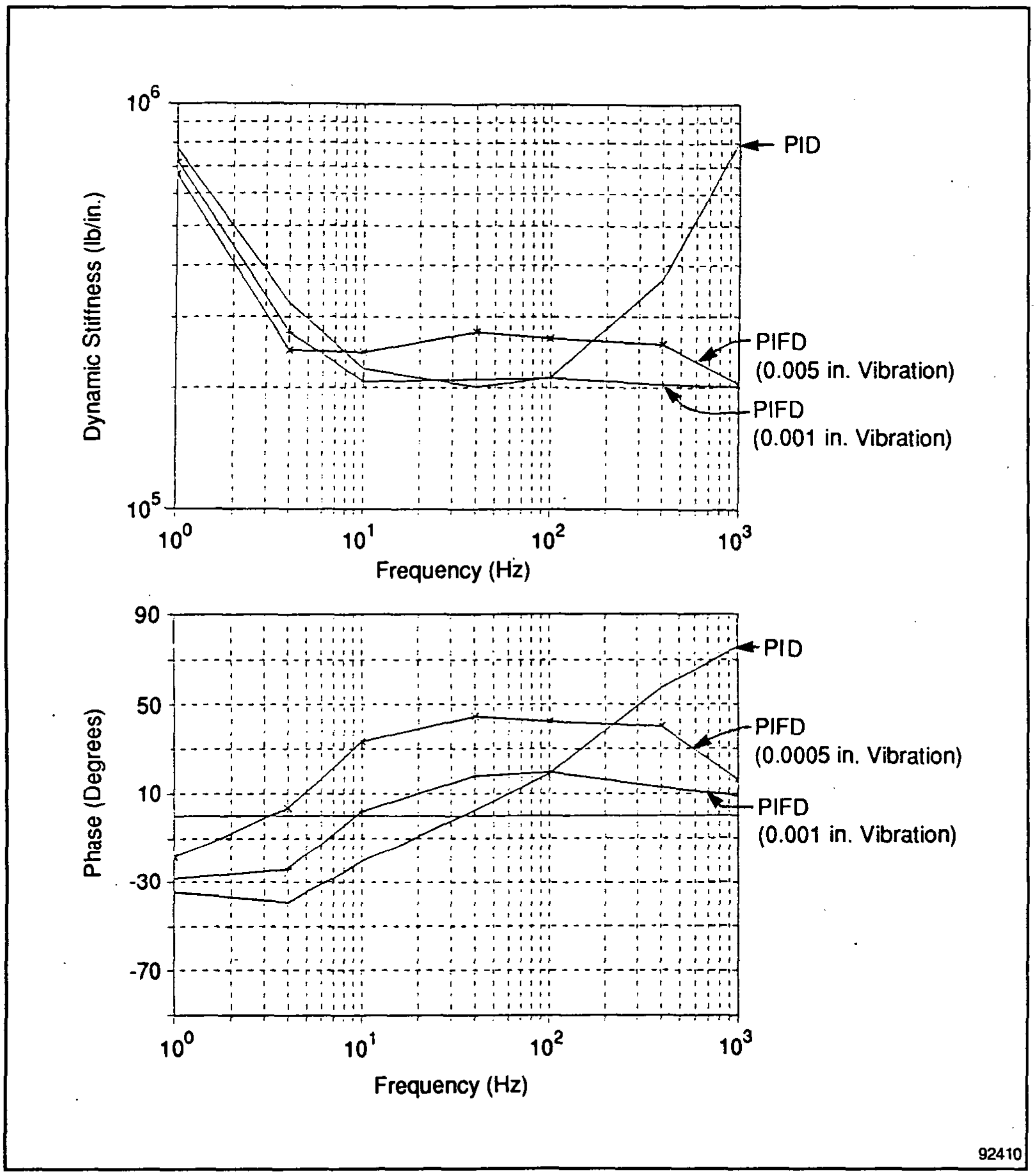

Figure 4. Comparison of Frequency Responses

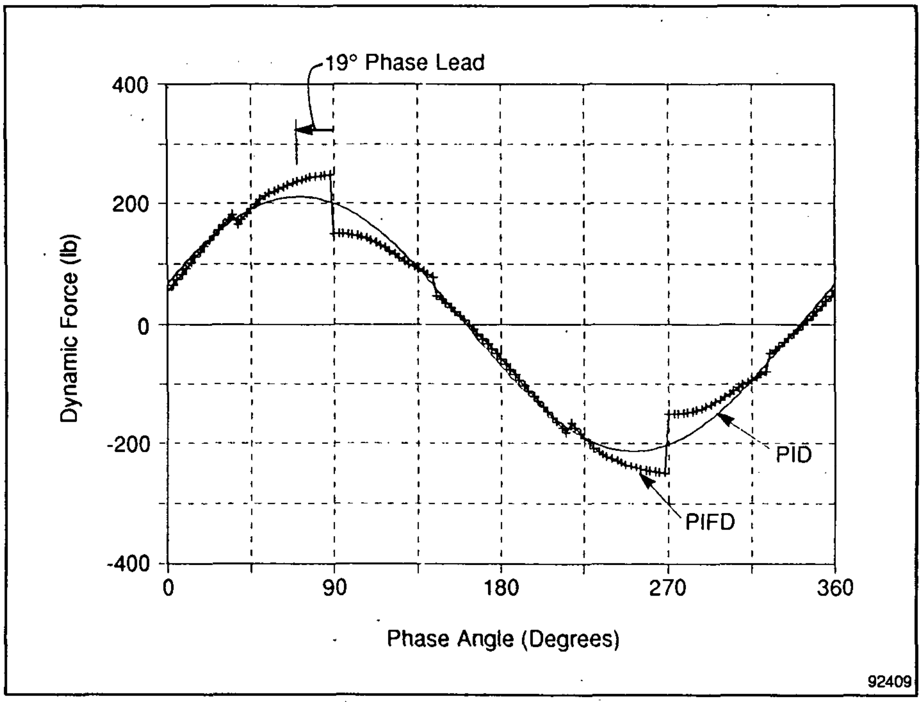

Figure 5. Comparison of Dynamic Control Forces
at 100 Hz and 0.001 in. Vibration

TABLE I. INTUITIVELY DERIVED DAMPING CONTROL MATRIX

<table>
<tr><td colspan="2" rowspan="2"></td><td colspan="2">Displacement |X|</td></tr>
<tr><td>Small</td><td>Large</td></tr>
<tr><td rowspan="2">Velocity |dX/dt|</td><td>Small</td><td>Large Damping Force</td><td>Large Damping Force</td></tr>
<tr><td>Large</td><td>Small Damping Force</td><td>Small Damping Force</td></tr>
</table>

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TABLE II. HARMONICS OF RULE-BASED CONTROL SYSTEM

Frequency (Hz)	Dynamic Stiffness (lb/in.) Harmonic Components		
	×1	×3	×5
0.1	0.1070E+07	0.1066E+05	0.1488E+05
0.4	0.9798E+06	0.1067E+05	0.1487E+05
1.0	0.7258E+06	0.1068E+05	0.1486E+05
4.0	0.2744E+06	0.1075E+05	0.1479E+05
10.0	0.2077E+06	0.1116E+05	0.1439E+05
40.0	0.2100E+06	0.1706E+05	0.1021E+05
100.0	0.2114E+06	0.1865E+05	0.1176E+05
400.0	0.2041E+06	0.3968E+05	0.1552E+05
1000.0	0.2013E+06	0.3380E+05	0.1358E+05

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