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Motion Characteristics of the VIRTTEX Motion System

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ABSTRACT

The motion characteristics of the VIRTTEX motion system were measured using a set of tests similar to those suggested in AGARD Report AR-144. The measurements were devised to measure the performance of existing motion systems, aid in the procurement of new motion systems, and help identify root causes of poor performance. Transfer functions (both within degree-of-freedom and between degree-of-freedom), ½ Hz noise levels, signal to noise contours, dynamic threshold measurements and hystersis measurements were all conducted. Before these tests, the VIRTTEX motion system was tuned with an emphasis on high bandwidth. The motion system as tuned, has a bandwidth greater than 13Hz on all degrees-of-freedom. The dynamics threshold is 15ms and the motion system has no significant hystersis. The transfer function measurements revealed larger than expected levels of cross talk between degrees-of-freedom. The motion system also exhibits turn-around bump and non-linearity at large velocities and accelerations. Further experience with the complex MTS controller is expected to lead to improvements in both cross talk and non-linearity. This measurement process will be repeated once planned improvements to the system are complete.

INTRODUCTION

Ford Motor Company recently installed the new VIRTTEX (VIRtual Test Track Experiment) driving simulator located at the Ford Scientific Research Laboratories in Dearborn, Michigan. VIRTTEX consists of a 24-foot diameter dome section mounted on top of a high performance six degrees-of-freedom hydraulic motion system. A vehicle cab is mounted inside the dome, with the driver's eye-point located at the center of the dome viewing volume. Five projectors are used to project a 180°x40° front and a 120°x25° rear image onto the inside surface of the dome. The inside surface of the dome is covered with a high-gain coating to provide a bright image at the drivers eye-point location.

VIRTTEX is a research tool for investigating driver-vehicle interactions. It will be used to investigate human factor related safety issues within the driving environment, driving dynamics and customer related brand identity issues. Given the current interest in telematic devices for road vehicles and the potential distraction they cause drivers, the initial VIRTTEX studies will involve driver distraction issues. As with any piece of research equipment it is imperative that the performance of the motion system is well documented. This allows researchers to assess the suitability of VIRTTEX for a given research project, and evaluate the effects of the motion system performance on experimental results.

This paper presents a comprehensive evaluation of the VIRTTEX motion system characteristics based on AGARD Report 144 (1) and the work of Reid and Grant (2). Describing functions are used to characterize the bandwidth of the system. Cross talk between degrees-of-freedom are characterized with cross talk describing functions. A set of ½ Hz noise level tests is used to describe the smoothness and non-linearity of the motion system. Signal-to-noise contours are used to depict the signal to noise ratio of the system as a function of amplitude and frequency. The response of the motion system to a step input is documented as a dead time and rise time. An estimate of the positional hysteresis of the motion system is also presented.

MOTION CONTROLLER TUNING

The MTS motion controller is a combination of a classical feedback and an unconventional feedforward controller (3). Unlike most hexapod controllers, the MTS system works in a world coordinate system and not in actuator space. The feedforward controller has eight mass properties that must be selected. The feedback controller has nine tunable parameters per degree-of-freedom. There are 9 additional tunable parameters per degree-of-freedom that can be used to shape the valve control signals in world coordinates. Ideally, the feedforward controller dominates the control loop with only a low-gain feedback position loop to control the low frequency positioning of the system.

The system was tuned as per MTS advice with an emphasis on achieving a minimum bandwidth of 13Hz. Significant feedback gain and lead were required to achieve the desired bandwidth. This likely contributed to the cross talk described later in the paper. Due to the complexity of the controller and our lack of tuning experience, it is unlikely that the tuning used for these tests is optimal. These motion measurements proved to be very useful for deciding how future tuning should proceed.

INSTRUMENTATION

The instrumentation that was provided by MTS as part of the feedback control system was used to measure the motion system performance. Nine Endevco 7290A-10 capacitive accelerometers are mounted to the actuator swivel blocks on the motion platform. The accelerometer data is filtered with a 300Hz 5 pole Bessel filter. The filtered data is then digitized using a 16-bit D/A converter. Actuator lengths are measured using 25bit Temposonics magnetostrictive position transducers. Translational and angular states of the platform are estimated using a combination of the nine accelerometers and six actuator length signals.

ANALYSIS AND RESULTS

Transfer Functions

Transfer function measurements were estimated by driving each degree-of-freedom with band-limited white noise. AGARD-144 and Reid and Grant (2) suggest a sum of sine waves to drive each degree-of-freedom, but current limitations in the MTS software interface made the use of sine waves impractical. For a linear system, the two methods produce identical results. The motion system was thought to be almost linear so band-limited white noise was deemed appropriate for determining the transfer functions.

For each degree-of-freedom six transfer functions were generated. The primary transfer function relates the response on the driven degree-of-freedom to the driving signal. The five other cross talk transfer functions relate the parasitic response on a non-driven degree-of-freedom to the driving signal.

For the translational degrees-of-freedom a 25Hz band-limited white noise source with a RMS of 0.25 m/s^2 was used to drive the system. For the rotational degrees-of-freedom the RMS of the 25Hz band-limited white noise was chosen to be 6 deg/s^2. The commanded acceleration and the six measured accelerations were sampled at 500Hz for 90 seconds. This provides a maximum frequency resolution of 250Hz.

The transfer function estimates were generated using the MATLAB transfer function estimator (4). This algorithm uses Welch's method to calculate the power spectral density, ϕ_{xx} of the driving signal and cross-spectral, ϕ_{xy} density of the driving signal and the motion base response. The transfer function is then found using:

$$L(jw) = \frac{\phi_{xy}}{\phi_{xx}} \quad (1)$$

A 500-point window was used with an overlap of 250 points. Each 500-point section was windowed with a 500 point Hanning window. The resulting frequency resolution is 1.0 Hz. This gave the best compromise between a high coherence and a good frequency resolution. The coherence between the driving signal and the motion base response was calculated using the MATLAB coherence estimator.

The primary transfer function for the X degree-of-freedom is shown in Figure 1. From the figure, it can be seen that the amplitude ratio is within 3dB out to 18Hz. The 90-degree phase lag point occurs at approximately 14.5 Hz. For a hydraulic hexapod of this size, this is an extremely high bandwidth.

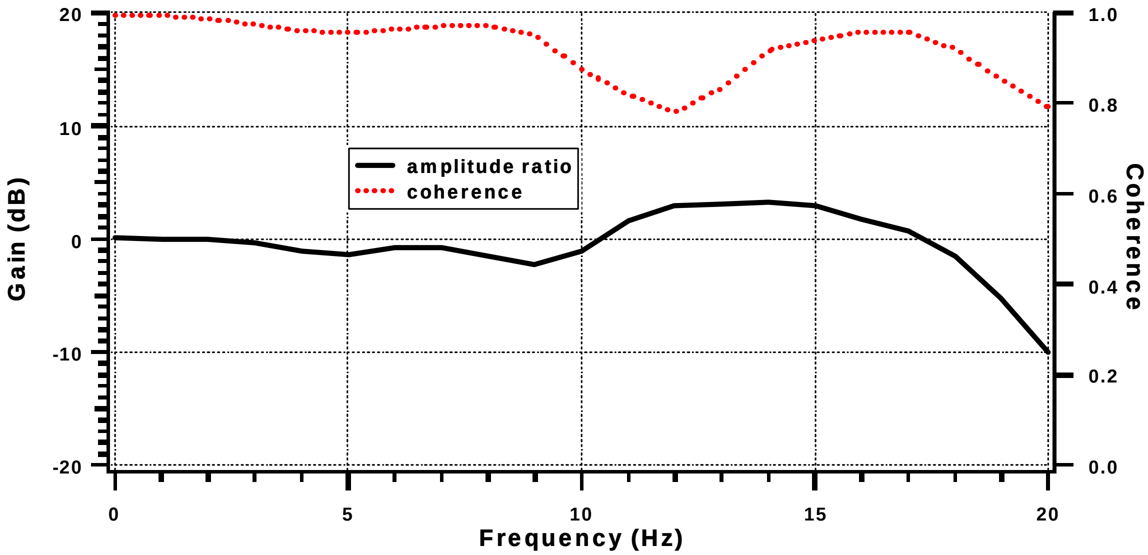

FIGURE 1a X-Transfer Function Amplitude Ratio and Coherence.

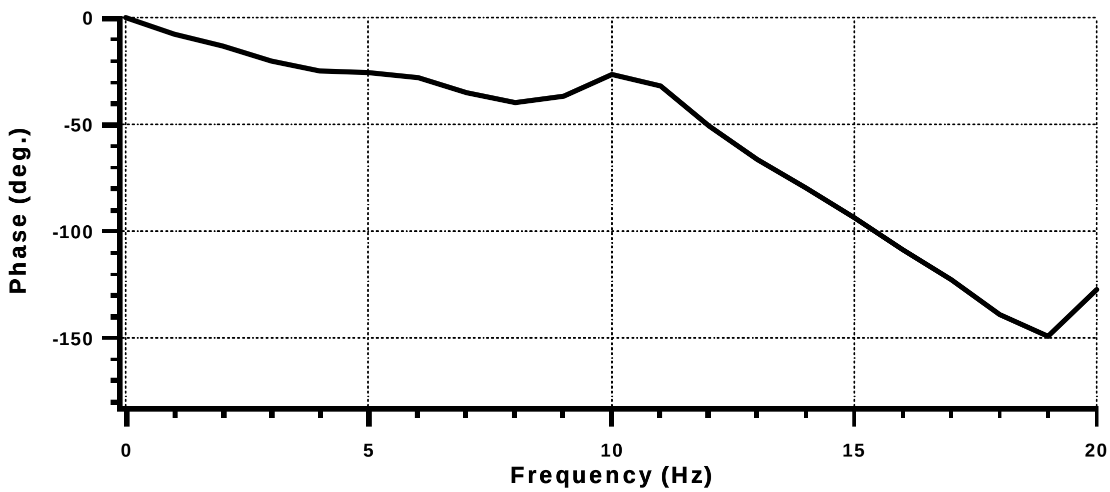

FIGURE 1b X-Transfer Function Phase Error.

The Y cross talk for the motion system driven in X is shown in Figure 2.

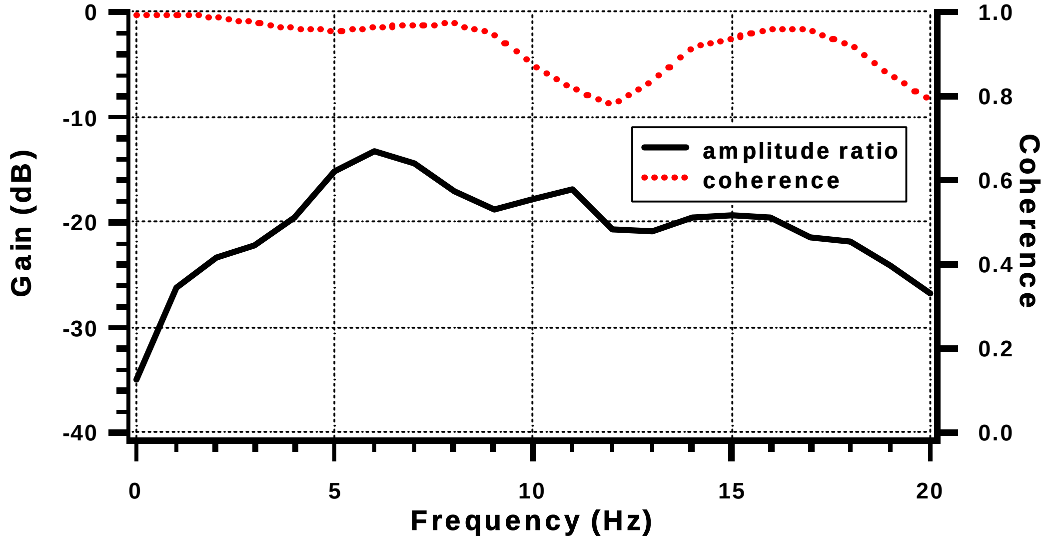

FIGURE 2 Y/X-Cross Talk Transfer Function.

For frequencies below 4Hz the cross talk is below –20dB. Between 4 Hz and 8Hz, the cross talk rises to almost –12dB at 6Hz. There is a second smaller rise at 11Hz, almost double the frequency of the first rise. The X and Y oil column resonance are approximately at 5.5Hz. It seems likely that the rise in the cross talk at 6 and 11 Hz are related to a lack of control near the oil column resonance and its first harmonic. The coherence is close to one over the frequency range of interest, indicating the transfer functions are statistically meaningful.

Similar results were obtained for the other degrees-of-freedom, with the exception of the cross talks between X and pitch and Y and roll. These cross talks were significantly larger than the cross talk between Y and X. This is due to the coupling induced by the high center of gravity of the motion payload. The feedforward model should be to control this coupling 34) but the current tuning may not take full advantage of the feedforward controller. Compared with the UTIAS flight research simulator motion base (2) the VIRTTEX motion system has much higher bandwidth but significantly more cross talk.

½ Hz Noise Level Tests

For the ½ Hz noise level tests the motion system is driven with a ½ Hz sine wave in each degree-of-freedom and the response on all the degrees-of-freedom is measured. Six different amplitudes were chosen to span the practical operational range of the simulator. Figure 3 shows the response of the motion system to a 1.28m/s^{2,} ½ Hz sine wave in the X degree-of-freedom.

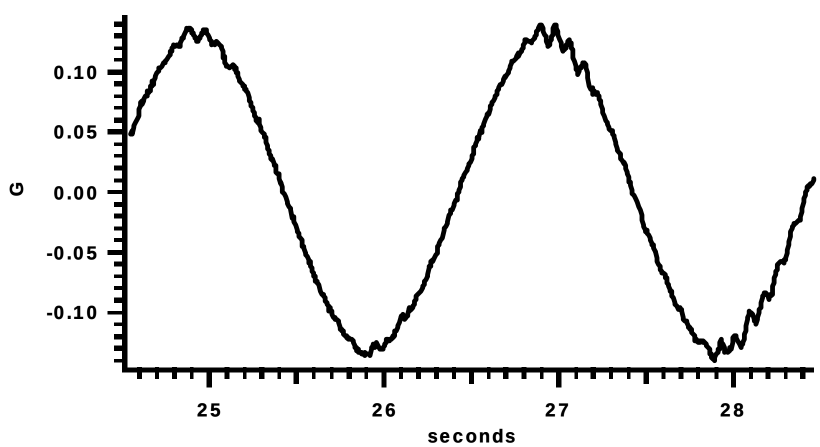

FIGURE 3 X response to ½ Hz 1.28 m/s^2 input.

The power spectrum of the signal was used to determine the noise in various frequency bands. The variance of the driving signal is found using

$$\sigma_d^2 = 2\phi_{xx}(i_f \Delta f)\frac{\sum w(j)^2}{(\sum w(j))^2} \quad (2)$$

where $i_f \Delta f$ is the frequency of the driving signal and w(j) is the value of the windowing function in the time domain. The total noise is estimated using:

$$\sigma_{tot}^2 = \sum 2\phi_{xx}(i\ \Delta f)\frac{\sum w(j)^2}{(\sum w(j))^2} - \sigma_d^2 \quad (3)$$

On non-driven degrees-of-freedom the driving variance is not subtracted from the total noise variance. The low-frequency non-linearity is the sum of the first two harmonics of the driving signal. It is only calculated for the driven degree-of-freedom. It is found using:

$$\sigma_{lfnl}^2 = \sum_{n=2}^{3} 2\phi_{xx}(ni_f \Delta f)\frac{\sum w(j)^2}{(\sum w(j))^2} \quad (4)$$

The high-frequency non-linearity is the sum of all remaining harmonics up to the maximum frequency in the power spectral estimate (100Hz in this case). It is found using:

$$\sigma_{lfnl}^2 = \sum_{n=4}^{N/2+1} 2\phi_{xx}(ni_f \Delta f)\frac{\sum w(j)^2}{(\sum w(j))^2} \quad (5)$$

where N is the number of samples in the window. Finally, the roughness is defined using:

$$\sigma_r^2 = \sigma_{tot}^2 - \sigma_{lfnl}^2 \quad (6)$$

The power spectrum was estimated using a MATLAB estimation routine (4) that employs Welch's method. For this analysis, the data was sampled at 200Hz and a window size of 10000 points was chosen to obtain a frequency resolution of 0.05Hz. A 10000-point Boxcar windowing function was used. Ninety seconds of data was collected for each run. These parameters were chosen to give the best estimate to the variance of the driving signals and its harmonics. These spectral estimate parameters will lead to poor estimates for any single value of the power spectrum that does not contain power from the driving signal (either at the driven frequency or one of its harmonics). Since we are only interested in the sum of the variances over the entire band excluding the harmonic frequencies, this is not a significant source of error. Comparisons between the total noise variance for non-driven degrees-of-freedom calculated using standard time domain analysis and those calculated in the frequency domain confirmed this method produces accurate results.

The peak noise was also estimated for all degrees-of-freedom. For the non-driven degrees-of-freedom it is simply the maximum absolute value in the time history of the signal. For the driven degree-of-freedom the system response at the frequency of the driving signal was subtracted from the time history. The system response at the driving frequency was determined using FFT analysis. The FFT of the time record was calculated and the amplitude and phase of the sine wave response of the motion system at the driving frequency was determined. This was subtracted off the time history and the peak noise is then the maximum absolute value of the resulting time history.

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Figure 4 shows $\sigma_{tot}, \sigma_{lfnl}, \sigma_{hfnl}$, and σ_r for the Z acceleration response plotted against the measured amplitude of the Z driving signal. The peak noise is also shown with the scale on the right hand side.

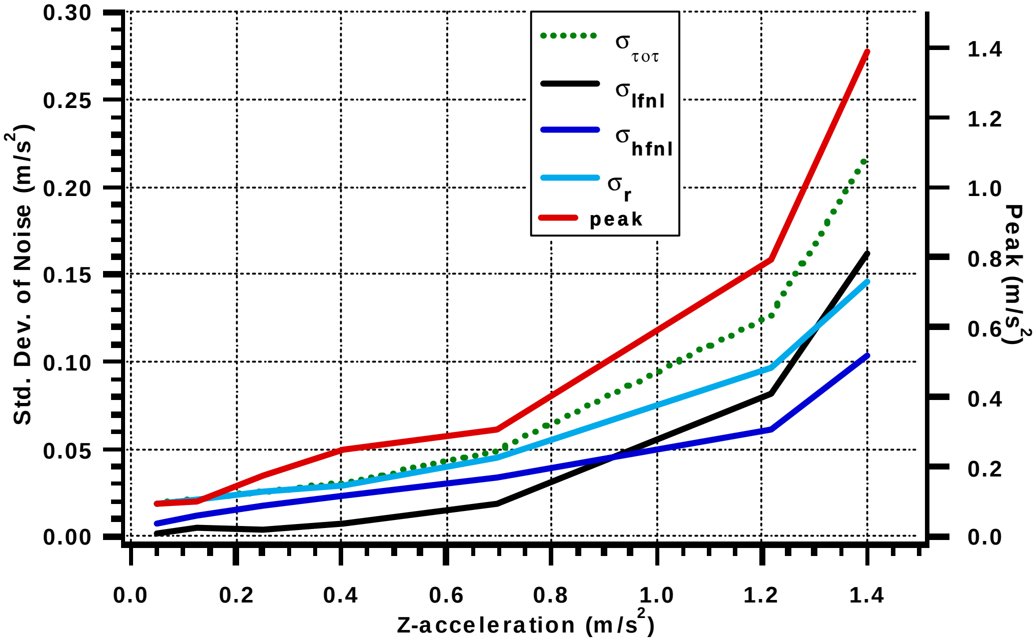

FIGURE 4 Z Noise Level Tests, Z-Commanded.

At low driving amplitudes, the noise is dominated by the roughness component. This suggests a background noise dominates the system response. The peak noise at low amplitudes remains constant out to 0.7 m/s^2. There is very little low frequency non-linearity at low accelerations. Examination of the time histories reveals that the peak noise for accelerations below 0.7m/s^2 is due to turn-around bump. This is demonstrated in Figure 5.

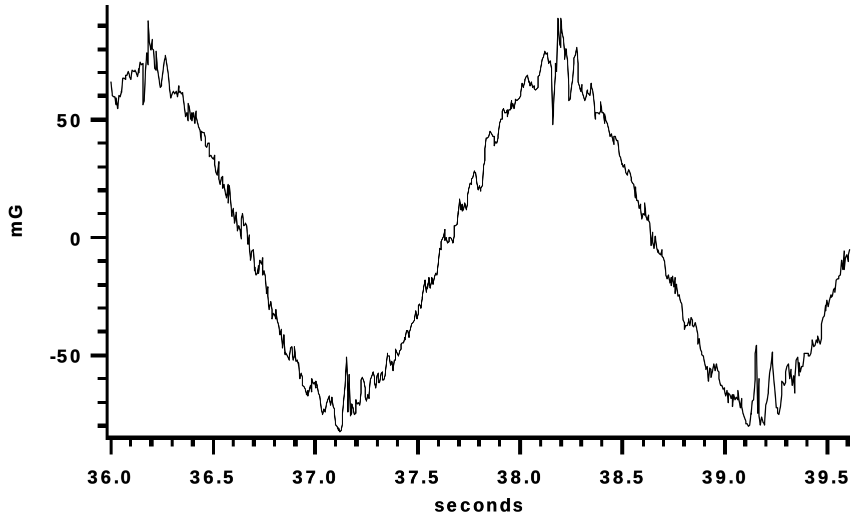

FIGURE 5 Z acceleration, 0.5Hz 0.07 m/s^2

This turn around bump may be caused by transient flow conditions in the valve that occur when the valve goes through zero opening, or by friction. As the velocity goes to zero the friction changes from sliding friction to stiction, this can generate a transient force on the actuators. From Figure 4 we can also see that there is a rather sharp rise in the low frequency non-linearity at high accelerations. Figure 6 shows the power spectrum for the 1.4 m/s^2 case.

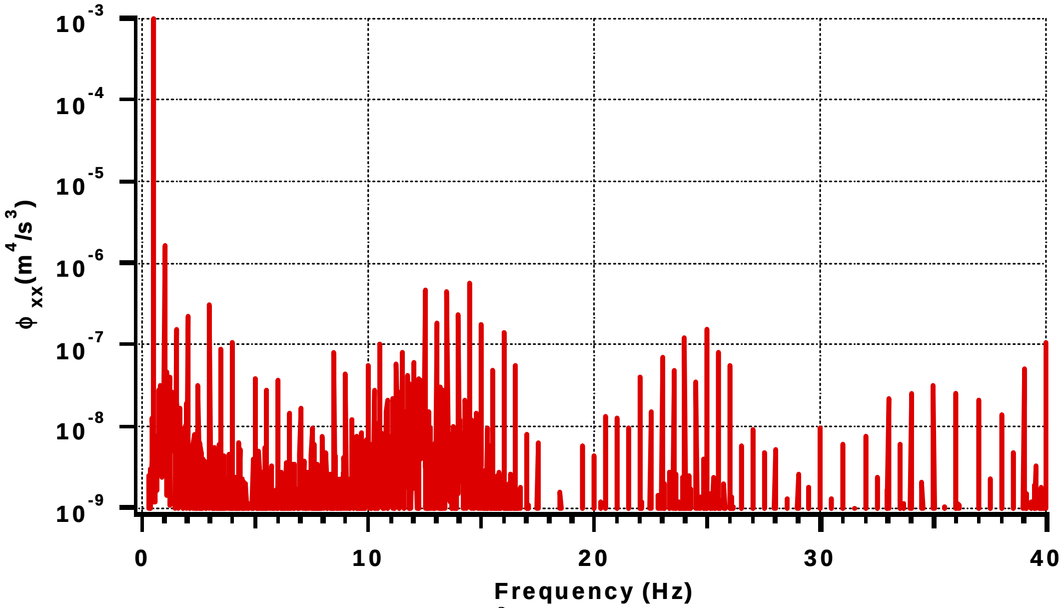

FIGURE 6 Z power spectrum, 0.5Hz 0.14 m/s^2.

As can be seen in this figure the first harmonic of the driving signal is quite large. There is significant power in most of the harmonics out past 40Hz. In general, the Z acceleration noise is large, with the peak noise being particularly large. The peak noise in heave at low acceleration is approximately half the magnitude of the driving signal and rises to almost equal the driving signal amplitude at the highest accelerations.

Figure 7 shows the peak and total noise measured in the Y degree-of-freedom for the base driven in Z.

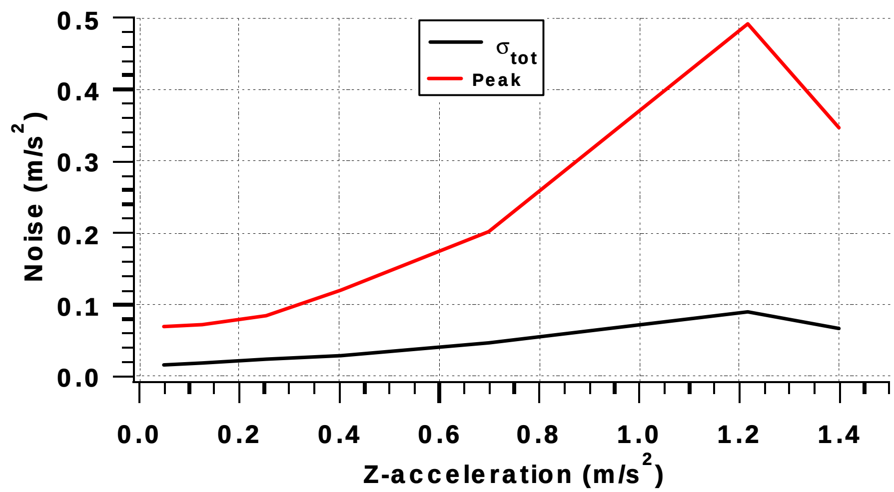

FIGURE 7 Y-noise for 0.5Hz Z-acceleration.

The total RMS noise is relatively small but the peak noise is quite large. This is not surprising since the disturbances associated with the turn-around bump occur along the actuator and not in a given degree-of-freedom.

To date only the Z and X degrees-of-freedom have been analyzed. As can be seen in Figure 8, the results for X are quite similar to the Z results for very low amplitudes but are significantly less noisy at the higher amplitudes.

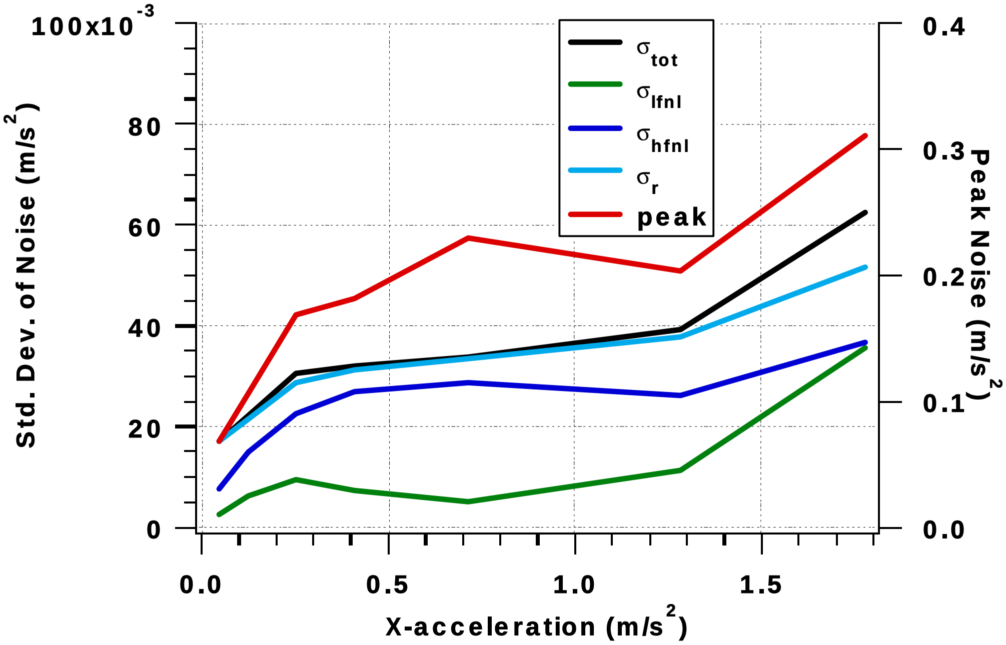

FIGURE 8 X Noise Level Tests, X driven.

This is somewhat expected as any anomalies in the actuator response will tend to cause larger disturbances in Z since all the actuators are contributing equally to the acceleration of the motion system. When the motion system moves in X some of the actuators have very little motion so thee disturbances from those actuators may be smaller; particularly if the disturbances are related to the amplitude of the actuator motion. It seems unlikely this can explain the entire difference between the X and Z results. It seems likely that the current tuning has exaggerated the noise in the Z-direction. The peak noise in X stays relatively constant out to 1.0 m/s^2 which suggest a friction phenomena is responsible for the turn-around bump in X. Flow conditions in the valve would tend to scale with the amplitude of the input. The peak noise in X for low and high amplitudes is significantly larger than required for high fidelity simulation.

Signal-to-Noise Contour Plots

Signal to noise contours are used to document the operating region of the simulator based on the total signal to noise ratio. An alternative approach for defining the operational region of the simulator is simply to use the position, velocity and acceleration limits of the motion system. As suggested by Reid and Grant (2), however, the only definitive limit is positional. The actuators have fixed lengths with piccolo cushions to decelerate the actuator as it approaches the end of its stroke. Velocity and accelerations limits of the motion system are less definitive and depend on the characteristic of the signal as well as the velocity and/or acceleration. Signal to noise contours are therefore a more useful way of defining the operational envelope of the motion system.

Thirty one signal-to-noise measurements were made over the frequency range from 0.1Hz to 11Hz for the X degree-of-freedom. The signal to noise contours for the remaining degrees-of-freedom will be measured in the near future. A sinusoidal input was used for each measurement and the signal to noise ratio defined to be:

$$SN = \frac{\sigma_d}{\sigma_{tot}} \quad (7)$$

Only the driven degree-of-freedom is analyzed. The data was sampled at 200Hz for 90 seconds. Power spectra were calculated as described in the previous section on ½ Hz noise level tests. Figure 9 shows the X signal-to-noise contours.

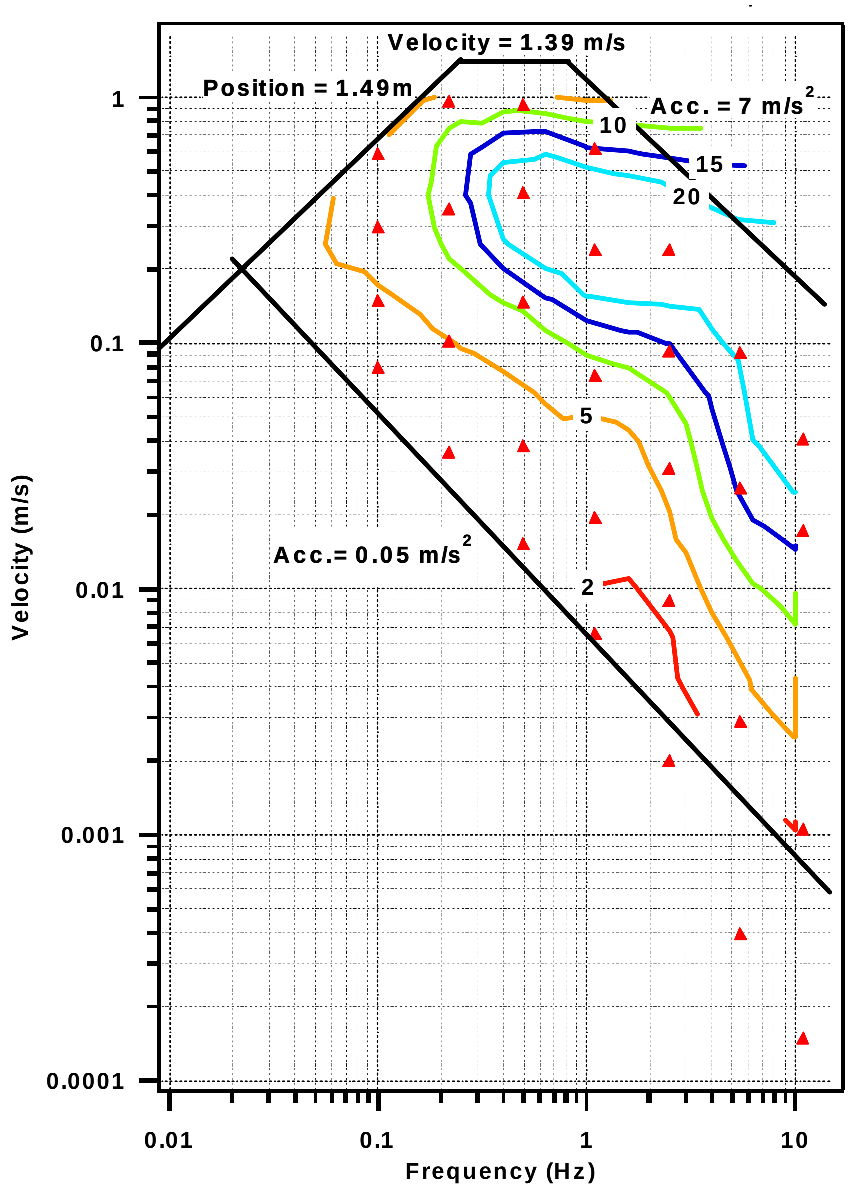

FIGURE 9 X-Signal to Noise Contours.

The contour lines were generated in two steps. First, the data was gridded onto a rectangular logarithmic grid using the MATLAB griddata routine (4). Linear triangulation was used to calculate the gridded data. The contour lines on the plot were then generated by using the IGOR-PRO contour plotting program, which employs linear triangulation. The operational envelope is bounded by the position limit, a maximum velocity limit, a maximum acceleration limit, and a lower acceleration threshold. The velocity limit of 1.39 m/s was determined from previously run step input tests. The acceleration limit of 7m/s^2 was based on structural concerns. The 0.05 m/s^2 acceleration limit corresponds to the motion detection capability of human (5). The area of highest signal to noise occurs in the center of the operating space of the motion system. Very small accelerations have poor signal to noise properties due to a fixed amount of background noise in the entire system. Inputs that generate high velocity lead to small signal to noise ratios. A potential cause for this is the current scavenge system, which cannot fully evacuate the oil from the backside of the

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actuator piston at sustained large velocities. This causes the actuator to bump against the stiff oil column resulting in an acceleration spike. This can been seen at 57.5s and 62s in the time history show in Figure 10.

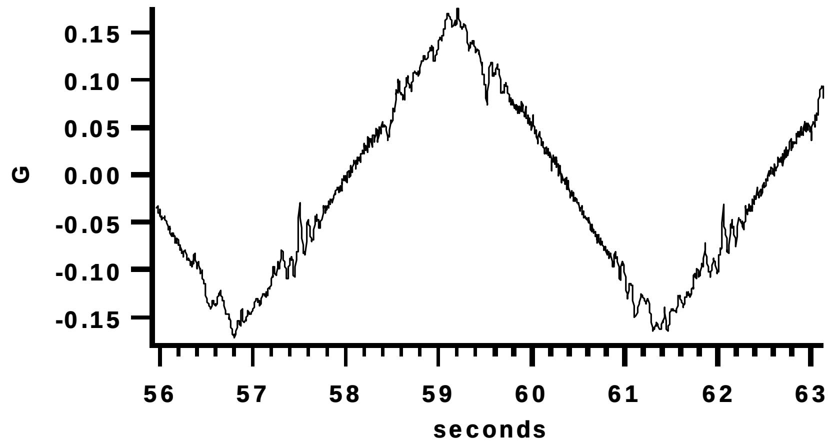

FIGURE 10 X-acceleration Response, 0.1Hz, 1.4 m/s^2

These bumps generally occur at or near the maximum velocity of the actuator, when the scavenge system needs to move the most oil. As was shown in the previous section high accelerations seem to lead to more harmonics of the driving signal that in turn leads to low signal-to-noise ratios.

Dynamic Threshold

The dynamics threshold was determined by driving the motion system with a 0.4 m/s^2 square wave in the X degree-of-freedom. The response and input are shown in Figure 11. The dynamic threshold is defined as the time required to reach 63% of the commanded input.

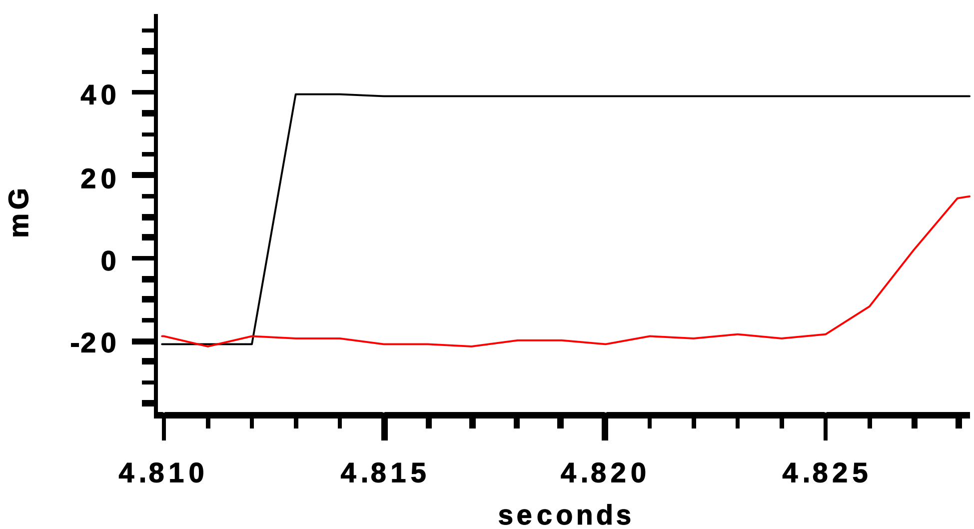

Figure 11 X-Dynamic Threshold Test.

The dead time is approximately 12ms and the dynamics threshold is approximately 15ms.

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Hystersis

The hysteresis was determined by driving the motion system with very low frequency sine waves and determining the maximum displacement error. Phase error in the system response will lead to displacement error but as shown by Grant (6) the error due to phase is roughly linear with frequency. By extrapolating the measured displacement error as a function of frequency to a frequency of zero, an estimate of the hysteresis can be made. The hysteresis was found to be negligible.

CONCLUSIONS

The VIRTTEX motion system has a bandwidth greater than 13Hz. With the current state of tuning, the system exhibits larger than expected cross talk, turn-around bump, and wide-band noise, particularly in the Z degree-of-freedom. The dynamic threshold is approximately 15 ms. The motion system has no significant hystersis.

The results of motion characterization suggest that the system should be re-tuned with a stronger emphasis on smoothness and cross talk. It is likely that significant improvements in the wide-band noise and cross talk could be achieved. Re-tuning is also likely to reduce the peak noise on the Z degree-of-freedom to levels close to that measured on the X degree-of-freedom. The turn-around bump on the X degree-of-freedom is unlikely to be significantly reduced by re-tuning. The bump is associated with either the flow characteristics of the servo-valve or friction in the actuator. Future work is planned to help identify the root cause of the turn-around bump and make the appropriate improvements. The scavenge system will be replaced by a larger capacity system in the near future, which should reduce some of the peaks in noise seen at the high actuator velocities.

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