Monterey Bay Aquarium Research Institute

PTO-Simulink Model Validation

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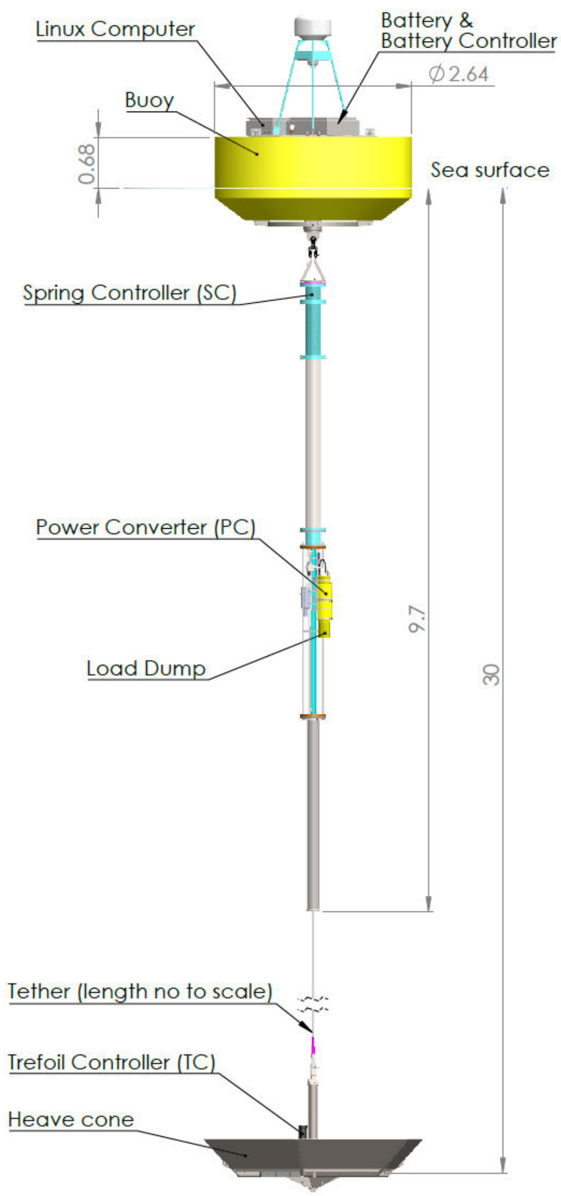
ABSTRACT

The development of the first of it's kind MBARI-WEC opened up the idea of powering automated underwater vehicles. This study investigates the validation of a Simulink, numerical model of the MBARI-WEC. During investigation a friction model was defined with the use of constant speed bench testing and linear regression models. A series of sinusoidal position based inputs were then used for bench testing and simulation to conduct a comparative analysis. The comparison validated portions of the Simulink model including the pneumatic and hydraulic systems and highlighted areas within the electrical model that required further development.

1 INTRODUCTION

In 2009, the Monterey Bay Aquarium Research Institute (MBARI), under the leadership of Andrew Hamilton, began research on a wave energy converter (WEC). The MBARI-WEC, as shown in Figure 1, is two-body point absorber. The two bodies are a surface expression buoy connected to a submerged heave-plate, by the electro-hydraulic power take-off [1]. Since the initial deployment in 2011, the MBARI-WEC has been modified based on data obtained from each deployment. The MBARI-WEC has the power generation capability of 300-400 W, when the wave states are average [2]. The long-term goal of this project is to provide a remote charging docking station for Automatic Underwater Vehicles (AUVs). For that goal to be achieved a more complex control system is being developed.

Figure 1:



MBARI-WEC diagram with all controllers labelled

This paper provides a numerical analysis of a Simulink MBARI-PTO model compared to bench test data of the physical model. First, the Simscape friction model will be discussed and how the coefficients were ascertained. Next, the MBARI-PTO will be compared to bench data using the root mean square error for a range of critical variables to validate the simulation results. The results from this analytically comparison, as well as further considerations will then be discussed.

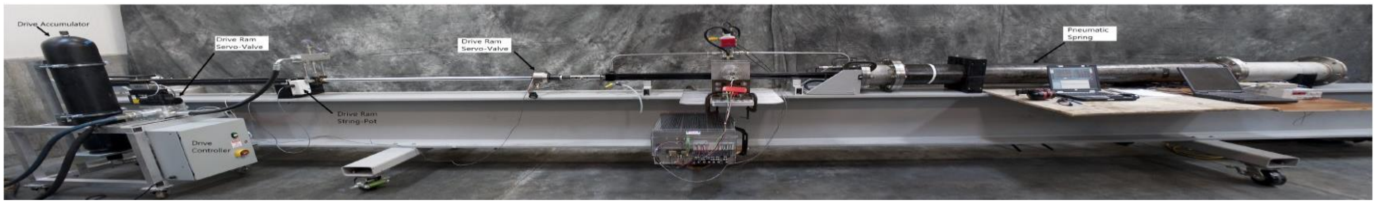
2 MATERIALS AND METHODS

The validation of the Simulink model developed for this project required an array of bench tests of the MBARI-WEC. Figure 2 shows how the bench test was set-up for the experiment. This section will lay out the details of all materials and devices used and the procedures taken to complete the validation.

2.1 MATERIALS

1. MBARI-WEC Tether as shown in Figure 2.
2. Test Machine Controller Delta RMC75E
    a. This is used to generate motion of the MBARI-WEC.
    b. Testing was done using both a position input and a velocity input
3. Linux Computer (Buoy Logger)
    a. This computer is used to log data from the different controllers on the MBARI-WEC.
    b. These controllers are shown in Figure 1.

Figure 2:



MBARI Laboratory PTO Bench Test Set-up.

4. Matlab
   a. Simulink/Simscape Model-
      The complete model developed using Simscape for different components of the MBARI-WEC and Simulink to connect each component. The MBARI-PTO model uses the *Foundation, Fluids* and *Electrical*, Simscape libraries.
   b. Matlab Scripts-
      All calculations and graphs were coded and produced using Matlab

2.2 METHODS

The first step in validating the Simulink model required determining the friction coefficient parameters for the pneumatic spring. After a universal friction block was created testing for the full model was conducted and results of the simulation were compared to that of the test data.

2.2.1 FRICTION BLOCK IDENTIFICATION

The MBARI-PTO model contains a Simscape rotational friction block component that requires coefficient input parameters. This was done by running the Delta RMC75E controller in velocity mode. The test machine extended and retracted the MBARI-WEC's piston rod at a range of constant velocities, at a single damping factor, as shown in Figure 2. The two data acquisition devices, both the buoy logger and the test machine logger, collected force and pressure measurements during this test. The test machine collected the total force on the MABRI-WEC, along with the upper and lower hydraulic pressure. Simultaneously, the buoy logger obtained the upper and lower pressure from the pneumatic piston. The areas of the of pistons, Table 1, are used to calculate hydraulic and pneumatics forces acting on the system for a given time during the test.

Table 2: The table provides the dimensions for the hydraulic and pneumatic spring pistons

Component	Area (cm^2)
Hydraulic Piston (HP)	8.87
Upper Pneumatic Piston (UPP)	126.7
Lower Pneumatic Piston (LPP)	115.2

The recorded pressures from the bench test and the piston areas were then used to calculate the pneumatic force (F_p) with the equation,

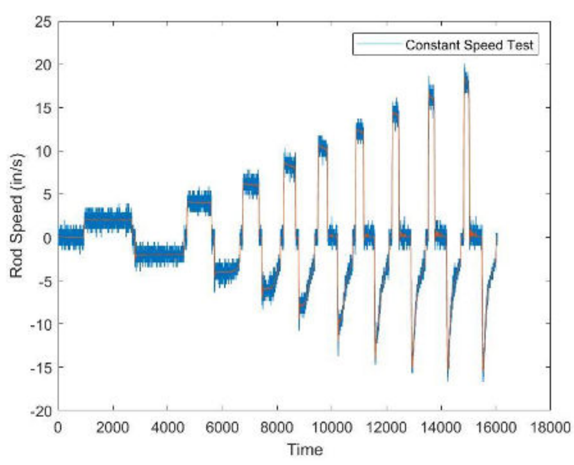
$$F_p = A_{Upp^*}SC_{upp} - A_{Lpp^*}SC_{Lpp} \qquad (1)$$

where A_{Upp} and A_{Lpp} are the upper and lower piston areas and SC_{upp} and SC_{Lpp} are the pneumatic (spring) controller pressure, upper and lower. The same method was used to calculate the hydraulic for (F_h) with the equation,

$$F_h = A_{HP}(TM_{uhp} - TM_{Lhp}) \qquad (2)$$

where A_{HP} is the area of the hydraulic piston and TM_{uhp} and TM_{Lhp} are the upper and lower hydraulic pressures. As previously stated, the Delta RMC75E controller also tracks and records the total test machine force (F_{TM}) that is being applied during bench testing.

Figure 3



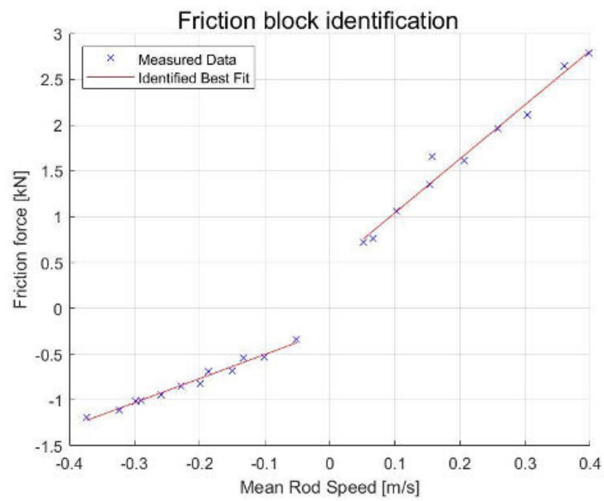
MBARI-WEC constant speed bench test 1

Bench test 1 was conducted at a variety of velocities, starting at ± 0.50 in/s, then ± 1.00 in/s, then from ± 2.00 in/s the increments changed to ± 2.00 in/s up until ± 24.0 in/s. The friction force (F_f) at each constant velocity was then calculated with the following equation,

$$F_f = -F_f - F_p - F_h \qquad (3)$$

By calculating friction force over a variety of constant speeds, the force associated with acceleration did not need to be factored into the total force. The blue portion in Figure 3 shows the sections where the piston rod velocity was constant. A Matlab script was written to separate each section of constant speed and the mean values of Pneumatic, Hydraulic, and total force was used for each section to determine the mean friction during that period. After the friction force was determined for a range of velocities, a best fit regression model was created to define the force from friction, as seen in Figure 4.

Figure 4:

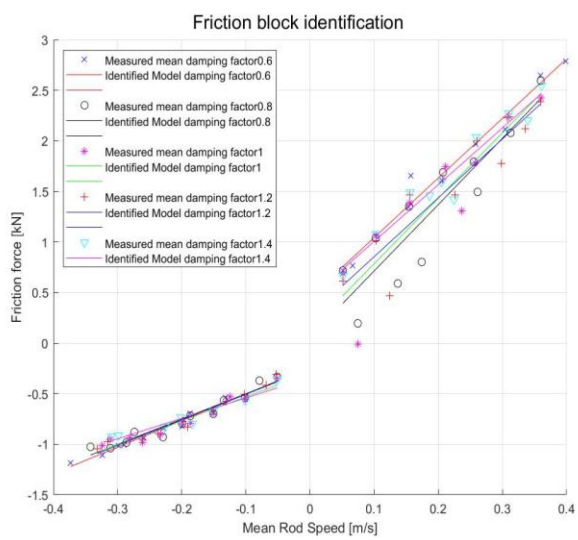


Bench test 1 mean friction plotted with best fit linear regression line.

2.2.2 UNIVERSAL FRICTION BLOCK

Initial friction block identification was conducted on a test, per test basis. Anytime a new bench test was performed a constant speed test was run to establish the friction block. The following methods were used to determine the sensitivity of friction block. The previous method of running a constant speed as the input function for the Delta RMC75E controller was followed, however, to develop a more complete look at the friction in the system the test was administer at five different damping scale factors. The five different constant speed tests were then used to find the friction force by the same method as described above. The five friction force linear regression models, as shown in Figure 5, were then averaged together to create a universal parameter for the friction block.

Figure 5:

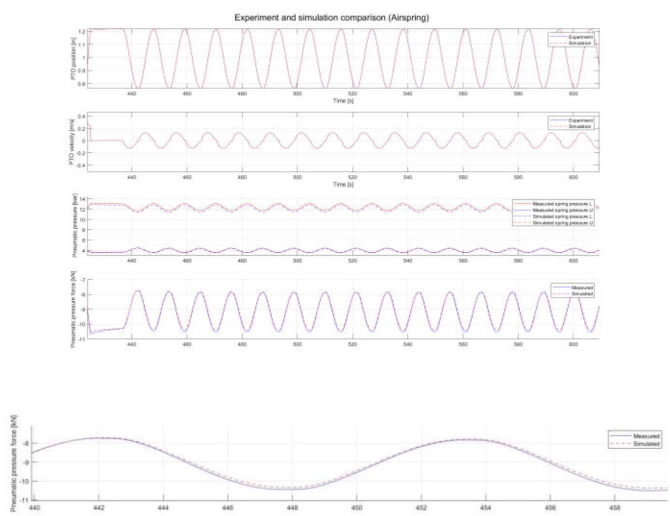


Bench test 2 variable damping scale factors and their corresponding mean friction plotted with best fit linear regression line.

2.2.3 FRICTION BLOCK COMPARISON

The defining of the friction block parameters allowed for the comparison of measured bench test data with that of results for the MBARI-PTO Simulink model. The first step was to determine that whether the universal friction parameters would impact the overall simulation. A basic cosine position function with nine frequencies was ran after the initial constant speed test and bench test 3 was used to run simulations for both friction block parameters and comparing the results.

Figure 6:



Top: Plot of simulated and measure data for position, velocity, pneumatic pressure, and pneumatic force.
Bottom: Zoomed in plot of pneumatic force.

The two simulations were evaluated with the data from a position input bench test. Several variables were analyzed from the bench test: pneumatic force, hydraulic force, friction force, and the upper and lower pressures of both the pneumatic spring and hydraulic system. The root mean squared error (RMSE) was used to determine the difference between the measured (bench test) and the simulated values using the following equation,

$$RMSE = \sqrt{\frac{(Measured - Simulated)^2}{N}} \qquad (4)$$

where N is the number of bench test samples collected. The RMSE was calculated for the variables listed above using both friction blocks. A simple error calculation using the following equation,

$$\%\, Error = \frac{RMSE_m - RMSE_I}{RMSE_m} * 100\% \qquad (5)$$

where $RMSE_m$ and $RMSE_I$ are the mean and individual RMSE values.

2.2.3 POSITION BASED INPUT FUCNTION

The next phase in testing was using the position based input function of the Delta RMC75E controller. This was conducted by using a sine function at a variety of frequencies and amplitudes, the damping scale factor was also changed. Table 2 shows the different test parameters for bench test 4. The higher amplitudes only tested the three lower frequencies. The velocity of the piston rod was then derived from the test machine sinusoidal position functions and used as the input into the MBARI-PTO Simulink model. After the position based sinusoid simulation was conducted the data was analyzed using Equation 4 for the previously listed variable as well variables related to the electrical: PTO RPM, Bus Current, Bus Voltage and Electrical Power.

Table 2: Parameters for bench test 4: 3 different damping factors, 2 different amplitudes and five different frequencies.

DAMPING FACTOR	FREQUENCIES (Hz)					AMPLITUDE (in)
	0.05	0.1	0.15	0.2	0.25	
0.6	X	X	X	X	X	12
	X	X	X			24
1	X	X	X	X	X	12
	X	X	X			24
1.4	X	X	X	X	X	12
	X	X	X			24

3 RESULTS

The first analysis that was conducted centered around the comparison of the RMSE values from the individual friction model and that of the universal friction model. After the friction model parameters were defined, we used the position based bench test to generate a velocity input for the Simulink Model. The simulation results and the bench test data were then analyzed.

3.1 FRICTION BLOCK

The data collected from running simulations with the individual and mean friction block parameters was evaluated using Equation (4) and Equation (5) the two simulation results were assessed. The results in Table 3 are on the RMSE for the entire simulation. A comparison broken up for each individual frequency test and graph of bench test 3 can be found in Appendix 1 and 2. The results shown in Table 3 indicated that there is no difference between the two simulations. Therefore, for the remainder of test the mean friction block parameters were used as the input for the MBARI-PTO Simulink model.

Table 3: RMSE comparison for individual and mean friction block parameters and the difference between the two values

	Pneumatic Force [kN]	Lower Gas Pressure [bar]	Upper Gas Pressure [bar]
Individual Parameters	1.17E-01	1.78E-15	0.00E+00
Mean Parameters	1.17E-01	1.78E-15	0.00E+00
Difference	0.00	0.00	0.00

3.2 BENCH TEST/SIMULATION

The final analysis performed to determine whether the MBARI-PTO Simulink model could be validated following the methods previously described in Section 2.2.3. The simulation failed at 600s after the fifth frequency of the first damping factor. This prevented a complete analysis of the fourth bench test. An analysis was carried out on the portion of the simulation that ran before the error. The results for this analysis can be seen in Table 4, 5 and 6. The three tables are broken up into different components of the MBARI-PTO, investigating the RMSE of the pneumatic, hydraulic and aspects of the electrical system.

Table 4: Pneumatic properties: force, upper and lower gas pressures. The table is of the RMSE for each frequency compared to the maximum value of each variable

	Pneumatic Force [kN]		Upper Gas Pressure [bar]		Lower Gas Pressure [bar]	
Frequency	RMSE	Max	RMSE	Max	RMSE	Max
0.05	0.03	7.00	0.07	4.70	0.21	13.00
0.1	0.03	7.00	0.08	4.70	0.22	13.00
0.15	0.04	7.00	0.08	4.70	0.22	13.00
0.2	0.05	7.00	0.08	4.70	0.23	13.00
0.25	0.06	7.00	0.08	4.70	0.24	13.00

Table 5: Hydraulic properties: force, upper and lower gas pressures. The table is of the RMSE for each frequency compared to the maximum value of each variable

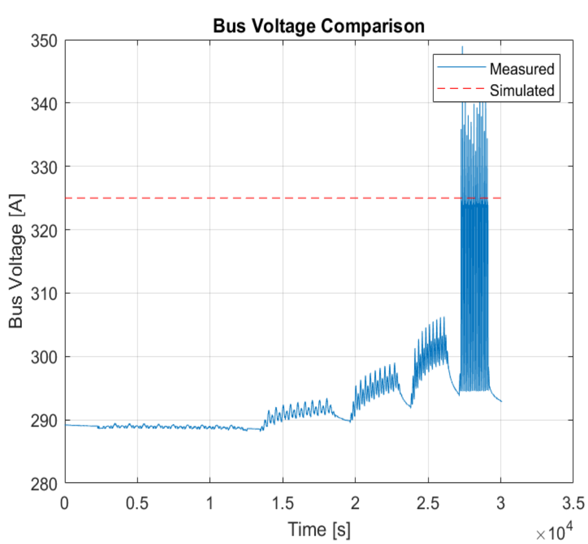
	Hydraulic Force [kN]		Lower Piston Pressure [bar]		Upper Piston Pressure [bar]	
Frequency	RMSE	Max	RMSE	Max	RMSE	Max
0.05	0.01	2.00	1.33	20.00	1.47	25.00
0.1	0.01	3.60	3.28	60.00	3.16	40.00
0.15	0.02	5.00	5.63	80.00	5.36	56.00
0.2	0.04	6.50	10.35	100.0	9.93	75.00
0.25	0.05	8.00	19.76	135.0	19.25	90.00

It should be noted at the time of this writing a thorough analysis was not conducted on the bus voltage. It was known at the time that the simulation voltage was not modeled with the same accuracy as other components. As shown in Figure 7, the simulation voltage was a constant of 325 V.

Table 6: Electrical properties; PTO RPM, bus current, electrical power. The table is of the RMSE for each frequency compared to the maximum value of each variable

	PTO RPM [rpm]		Bus Current [A]		Electric Power [W]	
Frequency	RMSE	Max	RMSE	Max	RMSE	Max
0.05	1.29	950	0.21	0.50	63.02	150
0.1	4.55	2,050	0.32	2.50	77.88	650
0.15	8.73	3,100	0.44	4.50	97.95	1300
0.2	11.68	4,300	0.57	7.00	122.17	2250
0.25	16.39	5,050	0.56	12.00	155.43	4050

Figure 7:



Comparison of Bus Voltage with simulation being modelled as a constant 325V.

A more accurate voltage model and other important results will be examined in the section 4.

4 DISCUSSION

A significant component of the MBARI-PTO Simulink model is the friction block. The bench testing conducted to helped define the parameters for this Simscape block allowed for an overall more accurate model. As shown in Figure 4, the friction force is not symmetrical about a mean rod speed of 0 m/s. Research into possible reasons for this asymmetry pointed to the type of seal in the pneumatic cylinder as a potential cause. According to a study on friction in pneumatic seals, by Azzi et al, "the seal geometry has a significant effect on friction [3]." The use of a lip seal in the pneumatic cylinder is likely the root of the asymmetry. This asymmetry showed in all the damping factor bench tests. The comparison of the individual and mean friction block parameters indicated that there is no differences in the two parameters. This is shown in Table 3, after using Equation (5) to compare the two simulations and allowed for the use of the mean parameters to define the model. While the constant speed mean friction parameters produce consistent results for the position based sinusoidal testing, further validation should be conducted for irregular functions. This would better reflect actual wave conditions

The results from the MBARI-PTO simulation, predominantly validates the Simulink model. As seen in the RMSE comparison for the pneumatics and hydraulic components (Tables 4 and 5), there is negligible difference between the simulation and the bench test data. However, as mentioned in Section 3.2 the simulation failed at 600 seconds, immediately following the fifth and highest frequency for the first set of damping factors. This error is most likely associated with the mass inertia of the piston rod in the MBARI-PTO Simulink model. When the simulation is run at that high of frequency, changing from high velocity to zero velocity in between tests creates an error in the model. This will need to be adjusted before complete validation can occur. The simulation hydraulic pressures as the frequency increased begin to deviate from the bench test data. This could suggest that in the real hydraulic circuit there are extra loads that haven't been considered in the model, such as pipe restrictions.

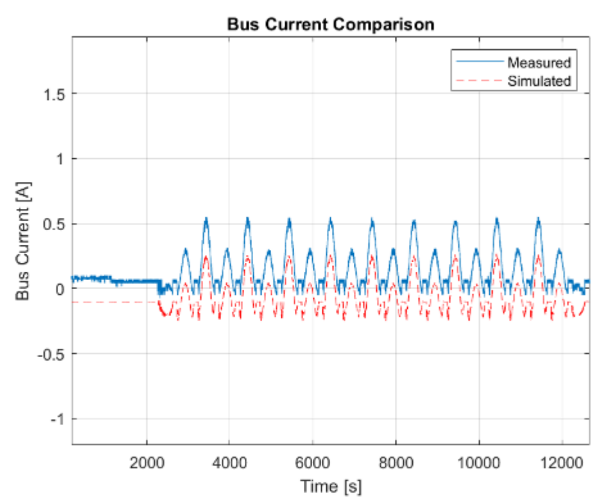
The RMSE values for the electrical components highlighted a discrepancy in the bus current. While the RMSE values were within a reasonable range but there was a noticeable scalar error in the simulation, shown in Figure 8. A closer look at the current parameters in the Simulink model will need to be investigated to determine the cause for this discrepancy.

5 CONCLUSIONS

The development of the MBARI-WEC is the first of its kind wave energy converter. Since the initial deployment in 2011, the WEC has undergone a series of iterations to help improve its performance. This study was completed to help validate the MBARI-PTO Simulink model in an ongoing attempt to design a more robust control system. The establishment of a friction block that produced accurate results for multiple test was an important step in completing the Simulink model. This allows for further simulations to be completed.

Overall, the Simulink model matched the test bench data, with a few areas of deviation. The RMSE for the different pneumatic variables was well within a margin or error and can be considered validated from this study. While the hydraulic force also had a reasonable margin of error for higher frequencies the different hydraulic pressures began to present larger errors. At this time further exploration can be made into the hydraulic circuit to find the cause of this error.

Figure 8:



Bench test vs Simulation comparison for bus current on first frequency from bench test.

The addition of a more accurate battery is another section that needs altered to validate the model completely. The constant voltage input while a reasonable assumption doesn't fully capture the true battery. Considering the assumptions that have been made and the complexity of all the different systems within the MBARI-PTO model, generally the model is an adequate representation of the fully MBARI-WEC.

Further simulation should be run with a force based input function to better represent the ocean conditions and act as further validation of the model.

6 ACKNOWLEDGEMENTS

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8 APPENDIX

Appendix 1- Individual vs mean friction block comparison for pneumatic force, lower and upper gas pressures**.**

	Pneumatic Force			Lower Gas Pressure			Upper Gas Pressure		
Input Test	Specific	Mean	Difference	Specific	Mean	Difference	Specific	Mean	Difference
1	0.011	0.011	-2.61E-09	0.037	0.037	-8.55E-09	0.015	0.015	4.47E-09
2	0.011	0.011	-6.95E-09	0.043	0.043	-7.44E-09	0.017	0.017	3.87E-09
3	0.011	0.011	1.02E-08	0.043	0.043	8.31E-09	0.016	0.016	-4.40E-09
4	0.017	0.017	-6.52E-08	0.048	0.048	-6.00E-08	0.018	0.018	3.16E-08
5	0.018	0.018	-1.46E-07	0.052	0.052	-9.79E-08	0.019	0.019	5.35E-08
6	0.017	0.017	-2.84E-07	0.054	0.054	-1.69E-07	0.019	0.019	9.76E-08
7	0.016	0.016	-2.60E-07	0.052	0.052	-1.55E-07	0.019	0.019	8.68E-08
8	0.025	0.025	-3.02E-07	0.054	0.054	-2.13E-07	0.016	0.016	1.11E-07
9	0.023	0.023	-2.76E-07	0.054	0.054	-2.00E-07	0.019	0.019	1.23E-07

Appendix 2- Pneumatic pressure vs Time and Pneumatic Force vs Time, Bench Test 2 for friction block comparison