A HYDRAULIC DAMPING APPROACH TO VARIABLE STROKE TREE SHAKER DESIGN

by

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INTRODUCTION

The labor shortage in the fruit and vegetable growing industry has intensified in recent years. Consequently, interest in the development of a satisfactory means for mechanically harvesting these crops has increased. Insufficient labor has affected the timeliness of harvest and a corresponding concern about possible quality deterioration of the market product. Mechanized harvest could greatly reduce these losses.

The harvesting problem has been particularly evident with the apple. Because of differing tree sizes and random locations of the fruit on the tree, it has been difficult to devise economical and efficient equipment to remove the fruit.

Several mechanical systems have been developed to handle the apple. All of these systems tend to bruise the fruit more than hand harvesting. Bruising, which is particularly critical in the fresh fruit market, is caused by relatively high forces incurred by the fruit during handling. This is the primary reason that completely successful mechanical harvesters have not been developed.

REVIEW OF LITERATURE

One practical approach to completely mechanized fruit removal is to shake the trunk or limb of the tree. The objective is to accelerate the fruit so that the inertia force developed will be greater than the bonding strength between the fruit and the tree. The published data available for optimum stroke length and shaking frequency is limited. No literature could be found for apples.

Adrian and Fridley (1)* have tested two types of shakers that produce motion in a tree limb. The first was a counter-rotating unbalanced mass assembly, while the second was a slider-crank arrangement. In the second system the slider is attached to the tree so that the reciprocation of the crank housing supplies the excitation force.

To mathematically describe a vibrating tree limb is difficult due to the many degrees of freedom of the limb. However, Adrian and Fridley have shown that reducing the limb to a single degree of freedom system is an acceptable procedure.

Adrian and Fridley (2) have stated that a vibrating cantilever beam is in many ways similar to that of a primary fruit tree limb. Other investigators (3, 4) have made similar assumptions.

In work conducted by Hamann (5), apple detachment appeared to be more dependent on length of stroke rather than on shaking frequency. This work showed that a high rate of detachment occurred at the beginning

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* Numbers in parentheses refer to appended references.

of the shaking operation. It appeared that some of the fruit bruising was due to many apples falling together.

Based on this apparent stroke dependence, a controlled rate of apple detachment from the tree might be attained with the use of a variable stroke shaking mechanism, where the stroke is gradually increased during shaking. There is no available data to support or deny this statement.

OBJECTIVES

To reduce bruising during free fall, a controlled rate of fruit detachment from the tree might be desirable. A variable stroke shaking mechanism appeared to be a reasonable method by which this can be achieved.

The objectives of this investigation were as follows:

1. To design and develop a variable stroke tree shaking mechanism based on the principle of variable hydraulic damping.
2. To perform laboratory tests for evaluation of the design.
3. To develop theoretical equations to describe the motion of the shaking mechanism.

DESIGN OF THE VARIABLE STROKE SHAKER

The Constant Stroke Machine

A Floating Boom Shaker, manufactured commercially by the Wagco Company of Meridian, California, served as the starting point for the design of the variable stroke shaker. This machine, shown in Figure 1, supplied a constant amplitude stroke for a given set of system parameters. The Wagco shaker consisted basically of two components, the supporting frame and the boom. The frame incorporated a standard three point tractor hitch, making the machine portable with the tractor. The boom was the vibrating portion of the machine and was supported at its center of gravity on the frame in a pendulum arrangement.

A hydraulically actuated, padded jaw clamp, located on one end of the boom, attached the boom to the tree. The vibrating source was connected to the other end of the boom. This source was similar to a piston-crankshaft arrangement, with the boom being analogous to the piston. The crankshaft was powered by a hydraulic motor and enclosed in a cast housing. A connecting rod attached the boom to the crankshaft. During operation the boom and housing were displaced in opposite directions.

The Variable Stroke Machine

To attain the variable stroke feature, a hydraulic cylinder was introduced into the boom to act as a dashpot. The cylinder barrel was attached to the crankshaft end of the boom by a polished slide, similar to the Wagco boom. The piston rod of the cylinder served as the boom,

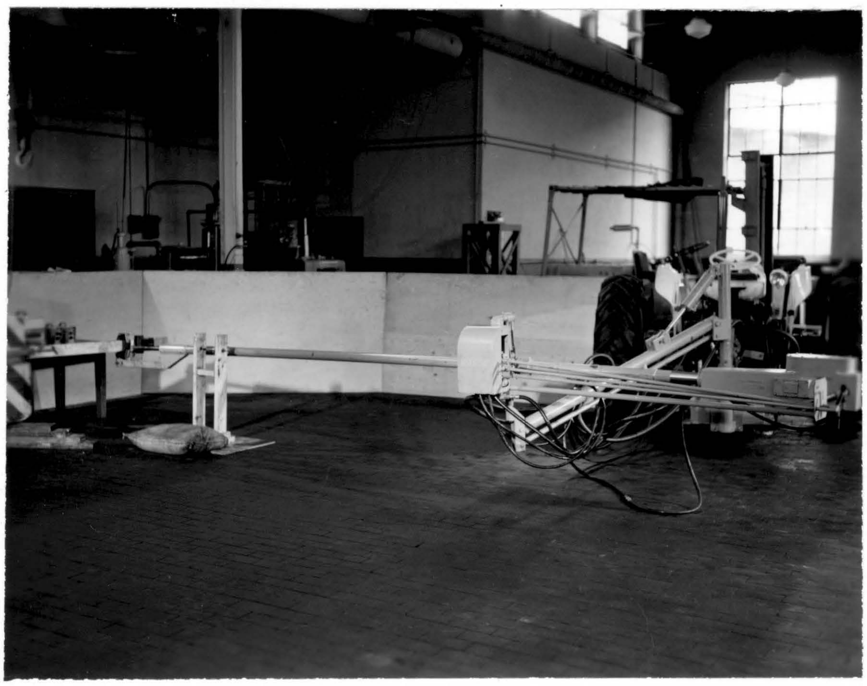


Figure 1. Wagco Shaker Mounted on Tractor in Position Tested.

extending to the clamp.

The cylinder was designed to function within the physical limitations of the Wagco machine. A sketch of the cylinder is shown in Figure 2 with a view of actual parts shown in Figure 3. The cylinder, assembled and mounted on the machine is shown in Figure 4.

Mild steel tubing was used to make the cylinder barrel and piston rod. The piston head and cylinder end caps were machined from gray cast iron to provide good wear and sealing characteristics with steel. O-rings were used for static and dynamic oil seals. Leather back up washers were used in addition to the O-ring for the dynamic seals. Machining tolerances and surface finishes were maintained within commercially recommended limits for O-ring use.

Fluid on both sides of the piston head was connected to an external valve. As the cylinder barrel was displaced by the rotating crankshaft, fluid tended to flow from one side of the piston head to the other because of the inertia of the piston rod. The external valve metered this flow. Varying displacements of the piston rod resulted, depending on the degree of restriction of the valve.

The valve was designed to be symmetric with respect to the direction of fluid flow. This caused an equal pressure drop across the valve for both directions of flow. The symmetric characteristic was felt to be mandatory to prevent the cylinder from extending or retracting unevenly during operation.

The valve consisted of a manually rotated spool and valve body. Each component had a fluid passage 1-3/8 inch in diameter. The variable

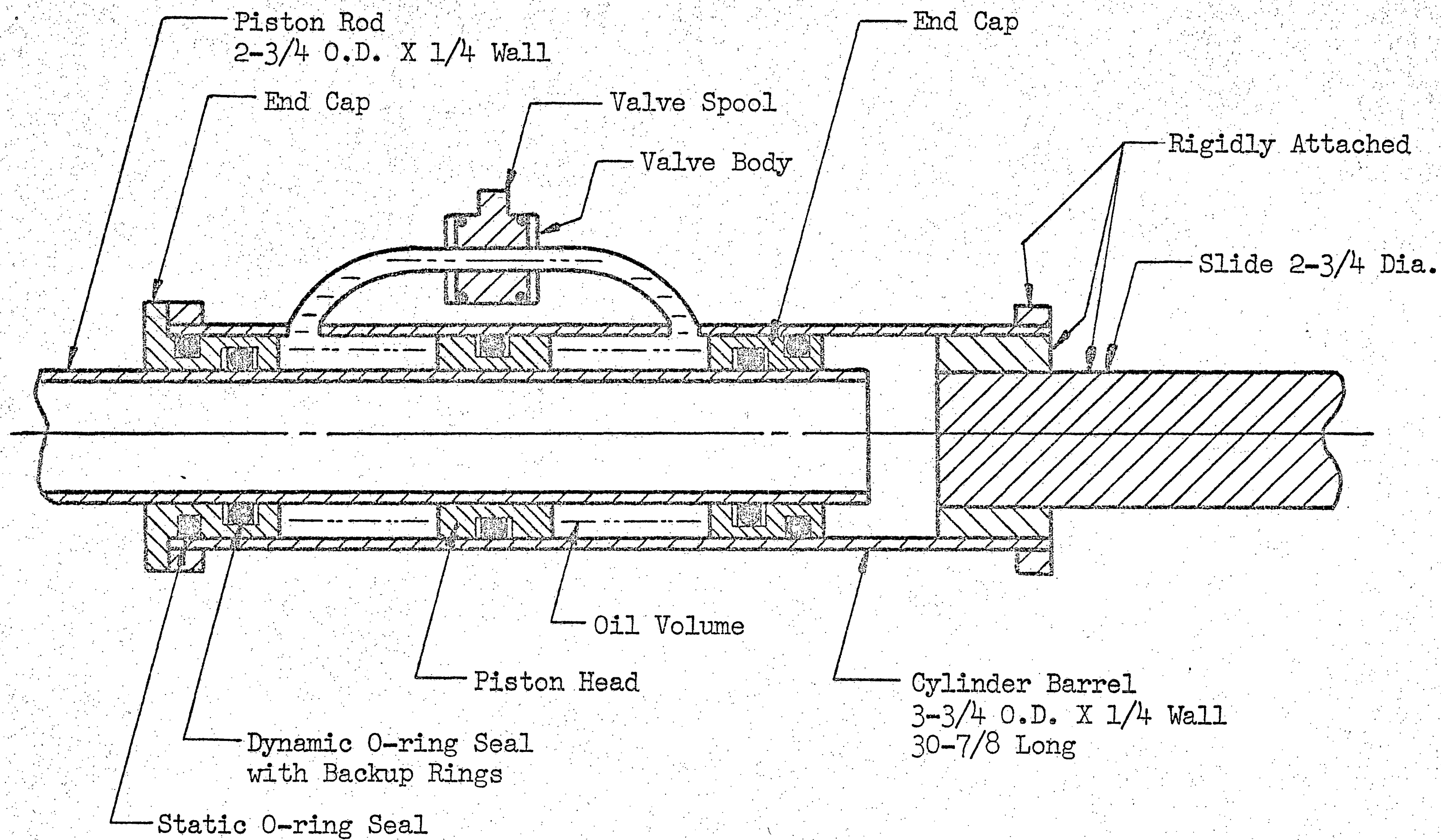


Figure 2. Sketch of Cylinder Used to Attain the Variable Stroke Feature.

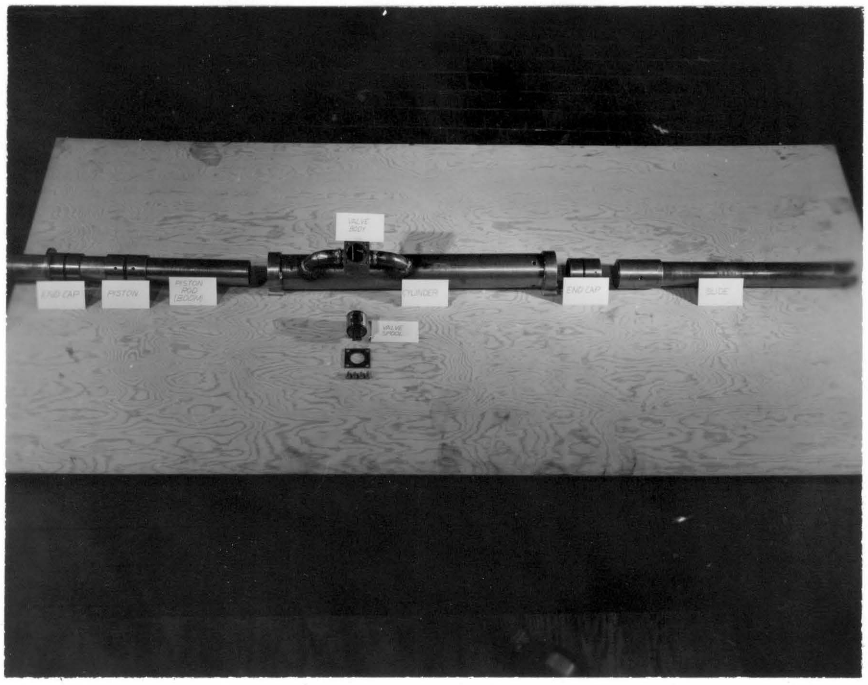


Figure 3. Damping Cylinder and Valve Parts in Disassembled Condition.

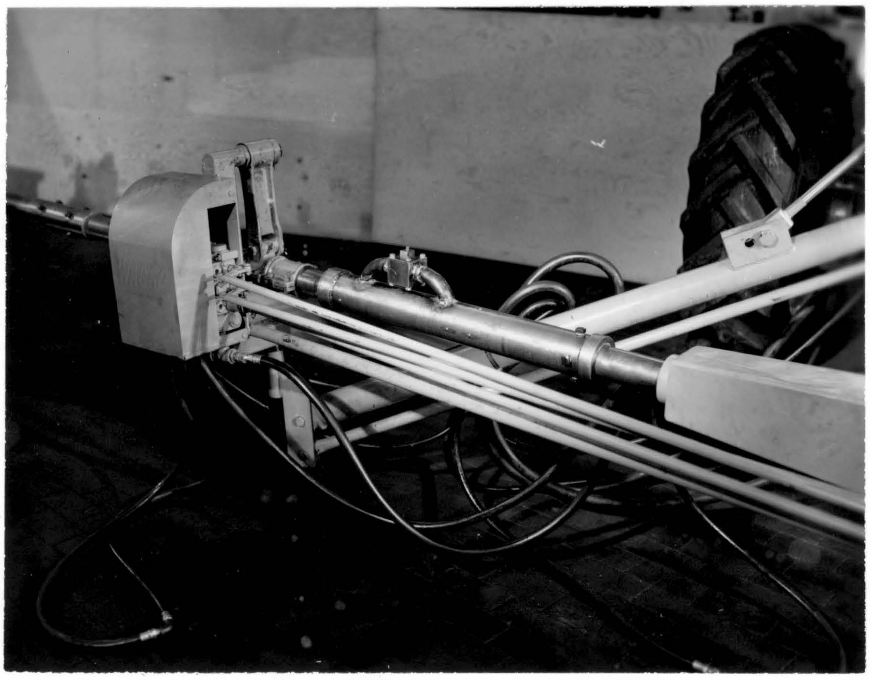


Figure 4. Damping Cylinder Mounted on the Machine.

area of the valve was controlled by the angular position of the spool passage with respect to the valve body passage.

The fluid used in the cylinder was Automatic Transmission Fluid, Type A. This fluid provided adequate lubrication, relatively low viscosity, and compatibility with the seals.

TEST PROCEDURE AND RESULTS

Displacement and acceleration of a simulated tree limb vibrated by the Wagco and variable stroke shakers were measured. Various limb spring constants, effective limb masses, and shaking frequencies were used to test these machines.

Test Apparatus

A cantilever 2 x 4 wooden beam was used to simulate a tree limb. Spring constants were varied by changing the distance from the boom clamping point to the fixed end of the beam. Steel blocks were clamped to the beam as close as possible to the boom clamp to simulate the effective mass of the tree limb.

A three point hitch tractor was used to mount and power the shaking mechanism. The shaking frequency was controlled by the tractor engine speed.

Acceleration and peak to peak displacement of the beam at the clamping point were the two quantities measured in testing. The acceleration was monitored by an Endevco Model 2219E accelerometer mounted to the boom directly under the clamp. The accelerometer output was amplified by a Kistler Model 504 charge amplifier and a Kistler Model 567 power amplifier. The resulting signal was recorded on a Minneapolis-Honeywell Visicorder, Model 1406. Peak to peak displacement of the beam was measured by a stationary stand and a steel arm rigidly attached to the boom. The stand consisted of two vertical wooden members mounted to a stationary base. A steel insert was pressed into each vertical member with a bolt threaded into each insert. The axes of the bolts were parallel to the base of the stand. Distance between the two bolt heads was varied by rotating the bolts. The stand was positioned so that the arm on the boom moved between the bolt heads. As the arm reciprocated with the boom, the bolts were rotated until they both reached the extremities of the arm movement. Peak to peak displacement of the beam was determined by measuring the distance between the bolt heads of the stand and from this value, subtracting the width of the arm.

A view of the test apparatus is shown in Figure 5.

The Constant Stroke Machine

To establish a base from which to evaluate the variable stroke shaker, the Wagco machine was tested. Three tree limb spring constants, two effective limb masses and five to six shaking frequencies were used to test this machine.

Results of the tests for the Wagco shaker are shown in Table 1. Typical acceleration recordings are shown in Figures 6 and 7. These tests established that the displacement of the beam was not greatly affected by the beam spring constant. Therefore, only one spring constant was used in testing the variable stroke machine. However, a change in the effective mass of the beam did produce an appreciable change in the displacement of the beam. Therefore, two mass levels were used in testing the variable stroke machine.

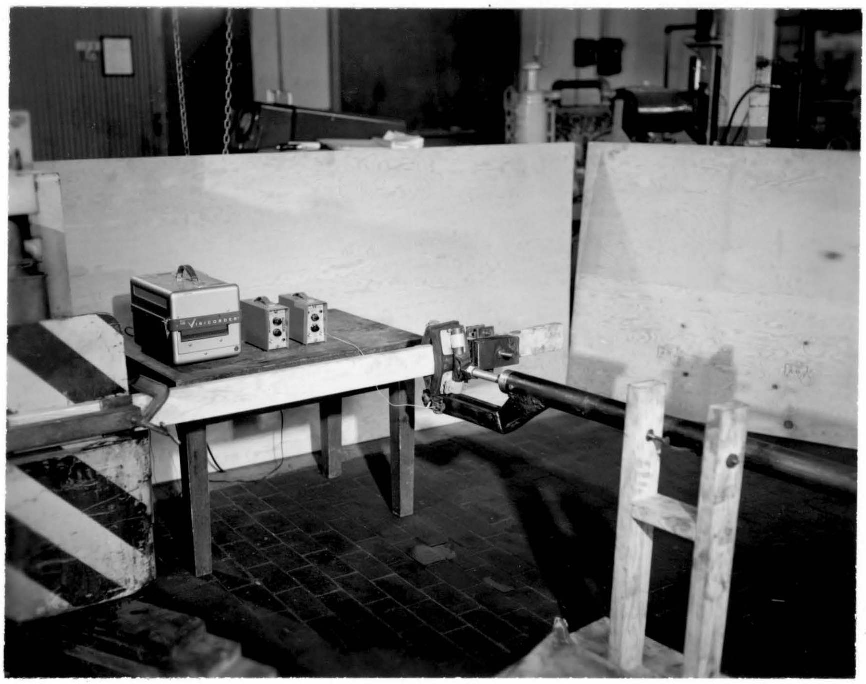


Figure 5. Overall View of Test Apparatus Showing Acceleration Readout Instruments, Padded Jaw Clamp, and Boom with Displacement Measuring Device.

Table 1. Displacement and Acceleration of Beam for Wagco Shaker

Shaking Frequency (cpm)	Beam Spring Constant (lb./in.)	Peak to Peak Displacement (in.)	Zero to Peak Acceleration (in./sec.^{2})
Attached Beam Weight - None			
325	38	1.75	2470
408	38	1.62	4150
457	38	1.62	5160
498	38	1.62	6260
535	38	*	*
600	38	1.56	9160
325	52	1.75	2530
408	52	1.69	3920
457	52	1.69	4830
498	52	1.69	6020
535	52	1.62	7240
600	52	1.62	8560
325	68	1.88	2160
408	68	1.69	4090
457	68	1.69	4950
498	68	1.62	5960
535	68	1.62	7240
600	68	1.62	8450
Attached Beam Weight - 35 lb.			
325	38	1.50	2040
408	38	1.37	3740
457	38	1.37	4460
498	38	1.31	5300
535	38	*	*
600	38	1.43	9650
325	52	1.50	2220
408	52	1.40	3860
457	52	1.37	4600
498	52	1.31	5550
535	52	1.25	6020
600	52	*	*
325	68	1.50	1870
408	68	1.40	3760
457	68	1.37	4460
498	68	1.28	5300
535	68	1.25	6260
600	68	*	*

* Not Recorded

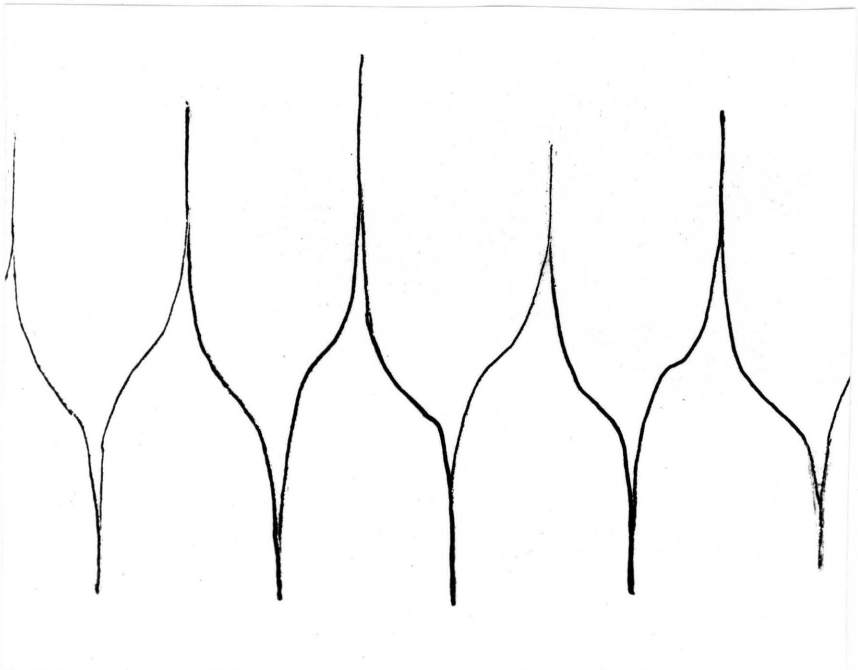


Figure 6. Acceleration Recording of Beam with Wagco Shaker for Shaking Frequency of 325 cpm, Beam Spring Constant of 68 lb./in. and No Attached Beam Weight.

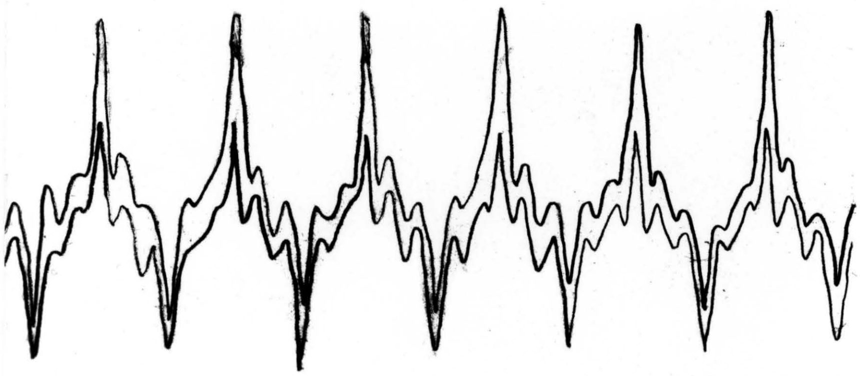


Figure 7. Acceleration Recording of Beam with Wagco Shaker for Shaking Frequency of 408 cpm, Beam Spring Constant of 38 lb./in. and 35 lb. Attached Beam Weight.

The Variable Stroke Machine

For each shaking frequency and mass level, eight angular valve spool settings were used in testing the variable stroke shaker. It was thought that the data collected from these tests would be adequate to analyze the dynamic behavior of the metering valve and cylinder assembly. Results of these tests are shown in Tables 2 and 3. The valve was fully open at the zero degree setting and fully closed at approximately 65 degrees. Three typical acceleration recordings are shown in Figures 8, 9 and 10.

Tables 2 and 3 show that as the orifice area is decreased, the displacement and acceleration of the limb is increased. A significant increase in these quantities occurs between the 50 degree to 70 degree valve setting range.

All desired data was not obtained for the Wagco shaker. This was due to erratic response of the system under certain conditions. It is believed that a resonance condition caused this problem. Erratic behavior was also noted for the variable stroke shaker. This behavior was more prominent at the higher valve settings and shaking frequencies. Larger than predicted beam displacements resulted, accompanied by excessive noise in the system. As a precautionary measure, some test conditions were omitted to minimize the possibility of a mechanical failure to some component of the machine.

Table 2. Displacement and Acceleration of Beam for Variable Stroke Shaker Without Attached Beam Weight

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Peak to Peak Displacement (in.)	Zero to Peak Acceleration (in./sec.^{2})
325	0	.31	353
	20	.31	340
	40	.34	409
	50	.43	409
	60	.97	625
	70	1.75	2410
	80	2.25 **	3140
	90	2.25	3620
408	0	.25	725
	20	.25	748
	40	.25	725
	50	.31	842
	60	.94	1060
	70	1.43	3380
	80	*	*
	90	*	*
457	0	.16	965
	20	.12	965
	40	.12	965
	50	.18	987
	60	.69	1060
	70	1.62	4100
	80	1.75	5320
	90	*	*
498	0	.12	987
	20	.12	1010
	40	.15	1010
	50	.15	1010
	60	.44	2040
	70	1.38	9400
	80	2.69 **	*
	90	*	*
535	0	.16	965
	20	.12	965
	40	.12	965
	50	.12	965
	60	.18	965
	70	2.75	*
	80	*	*
	90	*	*

* Not Recorded

** Erratic Behavior

Table 3. Displacement and Acceleration of Beam for Variable Stroke Shaker with 35 lb. Attached Beam Weight

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Peak to Peak Displacement (in.)	Zero to Peak Acceleration (in./sec.^{2})
325	0	.28	409
	20	.28	420
	40	.31	398
	50	.38	409
	60	.56	409
	70	1.62	4830
	80	1.56	3860
	90	1.68	3860
408	0	.12	649
	20	.12	772
	40	.18	772
	50	.22	772
	60	.62	772
	70	1.37	6750
	80	1.37	10900
	90	*	*
457	0	.10	795
	20	.10	772
	40	.12	868
	50	.18	868
	60	.47	868
	70	1.38	5790
	80	1.38	18800
	90	*	*
498	0	.12	1160
	20	.12	1160
	40	.12	1160
	50	.12	1160
	60	.18	1160
	70	1.38	17300
	80	*	*
	90	*	*
535	0	.06	963
	20	.06	963
	40	.10	963
	50	.10	963
	60	.12	963
	70	*	*
	80	*	*
	90	*	*

* Not Recorded

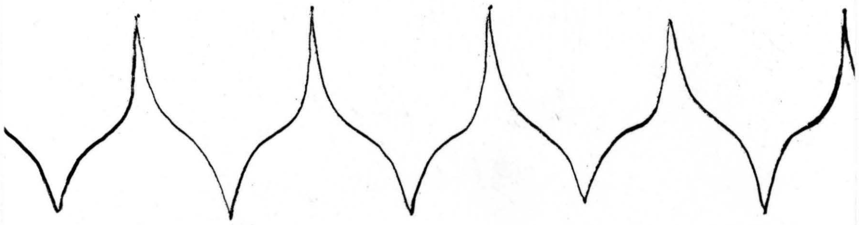


Figure 8. Acceleration Recording of Beam with Variable Stroke Shaker for Shaking Frequency of 325 cpm, Beam Spring Constant of 68 lb./in., No Attached Beam Weight and 20° Orifice Setting.

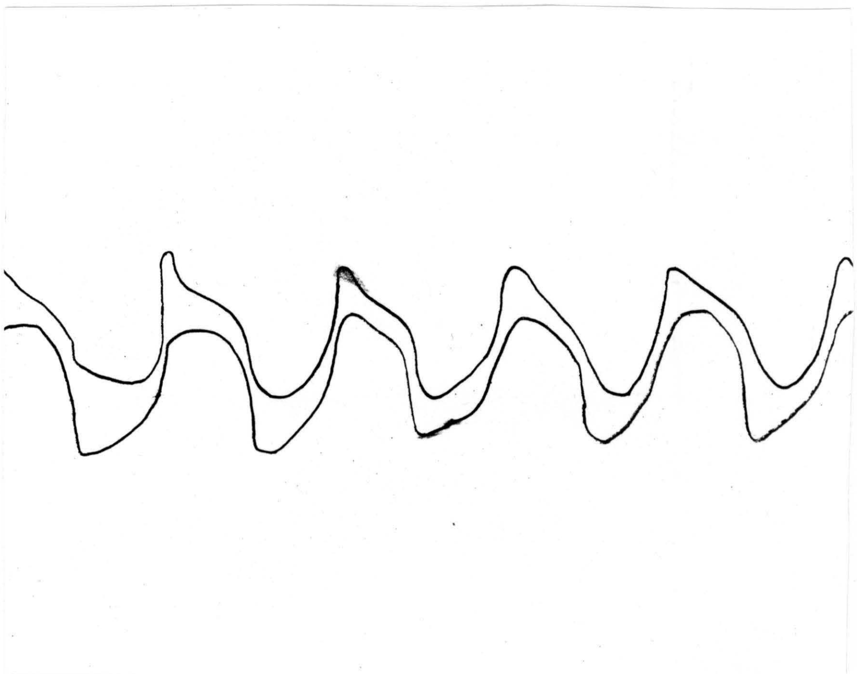


Figure 9. Acceleration Recording of Beam with Variable Stroke Shaker for Shaking Frequency of 325 cpm, Beam Spring Constant of 68 lb./in., No Attached Beam Weight and 50° Orifice Setting.

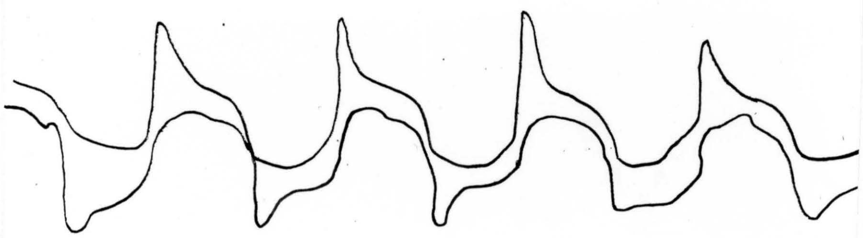


Figure 10. Acceleration Recording of Beam with Variable Stroke Shaker for Shaking Frequency of 325 cpm, Beam Spring Constant of 68 lb./in., No Attached Beam Weight and 70° Orifice Setting.

THEORETICAL ANALYSIS

The Constant Stroke Machine

A theoretical investigation of the Wagco shaker was performed to assure that certain assumptions were of acceptable accuracy when applied to the variable stroke machine.

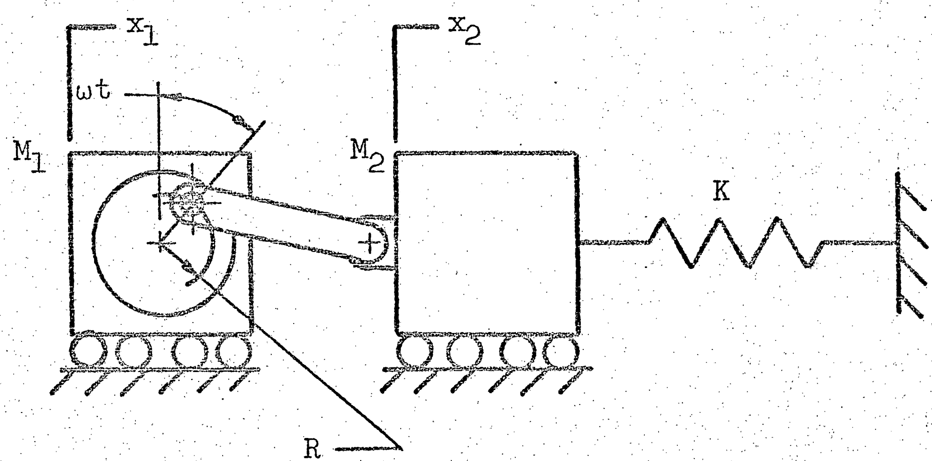
To analyze the Wagco shaker, the system was represented by two masses and one spring as shown in Figure 11. M_1 represents the mass of the crankshaft housing and internal parts and is 0.268 lb.-sec.^{2}/in. M_2 represents the mass of the boom and beam, which were assumed to act as one mass. The actual value of M_2 is either .443 or .508 lb.-sec.^{2}/in. depending on the effective limb mass. The spring constant, K, represents the spring constant of the cantilever beam. R, the crankshaft throw, is 2 inches. Sinusoidal motion was assumed for the system.

Two techniques were employed to analyze the Wagco shaker for the displacement and acceleration of the beam. The first was mobility theory (6) while the second was the exact solution of the differential equation of motion for the beam.

Before applying the mobility approach, the following displacement equation was written:

$$-x_1 + x_2 = R\sin\omega t. \qquad (1)$$

After taking the second derivative with respect to time and re-arranging terms, it was written as:



Definition of Symbols:
x_1 - displacement of M_1
x_2 - displacement of M_2
K - spring constant of cantilever beam
ω - angular frequency of crankshaft
t - time
R - radius of crankshaft throw

Figure 11. Theoretical Representation of Wagco Shaker.

$$\ddot{x}_1 = R\omega^2 \sin\omega t + \ddot{x}_2. \tag{2}$$

The Wagco shaker was further simplified as shown in Figure 12. The force F is the product $M_1\ddot{x}_1$. Using Equation (2) the force F was written in terms of $\ddot{x}_2$ and $\sin\omega t$, thereby eliminating the variable $\ddot{x}_1$.

The mobility diagram for the system was then drawn as shown in Figure 13. Using this diagram, the equations for the maximum displacement and acceleration of the beam were derived and are:

$$x_2 = \frac{Z_c M_1 R\omega}{i(1 + i\omega Z_c M_1)} \tag{3}$$

$$\ddot{x}_2 = -\omega^2 x_2 \tag{4}$$

where Z_c is the mobility of point c in the mobility diagram and i is the square root of minus one.

To solve for the exact solution of the differential equation of motion for the beam, the differential equation of motion for the system was written as:

$$M_1\ddot{x}_1 = -M_2\ddot{x}_2 - Kx_2 \tag{5}$$

Using Equation (2) to eliminate the variable $\ddot{x}_1$, the differential equation of motion for the beam was written as:

$$\ddot{x}_2 + \frac{K}{M_1 + M_2} x_2 = \frac{-M_1 R\omega^2}{M_1 + M_2} \sin\omega t. \tag{6}$$

The exact solution to Equation (6) is of the general form:

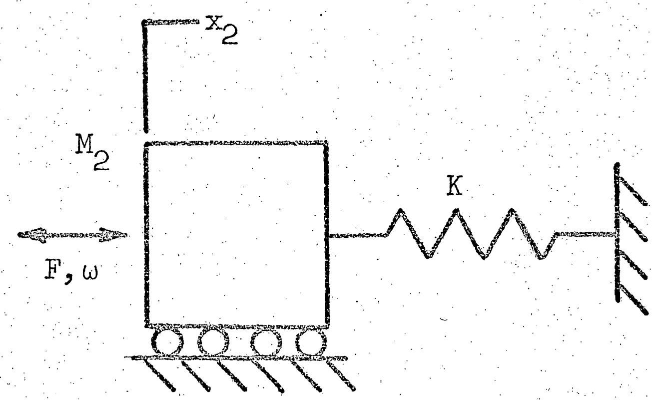


Figure 12. Simplification of Wagco Shaker with Mass One Removed.

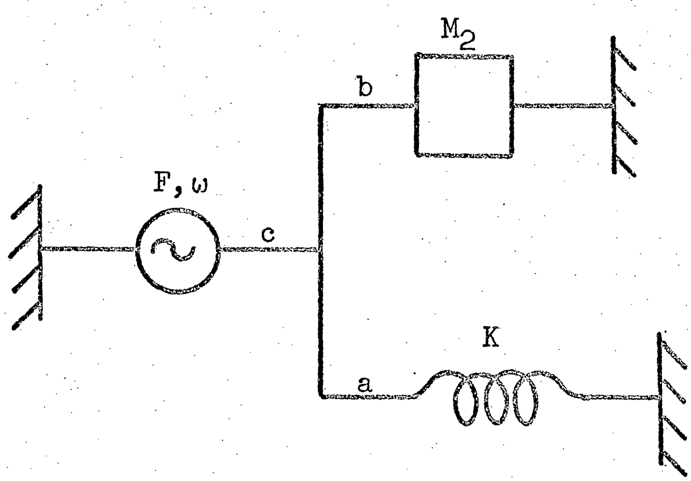


Figure 13. Mobility Diagram for Wagco Shaker.

$$x_2 = C_1 \cos\sqrt{\frac{K}{M_1 + M_2}}t + C_2 \sin\sqrt{\frac{K}{M_1 + M_2}}t + \frac{-M_1 R\omega^2 \sin\omega t}{K - \omega^2(M_1 + M_2)} \tag{7}$$

where C_1 and C_2 are arbitrary constants. To evaluate these constants, the following two conditions were employed:

at $t = 0$, $x_2 = 0$

at $\omega t = \pi/2$, $\dot{x}_2 = 0$.

Consequently, Equation (7) reduced to the final form of:

$$x_2 = \frac{-M_1 R\omega^2 \sin\omega t}{K - \omega^2 (M_1 + M_2)} . \tag{8}$$

Taking the second derivative of Equation (8), with respect to time, the acceleration of the beam was found to be:

$$\ddot{x}_2 = \frac{M_1 R\omega^4 \sin\omega t}{K - \omega^2 (M_1 + M_2)} . \tag{9}$$

Equations (3) and (8) and Equations (4) and (9) yield identical results for the displacement and acceleration respectively. These results are shown in Table 4.

The Variable Stroke Machine

The variable stroke shaker was similar to the Wagco shaker except

Table 4. Theoretical Displacement and Acceleration of Beam for Wagco Shaker

Shaker Frequency (cpm)	Beam Spring Constant (lb./in.)	Peak to Peak Displacement (in.)	Zero to Peak Acceleration (in./sec.^{2})
Attached Beam Weight - None			
325	38	1.58	912
408		1.55	1413
457		1.54	1762
498		1.54	2084
535		1.53	2399
600		1.52	3006
325	52	1.61	930
408		1.57	1430
457		1.55	1779
498		1.55	2101
535		1.54	2416
600		1.53	3024
325	68	1.64	949
408		1.59	1448
457		1.57	1797
498		1.56	2119
535		1.55	2434
600		1.54	3042
Attached Beam Weight - 35 lb.			
325	38	1.39	805
408		1.37	1249
457		1.36	1559
498		1.36	1845
535		1.35	2125
600		1.35	2664
325	52	1.41	818
408		1.38	1262
457		1.37	1571
498		1.37	1857
535		1.36	2137
600		1.35	2676
325	68	1.44	833
408		1.40	1276
457		1.38	1586
498		1.38	1872
535		1.37	2151
600		1.36	2690

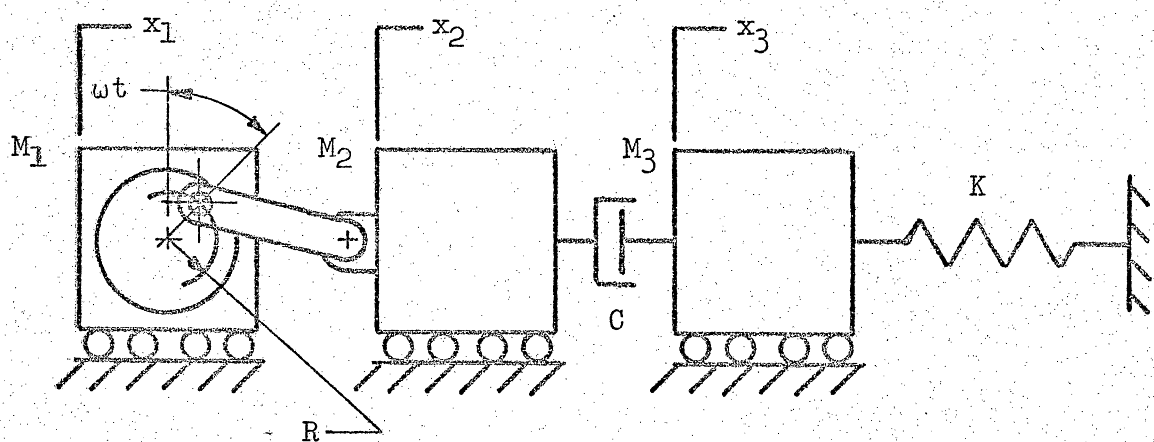
for the hydraulic cylinder-valve mechanism introduced into the boom. It was assumed that the cylinder valve combination provided viscous damping. A simplified sketch of this system is shown in Figure 14. M_1 is the same as it is for the Wagco shaker, while M_2 represents the mass of the slide portion of the boom and the cylinder barrel, and is .228 lb.-sec.^{2}/in. The coefficient of damping of the cylinder is represented by C. M_3 represents the mass of the piston rod or boom and the beam. Here again, the boom and beam were assumed to act as one mass with the actual value of M_3 being either 0.272 or 0.363 lb.-sec.^{2}/in., depending on the effective limb mass.

The mobility theory was used to calculate the coefficient of damping of the cylinder-valve combination. As a check, the computed damping coefficients were used in a numerical integration of the differential equations of motion for the system. Thus, it was convenient to compare the calculated displacements and accelerations of the beam to the corresponding test values.

Proceeding as in the case of the Wagco shaker, the system was simplified as shown in Figure 15. The force F is again the product $M_1\ddot{x}_1$. Using Equation (2), the force F was written in terms of $\ddot{x}_2$ and $\sin\omega t$, thereby eliminating the variable $\ddot{x}_1$.

The mobility diagram for the system was drawn as shown in Figure 16. With the use of this diagram, the equations for the maximum displacement and acceleration of the beam were derived and are:

$$x_3 = \frac{iZ_c Z_f M_1 R\omega}{Z_d\,(1 + i\omega Z_f M_1)} \qquad (10)$$



Definition of Symbols: x_1 - displacement of M_1
x_2 - displacement of M_2
ω - angular frequency of crankshaft
t - time
C - viscous damping coefficient of cylinder
x_3 - displacement of M_3
K - spring constant of cantilever beam

Figure 14. Theoretical Representation of Variable Stroke Shaker.

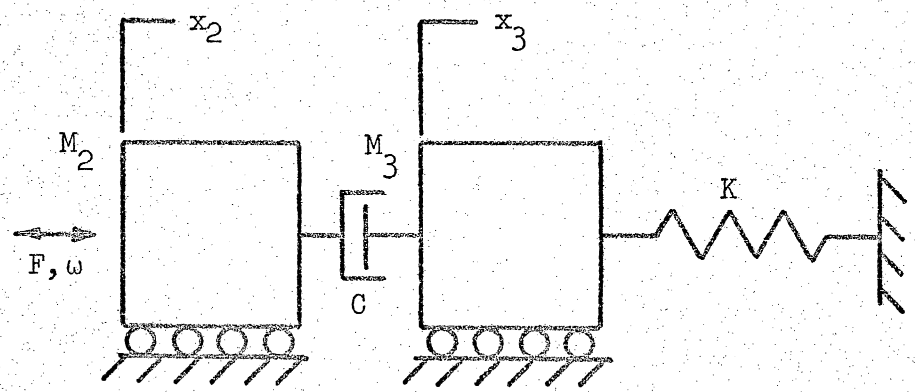


Figure 15. Simplification of Variable Stroke Shaker with M_1 removed.

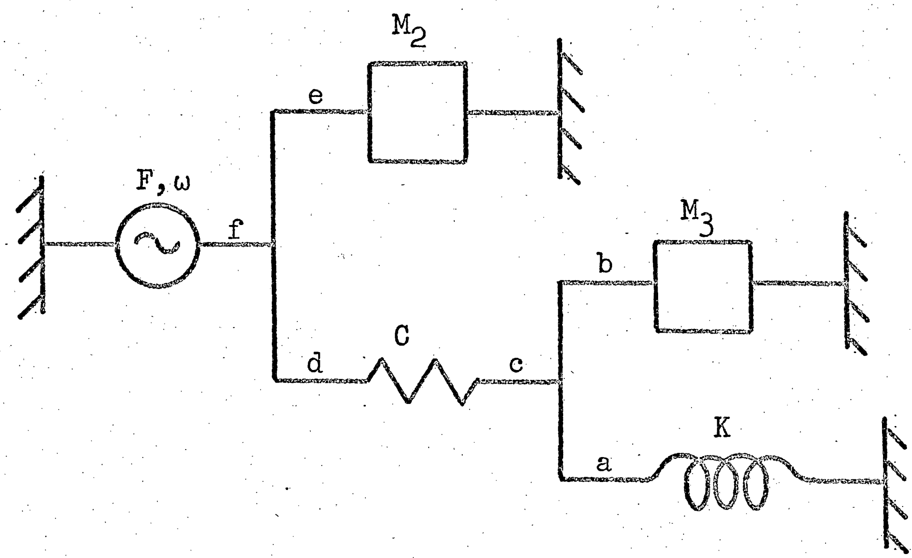


Figure 16. Mobility Diagram for Variable Stroke Shaker.

$$\ddot{x}_3 = -\omega^2 x_3 \tag{11}$$

where the subscript of each Z term refers to the mobility at that point in the mobility diagram.

A trial and error procedure was used to solve for the damping coefficients. A computer program was developed whereby the theoretical displacement of the beam, given by Equation (10), was compared to its test displacement for a given set of system parameters. The theoretical displacement was increased by incrementing the coefficient of damping until the difference between the theoretical and test displacements was less than or equal to 0.001 in. These results are shown in Tables 5 and 6. As mentioned previously, there were some conditions under which erratic action occurred. No coefficient of damping was calculated for the conditions where the measured beam displacement was higher than theory predicted.

The calculated coefficients of damping were employed to solve the differential equations of motion for the systems. This procedure was used to validate the calculated damping values.

The differential equations for the variable stroke system were written as:

$$M_1\ddot{x}_1 = -M_2\ddot{x}_2 - C(\dot{x}_2 - \dot{x}_3) \tag{12}$$

$$M_3\ddot{x}_3 = C(\dot{x}_2 - \dot{x}_3) - Kx_3. \tag{13}$$

Using Equation (2) to eliminate the variable x_1, Equation (12) was

Table 5. Coefficient of Damping of Cylinder-Valve Combination for Variable Stroke Shaker Without Attached Beam Weight

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Coefficient of Damping (lb.-sec./in.)
325	0	1.059
	20	1.059
	40	1.168
	50	1.503
	60	4.251
	70	*
	80	*
	90	*
408	0	1.140
	20	1.140
	40	1.140
	50	1.428
	60	5.529
	70	28.603
	80	*
	90	*
457	0	.815
	20	.607
	40	.607
	50	.920
	60	4.005
	70	*
	80	*
	90	*
498	0	.704
	20	.704
	40	.885
	50	.885
	60	2.733
	70	28.387
	80	*
	90	*
535	0	1.023
	20	.762
	40	.762
	50	1.154
	60	*
	70	*
	80	*
	90	*

Table 6. Coefficient of Damping of Cylinder-Valve Combination for Variable Stroke Shaker with 35 lb. Attached Beam Weight

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Coefficient of Damping (lb.-sec./in.)
325	0	1.364
	20	1.364
	40	1.519
	50	1.891
	60	2.946
	70	*
	80	*
	90	*
408	0	.746
	20	.746
	40	1.131
	50	1.391
	60	4.416
	70	*
	80	*
	90	*
457	0	.695
	20	.695
	40	.837
	50	1.270
	60	3.547
	70	*
	80	*
	90	*
498	0	.964
	20	.964
	40	.964
	50	.964
	60	1.461
	70	*
	80	*
	90	*
535	0	.509
	20	.509
	40	.862
	50	.862
	60	1.039
	70	*
	80	*
	90	*

* Not Calculated

written as:

$$\ddot{x}_2 + \frac{C}{M_1 + M_2}\dot{x}_2 - \frac{C}{M_1 + M_2}\dot{x}_3 = \frac{-M_1 R\omega^2}{M_1 + M_2}\sin\omega t. \tag{14}$$

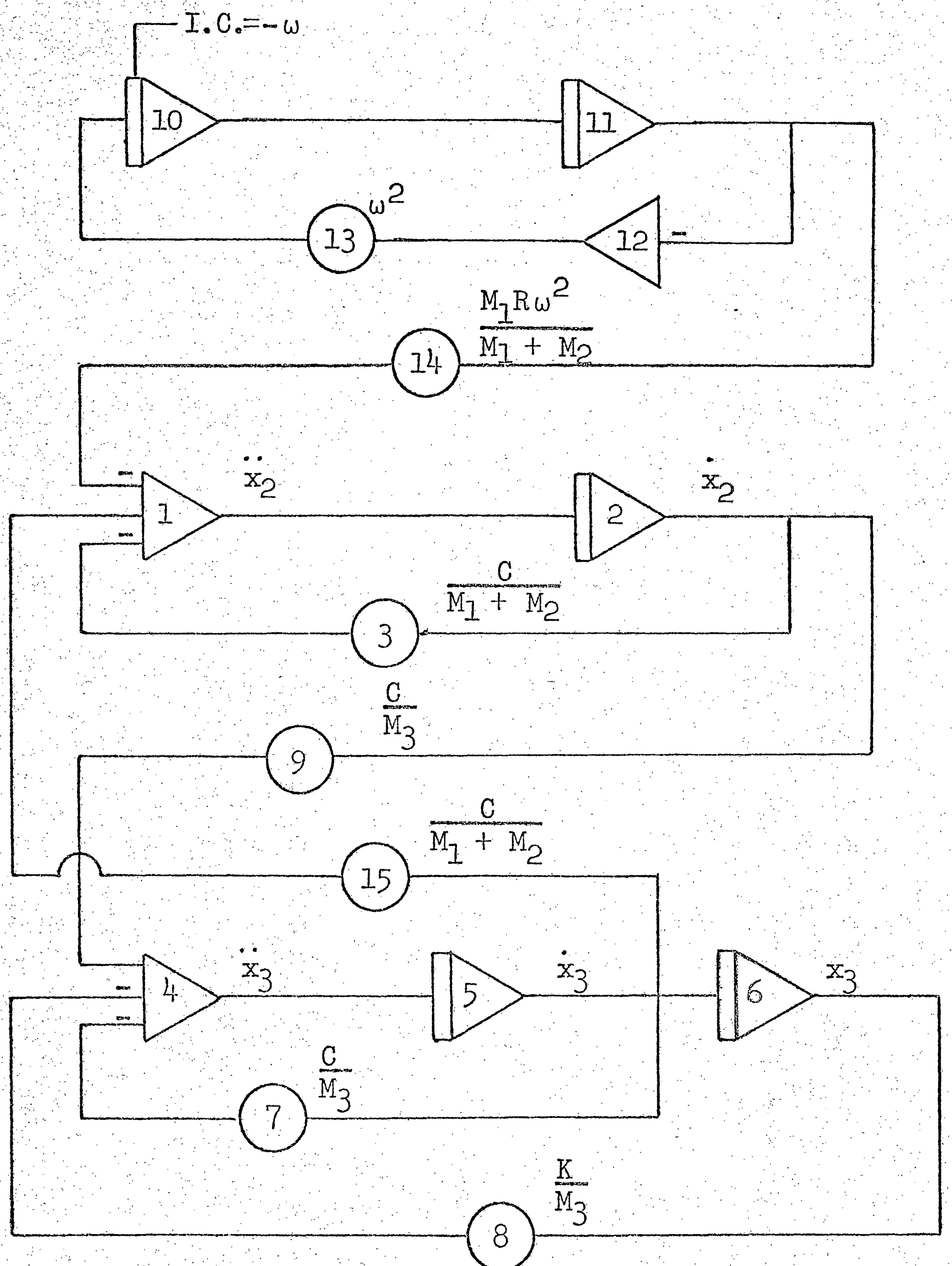
Equation (13) was re-written as:

$$\ddot{x}_3 + \frac{C}{M_3}\dot{x}_3 + \frac{K}{M_3}x_3 - \frac{C}{M_3}\dot{x}_2 = 0. \tag{15}$$

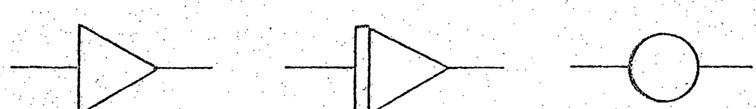
A computerized numerical integration solution was used to solve Equations (14) and (15). This program named Pactolus was also employed in the Wagco problem as a familiarization procedure. The input information to this program was the analog circuit for Equations (14) and (15) as shown in Figure 17. Each component was identified and the input and gain for each component was specified. The integration interval was one-thousandth of a second.

With all system initial conditions set to zero, as the actual values were unknown, transient effects appeared in the computed results, but were negligible after about four seconds of simulated operation. The maximum steady state values for displacement and acceleration of the beam were obtained from this program and are shown in Table 7.

These results show that as the value setting is increased, the displacement and acceleration of the beam is increased.



DESCRIPTION OF COMPONENTS:



Summing Amplifier

Integrating Amplifier

Attenuator

Figure 17. Analog Circuit for Differential Equations of Motion for the Variable Stroke Shaker

Table 7. Theoretical Displacement and Acceleration of Beam for Variable Stroke Shaker

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Peak to Peak Displacement (in.)	Zero to Peak Acceleration (in./sec.^{2})
Attached Beam Weight - None			
325	50	.41	238
325	60	.95	550
457	20	.10	111
Attached Beam Weight - 35 lb.			
457	60	.44	498

ANALYSIS OF RESULTS

The major purpose of this study was to develop a variable stroke tree shaker. A comparison of the test displacement of the beam for the Wagco and variable stroke shakers is shown in Table 8. From this table it can be seen that the objective was met. With low orifice settings, the displacement of the beam is lower for the variable stroke shaker than for the Wagco shaker. However, as the orifice setting is increased, displacement of the beam becomes larger.

Of further interest was the comparison of theory to test values for both machines. Test displacements and accelerations of the beam for the Wagco shaker are compared to theoretical results in Tables 9 and 10. Theoretical values were obtained using measured masses, frequencies and spring constants from Equations (3) and (4).

The test and theoretical displacement of the beam were felt to be in fair agreement. However, the test acceleration was consistently higher than theory predicted. It was thought that this difference was due to clearances at the wrist pin and/or main bearing of the vibrating source, thus causing a high short duration acceleration of the beam.

A comparison of theory and test values for the variable stroke shaker is shown in Tables 11 and 12. The theoretical displacement and acceleration were obtained from the numerical integration of the differential equations of motion for the system, using the coefficient of damping calculated from mobility theory. Here, the test and theoretical displacement of the beam were felt to be in good agreement, but the

Table 8. Displacement of Beam for Wagco and Variable Stroke Shakers

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Displacement (in.) Wagco Shaker	Variable Stroke Shaker
Attached Beam Weight - None			
325	0	1.88	.31
	20		.31
	40		.34
	50		.43
	60		.97
	70		1.75
	80		2.25
	90		2.25
408	0	1.69	.25
	20		.25
	40		.25
	50		.31
	60		.94
	70		1.43
	80		1.69
	90		*

* Not Recorded

Table 9. Displacement of Beam Obtained from Test and Theory for Wagco Shaker

Shaking Frequency (cpm)	Beam Spring Constant (lb./in.)	Peak to Peak Displacement (in.) Test	Theory
Attached Beam Weight - None			
325	38	1.75	1.58
408		1.62	1.55
457		1.62	1.54
498		1.62	1.54
535		*	1.53
600		1.56	1.52
325	52	1.75	1.61
408		1.69	1.57
457		1.69	1.55
498		1.69	1.55
535		1.62	1.54
600		1.62	1.53
325	68	1.88	1.64
408		1.69	1.59
457		1.69	1.57
498		1.62	1.56
535		1.62	1.55
600		1.62	1.54
Attached Beam Weight - 35 lb.			
325	38	1.50	1.39
408		1.37	1.37
457		1.37	1.36
498		1.31	1.36
535		*	1.35
600		1.43	1.35
325	52	1.50	1.41
408		1.40	1.38
457		1.37	1.37
498		1.31	1.37
535		1.25	1.36
600		*	1.36
325	68	1.50	1.44
408		1.40	1.40
457		1.37	1.38
498		1.28	1.37
535		1.25	1.37
600		*	1.36

* Not Recorded

Table 10. Acceleration of Beam Obtained from Test and Theory for Wagco Shaker

Shaking Frequency (cpm)	Beam Spring Constant (lb./in.)	Zero to Peak Acceleration (in./sec.^{2}) Test	Theory
Attached Beam Weight - None			
325	38	2470	912
408		4150	1413
457		5160	1762
498		6260	2084
535		*	2399
600		9160	3006
325	52	2530	930
408		3920	1430
457		4830	1779
498		6020	2101
535		7240	2416
600		8560	3024
325	68	2160	949
408		4090	1448
457		4950	1797
498		5960	2119
535		7240	2434
600		8450	3042
Attached Beam Weight - 35 lb.			
325	38	2040	805
408		3740	1249
457		4460	1559
498		5300	1845
535		*	2125
600		9650	2664
325	52	2220	818
408		3860	1262
457		4600	1571
498		5550	1857
535		6020	2136
600		*	2676
325	68	1870	833
408		3760	1276
457		4460	1586
498		5300	1872
535		6260	2151
600		*	2690

* Not Recorded

Table 11. Displacement of Beam Obtained from Test and Theory for Variable Stroke Shaker

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Peak to Peak Displacement (in.)	
		Test	Theory
Attached Beam Weight - None			
325	50	.43	.41
325	60	.97	.95
457	20	.12	.10
Attached Beam Weight - 35 lb.			
457	60	.47	.44

Table 12. Acceleration of Beam Obtained from Test and Theory for Variable Stroke Shaker

Beam Spring Constant - 68 lb./in.

Shaking Frequency (cpm)	Valve Setting (degrees)	Zero to Peak Acceleration (in./sec.^{2}) Test	Theory
Attached Beam Weight - None			
325	50	409	238
325	60	625	550
457	20	965	111
Attached Beam Weight - 35 lb.			
457	60	868	497

acceleration values varied widely for the same reason as in the Wagco shaker.

5

RECOMMENDATIONS

The variable stroke shaker developed for this study needs improvement. The design gave satisfactory performance for laboratory tests, but certain changes should be made before undertaking a rigorous field analysis.

Some suggested design improvements are:

1. The radial distance between the cylinder inside diameter and the piston rod outside diameter should be reduced from 1/4 inch to 3/16 inch. This change would eliminate the static oil seal and improve the design of the end caps.
2. The cylinder barrel and piston rod should be machined to size rather than polished from standard mill stock. This change would improve the straightness and reduce the ovalness of these parts.
3. Threaded fittings should be used to connect the metering valve to the cylinder.
4. Some means should be provided to prevent the cylinder assembly from extending or retracting excessively. This addition was not necessary in laboratory test. However, it should be incorporated prior to field work.
5. The rotation of the metering valve could be programmed to give a controlled rate of change of stroke.

SUMMARY AND CONCLUSIONS

A variable stroke tree shaking mechanism was designed and built. The variable stroke feature was attained with the use of a hydraulic cylinder and metering valve in the boom of the shaker. This assembly acted as a variable rate dashpot.

The design was tested in the laboratory under varying conditions of effective limb mass and shaking frequency. A wooden beam was used to simulate the limb. For comparison, a constant stroke shaker was subjected to similar tests.

Both the constant and variable stroke machines were subjected to a theoretical analysis. The constant stroke machine was analyzed by two theories, mobility theory and the differential equation of motion approach. Theory and test results were in good agreement for displacement of the beam but differed widely for acceleration.

For the variable stroke shaker, mobility theory was used to calculate the coefficients of damping due to the cylinder-valve combination. As a check these coefficients were used in the differential equations of motion for the system to solve for the displacement and acceleration of the beam.

The test results of this investigation were limited to the laboratory, but an effective variable stroke shaking mechanism was developed. Extensive field testing is needed to establish the feasibility of this type of shaker in the mechanical harvest of fruit trees.

The conclusions that can be drawn from this study are as follows:

1. The hydraulic damping approach is a valid and effective means to variable stroke tree shaker design.
2. Theory may be used to predict the displacement of a vibrating beam excited by a constant stroke tree shaking mechanism.
3. Theory may be used to predict the displacement of a vibrating beam excited by a variable stroke tree shaking mechanism where the variable stroke is provided by variable damping, if valid damping coefficients are known.

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A HYDRAULIC DAMPING APPROACH TO VARIABLE STROKE TREE SHAKER DESIGN

Norman Donald Kramer

Abstract

Much interest in mechanically harvesting apples has been generated, but no completely successful system has yet been developed. Bruising of the fruit has been a prime drawback to current mechanical harvesting methods.

One practical approach to mechanized apple harvesting is to shake the trunk or limb of the tree to remove the fruit. With this method, apple detachment appears to be more dependent on length of stroke rather than shaking frequency. Therefore, a variable stroke shaker appeared to be a reasonable approach to a controlled rate of detachment from the tree.

The objective of this project was to develop a variable stroke tree shaking mechanism. The variable stroke feature was attained with the use of a hydraulic cylinder and metering valve in the boom of the shaker with the cylinder acting as a variable rate dashpot. The design was tested in the laboratory under varying conditions of effective limb mass and shaking frequency. For comparison, a constant stroke shaker was subjected to similar tests.

Both shaking mechanisms were subjected to a theoretical analysis. For the constant stroke shaker, the theoretical and test displacements

of the limb were in good agreement, but the theoretical and test acceleration of the limb varied widely.

In the variable stroke shaker, the coefficient of damping of the cylinder was determined from test work and checked, using the differential equations for the system.

The testing performed in this investigation was limited to the laboratory, but for these conditions an effective variable stroke mechanism was developed.