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FABRIC RESEARCH LABORATORIES, INC.

FRL

413925

DEVELOPMENT OF A

COMPRESSION IMPACT TESTING MACHINE

FINAL SUMMARY REPORT

Submitted by

Fabric Research Laboratories, Inc.

for

Quartermaster Research and Engineering Command

U. S. Army

Contract Number DA-19-129-QM-1905

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I. INTRODUCTION

The test device covered by this contract was developed to provide means for dynamic testing of cushioning materials. Until recently designs for landing cushions of material rigged for parachute delivery were necessarily projected from static test data of the cushion material, and were proven by static test drops of the equipment item and its cushioning rigging There is an opportunity for increased efficiency and economy if a more precise knowledge of dynamic cushion material behavior can be made available, so that a more nearly optimum choice of and use of cushioning materials can supplant the present empirical design practices.

This new test facility will enable the Army to investigate the dynamic mechanical properties of various cushioning materials under controll laboratory conditions. Forearmed with knowledge gained in this manner, only materials showing a high potential for success need actually be drop-tested, thereby saving great amounts of time and expense.

Test machine specifications established by the Army required impact speeds in a range from 20 to 60 feet per second, and an energy level of up to 12,000 foot-pounds. The machine design conceived by FRL for this application has two welded steel cars aligned on horizontal rails. The cars are fitted with heavy impact plates, and the specimen to be tested is secured to the impact face of one car. In operation, the cars are initially at opposite ends of the 40-foot track, and are propelled toward a collision at the center by the action of co-ordinated pneumatic cylinders. Energy input to the system is controlled by adding or removing cast iron weights to adjust the mass of the moving cars, and speed is controlled by pressure in the pneumatic actuators. A separate pneumatic system operates friction brakes which stop the cars after the collision, and absorb resilient energy, if any, after rebound.

Instrumentation includes a shock accelerometer which senses deceleration force during the impact, and an electromagnetic velocity measurement system which measures relative velocities of the two cars before, during, and after the deformation of the test specimen. Outputs of the two measured variables are displayed and photographed on a dual beam oscilloscope.

This horizontal rail type machine eliminates the problems of repetitive impact, and the awkwardness of high drop towers and large masses required for gravity actuated devices. The use of two opposed, balanced, moving masses eliminates the shock loads which would be delivered to the building and surrounding equipment by a single-mass impact device of the same energy level.

The dynamic properties data obtained with the machine will provide a better understanding of the behavior of materials and structures under dynamic loading and will be useful in packaging, structural design, and other fields as well as the immediate problem of design of parachute landing cushions.

II. DESIGN CALCULATIONS

Design criteria for this device were established by provisions of the contract as follows:

"1. The contractor shall within 12 months after receipt of award, furnish necessary labor, materials, personnel, facilities and supplies, and do all other things necessary for, and shall design, fabricate, deliver, install, calibrate and demonstrate one completely assembled and operable compressive impact testing machine which shall include the furnishing of all instrumentation complying with the requirements and characteristics set forth below:

a. General Concept: The compressive impact tester shall essentially consist of a self-contained horizontally mounted thrust unit capable of accelerating a guided movable carriage assembly. The thrust unit shall provide sufficient energy to accelerate the carriage assembly throughout the required ranges of mass and velocity. The energy of the freely rolling carriage will be utilized to crush a specimen of energy dissipating material under test. Instrumentation shall be provided to record sufficient data for the complete determination of the stress-strain characteristics of test specimens.

b. Design Requirements:

(1) Thrust Unit: The thrust unit shall be designed to provide a variable carriage velocity range from 20 to 60 ft/sec., with a range of carriage energies from a design minimum to 12,000 ft/lbs at 30 ft/sec and from a design minimum to 12,000 ft/lbs at 60 ft/sec.

(2) Carriage system: The carriage system shall be designed to provide the following features:

(a) A minimum impact surface of 18 x 18 inches square (324 sq in).

(b) Means for varying carriage weight in nominal increments of 50 lbs to meet the ranges of energy requirements specified in (1) above.

(c) A delayed braking action; carriage braking system to be energized after the carriage has attained its maximum rebound velocit,.

(d) Means for returning carriage assembly to its starting position.

(e) Satisfactory means for maintaining carriage firmly in contact with thrust unit during acceleration stroke.

(f) Suitable device for loading and unloading carriage weights.

(3) Control Station: The system shall be designed to permit operation from one central control console. The control system shall include interlocking safety features to prevent the accidental firing of the thrust unit in order to provide the necessary protection to operating personnel and to the

equipment itself against injury or damage respectively. As applicable, interlocking safety controls shall be installed in the following areas to insure that:

(a) Carriage system is in firm contact with thrust unit prior to firing,

(b) A test specimen has been installed on thrust plate,

(c) Safety barriers are in a closed or locked position before firing,

(d) Braking system is fail-safe in case of malfunction,

(e) Failure of any other function or component pertinent to the design and/or operations of the machine is not hazardous to either operating personnel or to the machine itself.

Control circuitry shall include visual indication showing the position of the interlocking safety controls.

(4) Instrumentation, accurate to within plus or minus 5% shall be provided to record necessary data for relatively simple determinations of the following:

(a) Carriage velocity or energy prior to impact with specimen,

(b) Specimen crushing force and distance vs time; (specimen thickness will range from 3 to 12 inches, and crushing distance approximately 75% of specimen thickness).

(c) Test specimen resilience (rebound energy).

(5) Finish: All metallic surfaces subject to corrosion shall be protected by painting. Painting shall consist of a prime coat and a finish coat of Standard Army Olive Drab Color No. 24087 per Federal Standard No. 595.

(6) Workmanship: The finished assembly shall be clean, well made and free from any defect which may affect appearance or serviceability.

2. The contractor shall furnish one (1) set of reproducible manufacturing drawings of the machine, one (1) set of prints made from the reproducible drawings, and three (3) copies of instructions necessary for the operation, calibration, maintenance, instrumentation techniques and calculations, and circuit diagram at the time of delivery of the machine."

Design calculations to meet these requirements resulted in a machine design in accordance with the following:

A. Compressive Impact Machine, Design and Development

From Purchase Request Specifications:

Maximum Energy	=	12,000 ft.lbs.
Maximum Velocity	=	60 ft/sec.
Minimum Velocity	=	30 ft/sec.

For two equal carriages the kinetic energy of each is 6000 ft.lbs. at both 15 and 30 ft/sec.

1. Calculation of Carriage Weights

$$\text{Energy} = E = \frac{1}{2} m V^2 = \frac{1}{2} \frac{W}{g} V^2$$

$$\text{Weight} = W = \frac{2 E g}{V^2}$$

$$W_{15'/sec} = \frac{(2)(6000)(32.2)}{(15)^2} = 1720 \text{ lbs}$$

$$W_{30'/sec} = \frac{(2)(6000)(32.2)}{(30)^2} = 430 \text{ lbs.}$$

Assume a 3 foot long acceleration distance, and a near constant accelerating force. (Near constant force can be provided by large accumulator volume coupled to operating cylinder.)

$$\text{Acceleration} = a = \frac{V^2}{2s}$$

$$a_{15'/sec} = \frac{(15)^2}{2 \times 3} = 37.5 \text{ ft/sec}^2$$

$$a_{30'/sec} = \frac{(30)^2}{2 \times 3} = 150 \text{ ft/sec}^2.$$

2. Calculation of Accelerating Force

$$F = m a = \frac{W}{g} a$$

$$F_{15'/sec} = \frac{1720}{32.2} \times 37.5 = 2000 \text{ lbs}$$

$$F_{30'/sec} = \frac{430}{32.2} \times 150 = 2000 \text{ lbs.}$$

Or, 2000 lb force acting through three-foot stroke represents required input of 6000 ft.lbs. In practice, force requirement will be larger because of the following factors:

(a) Non-constant pressure of expanding gas will result in a non-constant accelerating force

(b) Internal friction of force cylinder

(c) Friction of carriage wheels and bearings

(d) Wind resistance of carriage

(e) Angular acceleration of carriage wheels.

Assume an 8-inch diameter cylinder. Then

$$\text{Area} = \frac{\pi}{4} 8^2 = 50 \text{ sq.in.}$$

$$\text{Pressure} = \frac{F}{A} = \frac{2000}{50} = 40 \text{ psi .}$$

This is too low for average air pressure requirement for maximum energy requirements. Try a smaller cylinder diameter; for a 6-inch diameter cylinder

$$A = \frac{\pi}{4} 6^2 = 28 \text{ sq.in.}$$

$$P = \frac{2000}{28} = 72 \text{ psi .}$$

72 psi is a reasonable level for low pressure operation. Reducing stroke from 36 inches or reducing cylinder diameter to 5 inches would both allow operation at a slightly higher pressure level.

A 6 inch or 5 inch diameter cylinder appears adequate for force requirements. Cylinder selection will also depend on inlet and outlet port size available to permit the gas velocities associated with the required carriage velocities.

6 inch cylinder @ 30 ft/sec.

$A_{cyl} = 28.27$ sq.in.

Inlet Port Pipe Size	Actual Diameter	Area	$\frac{A_{cyl}}{A_{port}}$	Discharge Velocity
3/4"	0.824"	0.533in^2	53.0	1590 ft/sec.
1"	1.049"	0.864in^2	32.7	980 ft/sec.
1-1/4"	1.380"	1.495in^2	20.8	625 ft/sec.

Velocity of sound in air = 1130 ft/sec @ 70°F

Velocity of sound in gas $\cong \sqrt{\frac{K}{\rho}}$

For Air, $K = 1.4$, $\rho = 1$

For Helium, $K = 1.66$, $\rho = 0.138$.

Velocity of sound in helium, compared to sonic velocity in air, assuming same temperature of 70°F, and $V_{air} = 1100$ ft/sec

$$V_{He} = V_{air} \frac{\sqrt{\frac{1.66}{.138}}}{\sqrt{\frac{1.4}{1}}}$$

$$= 1130 \frac{\sqrt{12}}{\sqrt{1.4}}$$

$$= 1130 \frac{3.46}{1.19} = 3290 \text{ ft/sec} .$$

6" cylinder with inlet ports shown above appears to be satisfactory with helium. A 3/4" port is too small with air. Suggest use of largest possible port size to minimize pressure drop.

For carriage weight range of 430 to 1720 lbs, and 50 lb weight increment,

$$\begin{array}{r} 1720 \\ 430 \\ \hline 1290 \end{array} = \text{Approx } 1300$$

$$\frac{1300}{50} = 26 \text{ incremental weights}$$

3. Carriage Design

Impact plate to be 18" square maximum (design requirement)

Assume a 1-1/2 inch thick steel plate

$$\text{Weight} = (1.5)^2 \times 61.2 = 138 \text{ lbs}$$

Then a trial weight balance of carriage =

Impact plate	140 lbs
Wheels, bearings, etc.	60
Frame	100
Impact bar and weight attachment system	100
Brake system	30
	430 lbs Total

For convenience, the same 50 lbs weight configuration as used for Air Force gun at FRL is employed.

Thickness = Approx 1-5/16"

Then for 26 weights, $\ell = 26 \times 1.31 = 34$ inches. Allowing additional length for impact plate, cap, nut, etc., overall length of carriage = approx 42 inches. Then an approximation of machine size is

Length of carriage =	Approx 42"
Length of cylinder (36" stroke, 10" stop tube)	56"
Specimen length	12"
Stroke	36"

Then

2 x 42"	=	7' 0"
2 x 56"	=	9' 4"
1 x 12"	=	1' 0"
2 x 36"	=	6' 0"
		23' 4"

This is total length without allowance for braking.

Assume 4' braking distance for each carriage, and 2' overtravel distance for each carriage, then

Length = 23' 4" plus 12' = 35' 4" .

Use 36' 0" for layout purposes.

For 4' braking distance, and a total <u>maximum</u> braking effort of 6000 ft.lbs.,

$$\text{Braking force} = \frac{6000}{4} = 1500 \text{ lbs.}$$

Assume a friction coefficient of 0.3, then

$$\text{Transverse brake force} = \frac{1500}{.3} = 5000 \text{ lbs.}$$

For 6" diameter brake cylinder

$$P = \frac{F}{A} = \frac{5000}{28} = 180 \text{ psi .}$$

4. Track Structure

Tracks in two identical 18' sections. Mounts for propulsion and brake cylinders at opposite ends. Brake cylinder location to be staggered to allow for non-concentric impact. Will require two separate 6-inch diameter brake cylinders. Legs are necessary to provide clearance height for brakes.

Assume a 6 x 6" WF 15.5 lb section, estimated weight of track structure is

Rails	72'	@ 15.5	=	1200
Columns	12 x 1.5	@ 15.5	=	278
Foot plates	12	@ 18	=	216
Cross members	6 x 3	@ 5.4	=	98
Fish plates, clips, bolts, etc.		@ 100	=	100
				1892

Say 1900 lbs.

(Does not include supports for brake and drive cylinders.)

5. Choice of Accumulator Configurations for Two Cylinder Propulsion System

a. Two welded steel tanks discharging through two separate valves into each of two operating cylinders.

b. One welded steel tank discharging through one valve with Y-branch to two operating cylinders.

c. One long tank (section of large pipe) discharging through separate valve at each end into each of two operating cylinders.

d. One welded steel tank, discharging through two valve and pipe systems to separate operating cylinders.

Of these, (b) appears most feasible, requiring no valve synchronization, and being most economical in use of components. Bifurcated inlet lines to cylinders must be generously sized for required flow rates, with consideration for added expansion volume downstream of valve, which will reduce efficiency of system somewhat, and require more gas to reach desired velocity.

6. Wind Resistance of Carriage

$$\text{Drag Force} = D = C_D \, q \, S$$

where C_D = drag coefficient

S = area (square feet)

$q = \frac{1}{2} \rho V^2$

ρ = density of air

At 100 mph (147 ft/sec), $q = 25.6$ lbs/sq.in.

At 30 ft/sec, $q = 25.6 \times \left(\frac{30}{147}\right)^2 = 1.07$ lbs/sq.ft.

At 30 ft/sec, C_D = approx 1.2

Then $D = 1.2(1.07)(1.5)^2$

$= 2.9$ lbs, negligible.

7. Impact Carriage - Wheel Acceleration

6" diameter wheel $=$ 1.57 ft circumference

8" diameter wheel $=$ 2.09 ft circumference

10" diameter wheel $=$ 2.51 ft circumference.

At 30 ft/sec, 6" wheel $=$ 19 rps $=$ 1140 rpm

8" wheel $=$ 14.4 rps $=$ 860 rpm

10" wheel $=$ 11.9 rps $=$ 715 rpm.

Estimated weights: 6" wheel $=$ 4-1/2 lbs
8" wheel $=$ 5-1/8 lbs
10" wheel $=$ 8-1/2 lbs.

Time for acceleration to 30 ft/sec in 3 feet:

$$t = \sqrt{\frac{2\,s}{a}} \qquad a = 150 \text{ ft/sec}$$

$$t = \sqrt{\frac{2 \times 3}{150}} = \sqrt{.04} = 0.2 \text{ sec} .$$

Acceleration torque of wheels $= T = \dfrac{Wr^2}{g}\,\dfrac{2\pi N}{60t}$ $\qquad N = \text{rpm}$, $t = \text{sec}$

Estimated Wr^2:

$$6" = 4.5 \times (2)^2 = 18 \text{ lb.in}^2 .$$

$$8" = 5.13 \times (3)^2 = 46 \text{ lb.in}^2 .$$

$$10" = 8.5 \times (3.7)^2 = 116 \text{ lb.in}^2 .$$

$$T_6 = \frac{18}{386}\,\frac{2\pi}{60}\,\frac{1140}{.2} = 28 \text{ in.lbs} \qquad F = \frac{T}{R} = \frac{28}{3} = 9.5 \text{ lbs}$$

$$T_8 = \frac{46}{386}\,\frac{2\pi}{60}\,\frac{860}{.2} = 54 \text{ in.lbs} \qquad F = \frac{T}{R} = \frac{54}{4} = 13.5 \text{ lbs}$$

$$T_{10} = \frac{116}{386}\,\frac{2\pi}{60}\,\frac{715}{.2} = 112 \text{ in.lbs} \qquad F = \frac{T}{R} = \frac{112}{5} = 22.5 \text{ lbs}$$

Total force for wheel acceleration, which is in addition to force for linear acceleration of carriage and friction forces, is for each carriage with four wheels:

6" wheels = approx 40 lbs

8" wheels = approx 54 lbs

10" wheels = approx 90 lbs

Thus smaller lighter wheels are to be favored if they can provide sufficient weight carrying ability.

8. Brake System

Assume carriage energy after impact at one-half of initial value of 6000 ft.lbs. Then

$$\text{Max velocity} = \frac{30}{\sqrt{2}} = 21.2 \text{ ft/sec.}$$

For braking to 0 ft/sec in 4 feet,

$$t = \frac{2\,s}{V} = \frac{8}{21.2} = 0.415 \text{ sec.}$$

$$\text{Braking Power} = \frac{6000}{.415} = 14{,}500 \text{ ft-lbs/sec.}$$

$$\text{Brake HP} = \frac{14{,}500}{550} = 26 \text{ HP}$$

Brake proportions for intermittent operation allows

$$PV \leq 1000$$

where V = average velocity

P = brake pressure.

The value of 1000 is for wood brake blocks, and can probably be safely exceeded with composition brake blocks.

If Max V = 30 ft/sec (full speed) or 21.2 ft/sec (1/2 energy)

V = 15 ft/sec (full speed) or 10.6 ft/sec (1/2 energy)

$$P = \frac{1000}{15} = 67 \text{ psi} \quad \text{or} \quad \frac{1000}{10.6} = 95 \text{ psi}$$

This indicates a maximum brake pressure of approximately 100 psi for wood blocks.

For composition blocks pressure will be limited to approximately 150 psi. Thus for braking forces of 5000 lbs the areas required are:

Brake Size	Area	Pressure
4" sq.	16 sq.in.	$\frac{5000}{16} = 313$ psi
5" sq.	25 sq.in.	$\frac{5000}{25} = 200$ psi
6" sq.	36 sq.in.	$\frac{5000}{36} = 140$ psi

Select 6" square brake block, operating on 6" wide brake strip.

9. Propulsion Cylinder Calculations

From Miller Catalog*, page 20, application is Fig. F_3H.

Stroke = "D" = 36"

Column Length = L = 4D = 144"

For 6" cylinder and 100 psi,

thrust = 2827 lbs.

The Catalog also provides that for "D" of 36" and L of 144", the length of the stop tube required is 10". The corrected L, with the stop tube is:

L = 144 + 10 = 154".

A 2-1/2" diameter piston rod is satisfactory in terms of column strength based on information furnished in the Catalog.

Use 6" diameter cylinder, 10" stop tube, 36" stroke, cushioned one end, oversize ports (1" diameter).

Cylinder is to be side lug mounted; centerline of cylinder to be 5" above tracks; centerline height of cylinder is 3-1/4". Therefore, mount cylinder on 1-3/4" thick cross members bolted or welded across top of tracks.

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* Miller Fluid Power Div. Flock-Reedy Corp., Bulletin No. AJH-104S, 1961

Overall length of cylinder (according to the Catalog) is

$$L = 56\text{-}1/16''.$$

Displacement of cylinders

$$= \frac{\pi}{4} (0.5)^2 \times 3 = 0.6 \text{ cu.ft. each.}$$

For 20' of 2" connecting pipe,

$$\text{Volume} = \frac{\pi}{4} \left(\frac{2}{12}\right)^2 \times 20 = \frac{20\pi}{144} = 0.45 \text{ cu.ft.}$$

Total expansion volume = approximately 1.7 cu.ft.

Assume accumulator volume = 3 cu.ft. Then

$$\text{Accumulator Pressure} = \text{Working Pressure} \times \frac{3 + 1.7}{3}$$

$$P_{Acc} = P_{Work} \times 1.56 .$$

(For 100 psi in cylinders, use 156 psi in tank, etc.)

Tank dimensions for 3 cu.ft. Assume 16" diameter x 26" long,

$$V = \frac{\pi}{4} D^2 L$$

$$= \frac{\pi}{4} (1.33)^2 \times \frac{26}{12} = 3 \text{ cu.ft.}$$

10. Maximum G Levels

For specimen thicknesses of 3" and 12", assume deformation = 75%, or 2.25" and 9". Assuming a constant deceleration,

$$V^2 = 2 A S .$$

At maximum velocity of 60 ft/sec,

$$60^2 = 2 A \frac{2.25}{12}$$

$$A_{3"} = \frac{(60)^2 \times 12}{2 \times 2.25} = 9600 \text{ ft/sec}^2$$

$= \text{approximately } 300 \text{ G's } (3" \text{ thickness})$

$$A_{12"} = \frac{(60)^2 \times 12}{2 \times 9} = 2400 \text{ ft/sec}^2$$

$= \text{approximately } 75 \text{ G's } (12" \text{ thickness})$

11. Deflection of Tracks with Loaded Carriage

6" I-beam is supported on 6' centers. Assume 900-lb load at mid-span,

$$I = 21.8$$

$$M = \frac{W\ell}{4} = \frac{900 \times 72}{4} = 16{,}200 \text{ in-lbs}$$

$$y = \frac{W\ell^3}{48EI} = \frac{900 \times 72^3}{48 \times 30 \times 10^6 \times 21.8}$$

$$= .001070$$

12. Maximum Velocity Performance

Acceleration time for maximum velocity $= \sqrt{\frac{2s}{a}} = \sqrt{\frac{2 \times 3}{150}} = 0.2$ seconds.

Maximum operating speed of ball valve operator may not permit operation of the ball valve in this time frame. This may or may not prove to be a limitation on maximum velocity performance of the equipment. Initial movement of the propulsion piston and initial acceleration of the carriage will require a reduced gas flow, as available through partially opened ball valve. Final acceleration and movement of piston will require full flow through wide open ball valve. There will be some lag between valve operation and carriage movement, so that if valve operating time is of the same order as the carriage accelerating time, then the valve operating speed should not prove an insufferable obstacle.

If, however, operating speed of the valve does prevent obtaining specified performance, then a maximum speed valving device (in the form of a burst diaphragm) can be designed and installed. This will provide low millisecond operating times, but has disadvantages of operation at relatively coarse stepped pressure levels; requires manufacture of expendable parts, and must be re-assembled for each operation. Fortunately, this has been proven to be unnecessary.

13. Consideration of Rotational Energy of Wheels, and Dissipation of This Energy During or After Impact

Consider case of maximum energy and maximum velocity impact on thin specimen. Assume

$$\text{weight (of each cart)} = 430 \text{ lbs}$$

$$\text{velocity (of each cart)} = 30 \text{ ft/sec.}$$

Assuming 3" specimen deforming to 1", then time of deceleration to zero of each cart from 30 ft/sec in 1":

$$s = \frac{1}{2} at^2$$

$$t = \sqrt{\frac{2s}{a}}$$

$$a = \frac{V^2}{2s} = \frac{(30)^2(12)}{2 \times 1} = 5400 \text{ ft/sec}^2$$

$$t = \sqrt{\frac{2s}{a}} = \sqrt{\frac{2}{(5400)(12)}}$$

$$= \sqrt{.0000311} = 0.0055 \text{ sec.}$$

For 6" wheels weighing 4.5 lbs each, the estimated Wr^2 is

$$Wr^2 = 18 \text{ lb.in}^2.$$

Deceleration Torque

$$= T = \frac{Wr^2}{g} \frac{2\pi N}{60t}$$

$$T = \frac{18}{386} \frac{2\pi 19}{0.0055}$$

$$= 1020 \times 4 \text{ wheels} = 4080 \text{ in.lbs.}$$

F on each wheel, at radius of 3" = 340 lbs.

Available force = 1/4 weight of carrier x friction coefficient

= approx 108 x 0.3 = 36 lbs.

Therefore, the wheel will skid.

Kinetic energy of wheels at 30 ft/sec

$$= \frac{1}{2} \frac{Wr^2}{g} \omega^2$$

$$= \frac{1}{2} \frac{18}{144 \times 32} (119)^2$$

$$= 27.8 \text{ ft.lbs.}$$

Kinetic energy of cart

$$= \frac{1}{2} \frac{W}{g} V^2$$

$$= \frac{1}{2} \frac{430}{32} (30)^2$$

$$= 6000 \text{ ft.lbs.}$$

$$\frac{4 \times 27.8}{6000} = 0.19\% \text{ -- Neglect.}$$

14. <u>Volume of Accumulator</u>

Assume thickness of heads = 3/8"

thickness of cylinder = 5/16".

Then from Lukans' handbook,

Volume of heads = 1.31 gallons

$$\text{Volume of cylinder} = \frac{\pi}{4} (15.375)^2 \times 21.75$$

$$= 4000 \text{ cu.in.}$$

$$\text{heads} = 1.31 \times 2 \times 231 = 608 \text{ cu.in.}$$

Total Volume = 4608 cu.in. = 2.6 cu.ft.

B. Instrumentation

1. Velocity Measurement

One of the difficulties of velocity measurement is the need for measuring and adding two separate simultaneous velocities to obtain the desired relative velocity. One way to accomplish this is to mount a pick-up device on one moving element, and a signal source on the other. An industrial type electro-magnetic pick-up can generate a strong signal voltage for direct relative velocity measurement.

The initial choice was Electro Products Laboratories' Model No. 3015, with a signal voltage of approximately 15 volts at 60 ft/sec and 0.005 clearance.

Operating against a 20 pitch rack attached to a lance, the following frequencies were generated:

$$20 \text{ pitch} = \frac{20}{\pi} = 6.37 \text{ teeth/in.}$$

Speed		Frequency	Cycles During Event Min	Cycles During Event Max
30 ft/sec	360 in/sec	2290 cps	11.5	172
40	480	3050	15.2	213
50	600	3820	19.1	286
60	720	4590	23.0	322

These appear reasonable for expected event durations (impact) in the range of 5 to 75 milliseconds. If lance pitches of other than 6.4 teeth per inch are desirable, then other lances with different pitches can easily be substituted.

2. Calibration of Velocity Transducer and Read-Out Equipment

Calibrate velocity measurement system with constant speed motor driving gear of same pitch as rack on velocity lance.

Use 20 pitch, 40 tooth, 2" pitch diameter gear

$$OD = 2.1 \text{ inches .}$$

3. Velocity Unbalance

Except at maximum velocities, the carriages will always carry extra weights. Therefore, if one carriage consistently shows higher velocities, it can be slowed down by dividing the weights asymmetrically, so that the otherwise faster carriage can be accelerated to a slower ultimate speed, thus allowing a closer control over velocity, and of the location of the collision.

4. Calibration of Shock Measurement System

Use Columbia #4000 Cathode Follower and #302 accelerometer and probe.

Effective capacitance of probe

$$= 25' \times 30 \ \mu\mu fd/1 \times (1 - 0.97)$$

$$= 750 \ (0.03)$$

$$= 22.5 \ \mu\mu fd \ .$$

5. Calibration of Accelerometer Read-Out System

$$S_1 = S \frac{C_a + C_c + C_1}{C_a + C_c + C_1 + C_p + C_x}$$

where
- S_1 = sensitivity of the system
- S = sensitivity of accelerometer = 125 pk mv/pk g
- C_a = capacitance of accelerometer = 400 $\mu\mu$fd
- C_c = capacitance of cable = 100 $\mu\mu$fd
- C_1 = capacitance of cathode follower = 37 $\mu\mu$fd
- C_p = effective capacitance of probe = 22.5 $\mu\mu$fd
- C_x = adjustable cathode follower shunt capacitance.

Therefore,

$$S_1 = S \frac{537}{560 + C_x}$$

C_x	Accelerometer mv/g	Maximum Voltage	System mv/g	Maximum G's
0	125	40	120.0	332
500 μμfd	125	40	63.3	630
1000 μμfd	125	40	42.9	935
0.01 μfd	125	40	6.37	6,300
0.1 μfd	125	40	0.67	59,900

III. HISTORY OF CONTRACT PROGRESS

This contract provided for the design, development, manufacture, test and delivery of a special compression test apparatus. During the initial months of the contract the design approach presented in the FRL® proposal was received, and engineering design calculations necessary to establish principal dimensions and operating conditions were performed. This data was presented to and approved by the Project Officer in October 1961. Next, design layouts were prepared and submitted for approval in December 1961, and this approval was secured in January, 1962.

Detailed drawings were prepared and approved in February, 1962, and orders for major sub-contracted parts were placed in March, 1962. Delivery of many of the major parts to the test site was made in April and May, 1962, and assembly and check-out of components and the entire unit as a system was started and continued through July, 1962.

First powered operation of the equipment was accomplished in August, and test operation of the equipment continued until the end of the contract. Minor mechanical modifications were performed as the need for them was indicated during the later phases of the test program.

Final check-out and testing was delayed, pending the installation of a permanent electric service at the test site. This was completed by the Army, and the bunker where the equipment is installed was equipped with heating and ventilating equipment, as well as adequate lighting and electrical power supply before the final check-out of the equipment.

After acceptance test runs were completed, and satisfactory operation of the equipment and reduction of the test output data was satisfactorily demonstrated, the complete equipment and documentation was turned over to the Army to complete requirements of the research contract. The acceptance test took place on 11 June 1963.

IV. CONCLUSIONS AND RESULTS

Completion of this contract has provided a test device which will allow compression impact testing of a variety of materials over a wide range of impact speeds and input energy levels. The machine provides an energy transfer system and an instrumentation system suitable for this wide range, and ruggedly designed and built to be capable of repeated operations in the shock environment imposed by the machine operation.

A program to utilize the equipment in a systematic test program of paper honeycomb and other potential energy absorbing cushioning materials, to characterize their mechanical properties under varying impact conditions, will provide the basis for more precise design analysis and more efficient utilization of these materials in future cushion designs.

APPENDIX I

OPERATING INSTRUCTIONS

COMPRESSION IMPACT TESTING MACHINE

Contract Number DA-19-129-QM-1905

FRL® Case Number C61066

Manufactured by

Fabric Research Laboratories, Inc.

for

U. S. ARMY QUARTERMASTER RESEARCH AND ENGINEERING COMMAND

Natick, Massachusetts

1. DESCRIPTION

This device is built in accordance with specifications established by the Quartermaster Corps, requiring impact speeds over the range of 30 to 60 feet per second, and an energy level of up to 12,000 foot pounds. The machine design conceived by Fabric Research Laboratories for this application comprises two welded steel carriages aligned on rails. The carriages are fitted with heavy impact plates, and the specimen to be tested is secured to the impact face of one carriage. In operation, the carriages are initially positioned at opposite ends of the 40 foot long track, and are propelled toward a collision at the center by the action of two co-ordinated pneumatic cylinders. Energy input to the system is controlled by regulating the pressure in the pneumatic actuators, and impact speed is controlled by adding or removing cast iron weights to adjust the mass of the moving carriages.

A separate pneumatic system operates friction brakes which stop the carriages after the collision and absorb the excess rebound energy.

Instrumentation includes a shock accelerometer, which senses deceleration force during the impact, and an electromagnetic velocity measurement system, which measures relative velocities of the two carriages before, during, and after the deformation of the test specimen. Outputs of the two measured variables are displayed and photographed on a dual beam oscilloscope.

This horizontal rail type machine eliminates the problems of repetitive impact and the awkwardness of high drop towers and large masses required for gravity actuated devices. The use of two opposed balanced moving masses eliminates the shock loads which would be delivered to the building structure and surrounding equipment by a single-mass impact device of the same energy level.

2. PREPARATION FOR TEST

In order to properly plan the test, with an optimum selection of controlled input variables, it is necessary to have some insight regarding properties and expected performance of the test specimen. Controlled parameters are impact speed and input kinetic energy. These are established by the selection of carriage mass, and accumulator gas pressure. Charts in Appendix II show the relationships of these four variables: input energy, carriage mass, impact velocity, and operating gas pressure. It is also necessary to select an operating pressure for the brake system;* this will always be a pressure considerably less than the driving pressure, and will be selected for compatibility with the expected energy absorption of the specimen. In general, if the brake pressure is too high in comparison with the specimen properties, then there will be no friction work done by the brakes, so that when the brake linkages are tensed, the carriages will be subjected to a second motion reversal, and another secondary collided impact. Conversely, if the brake pressure is too low, so that the braking frictional

* 5 to 15 psi depending on the nature of the test specimen.

energy is too little to halt them during the normal braking stroke, then the carriages will continue retreating to the battery position and will be arrested or reversed by compression of residual gas in the operating cylinders. However, most of the typical test specimens are efficient energy absorbers, with a consequence that the rebound energy available for dissipation in the two situations described above is so low that the actions obtained from incorrectly adjusted brake pressures are not violent. Also, it should be noted that the brake system has sufficient energy absorbing potential to safely halt the carriages after a completely elastic impact, in which the test specimen absorbs no energy.

When the operating variables have been selected there are several procedures which must be completed prior to running the test. First, load or remove weights from the two carriages to obtain the desired gross weights. Weight of both carriages should be equal. Use the davit and hoist provided with the equipment to facilitate lifting the weights. Remove the hoist mechanism from the carriages when the weights are in place. Secure the weights firmly against the forward end of each carriage, using the large wrench provided with the equipment.

Next, secure the specimen in place on one of the carriages. For uniform specimens, the center of specimen area should be symmetrical with the center of the carriage impact plate. Non-uniform, or stepped specimen samples should be mounted so as to minimize the eccentricity of the combined forces produced in deformation of the sample. Install the sample securely, using contact adhesive, paper tape, etc. If using adhesive, allow sufficient cure time before operation of the equipment to prevent slipping or lateral displacement of the specimen prior to impact.

3. IMPACT TESTING

Operation of the equipment during a typical impact test can best be described by a numbered check list of all the operations required for such a test:

A. Turn on all electronic equipment, and allow to warm up. This includes:

(1) oscilloscope
(2) cathode follower
(3) frequency converter

B. Retract operating pistons:

(1) Vent gas behind pistons by opening manual bleed valve in tee adjacent to firing valve.

(2) Close cylinder vent ball valves on top of each operating cylinder.

(3) Connect gas supply hose to tank of air compressor unit, or to nitrogen gas bottle. If using bottled gas as a pressure source, use a suitable pressure reducing gas pressure regulator.

(4) Retract the pistons by cracking open the manual valve in the cylinder return line. When both pistons are fully retracted to battery position, close this hand valve. Also close the manual vent valve in the cylinder supply line. Open the vent valves at the top of each operating cylinder.

C. Check to see that the specimen is securely mounted, the hoisting davit is removed from the carriages, the weights are secured in the carriages, there are no loose parts or tools in the carriages, and that the tracks and surrounding space are clear of tools, trailing wires, and other items.

D. Manually roll the carriages back to the battery position, with the padded rear face of the carriages in contact with the bumpers on the piston rods.

E. Check installation of shock and velocity transducers, and their trailing wire connecting cables.

F. Adjust the oscilloscope and associated electronic equipment to position the traces and to provide a time base and signal amplitude consistent with the experiment. (See Section 4 of this Appendix for discussion of instrumentation and calibration.)

G. Install an external triggering wire (or set the oscilloscope for internal triggering) as required for the particular experiment.

H. Check the camera for lens setting and film, and install on the oscilloscope.

I. Turn the control console power supply switch 'ON'. Carriage position and specimen indicator lights should come 'ON', indicating these parts in operating position.

J. Check the position of the brake strips. Friction strips in the brake cylinders should be at the beginning end of their stroke. Engaging strips should be up against their positioning stops.

K. Open the brake air supply valve with the switch on the console and pressurize the brake system to the predetermined pressure level. The yellow indicator light on the panel board should come on. If the brake system is overinflated, it can be vented through the manual valve on the side of the console. Pressure can be increased by opening and closing the brake inlet valve as required.

L. Open the accumulator air valve with the switch on the console, and pressurize the propulsion system. Note that this valve is inoperative unless the brake system has previously been pressurized, and the brake indicator light is 'ON'. Charge the propulsion system to the predetermined pressure level. Observe that the accumulator indicator light is 'ON'. <u>Note</u>: If the propulsion system is inadvertently overpressurized, <u>do not operate</u> the device until the pressure has been reduced to the required level. Excess accumulator pressure can conveniently be vented by manually opening the pressure relief valve in the top of the accumulator tank. This relief valve is pre-set to protect the tank and piping from dangerous overpressurization. Setting of the relief valve can be adjusted downward to prevent inflation of the accumulator tank to a level above that necessary for a particular series of tests, to provide a further safety restraint on the operation of the equipment.

Do not allow personnel to work in the area between the carriages when the accumulator has been charged. If this is necessary, manually move both carriages away from the actuating cylinders, so that an inadvertent operation of the cylinders cannot accelerate the carriages. Also, close the cylinder vent valves when the carriages are away from the cylinders, to protect the cylinders themselves from hammer blows which would be possible if they were fired without the external load offered by the carriages. Keep personnel clear of the equipment when charged and ready to fire!!

M. Make a final check before firing. Read brake and accumulator pressure gauges for correct pressure levels. Check that indicator lights on panel board are 'ON'. Check that vent valves on propulsion cylinders are OPEN with valve handles UP. Caution! This is important, for if one of these valves were closed, the carriages would be accelerated unequally, and impact of the carriages would not occur at the center of the track span. Check oscilloscope for location of traces and triggering action. Reset oscilloscope controls to operating position.

N. Open camera shutter.

O. Turn firing switch to FIRE position.

P. After impact, close camera shutter. Turn firing switch to RESET position.

Q. Develop and remove oscillograph from camera. Record operating data with any appropriate notes and identification of oscillograph. (Note: It is suggested that a consecutively numbered log of test machine shots be kept, to provide a history of operational data and allow reproducing test conditions and providing a timetable and record for maintenance and adjustment.)

R. Remove the test specimen.

S. Vent pressure in brake and propulsion cylinders.

T. Prepare for next succeeding test.

4. INSTRUMENTS AND CALIBRATION

General arrangement of the instrumentation system is shown in the block diagram. Refer to manufacturer's literature for data and instructions regarding the major instrument components, including camera, oscilloscope, cathode follower, accelerometer, etc. Discussion of the instrumentation will consider the following:

A. Force measuring system
B. Velocity measuring system
C. Displacement measuring system
D. Oscilloscope triggering
E. Camera operation
F. Calibration of systems.

A. Force Measuring System

Sensory unit for the force measuring system is a Columbia Model #302 accelerometer, which is mounted to the rear face of the impact pad of either of the two carriages by using the threaded stud and one of two threaded holes either above or below the large threaded rod. Observe the manufacturer's recommendations on installation torque of the accelerometer. Also mounted on the same carriage will be the CFP LN-25 cathode follower probe. (A cylindrical enlargement approximately 1" long and 1/2" diameter at the end of the extension cable.) This probe should be secured in place on the carriage, using glass reinforced tape, so that it will not be dislodged by the high impact shock forces. The accelerometer and probe should be connected with the special 3' long low noise cable assembly. The cable must be positioned to clear the carriage weights, and must also be securely taped in place to prevent cable whip, and also be supported to prevent inertia loads on the microdot cable connectors.

The extension cable from the probe will also serve as a trailing cable to the carriage. The end of this cable adjacent to the probe must be securely taped to the carriage to prevent inertia loads on the probe connection, and then a suitable length of the cable is festooned to an overhead support, and from there down to the stationary cathode follower. The cathode follower output signal is introduced into one channel of the oscilloscope with a suitably terminated shielded microphone cable. The cathode follower sensitivity adjustment permits attenuating the accelerometer signal so that large acceleration shocks can be observed. The cathode follower will also clip the top of input pulses with a voltage in excess of 80 volts so that the best system adjustment will utilize the least value of cathode follower shunt capacitance for signal attenuation that is possible without risk of clipping the signal pulse, and then use a maximum oscilloscope gain consistent with fitting the pulse height onto the available space on the oscilloscope tube. Recommended settings are shown in the calibration section.

A secondary force measuring device is a small, re-setting, spring-and-mass accelerometer, which mounts in a fuse clip on one carriage, and affords a rough measure of peak G level sustained during the impact. This can be re-set after use by un-threading the end plugs, and moving the indicator back to zero. When installing this sensor in the fuse clips, be sure that it is aligned properly end-for-end; if it is backwards it will record the maximum of either the acceleration produced by the propulsion cylinders or the acceleration produced by specimen rebound, whichever is the larger quantity.

B. Velocity Measuring System

The velocity transducer, like the force transducer, is a self-generating device not requiring external excitation or power supply. Primary sensor is an ElectroProducts Laboratories' model 3015-A magnetic pick-up, which generates an AC signal with voltage and frequency depending on the magnetic field variation which is seen by the transducer. One of the two moving carriages mounts the transducer, and the other mounts a "velocity lance" which is merely a toothed rack of magnetic material so arranged that relative motion of the two carriages drives the rack past the magnetic pick-up. Pitch of the rack is 20, resulting in a pitch length of π + 20, or 0.157". Thus the transducer output frequency is proportional to relative velocity, and, during the time of impact, the output wave-length is equal to 0.157" of specimen displacement. To improve the data presentation, the frequency signal from the transducer is fed to modifying circuitry in the frequency converter, which in turn generates a DC signal proportionate to the incoming frequency. This voltage appears on the oscilloscope tube as a voltage analog of relative carriage velocity.

If desired, the converter circuit can be by-passed, and the transducer output can be displayed directly on the oscilloscope. In this presentation, the velocity is determined by counting frequency of the displayed signal.

A second rack, with a pitch length of 0.375", has been provided to use with higher velocity impacts. This lance will allow a frequency output with a greater spacing between pips, to facilitate counting pips of higher frequency, and to enable measurement of velocity and displacement at higher operating speeds.

C. Displacement Measuring System

Best means of determining displacement is to display an unmodified AC signal from the velocity transducer. In that portion of the trace which corresponds to the impact event, each complete period of the wave form is equivalent to 0.157", or 0.375", of displacement, so that the total displacement can be obtained by counting periods of the wave form. If the frequency displayed on the oscilloscope photograph is too high to facilitate counting, then an easier to read, but less precise signal can be obtained by replacing the velocity lance with a coarser pitch rack, in which case the velocity and displacement calibration must be recalculated for the new conditions.

D. Oscilloscope Triggering

A simple scheme for oscilloscope triggering utilizes a switch which is actuated by passage of the carriages. A simple form for such a switch consists of two leaves of spring brass, normally separated by an air gap, and insulated from the track with Mylar or similar material. This forms a switch which can lay flat on the track for actuation by a passing wheel, is unharmed by the weight of the carriage, can readily be positioned along the track to most advantageously time the triggering action, and is thin enough to cause no undesirable transverse motion of the carriage.

Another triggering switch is a microswitch, operated by a blade-shaped extension. This switch is fitted with a magnet, to permit its mounting anywhere on the track length so that the blade is actuated by passage of the carriage wheel, and the position of the switch determines timing of the oscilloscope triggering.

A single sweep lock out mode of operation of the oscilloscope prevents undesirable multiple triggering and multiple sweep traces.

E. Camera Operation

Camera operation is best described by the manufacturer's literature, which provides information on lens settings, etc. The oscilloscope provided with the equipment has a P-11 phosphor, best for photographic purposes, and the graticule illumination should be adjusted for white light, rather than red. Good results were obtained with Polaroid 10 second film during check-out of the equipment.

F. Calibration of Systems

A calibration of the force measuring system, using the Columbia #4000 Cathode Follower, and #302 accelerometer and probe is established by the following procedure:

$$\text{Sensitivity of the system} = S_1 = S\,\frac{C_a + C_c + C_x}{C_a + C_c + C_1 + C_p + C_x}$$

where

S = manufacturer's calibration sensitivity of accelerometer = 125 peak mv/peak g

C_a = capacitance of accelerometer = 400 μμfd

C_c = capacitance of microdot cable = 100 μμfd

C_1 = capacitance of cathode follower = 37 μμfd

C_p = effective capacitance of probe

C_x = additional adjustable shunt capacitance of cathode follower.

The value of C_p, the effective capacitance of the probe and extension cable, is calculated as 25' x 30 μμfd/ft x (1 - 0.97)

$$= 750 \times (0.03)$$

$$= 22.5 \ \mu\mu fd \ .$$

Therefore $$S_1 = 125 \times \frac{537}{560 + C_x} \ .$$

Calibration of the system at various shunt settings is tabulated as follows:

C_x	Accelerometer mv/g	Maximum Voltage	System mv/g	Maximum g's at 40v
0	125	40	120.0	332
500 μμfd	125	40	63.3	630
1000 μμfd	125	40	42.9	935
0.01 μfd	125	40	6.37	6,300
0.1 μfd	125	40	0.67	59,900

This calibration data is tabulated at various oscilloscope gain settings to show overall system calibration, in terms of G's per cm of pulse height, in a separate table in the appendix.

The foregoing table is based on an unfiltered input to the oscilloscope. If the filter provided with the equipment is used to minimize the extraneous high frequency noise from the force signal, then the signal will be further attenuated to a signal voltage level shown in the following calibration table:

Acceleration System Calibration - G's per Cm
(With FRL Band Pass Filter, and based on Calibration Data of 20 June 1963)

Shunt	Oscilloscope Gain 100 mv/cm	200 mv/cm	.5 v/cm	1v	2v	5v	System mv/g
0	8.6	17.2	43	86	172	860	11.6
500 μμfd	16.1	32.2	85	170	322	1,610	6.2
1000 μμfd	24	48	120	240	480	2,400	4.2
.01 μfd	161	322	805	1,610	3,220	16,100	0.62
.1 μfd	1016	2030	5080	10,160	20,320	101,600	0.063

The velocity system is calibrated by mounting the magnetic transducer adjacent to a 2" diameter steel gear driven by a constant speed motor, and displaying the output voltage so produced on the oscilloscope. Linear velocity of the O. D. of the gear is 31.6 ft/sec. This provides for the direct calibration of the oscilloscope pulse height in feet per second, for any oscilloscope gain setting. Voltage adjustment on front of frequency converter is not a gain control, but is an adjustment for power supply voltage and should be set at 28 volts shown on the meter of the instrument.

5. MAINTENANCE AND ADJUSTMENT

Operating time of the moving parts of this device is short so that wear of parts in the usual sense is not a factor. However, the machine does produce considerable shock and vibration, and should be checked for loose bolts, etc. Maintenance instructions for the principal manufactured component parts are discussed in the manufacturer's data regarding these parts. In addition, a few items regarding adjustment and maintenance of the equipment should be noted.

A. Lubrication

The air cylinders will be adequately lubricated by oil from the compressed air line or present in the oil pumped nitrogen gas used for pressurizing the system, and should not require additional lubrication. Efficiency of the machine will be improved if the carriage casters and tracks and other surfaces bearing on the moving carriages are lightly oiled. The trailing brake arrester hook should run on an oiled brake strip, but care should be taken that oil or grease is kept off the friction brake strips which are engaged by the brake cylinders.

B. Adjustments

(1) When using bottled gas as a pressure source, the pressure reduction valve at the gas bottle should be adjusted so as to limit the pressure available for entrance into the machine system to a level not dangerously higher than the highest pressure contemplated for use in the tests in process.

(2) The pressure relief valve in the accumulator tank can be adjusted to provide an additional high pressure limit. Maximum design pressure of the pressure holding elements is 300 psi, which is considerably in excess of normal requirements for energy output of the machine. The relief valve can be adjusted to vent at any pressure up to this level. Have the piston rods extended when inflating the tank to set this valve, to eliminate any possibility of actuating the cylinders at an excessively high pressure level.

(3) The pressure switches in the rear of the console can be adjusted with a screw driver. A spring scale on the side of the switch shows the approximate adjustment level, and the setting can be checked more closely

by observing the appropriate gauge reading on the face of the console instrument panel.

(4) The remotely operated firing valve is a ball valve remotely operated by a pneumatic cylinder which is linked to the ball valve and turns it through a 90° arc. Adjustment screw stops are provided at both extremes of travel of the valves and should be adjusted to provide a maximum clear opening in one position, and zero leakage in the closed position of the valve.

(5) The main operating cylinders are fitted with oversized cushioned strokes at the rod ends of the cylinders. These are factory set to provide optimum deceleration of the moving piston during the working stroke of the cylinders, and have a needle valve adjustment which can be reset if desired. No cushions are provided or are necessary for the slow speed return stroke of these cylinders. The short stroke of the brake cylinders also does not require cushioning. Note that the ports on the rod ends of the brake cylinders are plugged. This traps a small volume of air on the rod side of the piston, and assures a positive release of the brakes when the brake pressure is released.

(6) The velocity lance should be positioned for easy entrance into the trumpet-shaped guide with the least possible transverse motion and vibration. The magnetic pick-up should be positioned as close as possible to the teeth of the rack, but without danger of an interference.

(7) Adjustment of the carriage mass is obtained by adding or removing fifty pound weights. Normally, the same number of weights are mounted on each carriage, in order to balance the device, and to provide for a collision at the center of the track span. However, if one carriage habitually leads the other, then this can be slowed down by adding one or a fraction of one additional weight to this carriage.

(8) The pressure gauges are fitted with a left-hand helicoid movement, which insures that a sudden reduction in pressure will not slap the pointer against the zero stop and lose the gauge calibration. If the gauges do become obviously out of calibration they should be removed from the panel and re-calibrated against a master gauge.

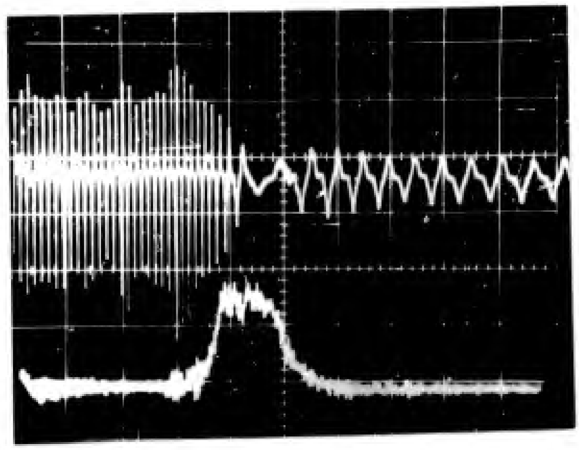
(9) The main pressure regulator should be set at 250 psi, to avoid overpressurizing the equipment. A second regulator in series with this is set at 100 psi, and limits pressure to the brake system and to the firing valve actuating cylinder. A small needle valve in the brake cylinder gas supply line limits the flow in this line so that it is possible to more slowly and conveniently raise the pressure in this small-volume system than would otherwise be possible.

C. Miscellaneous

(1) Drain the gas accumulator of condensation at intervals as necessary, using the stop-cock in the bottom of the tank. Any condensate accumulating at low places in the hose connections can be drained by disconnecting the hose and allowing the condensate to run out.

(2) The hose and pipe connections are subject to some shock and vibrations, and should be checked for tightness at intervals to prevent air leaks in the equipment.

(3) Means are provided for easy removal of the carriages from the rails to facilitate any needed repairs or maintenance. These fixtures are required: a wooden horse, a wooden extension beam, and a dolly with casters. To remove a carriage, first remove the operating cylinder and cylinder mounting bars at the appropriate end of the track. Position the horse about five feet from the end of the track, and place the beam in a level position between the horse and the cross member of the end track trestle. Place the wooden dolly on the inboard end of the beam. Roll the carriage over the dolly, and continue rolling carriage and dolly back until the carriage is supported on the beam free of the tracks and is accessible for service.



Typical Oscillograph from Impact Test Instrumentation

The above oscillograph was obtained from impact test of two pieces of expanded paper honeycomb, approximately 18" x 18" x 3" thick. Machine input and instrumentation settings are listed:

Acceleration Pressure	40 psi
Brake Pressure	10 psi
Carriage Weight (empty)	550 lbs each
Specimens	2 pieces 18" square paper honeycomb x 3" thick, both mounted on one carriage
Velocity fence tooth spacing	0.375"
Oscilloscope sweep rate	10 millisec/cm
Velocity trace gain setting	2V/cm
Acceleration trace gain setting	200 millivolts/cm
Cathode follower setting	1
Acceleration trace low pass filter	yes
Velocity trace frequency-to-voltage converter	not used

Test information is reduced to numerical form as follows:

1. Approach velocity.

a. Count the number of velocity pips per cm of oscilloscope grid in the portion of the trace prior to impact. In the example there are eight pips per centimeter.

b. Eight pips per centimeter at a sweep rate of 10 millisec per cm equals 800 pips per second.

c. 800 pips per second with a velocity lance tooth spacing of 3/8" equals a velocity of 300 inches per second, or 25 feet per second. This is the combined velocity of both moving carriages; each has an absolute velocity of approximately half this value.

2. Rebound velocity.

a. Pip spacing after the event is two pips per centimeter.

b. Two pips per centimeter at sweep rate of 10 ms = 200 pips per sec.

c. 200 pips per sec with a tooth spacing of 3/8" equals a velocity of 75 inches/sec, or 6.25 ft/sec. This is combined rebound velocity of the two carriages; each has a velocity approximately half of this total.

3. Input Kinetic Energy.

$$E = \frac{1}{2}\frac{W}{g} V_a^2$$

where W = two carriages at 550 lbs each = 1100 lbs

V_a = approach velocity of each carriage = 12.5 ft/sec

g = 32.2ft/sec^2

$$E = \frac{1}{2}\frac{1100}{32.2}(12.5)^2$$

$$= 2{,}660 \text{ ft-lbs}$$

4. Rebound Kinetic Energy.

$$E = \frac{1}{2}\frac{W}{g} V_r^2$$

$$E = \frac{1}{2}\frac{1100}{32.2}(3.13)^2$$

$$= 166 \text{ ft-lbs}$$

5. Energy absorbed in specimen equals the difference,

$$2660 - 160 = 2500 \text{ ft-lbs} .$$

6. Crushing Distance.

This is determined by counting the number of pips from the start of the impact until the system comes to rest. Time of the beginning of impact is determined from the initial rise point of the acceleration trace. Time of zero velocity is determined from break in frequencies of velocity curve. In the example, there are nine pips occurring during this crushing stroke, so that the crushing distance is $9 \times 3/8" = 3\text{-}3/8" \pm 3/16"$ (1/2 pip).

7. Crushing Force.

Average height of force signal in this example is 1.4 cm. Calibration of the force measuring system at the settings used is 17.2 g's per cm, so that the deceleration force on each of the two carriages is calculated at

$$1.4 \text{ cm} \times 17.2 \text{ g's/cm} = 24 \text{ g's} .$$

It should be noted that the crystal accelerometer used as a pickup in this system is subject to some variation in output signal with time, and must be recalibrated at intervals on a suitable shake table.

Force is calculated from deceleration as

$$F = ma = \frac{W}{g} a$$

$$F = \frac{550}{32.2} \times 24 \, (32.2)$$

$$F = 13{,}200 \text{ lbs} .$$

8. Average crushing stress, with 18" square specimen

$$= \frac{F}{A} = \frac{13{,}200}{2.25} = 5860 \text{ lbs/sq.ft.}$$

$$= 40.5 \text{ lbs/sq.in.}$$

APPENDIX II

A. List of Drawings and Tables

1. Photograph of Impact Tester
2. Operations Log Sheet
3. Curve-Energy vs Velocity
4. Piping Schematic - Drawing No. 61066-30
5. Wiring and Cabling Diagram - Drawing No. 61066-28
6. Control Schematic - Drawing No. 61066-20
7. Instrument Block Diagram - Drawing

B. <u>List of Manufacturer's Data - Component Parts</u>

(These documents submitted to Army Project Officer. Copies not available for inclusion in report.)

1. Instruction Manual - Type 502 Oscilloscope
2. Instruction Manual - Type 4000 Cathode Follower
3. Calibration Curves - A 1-1 Acceleration Sensor
4. Data Sheets - Miller Pneumatic Cylinders
5. Instructions - Helicoid Pressure Gauge
6. Instructions - G.E. Machine Tool Relays
7. Data Report on Unfired Pressure Vessels
8. Manufacturer's Data - Pioneer #TA-F Converter
9. Service Manual - Type 196-A Oscilloscope Camera
10. Lubrication Instructions - Sealmaster Bearings
11. Calibration Data - Columbia 302 Accelerometer

OPERATIONS LOG - COMPRESSI

TEST No.	SPECIMEN DESCRIPTION	DESIGN ENERGY	DESIGN VEL	COMB. WT	BR. PR.	ACC. PR.	VELOCITY MEASUREMENT					BR... MO...		...RCE	
							CAL	APP.	RBO	APP	RBO			PK	F
		Ft Lbs	Ft/Sec	LBS	PSI	PSI	Ft/sec/Cm	CM	CM	Ft/Sec	Ft/Sec			CM	G

\- COMPRESSION IMPACT TESTER

UREMENT		BRAKE	FORCE			'E'	'E'	E	COMMENTS	COMMENTS
APP	RBD	MOTION	CAL	PK	F	APP	RBD	ABS	SPECIMEN	MACHINE
Ft/SEC	Ft/SEC		Gs/CM	CM	G	Ft Lbs	Ft Lbs	Ft Lbs		

2

FABRIC RESEARCH LABORATORIES, INC.
DEDHAM, MASSACHUSETTS

FRL

DRAWING NO.

CLIENT

CASE NUMBER

DATE

DRAWN BY

CHECKED

APPROVED

SCALE

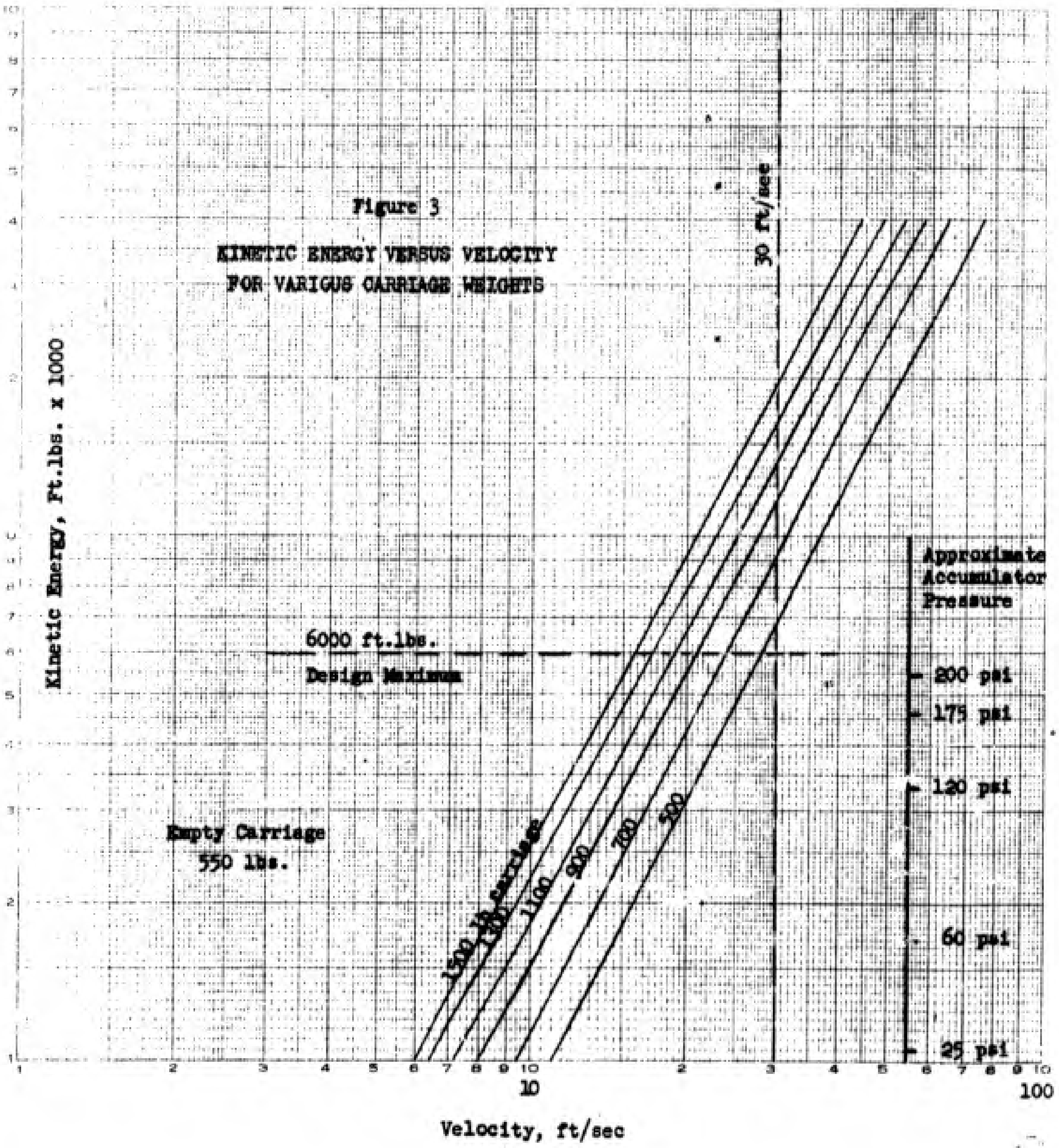
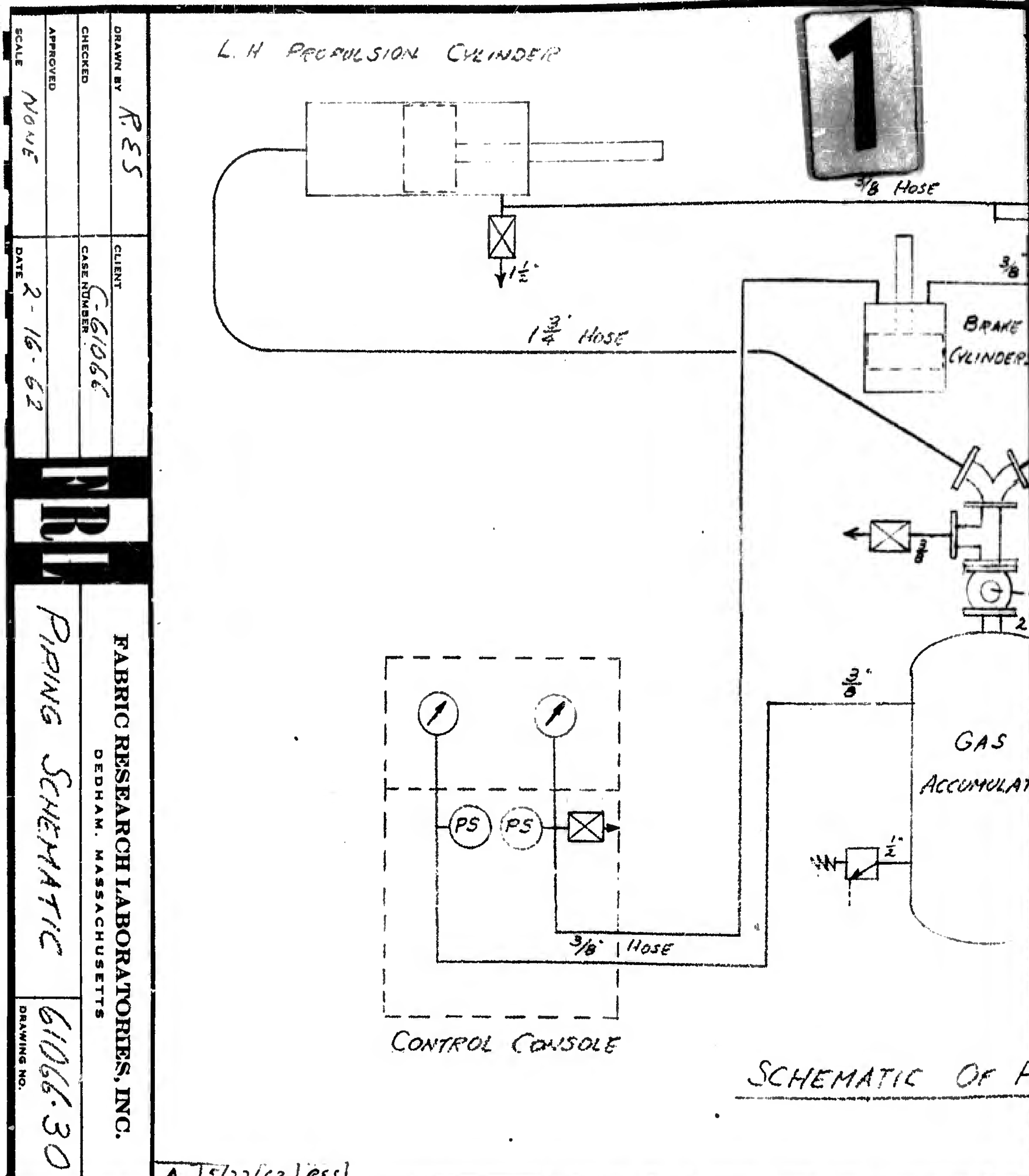
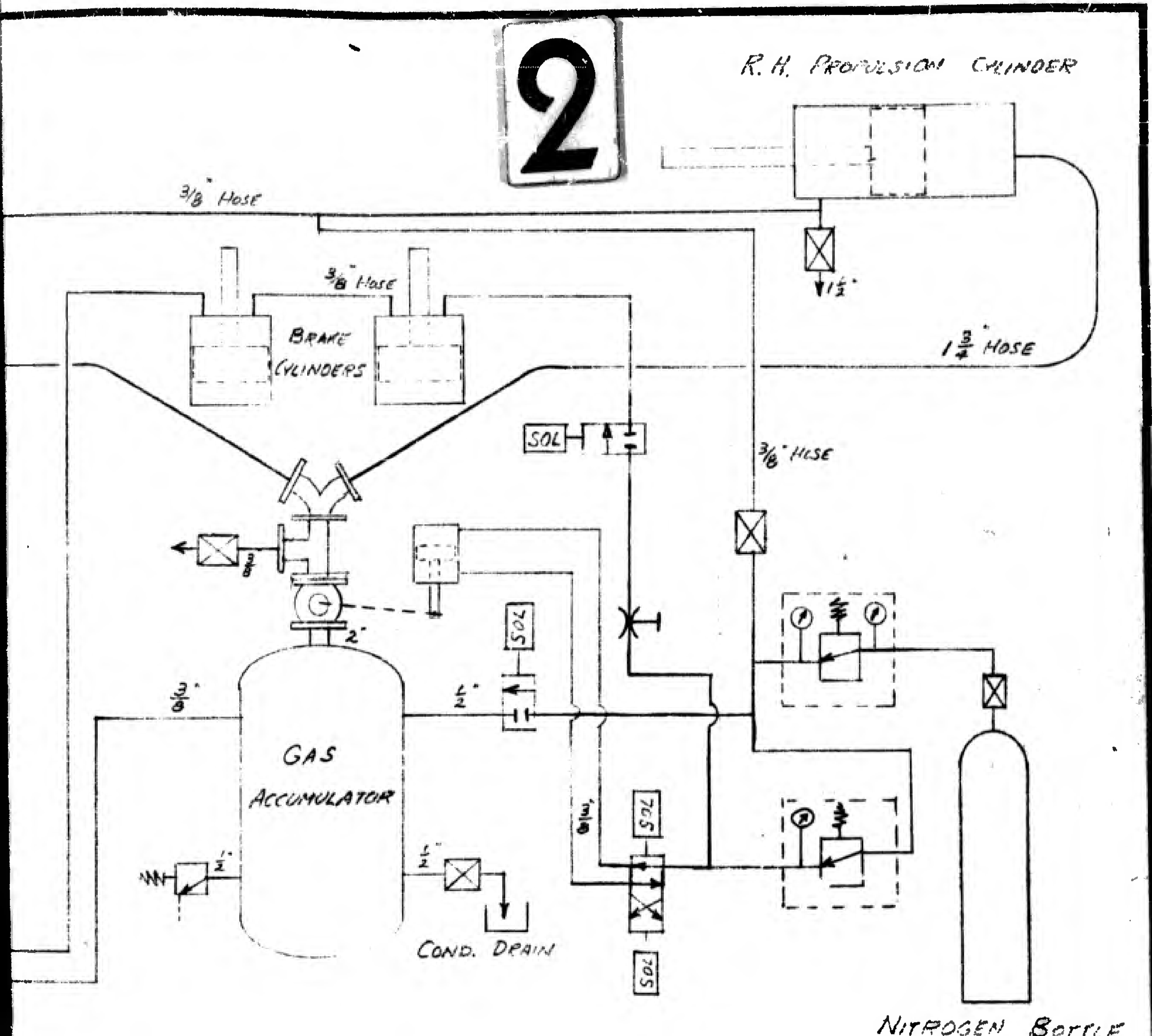


Figure 3

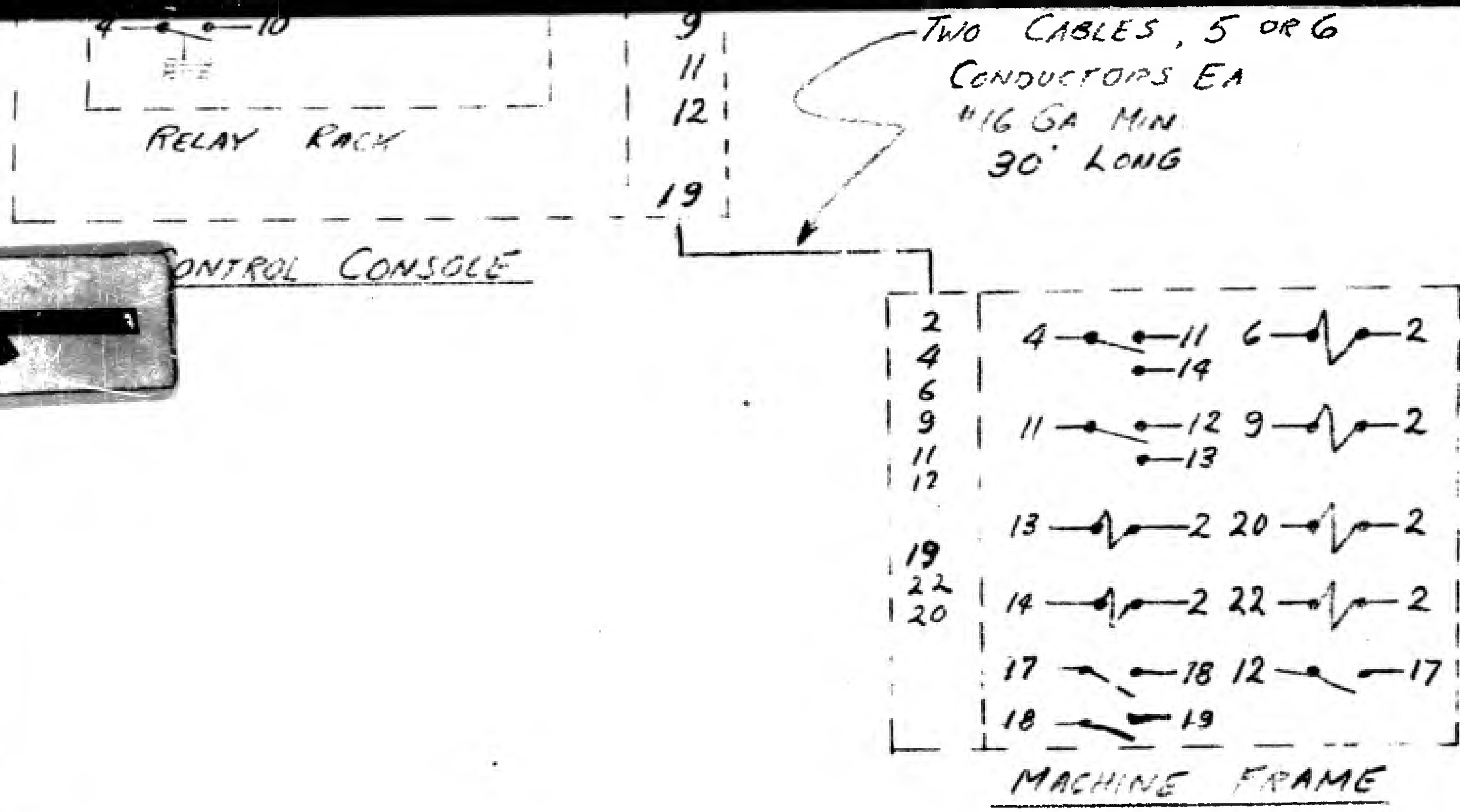
KINETIC ENERGY VERSUS VELOCITY FOR VARIOUS CARRIAGE WEIGHTS



1
L. H PROPULSION CYLINDER
3/8 HOSE
1 1/2"
3/8"
BRAKE CYLINDERS
1 3/4" HOSE
3/8
2
3/8"
GAS ACCUMULAT
1/2"
PS
PS
3/8" HOSE
CONTROL CONSOLE
SCHEMATIC OF P
A 5/22/63 RES
DRAWN BY R.E.S
CHECKED
APPROVED
SCALE NONE
CLIENT
CASE NUMBER C-61066
DATE 2-16-62
FRL
FABRIC RESEARCH LABORATORIES, INC.
DEDHAM, MASSACHUSETTS
PIPING SCHEMATIC
DRAWING NO. 61066-30

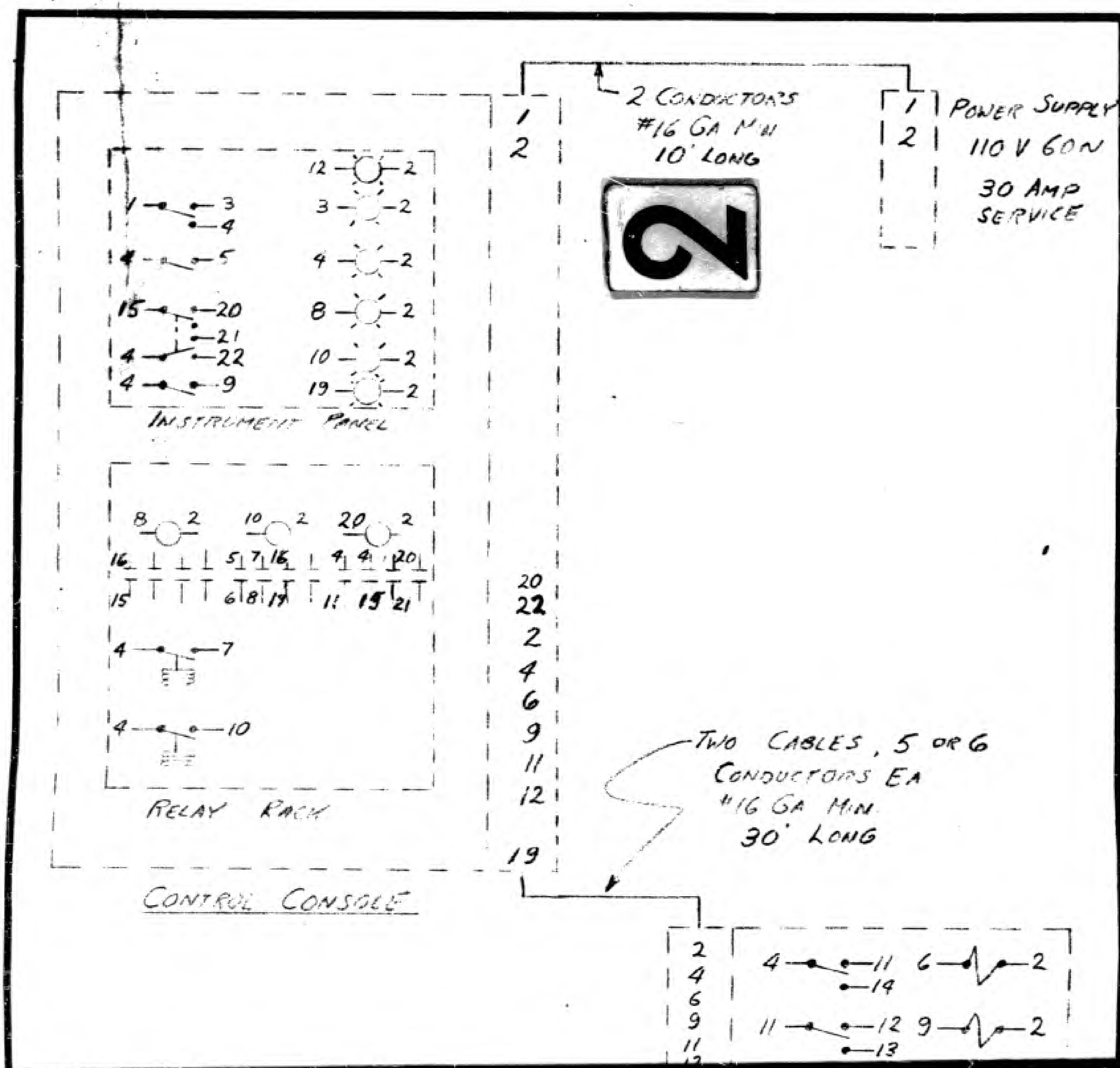


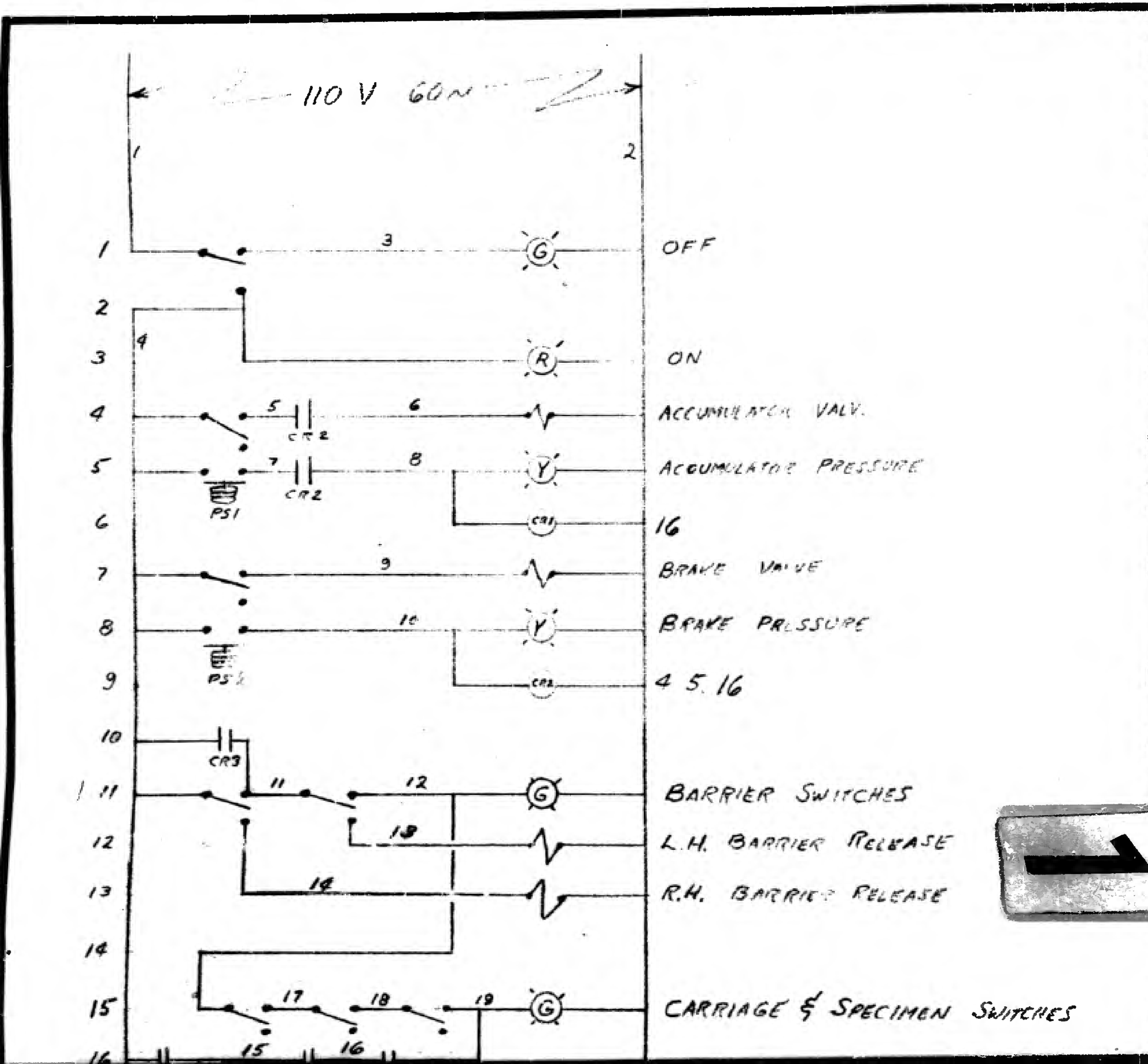
SCHEMATIC OF PNEUMATIC SYSTEM

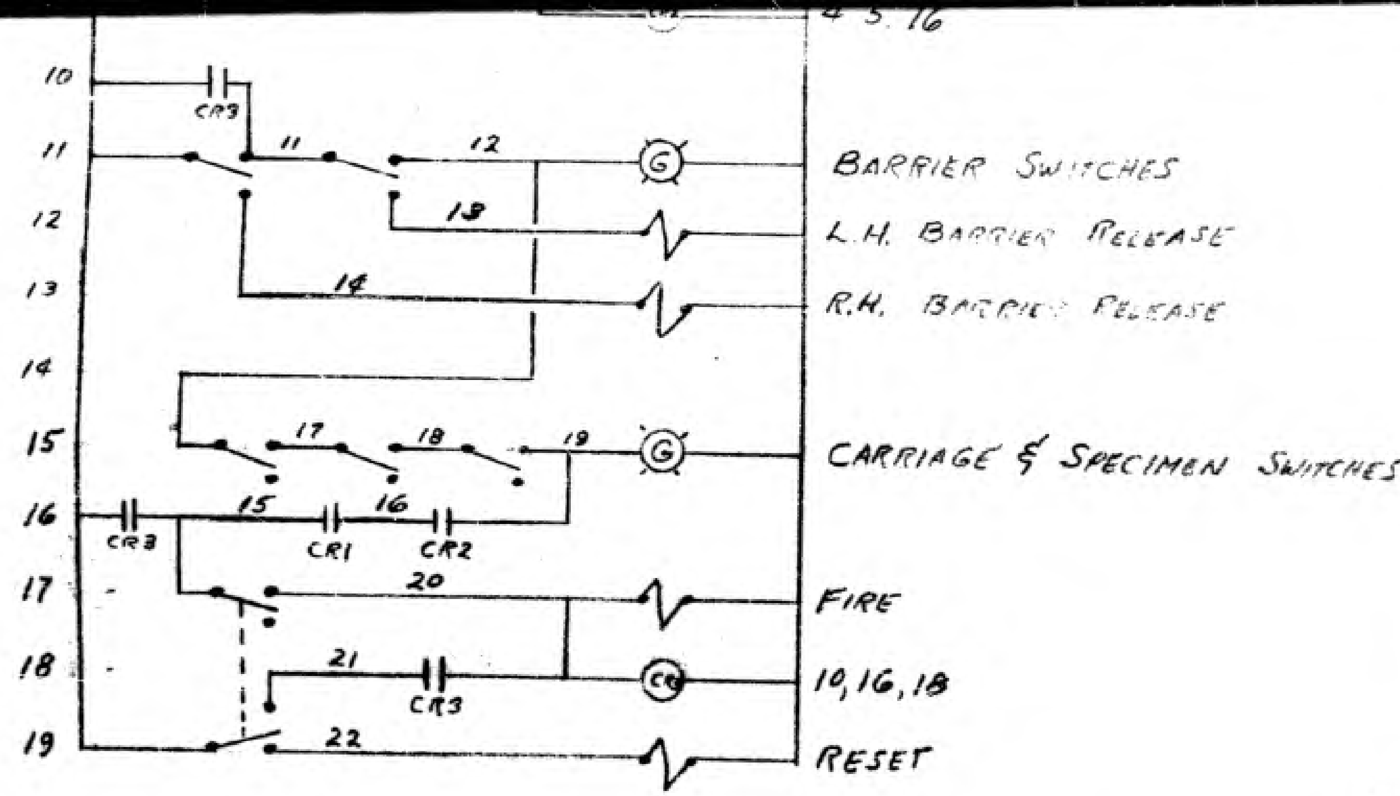


No	BY	DATE	CHANGE
(B)	RES	3/9/62	" "
(A)	RES	3/8/62	CHANGED CIRCUIT

DRAWN BY RES	CLIENT	**FABRIC RESEARCH LABORATORIES, INC.** DEDHAM, MASSACHUSETTS	
CHECKED	CASE NUMBER C-61066		
APPROVED		WIRING & CABLING	DRAWING NO. 61066-28
SCALE —	DATE 2/10/62		


2 CONDUCTORS #16 GA MIN 10' LONG
POWER SUPPLY 110 V 60 ~
30 AMP SERVICE
INSTRUMENT PANEL
RELAY RACK
CONTROL CONSOLE
TWO CABLES, 5 OR 6 CONDUCTORS EA #16 GA MIN. 30' LONG
2

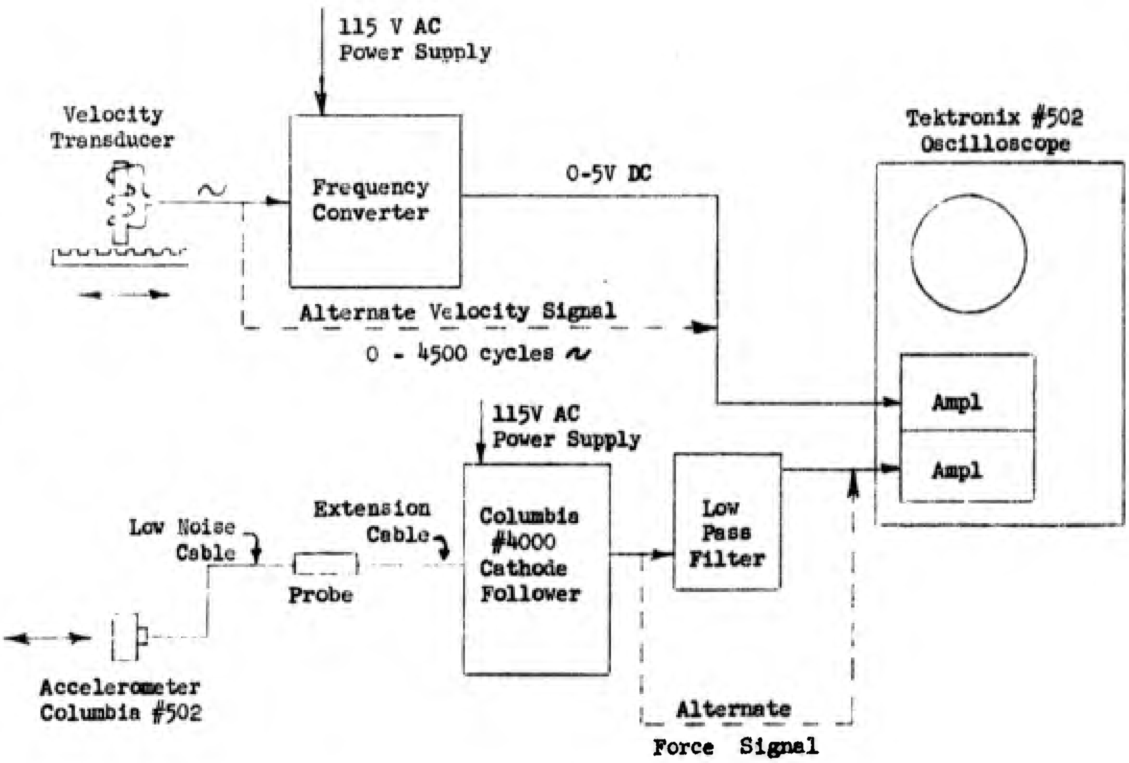

110 V 60~
OFF
ON
ACCUMULATOR VALV.
ACCUMULATOR PRESSURE
16
BRAKE VALVE
BRAKE PRESSURE
4 5 16
BARRIER SWITCHES
L.H. BARRIER RELEASE
R.H. BARRIER RELEASE
CARRIAGE & SPECIMEN SWITCHES
PS1
PS2
CR2
CR3



2

No	By	Date	Change
Ⓑ	RES	3/9/62	" "
Ⓐ	RES	3/8/62	Changed Circuit

DRAWN BY RES	CLIENT	FRL	FABRIC RESEARCH LABORATORIES, INC. DEDHAM, MASSACHUSETTS
CHECKED	CASE NUMBER C-61066		
APPROVED			CONTROL SCHEMATIC
SCALE	DATE FEB 8 1962		DRAWING NO. 61066-20



BLOCK DIAGRAM

OF INSTRUMENTATION SYSTEM

Figure 7

UNCLASSIFIED

UNCLASSIFIED