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Basic Design Features of a Nonimpacting, Pneumatically Driven, Hydraulically Damped High Speed Tester

R. D. SPANGLER

Eastern Laboratory, Explosives Department, E. I. du Pont de Nemours & Company, Inc., Gibbstown, New Jersey

INTRODUCTION

In the design of a high speed constant velocity tensile tester, one is faced with a number of problems not the least of which is how one attains the desired high velocity V and minimizes the effect of the transition velocity from 0 to V on the test results. In such designs one of two approaches is usually followed: (*1*) The machine is brought up to speed and one end of the test specimen is engaged after attainment of the desired velocity. An example is the flywheel-type tester. (*2*) In the second approach the test specimen is placed without slack in a machine and one end of the specimen brought to the desired velocity V by a machine starting from rest. This is often done by driving a piston pneumatically or hydraulically. The time required to attain constant velocity in such a machine is important. This time may be short enough to constitute impact loading or it may be so long that the test is over before constant velocity is attained. This time depends on the nature of the material and the length of the test specimen. It is the purpose of this paper to assess some of the factors affecting the velocity rise time in a pneumatically driven, hydraulically damped high speed tester.

THE ROLES OF MASS, FORCE, AND DAMPING IN TESTER DESIGN

Consider the basic elements of such a machine, Figure 1, consisting of a piston of area A, a draw rod, and a damping plate of total mass M' driven by a constant pressure P, and damped by a fluid of viscosity η. If x is the displacement of the piston measured downward the equation of motion of the piston is

$$M(d^2x/dt^2) + C(dx/dt) = F \qquad (1)$$

where M is the total mass set in motion including the effective mass of the fluid in the holes of the damping plate which is at a velocity greater than the plate itself (this point will be discussed later); C is the damping constant which depends on the viscosity of the fluid and on the number, length, and area of the holes in the damping plate (see following section); and F is the resultant force acting downward on the piston. Thus, $F = PA - f'$ where f' includes all frictional forces.

By setting $v = dx/dt$, and $dv/dt = d^2x/dt^2$, eq. (1) becomes

$$M(dv/dt) + Cv = F \qquad (2)$$

or $dv/(F/C - v) = (C/M)dt$ which on integrating and setting $v = 0$ when $t = 0$ becomes

$$v = F/C(1 - e^{-(C/M)t}) \qquad (3)$$

This equation furnishes the basis for the design of a high speed tester. A constant velocity $V = F/C$ will be attained when $(C/M)t$ becomes large. Thus the action of the machine is completely determined by the three parameters F, C, and M over which the design engineer has considerable choice. The terminal constant velocity is given by the ratio F/C. The time to acquire this velocity depends on the ratio M/C. When $t = M/C$ the velocity v becomes 63.3% of V. This value of t is defined as the time constant τ of the machine.

Figure 2 is a plot of eq. (3) for reasonable attainable values of C, F, and M. Also included in Figure 2 is a plot of a displacement-time curve for the same values of the parameters. The equation for this displacement-time curve is obtained by replacing v with dx/dt in eq. (3) and integrating. The equation is

$$x = F/C\{t - (M/C)[1 - e^{(-C/M)t}]\}$$
$$= (F/C)t - (M/C)v \qquad (4)$$

These curves show that it is possibie to build a machine which will attain essentially constant ve-

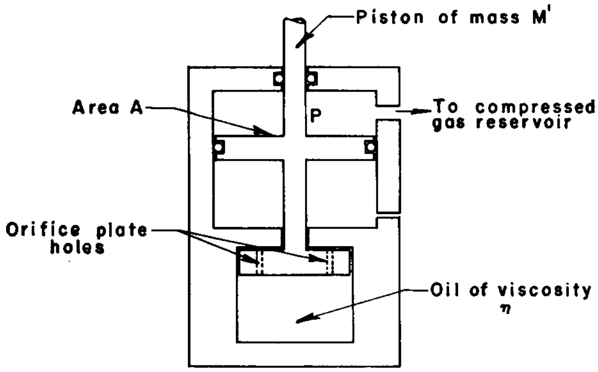

Fig. 1. Elements of a high speed tester.

locity of about 50 ft./sec. (36,000 in./min.) in about 0.3 msec. and that the piston of the machine will have traveled about 0.06 in. before attaining this velocity.

THE DAMPING CONSTANT

Damping can be obtained by driving a fluid of viscosity η through small holes in a plate such as illustrated in Figure 1 and pictured in Figure 3. For such a case the damping constant is given by the equation

$$C = 8L\eta A_1^2/r^2A \tag{5}$$

where L is the length of the holes in the plate, A_1 is the area of the plate, r is the radius of the holes, and A is the total area of the holes. This equation follows from a consideration of Poiseuille's law which states that the pressure p between the ends of a tube of length L and radius r transmitting a volume q of fluid per unit time is

$$p = 8\eta Lq/\pi r^4 \tag{6}$$

The resultant force on the plate required to produce this flow is

$$F = pA_1 = 8\eta LqA_1/\pi r^4 \tag{7}$$

If there are N holes in the plate the total rate of flow is

$$Q = Nq = A_1V \tag{8}$$

where V is the velocity of the piston, Figure 1. From eq. (8),

$$q = A_1V/N \tag{9}$$

Also

$$N\pi r^2 = A \tag{10}$$

so that

$$F = \frac{8\eta LA_1^2V}{Ar^2} \tag{11}$$

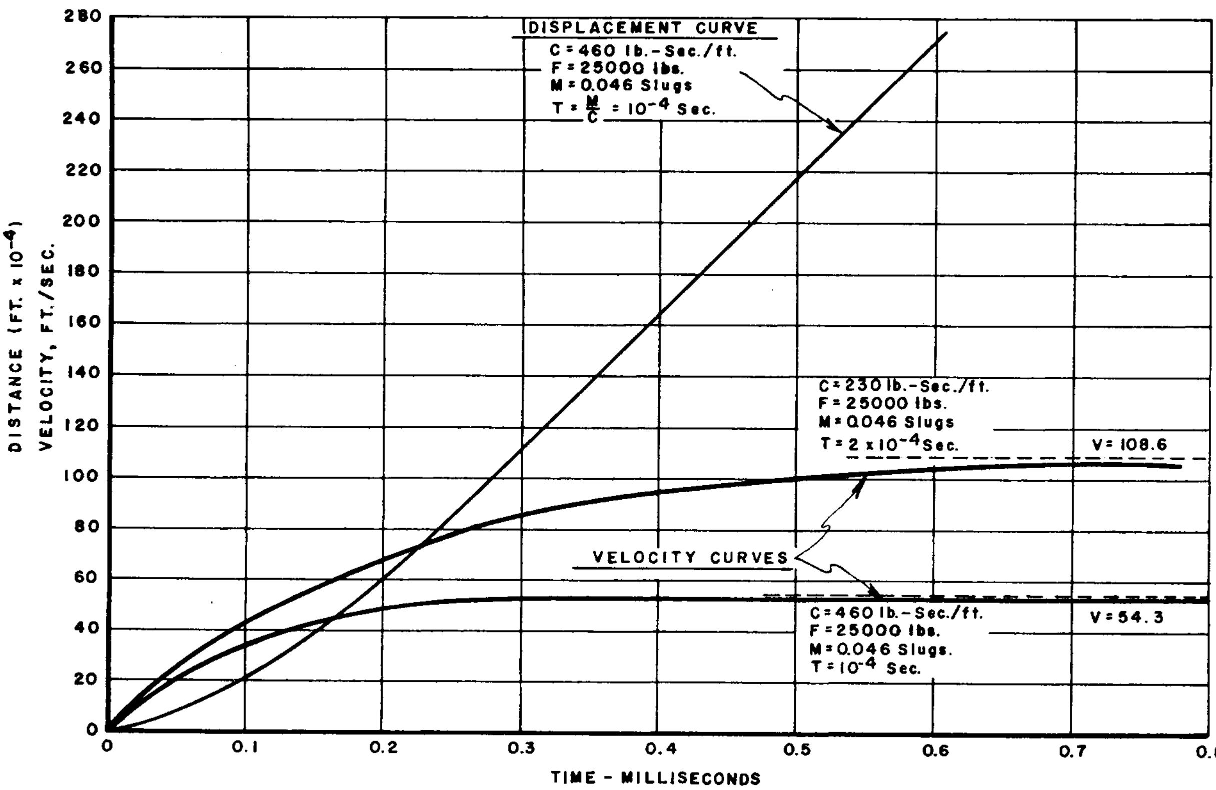

Fig. 2. Velocity and displacement vs. time curves (theoretical).

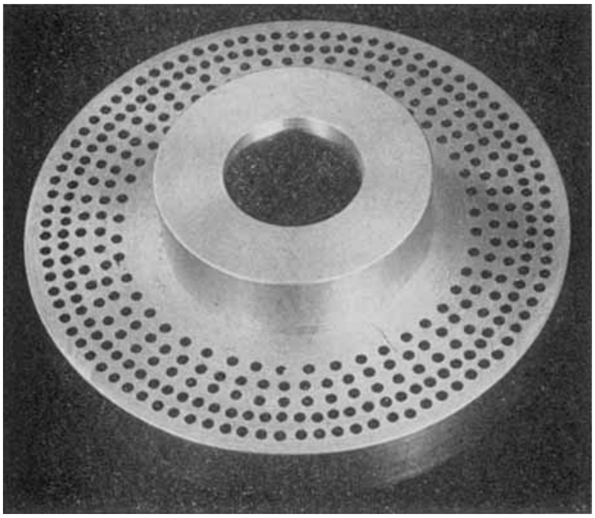

Fig. 3. Orifice plate.

But by definition

$$C = \frac{F}{V} = \frac{8\eta L A_1^2}{A r^2} \qquad (12)$$

Two additional considerations entered into the design of a damping plate.

1. The damping tubes and viscosity of damping fluid must be such as to avoid turbulence. The limiting velocity is given by

$$U_C = \frac{1000\eta}{Dd} \qquad (13)$$

where D is the diameter of the tube, η is the viscosity, and d the density of the fluid.

2. The tubes must be such that steady state flow is established within the time constant of the machine. This is governed by the formula[1]

$$r \leqq \sqrt{\eta/dT} \qquad (14)$$

where r is the radius of the tube and T the time to reach steady flow.

Using the above formula, a machine was designed and built aimed at attaining a steady velocity of about 50 ft./sec. in about 3×10^{-4} sec. To meet the requirements the moving parts were made of light weight alloys, the damping fluid was of a very high (12,500 cstoke) viscosity, and the damping chamber was designed so a minimum mass of fluid was set in motion. This was accomplished by driving, as sketched in Figure 1, the orifice plate through the fluid instead of setting all of the fluid in motion at one time. A displacement time record for one stroke of such a machine is shown in Figure 4 where the line is reasonably straight indicating a velocity of 24 ft./sec. after 2×10^{-4} sec. With a higher pressure the goal would have been attained.

EFFECTIVE MASS OF A FLUID IN A DAMPING ORIFICE

In the type of machine we are discussing the fluid in the holes of the orifice plate will by necessity have a greater acceleration than the other moving parts. This can be handled in the following way. The effective mass of the fluid in an orifice is the actual mass of the fluid in the orifice times $(A/A')^2$ where A' is the cross-sectional area of the orifice

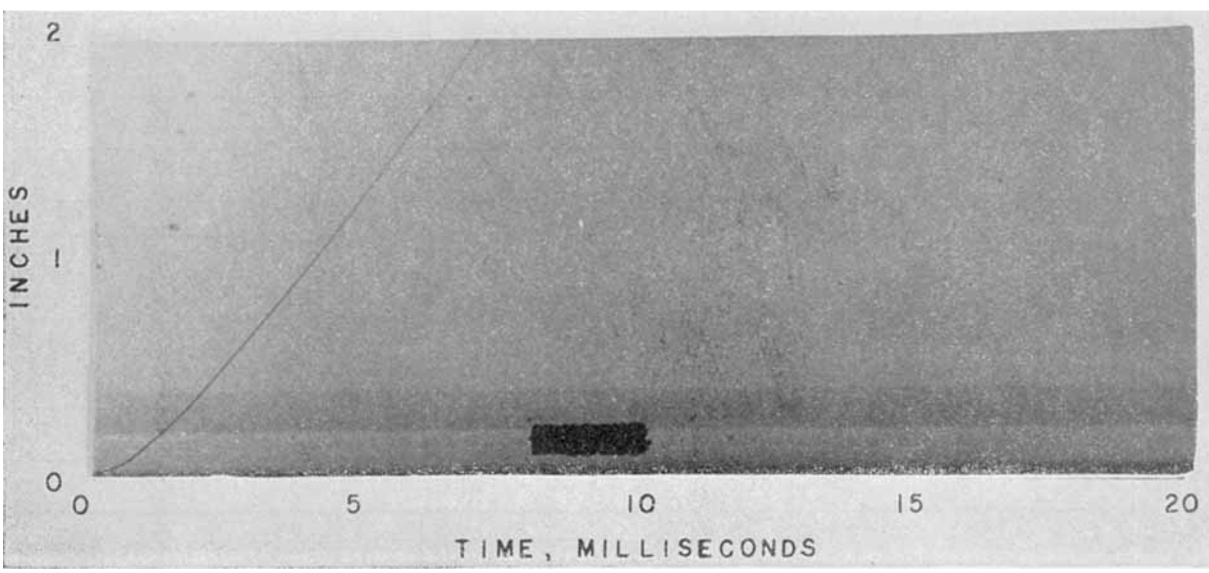

Fig. 4. Displacement vs. time curve (experimental).

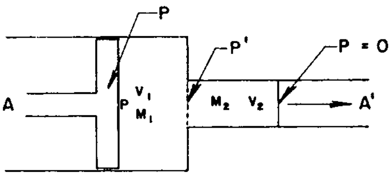

Fig. 5. Accelerating fluid in an orifice.

and A is the corresponding area of the fluid stream leading to the orifice.* This may be shown as follows. Consider a fluid being accelerated in a tube in which the cross-sectional area changes from A to A' as sketched in Figure 5. Let M_1 be the mass of the fluid in A and M_2 that in A'. A piston P accelerates the fluid M_1 in the direction indicated by the arrow by an amount dv/dt. This gives rise to a pressure p at the piston fluid interface, and a force

$$F = pA \qquad (15)$$

at the interface which is the accelerating force for the fluid. Frictional forces are neglected. A pressure p' will be produced at the junction of the two tubes. From this we identify the force accelerating the fluid M_2 as

$$F_2 = p'A' = M_2(dv_2/dt) \qquad (16)$$

Also p' gives rise to a backstream force F' on M_1 given by

$$F' = p'A \qquad (17)$$

The equation of motion for M_1 is then

$$F - p'A = M_1(dv_1/dt)$$

or

$$F = p'A + M_1(dv_1/dt) \qquad (18)$$

From eq. (16)

$$p' = (1/A')M_2(dv_2/dt)$$

Also

$$Av_1 = A'v_2 \quad \text{or} \quad (dv_2/dt) = A/A'(dv_1/dt)$$

hence

$$p' = (1/A')M_2(A/A')(dv_1/dt) \qquad (19)$$

Substituting in (18)

$$F = [(A/A')^2M_2 + M_1]dv_1/dt$$

showing that the effective of the mass M_2 in the equation of motion is $(A/A')^2M_2$.

* This fact was pointed out to the writer by Prof. D. S. Clark of the California Institute of Technology.

LIMITING VELOCITY OF A GAS THROUGH AN ORIFICE

In designing a pneumatically driven machine the thermodynamic principle must be kept in mind that regardless of the difference of pressure across an orifice the velocity of the gas in the orifice will not exceed the velocity of sound for the gas. This may limit the velocity of a machine if gas is admitted to a cylinder by a valve.

IMPACT VS. NONIMPACT LOADING

Impact and nonimpact loading are distinguished as follows.

Consider a tensile specimen supported at one end. If it is loaded at the other end, a stress wave travels along the specimen with a velocity equal to the velocity of sound for the material of which the specimen is made. In general, depending on the attenuation in the material and the nature of the support, the wave will be reflected from the supported end and may travel along the specimen several times. Eventually the stress wave dies out and the specimen is under uniform tension from one end to the other. While the waves are traveling the specimen will not be under uniform tension along its length. Depending on the rate at which the load is applied, the tension can be anything from the full value of the load at one end and no tension at the other, to a value where the difference cannot be detected along the length of specimen. The former would be impact loading, the latter nonimpact.

In impact loading the specimen may break at one end before the stress is felt at the other. In such a case strain measurements based on the length of the specimen would mean nothing since the elongation over part of the specimen length would be zero.

In nonimpact loading the specimen would fail at the weakest point and elongation would be uniform along the length of a homogeneous specimen of uniform cross section.

It is seen from the above that the distinction between impact and nonimpact loading depends on the length of the specimen as well as the rate of loading. Rates of loading that will give impact or nonimpact loading can be calculated for a specimen of a given material and a given length.

Consider a plastic specimen $^1/_2$ in. in length. An average value for the velocity of sound in plastics is approximately 80,000 in./sec. The time required for a stress wave to travel from one end of the specimen to another is as given in the following:

$$t = \frac{0.5}{8 \times 10^4} \text{ sec.} \cong 6 \times 10^{-6} \text{ sec.} \cong 10^{-5} \text{ sec.}$$

If a $1/2$-in.-long specimen is loaded in 10^{-5} sec. or less, then one would classify the loading as impact. Allowing 10 passages of the wave along the specimen for the waves to damp out one should be fairly sure of nonimpact loading of a $1/2$-in. plastic specimen if the loading time is greater than 10^{-4} sec.

Reference

1. Clark, D. S., and D. S. Wood, *Proc. Am. Soc. Testing Materials*, **49**, 717 (1949).

Synopsis

The significance of the time to acquire constant velocity in high speed testing is discussed. It is pointed out that the border line between impact and nonimpact testing depends on this time as well as the length of the test specimen and the velocity of sound in the material being tested. Equations relating force, mass, velocity, damping, and time are worked out for a pneumatically driven, hydraulically damped piston-type machine. These equations afford a basis for the design of a high speed tester. Practical values of parameters are given for a machine which will acquire a uniform velocity of 50 ft./sec. in about 3×10^{-4} sec. Design data for a damping system are also given.

Résumé

La signification du temps requis pour atteindre une vitesse constante au cours de tests rapides est discutée. On a mis en évidence que la limite entre les tests par impact et sans impact dépend autant de ce temps que de la longueur du spécimen testé et de la vitesse du son dans le matériau soumis au test. Des équations reliant la force, la masse, la vitesse, l'amortissement et le temps ont été élaborées pour des machines fonctionnant pneumatiquement ou par piston amorti hydrauliquement. Ces équations constituent une base pour un modèle d'appareil pour tests à haute vitesse.. Les valeurs pratiques des paramètres sont données pour une machine qui doit acquérir une vitesse uniforme de 50 pieds/sec. en 3×10^{-4} sec. environ. Les données pour un modèle de système d'amortissement sont également fournies.

Zusammenfassung

Die Bedeutung der Einstelldauer einer konstanten Geschwindigkeit für Hochgeschwindigkeitstests wird diskutiert. Es wird betont, dass die Grenze zwischen Schlag- und Nichtschlagtest sowohl von dieser Dauer als auch von der Länge der Testprobe und der Schallgeschwindigkeit im getesteten Material abhängt. Beziehungen zwischen Kraft, Masse, Geschwindigkeit, Dämpfung und Zeit werden für eine pneumatisch getriebene, hydraulisch gedämpfte Kolbenmaschine ausgearbeitet. Diese Beziehungen liefern eine Grundlage für den Entwurf eines Hochgeschwindigkeits-Testers. Praktisch verwendbare Werte für die Parameter werden für eine Maschine mitgeteilt, die in etwa 3×10^{-4} sek eine einheitliche Geschwindigkeit von 50 ft/sec annimmt. Ebenso werden Daten für den Entwurf eines Dämpfungssystems angegeben.

Discussion

Question: In the equation $V = \sqrt{E/D}$, what is D?

Answer: That is the density. This is an equation for the velocity of a stress wave in the material being tested.

Question: It is the transverse wave?

Answer: No, it is a longitudinal wave.

Question: Did you consider using adjustable orifices in order to get your speed?

Answer: No, I didn't. After working out these equations and looking at them, I became very conscious of mass, and particularly conscious of any mass that moves faster than the rest of the parts. I decided that instead of pushing a whole chamber of liquid ahead of a piston (this would involve quite a bit of mass) I would do better to push a piston with some holes in it through a chamber. Now, the only mass that is moving is the mass of the liquid in the orifice plates. When I built the machine, I learned it wasn't really very versatile because in a machine designed to hold a pressure of 50,000 lb. it is rather awkward to change the pressure and to change the damping fluid in the orifice chamber. This reduced the flexibility of the machine, and as far as flexibility is concerned, it should be done some other way. However, this works pretty well.

Question: What is the relative importance of the amplitude of the stress wave compared with the transit time throughout the length?

Answer: If you get enough transit time, the amplitude is of no importance as long as it isn't great enough to damage the specimen in the beginning. This stress wave, of course, is applied as an exponential curve.

Question: When you go to long strokes with a consequent large volume in the piston, will the true measurements of the dry mass by adiabatic expansion lower your driving force by large amounts?

Answer: I don't know. This is certainly possible if the damping material is heated in the process. I used a silicone liquid of some million centistokes' viscosity.

The maintaining of pressure on the piston, however, must be considered and balanced against the heating up of the viscous damping material. There was not enough work done for a real sorting out of all of the variables.

Question: Can you give us some idea of the displacement involved before the attainment of a velocity of 10 ft./sec.? It appeared to be one tenth of an inch.

Answer: No, the distance scale is in terms of 10 to the fourth power, so we have 20 divided by 10 to the fourth, which is 2 mils. It is two thousandths of a foot.

Question: In your original equation you didn't have the external force (the force of the sample) and if that force changed with time, the position you have at C would be changed in time.

Answer: Yes. The force that entered into the equation would, of course, be equal to the pressure times the area minus certain other forces, one of which would be the resistance of the sample; others would be frictional forces in the machine. In that case you simply design so that it is small in comparison.