PLAINTIFF'S
EXHIBIT
JMMC-219

COOK TECHNICAL REVIEW

VOLUME 2 AUGUST, 1955 NUMBER 2

ANALYSIS AND CONTROL OF HYDRAULIC SURGE

The COOK TECHNICAL REVIEW is published at frequent intervals by the Cook Research Laboratories, a division of Cook Electric Company, 2700 N. Southport Avenue, Chicago 14, Illinois, in cooperation with the other company divisions:

Air Mod
Diaphlex
Electronic Systems
Inland Testing Laboratories
MagniLastic
Wirecom

W. C. Hasselhorn
President

John S. Baker
Vice President in Charge of Engineering

C. E. Lundeen
Vice President-Treasurer

Bert Erikson
Secretary-Controller

EDITORIAL BOARD

J. R. Downing, Director, Cook Research Laboratories
J. C. Bellamy, Associate Director, Cook Research Laboratories

EDITORIAL STAFF

Leo G. Killian, Editor
Harry R. Thornton, Circulation Manager

Price in the United States and Canada $1.00 per copy; in other countries $1.25. Checks should be made payable to Cook Electric Company and addressed to the Circulation Manager.

Printed in U. S. A.

Copyright 1955 by Cook Electric Company

Volume 2 August, 1955 Number 2

ANALYSIS AND CONTROL OF HYDRAULIC SURGE

by

Howard Moore
Project Engineer

MAGNILASTIC

A Division of

COOK ELECTRIC COMPANY

2700 North Southport Avenue

Chicago 14, Illinois

PREFACE TO SECOND ISSUE

In 1954 a few copies of this manual were made up in mimeographed form and distributed as Issue No. 1 to selected organizations and engineers engaged in hydraulics and related fields. Since that time additional tests have been conducted, data analyzed, and studies implemented. These investigations were primarily concerned with a determination of the relationship of n (the polytropic exponent of gas expansion) with respect to duration of the surge cycle, a theoretical approach to and development of formulas related to the hydraulic energy balance, and the inclusion of some tabular material previously unavailable. The typography has been corrected and the plates prepared in a more legible fashion. A number of comments and suggestions received from users of the manual have been incorporated. The critical review undertaken by Messrs. Elmer A. Beck, Jr. and Ralph Weaver, of the Cook Research Laboratories, has been of great value in preparing this revised edition.

It is hoped that this volume will resolve the current divergencies in water hammer and surge analysis and will provide a convenient reference for pertinent data previously obtainable only from a number of separate sources.

Chicago, Illinois
August, 1955

Howard Moore

PREFACE TO FIRST ISSUE

This manual has been prepared for the practicing engineer who, always pressed for time, seeks expeditious and logical explanations for surge phenomena, universally applicable formulas expressed in common engineering units, and methods of surge analysis formulated for rapid use. To achieve the goals of usability and simplicity the mathematics employed have been limited to algebraic derivations from energy concepts and thermodynamics. To further simplify and accelerate the solution of water hammer and surge control problems, several plots and nomographs have been prepared. Included in the tabular material are data of sufficient scope to permit the complete surge analysis of most piping systems.

This manual covers the fields of water hammer, surge suppressing devices, surge control with pneumatic suppressors, pulsation control and resonance, accumulators, and thermal expansion compensation.

The assistance and encouragement received from various individuals and organizations who supplied information and advice during the preparation of this manual is gratefully acknowledged. In particular, a word of appreciation is in order to the Pelton Water Wheel Company, McGraw-Hill Book Company, and the American Society of Mechanical Engineers for permission to reproduce published figures and drawings. Staff members of the Cook Electric Company and the Cook Research Laboratories who aided materially in the preparation and finalization of the manual are Messrs. Harry R. Thornton, advertising manager, Glen Juergensen, chief engineer, MagniLastic Division, and Leo G. Killian, Director of Technical Publications, Cook Research Laboratories. The interest of Messrs. Walter C. Hasselhorn, President of Cook Electric Company, and Robert L. Sigwell, Manager of its MagniLastic Division, has greatly benefitted the surge investigations and preparation of this manual.

Chicago, Illinois
August, 1954

Howard Moore

CONTENTS

	Page
Preface to Second Issue	v
Preface to First Issue	vi
Symbols	x
Definitions	xii

Introduction

Historical Background of Water Hammer Studies	1
Classification of Water Hammer	1
Scope of Available Literature	1
Need for Empirical Data	1
Range of Tests Conducted	1

Part I—The Phenomena of Water Hammer

The Case of Rapid Fluid Deceleration	4
Sequence of Physical Events	4
Elastic Properties of the Fluid and Conduit	6
Nature and Velocity of the Pressure Wave	7
Pressure-Velocity Relationship	9
Pressure-Excess Volume Relationship	9
Effective Velocity in Complex Conduits	9
The Case of Slow Fluid Deceleration	9
The Time Parameter, N	9
Pressure-Velocity Relationship	11
Allievi's Theory of Water Hammer	11
Surge Wave Shape for $T > \mu$	15
Effective Velocity of Pressure Wave	15

Part II—Surge Suppressing Devices

By-Pass Type Suppressors	16
Relief Valve	16
Automatic Surge Suppressor	16
Storage Type Suppressors	17
Surge Tanks	17
Air Chambers	17
Gas-Liquid Separating Media	17
Piston-Type Suppressors	18
Bladder-Type Suppressors	18
Diaphragm-Type Suppressors	19
Sleeve-Type Suppressors	19
Pneumatic Bellows-Type Suppressor	20
Evaluation of Storage-Type Suppressors	22

Part III — Surge Control With Pneumatic Suppressors

	Page
Effect of Suppressors	22
The Energy Balance	23
Complex Systems	26
Orifice Filling Control	27
Series Systems	30
Effect of Check Valves on Surge Characteristics	31
Suppressor Placement	31
Uphill Pumping Systems	31
Pulsation Control in Reciprocating Pump Lines	35
Resonance	36
Accumulators	37
Thermal Expansion Compensation	38
Hydraulic Energy Balance	39
Table I. Conversion Factors and Formulas	44
Table II. Viscosity Conversion Factors	45
Table III. Properties of Liquids	46
Table IV. Properties of Conduit Materials	47
Table V. Pipe Dimensions	48
Table VI. Logarithms	50
Bibliography	54

ILLUSTRATIONS

		Page
Plate I.	Pump and Engine Assemblies	2
Plate II.	500-Foot Pipeline Loops	2
Plate III.	Controlled Closing Plug Valve	2
Plate IV.	Suppressor Filling Control Orifice Plates	3
Fig. 1a, 1b.	Schematic Representation of Water Hammer Cycle	4
Fig. 2.	Simultaneous Pressure-Time Records of Water Hammer (Theoretical)	5
Fig. 3.	Simultaneous Pressure-Time Records of Water Hammer (Actual)	5
Fig. 4.	Nomograph for a	8
Fig. 5.	Nomograph for Pipe Line Parameter K	10
Fig. 6.	Nomograph for Time Parameter N	12
Fig. 7.	Chart for the Determination of Pressure Ratio Z^2	13
Fig. 8.	Allievi Chart—Pressure Rise for Uniform Gates and Simple Conduits. Small K and N	14
Fig. 9.	Allievi Chart—Pressure Rise for Uniform Gates and Simple Conduits. Large K and N	14
Fig. 10.	Simultaneous Pressure-Time Records for Slow Closure	15
Fig. 11.	Relief Valve at Shutoff Valve	16
Fig. 12.	Relief Valve in Pump Discharge Line	16
Fig. 13.	Automatic Surge Suppressor	16
Fig. 14.	Surge Tank Installation	17
Fig. 15.	Air Chamber	17
Fig. 16.	Floating Piston Type Suppressor	18
Fig. 17.	Bladder Type Suppressor	19
Fig. 18.	Diaphragm Type Suppressor	19
Fig. 19.	Sleeve Type Suppressor	19
Fig. 20.	Nonpressurized Hydropad	20
Fig. 21.	Pressurized Hydropad	20
Fig. 22.	Pressurized Hydropad with Bellows Extended	20
Fig. 23.	Stainless Steel-Trimmed MagniLastic Hydropad	21
Fig. 24.	55,006 Cubic-Inch Hydropad	21
Plate V	Hydropad Installation, San Pedro, California	21
Fig. 25.	Simultaneous Pressure-Time Records for a System Incorporating Suppressors	22
Fig. 26.	$\frac{p_m^\circ}{p_o^\circ}$ vs. Capacity Factors	24
Fig. 27.	$\frac{p_o^\circ}{p_m^\circ}$ vs. $\left(\frac{p_o^\circ}{p_m^\circ}\right)^{\frac{1}{n}}$	26
Fig. 28.	$\frac{p_m^\circ}{p_o^\circ}$ vs. Y	27
Fig. 29.	Pressure-Time Records for a System Incorporating Suppressors and Orifice Filling Control	29
Fig. 30.	System Diagrams and Superimposed Pressure-Time Records	31
Fig. 31a.	Flow, Pressure, and Pump Speed Curves for Power Failure in Uphill Pumping System Without Check Valve	32
Fig. 31b.	Flow, Pressure, and Pump Speed Curves for Power Failure in Two Uphill Pumping Systems Incorporating Check Valves	32
Fig. 32.	Relation Between Friction Factor and Reynolds' Number for Flow in Circular Pipes, With Added S Number Coordinates	33
Fig. 33.	Nomograph for Kinetic Energy per Unit Length	40
Fig. 34.	Nomograph for Elastic Energy per Unit Length	41
Fig. 35.	Bulk Modulus of Hydrocarbons	42
Fig. 36.	Bulk Modulus and Specific Weight of Water	43

SYMBOLS

Symbol	*Unit*	*Description*
A	sq ft	Flow area of pipe
a	fps	Velocity of pressure wave
a_e	fps	Effective velocity of pressure wave
a_t	fps	Velocity of pressure wave at temperature t_1
b	in.	Bore of reciprocating pump cylinder
C	–	Coefficient of orifice discharge
C_0	cu ft	Capacity of suppressor: volume occupied by gas at initial charging pressure p_0°
c_0	cu in.	
C_m	cu ft	Volume occupied by gas within suppressor at pressure p_m°
C_x	cu ft	Volume occupied by gas within suppressor at pressure p_x°
c_x	cu in.	
D	ft	Inside diameter of pipe
d	in.	Inside diameter of pipe
d_1	in.	Expanded internal diameter of pipe
d_2	in.	Orifice diameter
E	psi	Modulus of elasticity (Young's) of pipe material
e	in.	Thickness of pipe wall
F	lb	Force
F_1	–	Reciprocating pump volumetric factor
f	–	Pipe friction factor
g	ft/sec^2	Acceleration due to gravity
H_0	ft	Pressure head for steady conditions
H_s	ft	Static head
h	ft	Pressure rise above normal at any time
h_{max}	ft	Maximum pressure rise
h_r	ft	Pressure rise for instantaneous closure $h_r = av_0/g$
h_w	inches of water	Differential pressure across orifice
K	–	Pipe line parameter $= av_0/2gH_0$
k	psi	Bulk (volume) modulus of liquid
L	ft	Length of any uniform section of pipe
L_1	ft	Distance between valve and suppressor
L_2	ft	Resonant length of "closed end" pipe
L_3	ft	Resonant length of "open end" pipe
M	–	Any integer
M_1	–	Any odd integer
N	–	Number of time intervals $= aT/2L$
N_R	–	Reynolds' number
N_1	–	Number of time intervals $= aT/2L_1$
n	–	Polytropic exponent of expansion of a gas
n_1	cps	Frequency of pulsations
P	psfg	Unit pressure
P_0	psfg	Initial pressure
P_1	psfg	Maximum pressure
p	psig	Unit pressure
p'	psf per ft	Pressure gradient
p_a	psia	Atmospheric pressure
p_f	psi	Pressure drop in conduit during steady flowing conditions
p_m	psig	Maximum allowable surge pressure
p_m°	psia	Maximum allowable surge pressure, absolute $= p_m + p_a$
p_{max}	psi	$p_1 - p_0$

Symbol	Unit	Description
p_0	psig	Initial gauge pressure
p_0°	psia	Initial absolute pressure $= p_0 + p_a$
p_p	psig	Pump discharge pressure
p_r	psi	Pressure rise
p_s	psig	Static or elevation pressure
p_x	psig	Reduced pressure
p_x°	psia	Reduced pressure, absolute $= p_x + p_a$
p_1	psig	Maximum water hammer pressure, or locked up pressure at temperature t_1
p_1°	psia	$p_1 + p_a$
p_2	psig	Pressure at orifice inlet when full flow is established
p_3	psig	Minimum pressure at instant of flow reversal
S	—	Johnson's parameter, equations (44) or (45)
s	psi	Hoop stress
s_1	in.	Piston stroke of reciprocating pump
T	sec	Time of valve closure
T_c	sec	Time after valve closure
t	sec	Time
t_o	°F	Initial temperature
t_1	°F	Final temperature
V	cu ft	Volume
V_C	cu ft	Expanded volume of conduit
V_d	cu ft	Volume of liquid entering suppressor
V_d'	cu in.	Volume of displaced fluid
V_L	cu ft	Expanded volume of liquid
V_0	cu ft	Initial volume
V_p	cu ft	Excess volume stored in pipe
V_1	cu ft	Final volume
v, v_1, v_2	fps	Instantaneous velocity of liquid
v_e	fps	Effective velocity of liquid
v_0	fps	Velocity of liquid under initial steady-state conditions
W_L	ft-lb	Elastic energy stored in compressed liquid
W_L'	ft-lb/ft	Elastic energy stored in compressed liquid per unit length
W_0, W_1, W_2	ft-lb	Kinetic energy
W_0'	ft-lb/ft	Kinetic energy per unit length
W_p	ft-lb	Elastic energy stored in stretched pipe
W_p'	ft-lb/ft	Elastic energy stored in stretched pipe per unit length
W_S	ft-lb	Elastic energy stored in stretched pipe and compressed liquid
W_S'	ft-lb/ft	Elastic energy stored in stretched pipe and compressed liquid per unit length
w	lb/cu ft	Specific weight of liquid
w_0	lb/cu ft	Specific weight of liquid at base temperature t_0
w_t	lb/cu ft	Specific weight of liquid at temperature t_1
Y	—	Pressure ratio factor: equation (37)
Z^2	—	Ratio of total water hammer pressure to initial pressure $= p_1/p_0$
α	ft-lb	Potential energy stored in suppressor
β	—	Orifice ratio d_2/d
γ, γ_T	ft-lb	Potential energy or flow work of liquid entering suppressor
$\delta; \delta_1, \delta_2$	ft-lb	Elasic energy stored in stretched pipe and compressed liquid
δ'	ft-lb/ft	Elastic energy stored per unit length
ε_C	in./in.-°F	Coefficient of linear expansion (conduits)
ε_L	cu in./cu. in.-°F	Coefficient of cubical expansion (liquids)
λ	ft	Wave length
μ	sec	Period of pipe $= 2L/a$

Symbol	*Unit*	*Description*
μ_1	slugs/ft-sec	Dynamic viscosity
μ_2	centipoise	Dynamic viscosity
ν	centistokes	Kinematic viscosity
ρ	slugs/cu ft	Mass density of liquid = w/g

DEFINITIONS

For purposes of this manual, the following definitions are used:

Pressure SURGES are water hammer phenomena caused by liquid deceleration or unsteady flow.

The CAPACITY of a surge suppressor is the volume occupied by the pressurizing gas at the initial charging pressure.

INTRODUCTION

Historical Background of Water Hammer Studies

The first significant treatises on the phenomenon of water hammer were those of Michaud (1878), Joukowsky (1904) and Allievi (1903 and 1913); Michaud (Bibliography: Ref. 1) noting the pressure oscillations characteristic of water hammer, as affected by the combined elasticities of the conduit walls and confined liquid, regarded this elastic structure as analogous to an air chamber of variable capacity. Joukowsky (Ref. 2) first established the rate of propagation of the pressure wave, a, and derived the expression for maximum pressure rise in a water conduit, $h = av_0/g$ (ft), following closure of discharge valve within the period $2L/a$ (sec). Allievi (Ref. 3) presented in his Notes I to V and accompanying figures a mathematical analysis and convenient method for determining the maximum pressure rise for uniform valve closures—both rapid and slow—in simple conduits. He also considered the surges resulting from acceleration of flow and resonant conditions in a conduit.

In 1933 a symposium on water hammer was held in Chicago under the auspices of the A.S.C.E. Power Division and the A.S.M.E. Hydraulic Division. Existing theories were reviewed and compared, corroborating evidence was presented from test data, and a comprehensive bibliography was compiled. In the publication of these papers (Ref. 4), an excellent background was provided for the analysis of water hammer. Worthy of note is the high esteem which the A.S.M.E. Committee on Water Hammer had for the analyses of Allievi. Referring to his *Theory of Water Hammer* (Ref. 3), the statement is made: "This treatise was translated in many languages and forms the basis for virtually all of our theory of water hammer."

Classification of Water Hammer

In the *Symposium* (Ref. 4) the A.S.M.E. Committee on Water Hammer defined five basic conditions for water hammer investigations and analysis:

- Case I —Conduits of Uniform Thickness and Diameter (Simple Conduits)
- Case II —Conduits of Variable Thickness and Diameter (Complex Conduits)
- Case III —Conduits Having Branch Pipes (Compound Conduits)
- Case IV —Water Hammer in Pump Discharge Conduits (Power Failure of Centrifugal Pump in an Uphill Pumping System)
- Case V —Surge Tanks (Stand Pipes Open to the Atmosphere)

To these may be added the following conditions:

- Case VI —Pulsation in Reciprocating Pump Discharge lines
- Case VII —Surge Control by Relief and Bypass-Type Suppressors
- Case VIII—Surge Control by Pneumatic Surge Suppressors

Scope of Available Literature

Cases I through V are adequately analyzed in the papers and references cited in *Symposium on Water Hammer* (Ref. 4). Additional recommended material is listed in the Bibliography Page 54. It is chiefly with Cases VI through VIII that the present Manual is concerned. A survey of all available literature revealed a surprising misconception and misapplication of the basic principles on the part of many authors. The range of analyses and sizing formulas pertinent to Case VIII in particular, produced such a scattering of results from given data as to be wholly unacceptable.

Need for Empirical Data

It was desirable that a comprehensive series of full-scale tests be conducted, employing flow and pressure recording instrumentation of high accuracy and negligible inertia. From the records so obtained, the correct application of physical principles, the adequacy of methods and formulas, and the values of certain basic constants could be determined.

Range of Tests Conducted

In order to examine as many variables as possible in the investigation of surge phenomena, it was necessary to set up a liquid transferring system wherein line length, flow velocity, deceleration rate, and pressure of the flowing liquid could be held constant or varied individually over a sufficiently wide range to insure obtaining definitive data. If the four independent variables—pipeline length, fluid velocity, valve closing time, and fluid head—could be controlled at will, it would be possible to produce any

desired values of the basic water hammer parameters K and N.

A hydrant refueling system for aircraft was used in the surge investigation. The following components were used from this system: pump and engine assemblies, check valve, gate valves, tubing and fittings, grooved-type couplings with gaskets, pressure regulating valve, and surge suppressors.

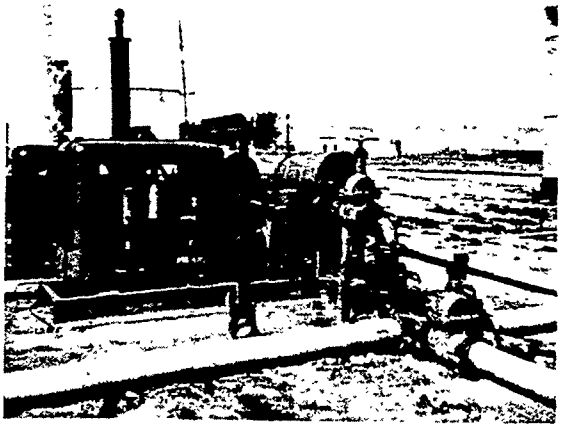

Plate I—Pump and Engine Assemblies

Two of the skid-mounted pump and engine assemblies (Plate I), each equipped with a Marlow Model 6G2 self-priming centrifugal pump driven by a Continental Model T-427 engine, were connected in series. Depending upon engine speed, the units supplied fluid at flow rates of from 0 to 620 gpm at pressures up to 150 psi. A 6-inch Powell horizontal swing check valve with a cast iron body was used for some of the tests at the discharge of the second-stage pump. Gate valves used were line size (6-inch or 4-inch) with cast aluminum bodies and bronze trim. The valves were manufactured by the Milwaukee Valve Company and conformed to their Fig. P-2654 except that the ends were grooved for couplings. The pipe line for the test system was laid out in long loops (Plate II). The lengths of 6-inch aluminum tubing and malleable iron fittings used for the pipeline conformed with Specifications MIL-T-10794 and MIL-P-10388, respectively. The malleable iron couplings, complete with synthetic rubber gaskets, conformed with Specification MIL-C-10387. The 6-inch pressure regulating valve was a Cla-Val No. 90-AF-2 with an aluminum body and having an adjustable discharge pressure range of from 40 to 60 psi. Eight surge suppressors from the portable system were available. These suppressors were the Cook Electric Company's Model HJ-350 with a pressurizing gas capacity of 200 cubic inches each. Additional suppressors, furnished by various manufacturers, provided a total suppressor capacity of nearly 7,000 cubic inches.

Components for measuring and controlling the fluid flow were added to the basic system. A flow section consisting of a 20-foot length of 6-inch steel pipe fitted with a pair of 6-inch, 300-pound, orifice flanges was used with square-edged orifice plates to

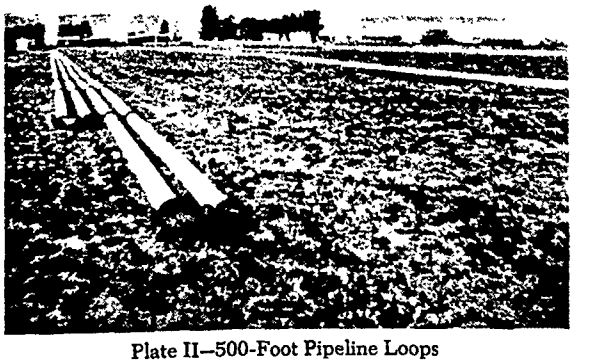

Plate II—500-Foot Pipeline Loops

measure and record flow rates. Three calibrated orifice plates constructed of stainless steel, and having bores of 3.750, 4.500 and 4.750 inches, were used. The orifice plates were calibrated using water as the test fluid. Variable speed shutoff of the flow of fluid in the system was provided by a 3-inch Nordstrom plug valve (Plate III) actuated by an air cylinder whose closing time could be regulated by an air control valve. Flow control was furnished by a 6-inch Grove Flex-Flo valve whose jacket pressure was remotely controlled by a Grove Loader supplied with compressed air.

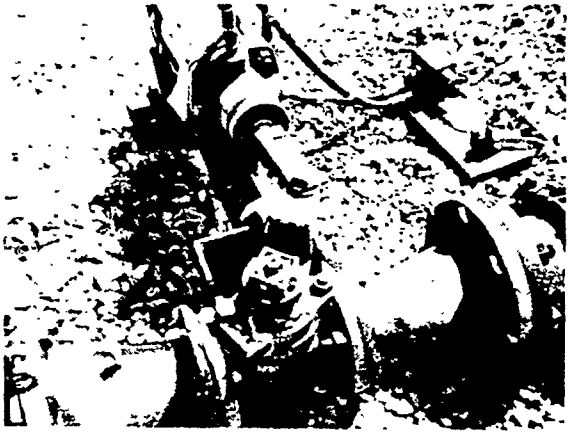

Plate III—Controlled Closing Plug Valve

Square-edged orifice plates for suppressor filling control (Plate IV) having bores ranging from 0.45 inches to 1.80 inches were inserted in a pair of 6-inch orifice flanges located between the suppressors and the main flow line.

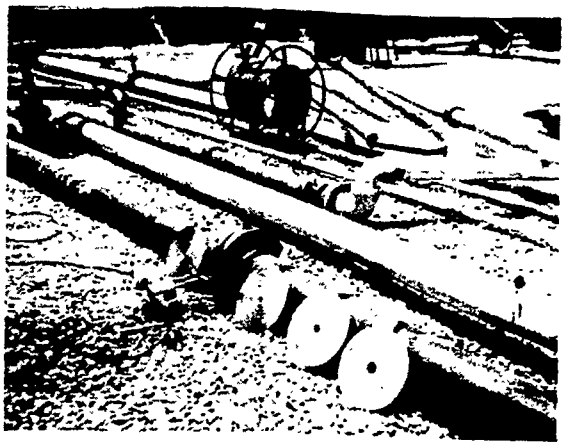

Plate IV—Suppressor Filling Control Orifice Plates

Pressures were sensed by two Statham transducers, one having a range of 0—300 psig and the other a range of 0—500 psig. The output of each pickup was fed to a Brush amplifier and continuously recorded by a 2-channel Brush oscillograph. For some tests where synchronization of transient measurements was not required, a Bristol pressure recorder was employed in addition to the Statham-Brush combination. In some tests indicator gauges were placed at various locations in the system for visual observation of pressures.

Flows at the orifice were recorded by a Bristol differential pressure recorder. The time record of the temperature of the nitrogen gas pressurizing a surge suppressor was obtained by the insertion of an iron-constantan thermocouple into the gas chamber and the continuous recording of the output voltage by means of a Bristol Dynamaster temperature recorder. Specific gravity and temperature of the test fluid were determined with a laboratory hydrometer and a thermometer.

For some tests additional pressure measuring and recording equipment was available. This instrumentation consisted of four Statham pressure transducers, four Brush amplifiers, and two 2-channel Brush oscillographs. During test runs for which this additional equipment was available, six pressures at six locations in the test system could be recorded simultaneously.

To indicate the closing time of the Nordstrom valve on the Brush pressure record charts, the circuit controlling the event-marker pen was energized by the voltage supplied to the small solenoid pilot which operated the air cylinder attached to the Nordstrom valve. At the instant of complete closure the event-marker pen circuit was momentarily de-energized by the action of a secondary circuit containing switch contacts located on the stem and body of the Nordstrom valve.

Calibration of the transducers and recording equipment by means of a dead-weight tester was performed at the beginning of every series of test runs and at intervals between runs in the series. An over-all accuracy of $\pm 2\%$ in pressure measurement and $\pm 3\%$ in flow measurement could be reasonably expected from this instrumentation.

The tests were made using either water or Grade JP-4 jet engine fuel with a specific gravity of 0.815, as required by the objective of a particular test. Among the conditions studied by means of the test setup, the following are pertinent to the present study:

Measurement of water hammer phenomena
Reduction of surge pressures by suppressors located at the shutoff valve
Reduction of surge pressure by suppressors located at various distances from the valve
Surge reduction by suppressors located at the valve and having orifice filling control
The influence of a check valve on the resonance of a system
Heat transfer characteristics of various pneumatic suppressors during energy absorption.

PART I—THE PHENOMENA OF WATER HAMMER

The Case of Rapid Fluid Deceleration

Sequence of Physical Events

The physical concepts of the water hammer cycle can be most clearly illustrated by consideration of the simple conduit, supply reservoir, and shutoff valve illustrated in Fig. 1a and 1b. Diagram (1) shows the steady state conditions existing before valve closure. The fluid is flowing from left to right at an inital, constant velocity, v_0. Because of friction losses, pressure in the conduit diminishes uniformly from reservoir to valve, where its value is H_0. Now let the valve be closed completely and instantaneously. The mass of fluid immediately adjacent to the valve has had its velocity, v, reduced from v_0 to 0, and its kinetic energy, per unit length.

$$(w/2g)Av_0^2 \quad \text{ft-lb}$$

converted into elastic energy stored in the stretched pipe, per unit length,

$$\frac{A\,w^2\,h^2\,d}{288eE} \quad \text{ft-lb}$$

and in the compressed fluid, per unit length,

$$\frac{A\,w^2\,h^2}{288k} \quad \text{ft-lb}$$

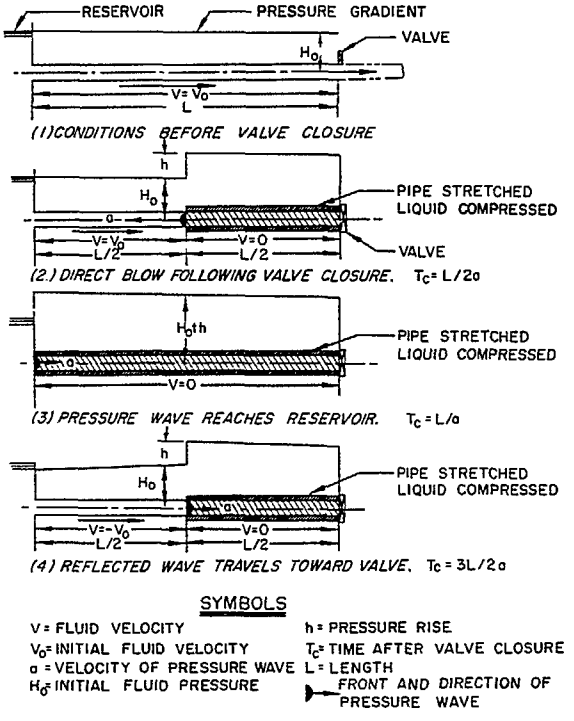

Fig. 1a. Schematic Representation of Water Hammer Cycle

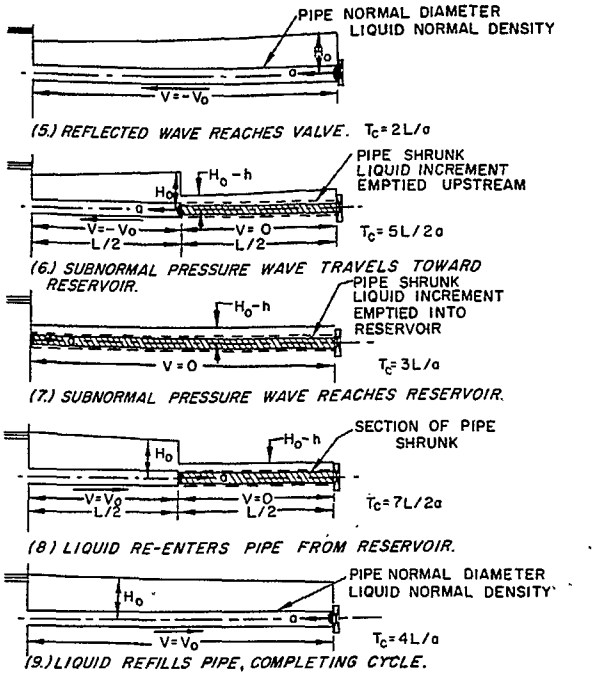

Fig. 1b. Schematic Representation of Water Hammer Cycle

While the fluid in the conduit upstream from the region of compression continues to flow in the initial direction and at the initial velocity, v_0, the front of this region of compression travels very rapidly toward the reservoir. Its speed of propagation, designated as the "velocity of the pressure wave," is represented by the constant a.

Conditions existing at the time when the pressure wave has travelled upstream half the distance to the reservoir, during the time following valve closure, $T_c = L/2A$, are depicted in Diagram (2). Fluid in the

upstream section is still flowing with the initial velocity, v_0, and at the initial pressure, H_0. All fluid in the downstream section is at rest and at a pressure of $H_0 + h$.

At the time L/a, the region of compression will reach the reservoir. As shown in Diagram (3), the entire column of liquid is at rest and under a pressure of $H_0 + h$. It should be noted that the value of H_0 is variable between the reservoir and valve, depending upon the friction losses when $v = v_0$. The pressure rise, h (in feet of fluid), is constant from energy considerations, but algebraic addition of H_0 and h will produce the sloping pressure gradient shown in Diagram (3).

Because of the excess pressure h in the conduit, the adjacent fluid empties into the reservoir, the energy stored in the stretched pipe and compressed fluid being reconverted into kinetic energy to produce a velocity equal to v_0, but in the opposite direction. The zone in which this reconversion of energy takes place starts at the reservoir and travels toward the valve with the constant velocity a. Diagram (4) illustrates conditions when this reflected wave has travelled half the distance to the valve. Fluid in the upstream section is flowing at velocity $-v_0$ into the reservoir at the initial pressure, H_0. Fluid in the downstream section is at rest and subjected to a pressure of $H_0 + h$.

When all the stored energy has been reconverted and the reflected wave has reached the valve, the conditions are those depicted in Diagram (5), at $T_c = 2L/a$. All the fluid is at the initial density and is moving toward the reservoir at constant velocity $-v_0$; the pressure in the conduit is H_0.

As flow continues toward the reservoir, Diagram (6), the quantity of fluid leaving the downstream section of the pipe is sufficient to result in a pressure reduction of $-h$. The fluid remaining, having imparted its total kinetic energy to the upstream fluid column, comes to rest. At time $T_c = 3L/a$, the whole column is at zero velocity, as illustrated in Diagram (7). Line pressure at this instant is theoretically $H_0 - h$, but in actual practice $H_0 - h$ can never be less than the vapor pressure of the liquid.

Unbalanced pressures at the reservoir now restore the flow to v_0 in the original direction and at the original pressure, H_0, as shown in Diagram (8).

The region of transition, where the head of the moving column of fluid enters the section of pipe which has been at reduced pressure, travels at the velocity, a. When this region reaches the valve, at time $T_c = 4L/a$, as shown in Diagram (9), conditions are identical to those existing at the instant of valve closure, $T_c = 0$. All the fluid is flowing toward the valve at the initial velocity, v_0, and pressure, H_0. The cycle, completed in $4L/a$ seconds, is now repeated at the same rate but with diminishing intensity until damped out by pipe friction and the limitation that $(H_0 - h)$ must be at least equal to the vapor pressure of the liquid.

The square-topped pressure-time wave pattern, characteristic of the water hammer cycle, is illustrated in Fig. 2. Here theoretical pressure-time measurements are presented for three locations in the pipeline: at the valve, at the mid-point, and at the reservoir outlet. Note the sloping gradient resulting from recovered friction head. Tracings of electronically recorded pressure-time measurements made under controlled conditions on a test system containing 3078 feet of 6-inch aluminum tubing and handling JP-4 jet engine fuel at an initial flow rate of 655 gpm are reproduced in Fig. 3. Pressure pickups were located at the shutoff valve, at the mid-point in the piping system, and at the outlet of the centrifugal

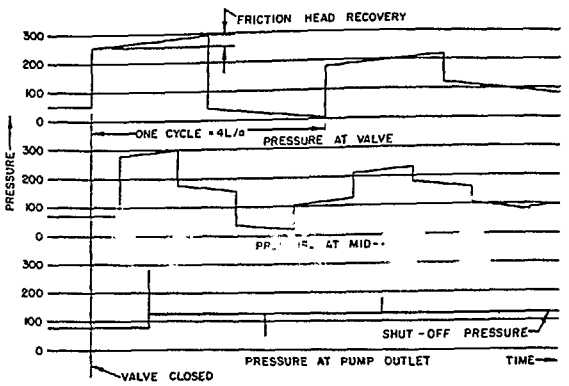

Fig. 2. Simultaneous Pressure-Time Records of Water Hammer (Theoretical)

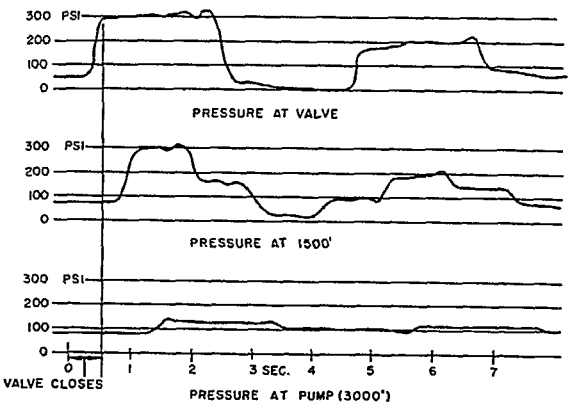

Fig. 3. Simultaneous Pressure-Time Records of Water Hammer (Actual)

pump which served as the reservoir. Some minor pressure reflections will be noted, as well as a drop in the pressure curve just before the maximum value is reached. These irregularities may be partially explained by the configuration of the system, which was laid out in long loops of 500 feet or more, and the presence of the valves and fittings required to control the total effective line length by permitting flow through selected loops. Although the initial pressure rise at the valve, $p_r(=wh/144)$, was 240 psi, the subsequent value of initial pressure reduction (theoretically equal to $-p_r$) was only 30 psi, and $H_0 - h$ never dropped below atmospheric pressure. The true vapor pressure of JP-4 at the temperature recorded during the run was approximately 2 psia.

Elastic Properties of the Fluid and Conduit

Although regarded as incompressible in the analysis of steady flow problems in fluid mechanics, all liquids are compressible. It is the compressibility of the liquid which, combined with the elasticity of the conduit walls, directly determines the pressure rise resulting from converted energy, and the velocity of propagation of the pressure wave, *a*.

Defined as the reciprocal of compressibility, the bulk modulus, or modulus of volume elasticity,

$$k = \frac{p_1 - p_0}{\frac{V_0 - V_1}{V_0}} \text{ psi} \qquad (1)$$

where p_0 and p_1 are the initial and final pressures, psi, and V_0 and V_1 are the corresponding volumes, in any convenient unit.

The bulk modulus of water is 280,000 psi at 32°F (See Fig. 36). The physical significance of Formula (1) is that for each psi of applied pressure the confined volume of water at 32° is reduced by $\frac{1}{280,000}$. Were it not for the fact that the bulk modulus of a liquid increases with an increase of pressure (at 2500 atmospheres the bulk modulus of water at 32°F is 567,000 psi), it might be logical to assume that when a pressure equal to the bulk modulus was reached, the volume, V_1, occupied by the liquid would have been reduced to zero.

The energy stored by elastic compression of the liquid is the volume change times the average value of the pressure change producing the new volume, or

$$W_L = 144(V_0 - V_1)\frac{(p_1 - p_0)}{2} \text{ ft-lb} \qquad (2)$$

where V_0 and V_1 are initial and final volumes in cubic feet and p_0 and p_1 are corresponding pressures in psi. From Equation (1) a value of $V_0 - V_1$ is obtained:

$$V_0 - V_1 = \frac{V_0(p_1 - p_0)}{k} \qquad (1a)$$

Substitution into Equation (2) yields

$$W_L = \frac{72V_0(p_1 - p_0)^2}{k} \text{ ft-lb} \qquad (3)$$

For unit length the expression becomes

$$W_L' = \frac{72A(p_1 - p_0)^2}{k} \text{ ft-lb/ft} \qquad (3a)$$

where A is the flow area of the conduit in square feet.

The elasticity of the pipe is expressed in the concept and value of Young's modulus of elasticity, defined as the ratio of stress to strain, or

$$E = \frac{s}{\frac{d_1 - d}{d}} \text{ psi} \qquad (4)$$

where E = Young's modulus, psi

s = hoop stress, psi

d = initial pipe diameter, inches

d_1 = expanded pipe diameter, inches

Inside diameters may be used without significantly affecting the accuracy of calculations. For a pipe the increment of stress, ds, resulting from an internal pressure rise from p_0 to p_1, is

$$ds = \frac{(p_1 - p_0)d}{2e} \text{ psi} \qquad (5)$$

where e is the pipe wall thickness, inches.

The energy stored in the stretched pipe walls is expressed by the product of the volumetric change and the average value of the pressure change, or

$$W_p = 144(V_1 - V_0)\frac{(p_1 - p_0)}{2} \text{ ft-lb} \qquad (6)$$

in which

$$V_1 - V_0 = \frac{L}{144}\,\frac{\pi}{4}\,\frac{2d^3}{2eE}(p_1 - p_0) \text{ cu-ft} \qquad (7)$$

and L is the length in feet. Substitution and simplification yield

$$W_p = \frac{72ALd(p_1 - p_0)^2}{eE} \text{ ft-lb} \qquad (8)$$

For unit length

$$W_p' = \frac{72Ad(p_1 - p_0)^2}{eE} \text{ ft-lb/ft} \qquad (8a)$$

The total energy stored by compression of the liquid and expansion of the pipe walls is

$$W_K = W_L + W_p = 72AL(p_1 - p_0)^2\left(\frac{1}{k} + \frac{d}{eE}\right)\text{ft-lb} \tag{9}$$

For unit length:

$$W_S' = 72A(p_1 - p_0)^2\left(\frac{1}{k} + \frac{d}{eE}\right) \quad \text{ft-lb/ft} \tag{9a}$$

Nature and Velocity of the Pressure Wave

The increment of volume, a thin cross-sectional slice of liquid in which, at any instant, the steady flow within the pipe is undergoing an abrupt change in velocity, is known as the front of the pressure wave. Like any fluid disturbance, this wave front travels with the speed of sound, reflects from branches, dead ends, and changes in section. It travels in either direction with undiminished intensity. Two pressure waves, starting at the opposite ends of the pipe, will pass through each other without mutually interfering, and progress until reflected. Pressures in their wakes will be the algebraic sum of the individual pressures.

The velocity of propagation, a, of the pressure wave can be derived quite simply from logical reasoning. If liquid is entering a pipe of constant cross-section A (square feet) at a constant velocity v_0 (feet per second) and there is no outlet, in t seconds the quantity Av_0t (cubic feet) will have occupied the increment of volume created by compression of the fluid and expansion of the pipe walls throughout the length L of the pipe. The rate at which this volume is being occupied is obviously L/t, which ratio we shall define as a, the velocity of propagation of the pressure wave. Setting the added volume equal to the space provided by a pressure increase from p_0 *to* p_1.

$$Av_0t = AL(p_1 - p_0)\left(\frac{1}{k} + \frac{d}{eE}\right) \quad \text{cu ft} \tag{10}$$

and letting

$$a = \frac{L}{t} = \left(\frac{v_0}{p_1 - p_0}\right)\left(\frac{1}{\frac{1}{k} + \frac{d}{eE}}\right) \quad \text{ft/sec} \tag{11}$$

variables can be eliminated by equating the total kinetic energy before valve closure:

$$W_0 = \frac{wALv_0^2}{2g} \quad \text{ft-lb} \tag{12}$$

where w is the specific weight, lb/cu ft

g is the acceleration of gravity: 32.2 ft/sec^2 to the total strain energy stored after valve closure:

$$W_S = 72AL(p_1 - p_0)^2\left(\frac{1}{k} + \frac{d}{eE}\right) \quad \text{ft-lb} \tag{9}$$

from which

$$\frac{v_0^2}{(p_1 - p_0)^2} = \frac{144g\left(\frac{1}{k} + \frac{d}{eE}\right)}{w} \tag{13}$$

Substitution in (11) readily yields

$$a = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{k} + \frac{d}{eE}\right)}} \quad \text{ft/sec} \tag{14}$$

where d = inside diameter of pipe, inches

e = pipe wall thickness, inches

Using this value of a, Formula (9) becomes

$$W_S = \frac{10{,}368ALg(p_1 - p_0)^2}{a^2w} \quad \text{ft-lb} \tag{9b}$$

For convenience in determining the value of a the nomograph, Fig. 4, has been prepared. An example is provided to illustrate its use.

Given data: pipe I.D., $d = 10.020$ in.
wall thickness, $e = 0.365$ in.
Young's modulus for pipe material, $E = 30{,}000{,}000$ psi
Fluid contained: water, with bulk modulus $k = 294{,}000$ psi
specific weight of water, $w = 62.4$ lb/cu ft

Find the ratio $d/e = 10.020/0.365 = 27.5$. Using a straightedge connect 30,000,000 on the E scale with 27.5 on the d/e scale. Mark the intersection on axis (1). Connect this intersection with point (2) and mark the intersection with curve (3). Extend a straight line from point (4) through the last intercept on curve (3) to intersection with line (5). Connect this point with 294,000 on the k scale and mark the intercept on line (6). Connect this intercept with point (4) and mark the intersection with curve (7). Extend a straight line from point (8) through the intercept on curve (7) until it intersects line (9). Connect this point with 62.4 on the w scale and read 4160 fps at the intersection with the a scale. An accuracy of $\pm\frac{1}{2}\%$ can be reasonably expected from careful use of this nomograph. The actual calculated value of a for the example just given is 4157 fps, or an error of 0.075%.

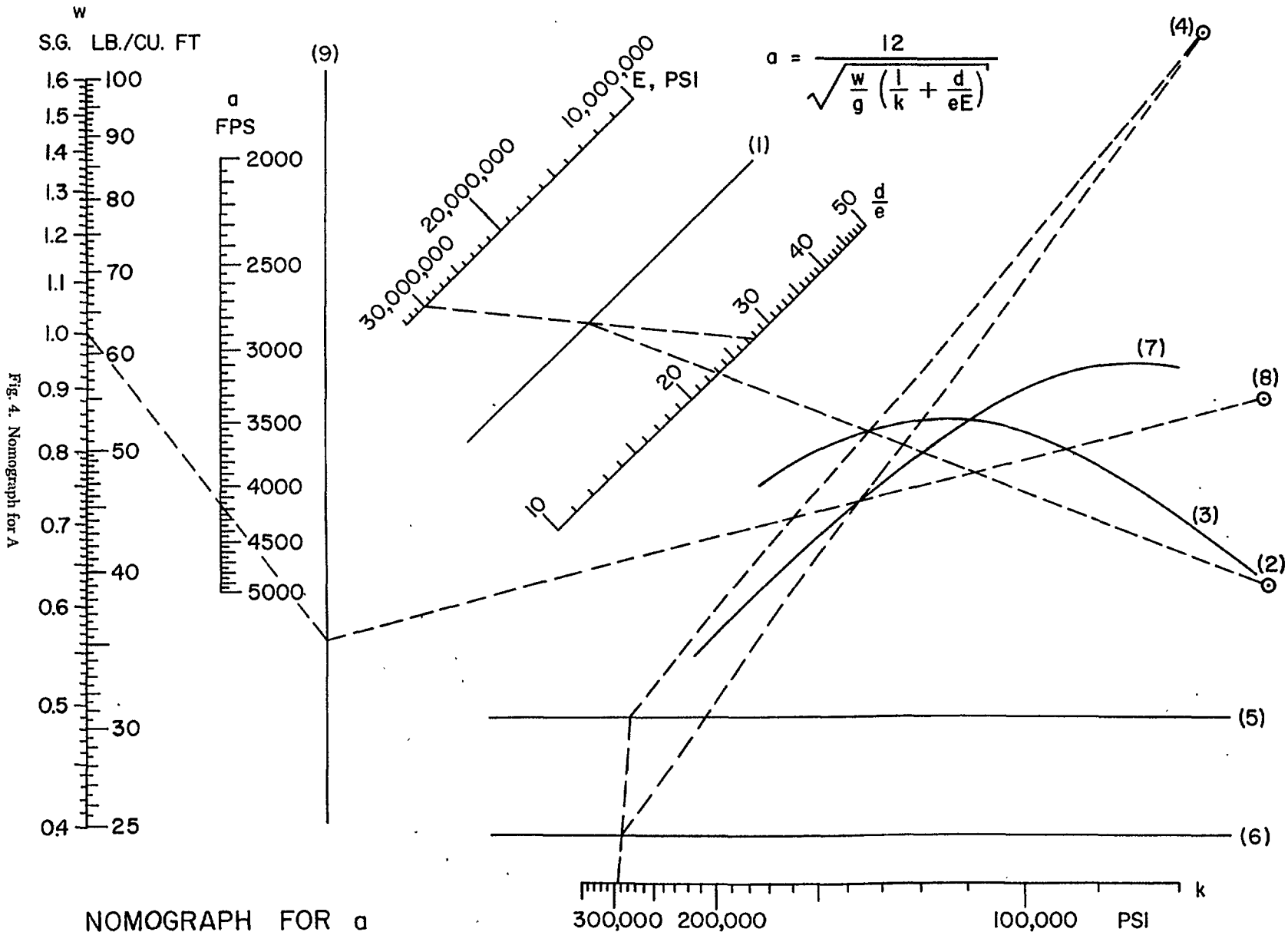

Fig. 4. Nomograph for A

Pressure-Velocity Relationship

By combining Equations (13 and (14) and solving for the pressure rise, $p_1 - p_0$, the celebrated Law of Joukowsky (Ref. 2) can be expressed in common engineering units:

$$p_1 - p_0 = \frac{v_0 w a}{144g} \text{ psi} \qquad (15)$$

Derivation of Joukowsky's Law may also be made by applying the momentum relationship expressed in Newton's Second Law. Since impulse Ft equals momentum Mv_0 the increased force F caused by fluid deceleration is equal to the time rate of change of momentum: $F = d(Mv_0)/dt$. Because the change in momentum is equal to the rate at which the mass of the fluid is being brought to rest, $F = v_0(dM/dt)$. The differential of mass, dM, may be thought of as a thin slice, dL feet thick and A square feet in cross section, having a volume AdL cubic feet and a mass of $AdLw/g$ slugs. Since $dL/dt = a$, the expression may now be written

$$F = v_0\left(\frac{wAdL}{gdt}\right) = \frac{v_0 w A a}{g} \text{ lb}$$

Solving for the pressure rise,

$$\frac{F}{A} = P_1 - P_0 = \frac{v_0 w a}{g} \text{ psf} \qquad (15a)$$

or,
$$p_1 - p_0 = \frac{v_0 w a}{144g} \text{ psi} \qquad (15)$$

A convenient nomograph is provided in Fig. 5 for the rapid solution of this formula. The scales to be used are a and v_0, which give the pressure rise in feet of fluid on the h_{max} scale. From the relation,

$$p = \frac{wh}{144} \text{ psi} \qquad (16)$$

where h = pressure expressed in feet of fluid, conversion to $(p_1 - p_0)$, shown on the nomograph as the p_{max} scale, is accomplished by intersecting the proper value for w and the value of h_{max} already determined and reading $(p_1 - p_0)$ on the p_{max} scale.

The dashed lines (Fig. 5) illustrate the procedure. To determine the pressure rise resulting from sudden valve closure in a pipe carrying water (w = 62.4 lb/cu ft) at an initial velocity v_0 of 12.7 feet per second where the velocity of the pressure wave a is 4000 feet per second, place a straightedge from 4000 on a to 12.7 on v_0. Read 1570 on h_{max}. Intersect 1570 on h_{max} and 62.4 on w. Read 680 psi on p_{max} ($= p_1 - p_0$).

The maximum pressure experienced at the valve will be given by

$$p_m = p_1 + p_f/\sqrt{2} \text{ psi} \qquad (17)$$

where p_f is the pressure drop in the system (psi) during initial flowing conditions p_0 and v_0. (Ref. 21)

Pressure-Excess Volume Relationship

From Equations (10) and (14) an expression for the pressure rise resulting from the presence of an excess volume of fluid, $V_1 - V_0$, is obtained:

$$p_1 - p_0 = \frac{(V_1 - V_0)a^2 w}{144ALg} \text{ psi} \qquad (18)$$

or,
$$p_1 - p_0 = \frac{V_1 - V_0}{AL\left(\frac{1}{k} + \frac{d}{eE}\right)} \text{ psi} \qquad (18a)$$

If Equation (18a) is solved for k an unknown bulk modulus may be determined in a simple laboratory setup requiring only a pressure gauge and a hand pump of known displacement.

Effective Velocity in Complex Conduits

The initial flowing velocity v_0 may be found from the relations,

$$v_0 = \frac{0.4085Q_0}{d^2} = \frac{0.002228Q_0}{A} \text{ fps} \qquad (19)$$

where Q_0 = initial flow rate, gpm
d = inside diameter of pipe, inches
A = flow area of pipe, square feet

Although pressure reflections occur at points where the pipe changes thickness or diameter, these reflections do not appreciably affect the pressure rise during the first water hammer cycle. For all practical purposes the effective flowing velocity of the fluid v_e may be derived from energy and continuity considerations and expressed:

$$v_e = \sqrt{\frac{L_A v_A + L_B v_B + \cdots L_N v_N}{\frac{L_A}{v_A} + \frac{L_B}{v_B} + \cdots \frac{L_N}{v_N}}} \text{ fps} \qquad (20)$$

where L = pipe length, feet
v = flowing velocity of fluid, fps
subscripts $A, B, \ldots N$ refer to sections of the complex piping system.

The Case of Slow Fluid Deceleration

The Time Parameter, N

In our preceding analysis we have been considering instantaneous closure of the valve. All the phenomena described and mathematical relationships derived hold true if the valve is closed in less time than it takes the pressure wave to travel from the

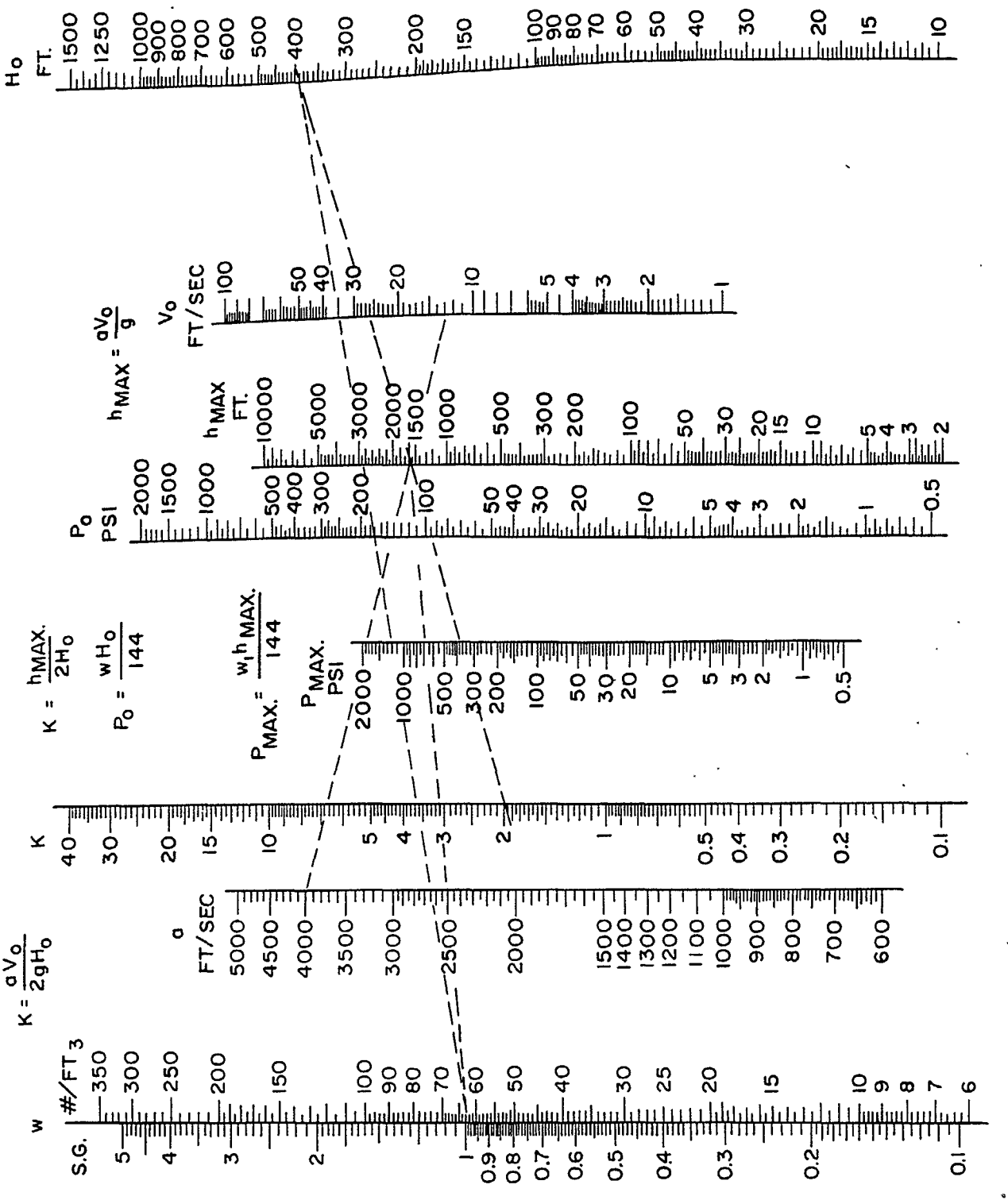

Fig. 5. Nomograph for Pipe Line Parameter K

valve to the reservoir and back to the valve, a time equal to $2L/a$ seconds, and designated as μ, the period of the pipe.

If, however ,the valve closes in T seconds, an interval longer than μ, the ratio of its closing time to the period of the pipe is expressed by the symbol N, the time parameter of the system.

$$\mu = \frac{2L}{a} \text{ sec} \qquad (21)$$

$$N = \frac{T}{\mu} = \frac{aT}{2L} \text{ intervals} \qquad (22)$$

For rapid determination of N a nomograph is presented as Fig. 6. The dashed lines exemplify its use. In a system containing 1000 feet of pipe, where the velocity of the pressure wave is 4000 feet per second, find N for a valve closure time T of 1 second. Using a straightedge, intersect 4000 on a and 1000 on L. Read $\mu = 0.5$ sec. Now intersect 0.5 on μ and 1 on T. Read 2 on N.

Pressure-Velocity Relationship

When the valve is only partially closed during the first interval μ, only a portion of the original liquid velocity has been converted into strain energy. If v_0 and v_1 are the initial and reduced velocities and p_0 and p_1 the corresponding pressures, the relationship may be derived from an extension of Joukowsky's Law, Formula (15):

$$p_1 - p_0 = \frac{wa}{144g}(v_0 - v_1) \text{ psi} \qquad (23)$$

For a valve closure time $T > \mu$ and therefore $N > 1$, at the beginning of the second interval two pressure waves will be starting from the valve—a positive wave of magnitude proportional to the fluid velocity destroyed during the first interval, and a negative wave which, originating at the instant of valve movement, has already traveled as a positive wave to the reservoir and returned to the valve during the first interval μ. Algebraic addition of these two pressures gives the resultant pressure rise at the valve for the instant considered.

Actual calculation of the pressure rise for any condition of fluid velocity, pipe length, and time of valve closure ($T > \mu$) is a matter of considerable complexity. Several authors have presented a variety of solutions, chief among which should be mentioned the formulas and arithmetic integration method of N. R. Gibson (Ref. 5), the graphical solutions of M. L. Bergeron (Ref. 6), Dawson and Kalinske (Ref. 7), and Angus (Ref. 8).

Allievi's Theory of Water Hammer

For ease of application the theory and charts of Lorenzo Allievi (Ref. 3) have found wide acceptance. A considerable number of full scale tests has proven the validity and accuracy of Allievi's theory for simple conduits and uniform valve closure. By expressing the flowing conditions of head and velocity within the pipe by the dimensionless parameter K, such that K is equal to one-half the square root of the ratio of the kinetic energy to the initial stored energy within the pipe, the conduit parameter K may be written:

$$K = \frac{1}{2}\sqrt{\frac{W_0}{W_s}} \qquad (24)$$

Substituting the values of W_0 and W_s from Equations (12) and (9b), and using in place of the pressure rise $p_1 - p_0$ the initial pressure p_0

$$K = \frac{awv_0}{288gp_0} \qquad (25)$$

Since $p_0 = wH_0/144$ psi (16a)

where H_0 = pressure in feet of fluid for initial conditions, the conduit parameter can be reduced to Allievi's original form:

$$K = \frac{av_0}{2gH_0} \qquad (25a)$$

The nomograph previously cited, Fig. 5, provides for rapid determination of K. As an extension of the example given to illustrate the solution for rapid fluid deceleration pressure rise, consider an initial pressure p_0 of 173 psi at the valve. Find K. Using the relationships $p_0 = wH_0/144$ and $K = h_{max}/2H_0$ first solve for H_0 by intersecting 62.4 on w and 173 on p_0. Read 400 on H_0. Now using the value of h_{max} previously found (1570 ft.) intersect 400 on H_0 and 1570 on h_{max}. Read 2 on the K scale.

Employing the parameters N and K as the axes of a Cartesian diagram, Allievi plotted the ratios Z^2 of the total surge pressures to the initial flowing pressures as a series of diagonal lines. A reproduction of the Allievi chart is included as Fig. 7. An enlarged section showing small values of K and N is given in Fig. 8. Figures 7 and 8 are reprinted from the *Symposium on Water Hammer* through the courtesy and with the permission of the American Society of Mechanical Engineers. For large values of K and N the chart, Fig. 9, reproduced by permission of McGraw-Hill Book Company from Rich's *Hydraulic Transients*, copyright 1951, is presented.

The quantity Z is defined by the equation

$$Z^2 - \frac{K}{N}Z - 1 = 0 \qquad (26)$$

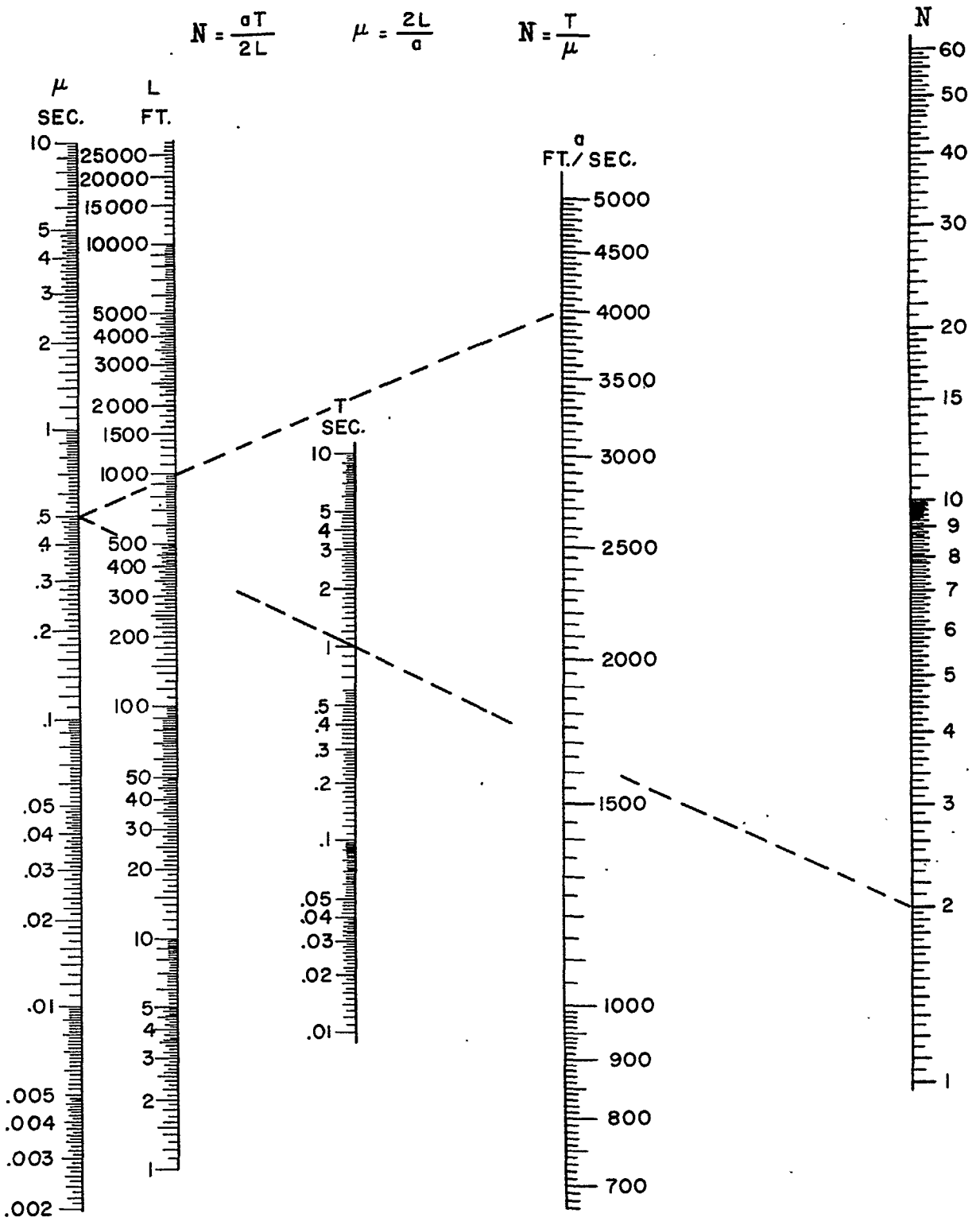

Fig. 6. Nomograph for Time Parameter N

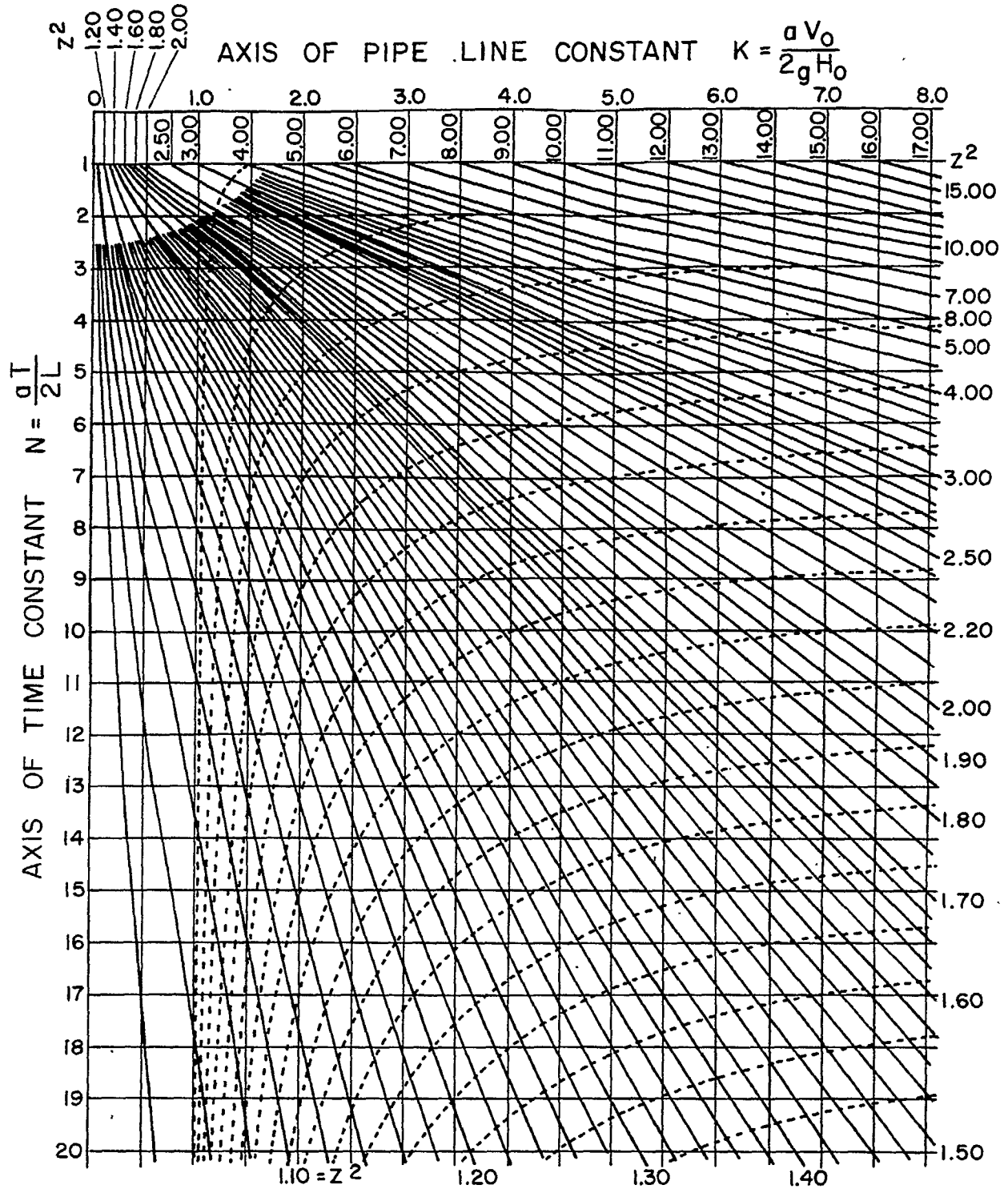

Fig. 7. Chart for the determination of pressure ratio Z^2 for uniform conduits, from L. Allievi's *Theory of Water Hammer*. Reproduced from *Symposium on Water Hammer* by permission of The American Society of Mechanical Engineers. (Dotted lines indicate interval at which maximum pressure occurs.)

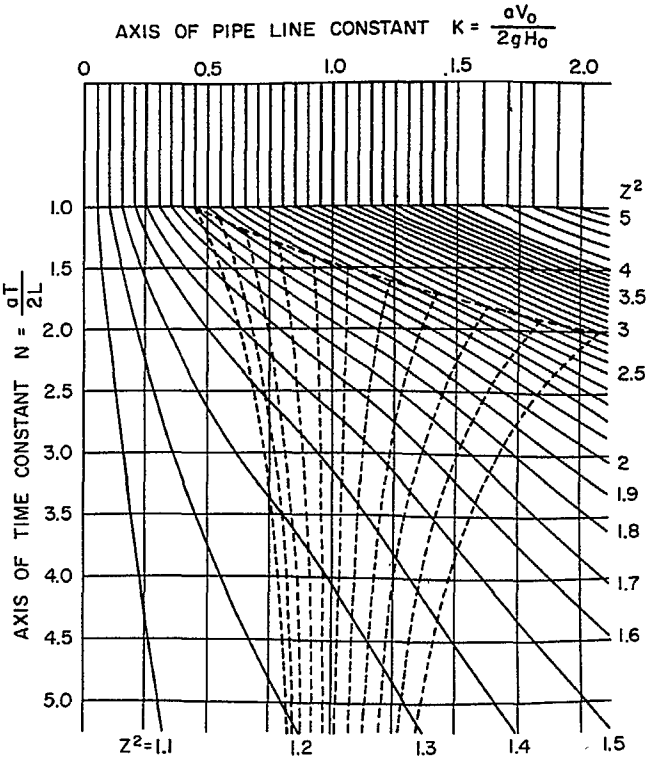

Fig. 8. Allievi chart—pressure rise for uniform gates and simple conduits—small K and N. Reproduced from *Symposium on Water Hammer* by permission of The American Society of Mechanical Engineers.

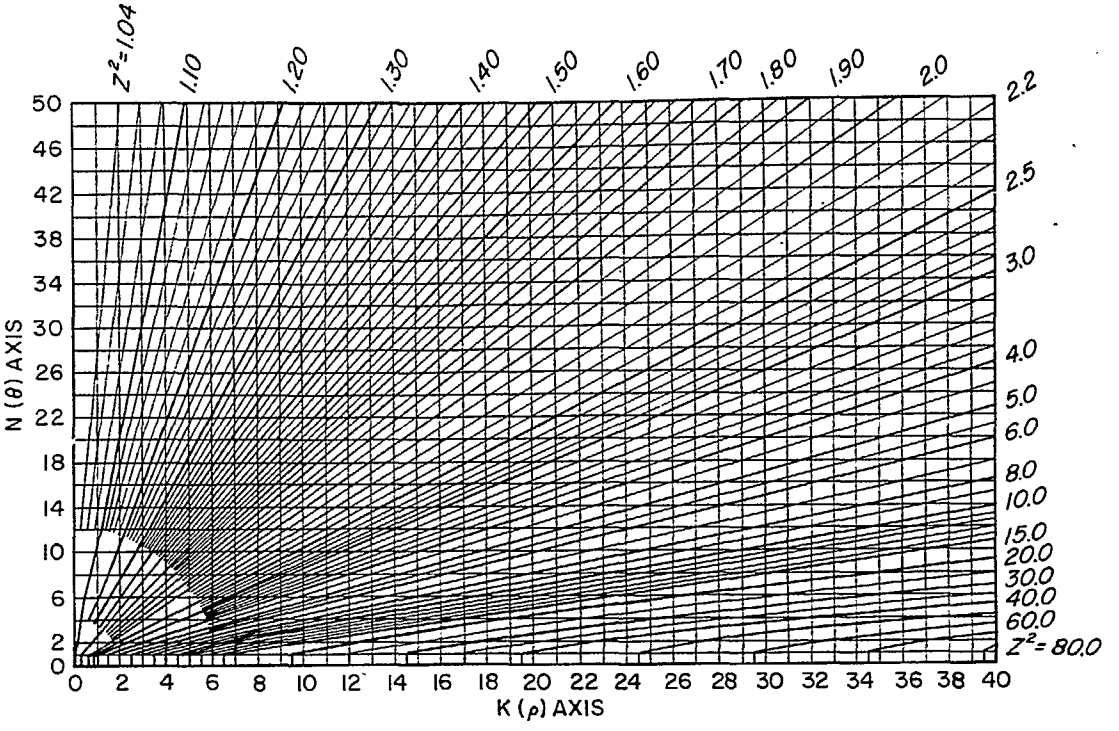

Fig. 9. Allievi chart—pressure rise for uniform gates and simple conduits. Large K and N. *(ASME Paper No. 44-A-42.)* Reproduced by permission of McGraw-Hill Book Company from Rich's *Hydraulic Transients*, copyright 1951. Further reproduction not authorized.

To illustrate the use of these charts let us further extend the examples previously cited. Given data:

Pipe length, $L = 1000$ ft
velocity of pressure wave, $a = 4000$ fps
valve closure time, $T = 1$ sec
initial pressure, $p_0 = 173$ psig
specific weight of fluid, $w = 62.4$ lb/cu ft
flowing velocity, $v_0 = 12.7$ fps
Find the total surge pressure, p_1.

Values for $N\ (=2)$ and $K\ (=2)$ have already been determined. Using the enlarged chart, Fig. 8, at the intersection of $K = 2$ (horizontal axis) and $N = 2$ (vertical axis) read $Z^2 = 2.8$.

Since

$$Z^2 = \frac{\textit{total surge pressure}}{\textit{initial pressure}} = \frac{p_1}{p_0} \qquad (26a)$$

the maximum pressure p_1 for the example given will be $2.8 \times 173 = 484$ psig.

Surge Wave Shape for $T > \mu$

Tracings of three simultaneously recorded pressure-time measurements made at the valve, at the mid-point of the pipe, and at the centrifugal pump discharge in a fuel transferring system are shown in Fig. 10. The rounded wave shape is characteristic of valve closure times $T > \mu$. Test conditions were the following:

Pipe length, $L = 2066$ ft
Velocity of pressure wave, $a = 2780$ fps
Specific weight of liquid, $w = 50$ lb/cu ft
Initial flowing velocity, $v_0 = 6.8$ fps
Valve closure time, $T = 3.6$ sec
Initial pressure at valve, $p_0 = 20$ psig
Total surge pressure, $p_1 = 130$ psig
Static no-delivery head of pump = 50 psig

From these data K and N are determined to be 5.10 and 2.32 respectively. Application of the chart, Fig. 7, yields a theoretical value of 6.5 for Z^2, which is in excellent agreement with the measured pressure ratio.

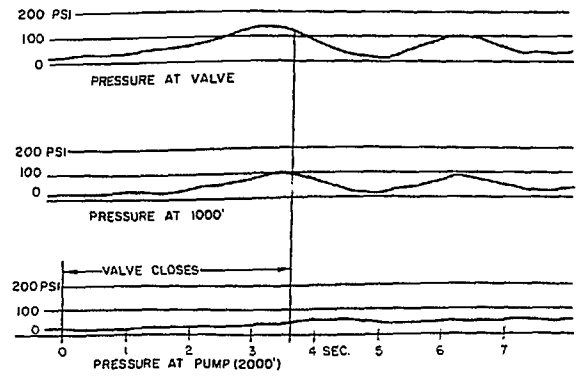

Fig. 10. Simultaneous Pressure-Time Records for Slow Closure

It is worthy of note that during the half cycle of reduced pressure the minimum pressure recorded at the valve was 12 psig. As the pressure reduction, 20-12 psi, is a measure of the reverse flow velocity, $-v$, of the liquid which is flowing toward the pump during the interval between L/a and $2L/a$, it is obvious that $-v$ is considerably less than the initial velocity v_0. The maximum pressure will therefore occur during the first cycle under these and similar conditions. Dampening of the pressure oscillations will be rapid, the asymptotic value being the static no-delivery head of the pump.

Effective Velocity of Pressure Wave

In a complex piping system the effective value of a will be that of the section of pipe nearest the valve when $T < \mu$.

When $T > \mu$, or in systems employing suppression equipment, the effective velocity of the pressure wave,

$$a_e = \frac{L_A + L_B + \ldots L_N}{\frac{L_A}{a_A} + \frac{L_B}{a_B} + \ldots \frac{L_N}{a_N}} \text{ fps} \qquad (27)$$

where subscripts $A, B \ldots N$ refer to the individual section of pipe.

PART II—SURGE SUPPRESSING DEVICES

Based on an understanding of the phenomena of water hammer and the application of the principle that the pressure rise is proportional to the rate of deceleration of the fluid, a number of surge-controlling devices have been developed to bring the column of moving liquid to a more gradual stop. These devices may be divided into two basic types: those which by-pass some of the fluid around the shutoff valve and those which temporarily store some of the fluid in a tank or compression chamber.

By-Pass Type Suppressors

By-pass suppressors reduce surge pressures by effectively increasing the time of valve closure, T, with resultant proportional increase in the value of N. For any given value of K, the ratio, Z^2, of the maximum pressure to the initial pressure will be reduced accordingly, as graphically illustrated on the Allievi charts, Figs. 7-9.

Relief Valve

The simplest form of a by-pass suppressor is the relief valve, which may discharge either to atmosphere or into the piping downstream from the shutoff valve, according to the requirements of the system. Two schematic diagrams showing possible

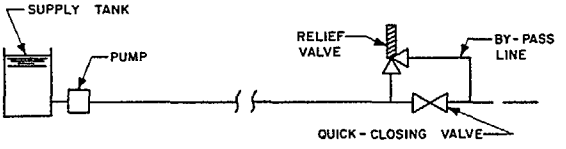

Fig. 11. Relief Valve at Shutoff Valve

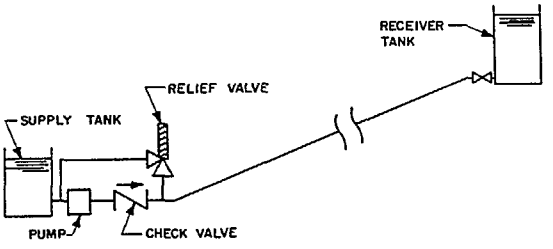

Fig. 12. Relief Valve in Pump Discharge Line

arrangements of systems incorporating relief valves are presented in Figs. 11 and 12. In Fig. 12 an uphill pumping system is shown. With power failure, the column of liquid slows down and stops when its forward momentum is converted into friction losses and potential energy. It then surges back toward the pump, slamming the check valve closed, and, if relief were not provided, producing severe water hammer. A more detailed description and analysis of uphill pumping systems will be found in Part III.

The chief advantages of a relief valve are its small size, ease of installation, and low first cost. Its limitations often outweigh these considerations. It cannot store the fluid in the system. It may require frequent inspection and maintenance. The inertia of its moving parts may cause it to open and close in unison with the arrival and departure of the surge wave,thus magnifying the conditions it is intended to control.

Automatic Surge Suppressor

These objections have been overcome in the Automatic Surge Suppressor, shown in Fig. 13 and re-

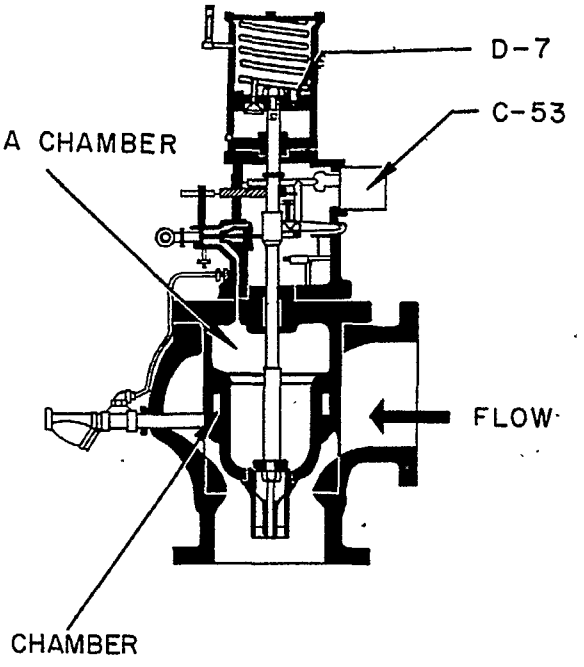

Fig. 13. Automatic Surge Suppressor (Courtesy Pelton Water Wheel Co.)

produced through the courtesy of the Pelton Water Wheel Company. Power failure in an uphill pumping system de-energizes the solenoid control, *C-53*, allowing system pressure in *B* chamber to open the main valve. When the valve is fully open, a reversing mechanism admits system pressure to *A* chamber and the valve slowly closes, its rate depending upon the adjustment of needle valve *D-7* in the oil-

filled dashpot. Automatic Surge Suppressors of this type have found their greatest acceptance in water supply and hydroelectric systems, where they are installed at the outlet of the check valve in uphill pumping systems, or on the upstream side of the turbine control valve in hydroelectric systems.

Storage Type Suppressors

Surge Tanks

The oldest surge suppressor in use is the surge tank, shown schematically in Fig. 14. It is a large

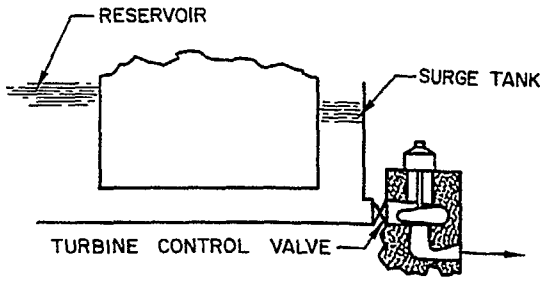

Fig. 14. Surge Tank Installation

standpipe open to the atmosphere and having a cross-sectional area several times that of the line it protects. With rapid closure of a downstream control valve, as in a hydroelectric system, the water flowing in the supply line is diverted into the surge tank until the kinetic energy of the moving water column has been converted into added potential energy in the standpipe. A summary of the methods of analysis developed by Gaden and Calame and translated by Eugene E. Halmos is given in Mr. Halmos' article, "The Effect of Surge Tanks on the Magnitude of Water Hammer in Pipe Lines," in *Symposium on Water Hammer* (Ref. 4).

Improved performance at lower installation cost is obtained by the use of an internal riser pipe and a two-directional orifice control in the Johnson Differential Surge Tank. The analysis developed by R. D. Johnson will be found in References 9 and 10.

The advantages of a surge tank are its over-all efficiency and negligible maintenance requirements. When a surge tank is planned and constructed during the installation of the associated piping system, its first cost may be reasonable. Addition of surge tanks to existing systems, however, may involve considerable modification and expense. The use of surge tanks is limited to water handling systems.

Air Chambers

If the top of the surge tank were closed, thus preventing the escape of the air entrapped above the liquid, a significant reduction in the size of vessel required to limit the maximum developed pressures would be obtained. Pressure vessels designed for this function are referred to as air chambers, and are shown schematically in Fig. 15. The volume occupied by the entrapped air at the pressure existing

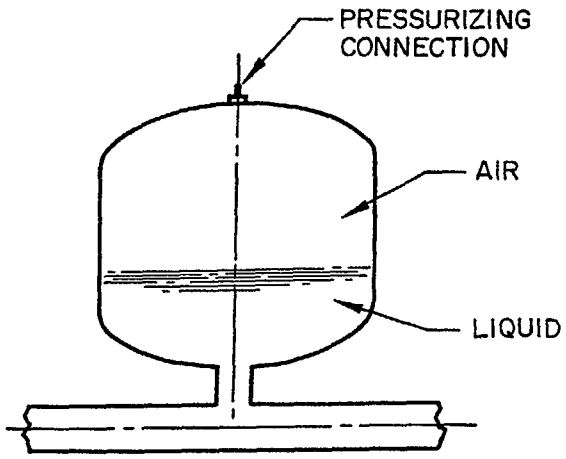

Fig. 15. Air Chamber

in the main flow line during steady state conditions is defined as the CAPACITY of the air chamber. Assuming isothermal conditions, the air volume varies inversely as the absolute pressure. When flow in the main line is stopped, surge reduction is effected by conversion of the kinetic energy in the system into elastic energy stored in the compressed air as well as in the pipe and liquid.

Air chambers possess the following advantages over open-top surge tanks: reduction in size, particularly in the required height, and a simpler and less costly installation. Opposed to these benefits, air chambers exhibit certain serious shortcomings. If the pressure in the main system is ever reduced below the design pressure for steady flow conditions, the air chamber will lose some of its entrapped air and, as a result, its capacity will be reduced. The solubility of air in water and various other liquids makes recharging the air chamber imperative at frequent intervals. In some systems a small air compressor is installed to perform this service. The number of liquids which may be satisfactorily handled with an air chamber is limited.

Gas-Liquid Separating Media

Correction of the deficiencies inherent in the air chamber could obviously be accomplished by separating the air or other gas from the liquid surface by a movable piston or by some flexible material capable of stretching like a sleeve or diaphragm,

collapsing like a bladder, or expanding like a bellows. Once the type had been established, the material must be chosen to meet the requirements of flexibility throughout the operating temperature range, fatigue strength for the expected life cycling of the device, and chemical inertness to the liquids handled by the associated system. Consideration should be given to the strength-to-weight ratio of the material, since its section modulus will govern its sensitivity to the differential pressures imposed. Insertion of any gas-liquid separating medium immediately affects the rate of heat transfer during compression of the gas. It was assumed that isothermal conditions prevailed in the true air chamber device. Test results have indicated the validity of this assumption.

Thick diaphragms or gas-filled bladders of synthetic rubber possess insulating properties which reduce the transfer of heat during the compression cycle and more nearly approach adiabatic conditions.

From the basic gas laws

$$p_0{}^\circ C_0{}^n = p_m{}^\circ C_m{}^n \qquad (28)$$

where $p_0{}^\circ$ and $p_m{}^\circ$ are the initial and final absolute gas pressures, C_0 and C_m are the corresponding gas volumes, and n is the polytropic exponent of gas expansion, for a required volume, C_m, the value of C_0 depends upon the value of n. For the complete transfer of generated heat, isothermal conditions prevail, $n = 1$, and Equation (28) reduces to

$$p_0{}^\circ C_0 = p_m{}^\circ C_m \qquad (28a)$$

When no heat is transferred, n is equal to the ratio of specific heats ($n = 1.4$ for nitrogen gas). When some heat transfer occurs the value of n lies between 1 and 1.4. It is obvious from the foregoing discussion that for a specified volume at maximum pressure, C_m, the smallest initial volume or capacity C_0 will be required when the value of n approaches 1. Suppressors having the lowest value of n will thus be most efficient.

Actual values for n applicable to pneumatic suppressor operation are not constant, but vary inversely as a function of the time interval during which the suppressor receives the volume of liquid $C_0 - C_m$ displaced from the associated piping system. In the following discussions the cited values for n apply to systems protected by adequately sized suppressors whose filling time—a function of line length, suppressor capacity and proper orificing—is three seconds or longer. For systems having line lengths less than 3000 feet or for a pressure ratio $p_m{}^\circ/p_0{}^\circ$ greater than 5 the suppressor will fill in less than three seconds and n will approach the adiabatic value.

Piston-Type Suppressors

A cross-sectional sketch of a floating piston-type suppressor is shown in Fig. 16. Close machining tolerances are required to allow easy movement of the plunger. Piston friction and inertia may result in delayed response to pressure differences with the result that the piston, acting like a dead end, may

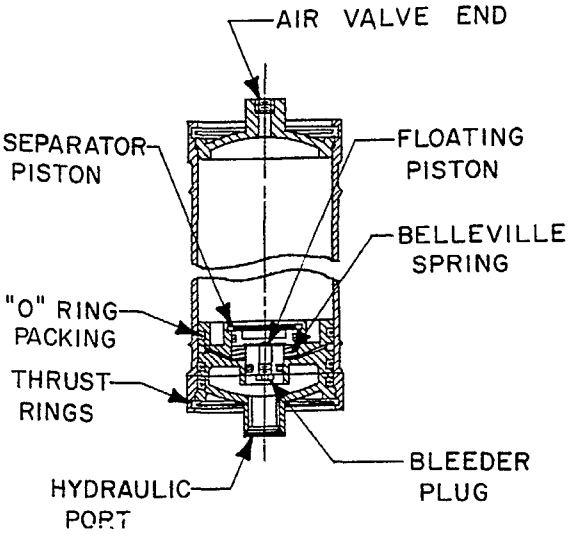

Fig. 16. Floating Piston Type Suppressor

reflect the front of a surge wave instead of absorbing it. The principal drawback to a piston suppressor is the slow but constant leakage of the pressurizing gas past the O-rings. Frequent maintenance and inspection are necessary. The O-ring material is critical, since it must withstand abrasion, chemical attack, and remain dimensionally constant throughout the temperature range imposed by the fluid handled in the system. For piston-type suppressors, a value of n between 1.2 and 1.3 may be employed.

Bladder-Type Suppressors

The generic type referred to as a bladder-type suppressor is illustrated diagramatically in Fig. 17. Pressurizing gas—air or nitrogen—expands the bladder against the walls of the external housing. The volume of liquid entering from the associated system collapses the bladder and compresses the gas.

For suppressors of this type, the value of n depends upon the material and thickness of the bladder. In most instances it will be between 1.15 and 1.25, as determined from reliable test data.

Among the advantages claimed for the bladder-type suppressor are ease of installation and reasonable first cost.

The principal disadvantages stem from the bladder itself. When the device is not in active use—that is, when surge pressures are not being applied—the fibers of the bladder are under a constant strain from the internal gas pressure. Continued flexing and abrasion from the housing walls weaken these

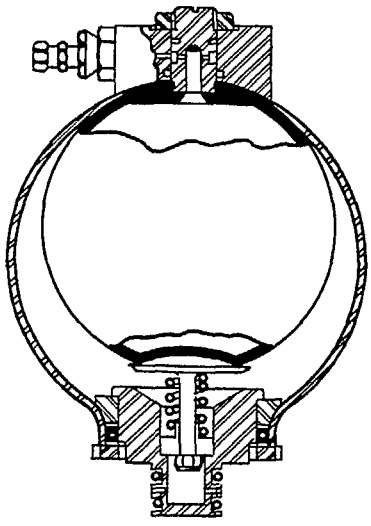

Fig. 17. Bladder Type Suppressor

fibers and result in premature failure. The material itself—usually neoprene or Buna N—has a definitely limited field of applications, since it is attacked by many fuels and most chemicals in common use. Its usable temperature range is from about $-20°$ to $+130°$ F. Because of the critical nature of the bladder, frequent inspection and replacement may be required.

Diaphragm-Type Suppressors

Figure 18 illustrates a suppressor of the diaphragm type. Its similarity to the bladder type is apparent. Since the diaphragm is usually made of neoprene or Buna-N, the evaluation of the bladder suppressor will apply as well to the diaphragm type.

Sleeve-Type Suppressors

An interesting variant of the bladder type suppressor is shown in Fig. 19. Liquid in the system passes through the central pipe which is perforated with small holes along its entire length. A synthetic rubber sleeve, attached at both ends, encompasses the pipe. The outer housing contains the pressurizing nitrogen gas. Increased liquid pressure inside the central pipe causes the sleeve to swell, compressing the gas in the housing.

Values of n ranging between 1.15 and 1.2 are applicable to suppressors of this construction.

Because its configuration permits the suppressor to be installed either in the main flow line or on a branch connection (with one end capped), a series of tests was conducted as one phase of the cited in-

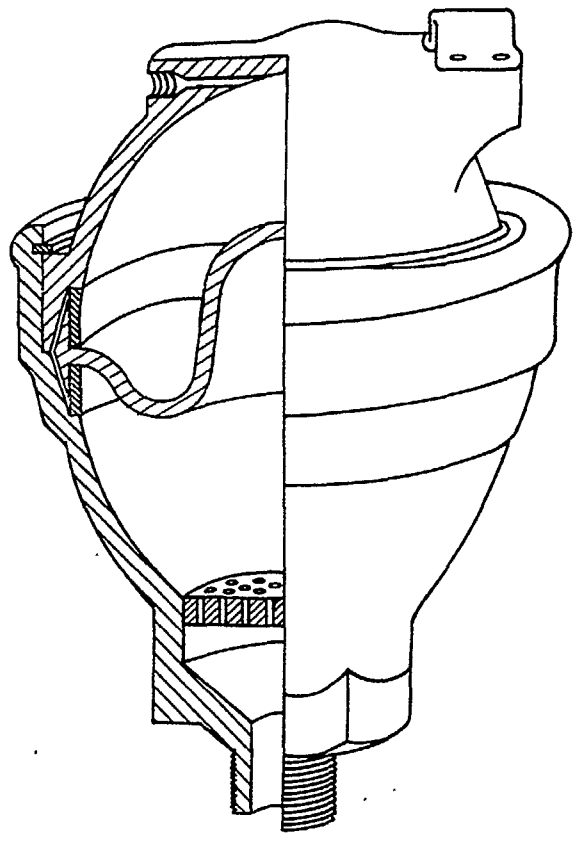

Fig. 18. Diaphragm Type Suppressor

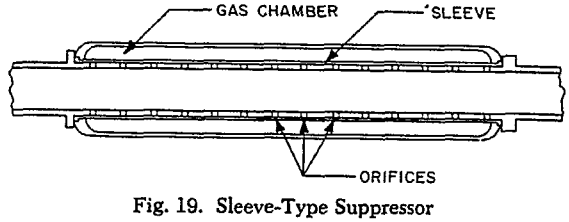

Fig. 19. Sleeve-Type Suppressor

vestigation to determine the relative surge reducing characteristics of the alternate installations. Over a wide range of variables—flow rates, system pressures, line lengths—no measurable difference could be determined. It was concluded that an "in-line" suppressor was equivalent to an "appendage" suppressor and its position in the system should be governed by piping and space requirements.

The chief advantages of the sleeve-type suppressor are its simplicity of construction, optional positioning for installation, low ratio of weight to capacity, and usually reasonable first cost.

The disadvantages are mainly attributable to the synthetic rubber sleeve, which has a limited flexural life, and which becomes brittle below − 30°F and tacky above +120° F. Even at moderate ambient temperatures, between + 50°F and + 100°F, the sleeve may allow the pressurizing gas to escape. Synthetic rubber is attacked by aromatic hydrocarbons, by most acids and bases, and by many salts.

Pneumatic Bellows-Type Suppressor

Suppressor construction which successfully overcomes temperature limitations, chemical attack, loss of pressurizing gas and flexural fatigue is accomplished in the stainless steel bellows design, one

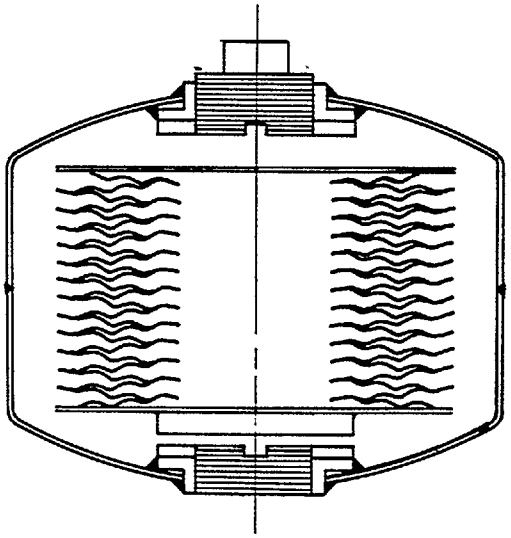

Fig. 20. Non-Pressurized Hydropad

form of which is shown in Fig. 20. This unit, designed and fabricated by the MagniLastic Division of the Cook Electric Company, and designated the "Hydropad," contains an inner collapsible chamber filled with air or nitrogen gas at atmospheric pressure. Liquid from the system fills the outer housing and surrounds the bellows assembly. Surge pressures originating in the system are transmitted to the liquid in the housing and absorbed by compression of the gas within the bellows. Because of its low first cost and the simplicity of obtaining any required capacity by connecting several Hydropads in series, this unit is especially adaptable to low pressure systems (50 psi and below).

For application at pressures above 50 psi, a nitrogen-pressurized suppressor of the type shown in Fig. 21 was developed by MagniLastic. Gas, charged into the domed housing to the pressure maintained in the system during initial steady flow conditions, compresses the bellows assembly. Liquid displaced

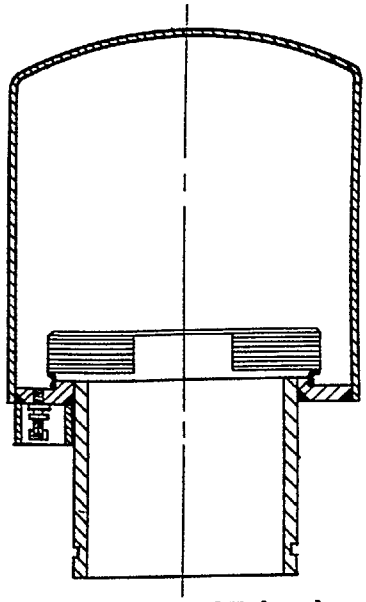

Fig. 21. Pressurized Hydropad

from the system during surge conditions enters the bellows, extending them and compressing the gas (Fig. 22). In the configuration shown in Figs. 21 and 22 the top plate terminating the bellows assembly, the bellows diaphragms, and the lower attach-

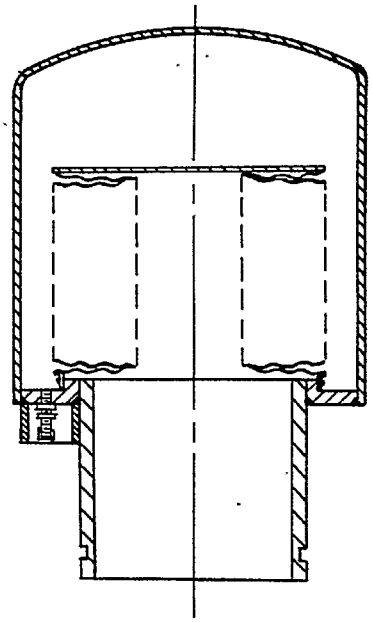

Fig. 22. Pressurized Hydropad with Bellows Extended

ing ring are all fabricated from Type 347 stainless steel. Where all suppressor parts in contact with the liquid must be corrosion- or temperature-resistant, the entrance connection and a lower ring liner are also made of stainless steel. Such a design is shown in Fig. 23. Suppressors of this construction have an operating temperature range from −85° F to +750° F and have given years of trouble-free service in fuel, chemical, beverage and water handling systems.

For greater temperature extremes (−325° to +1000° F) an all-stainless steel model is recommended. Such a unit would find application in liquid nitrogen or liquid oxygen transfer systems and high temperature chemical processes.

Standard models are produced in capacities ranging from 60 cubic inches to 50,000 cubic inches in 200-, 500-, 1000-, and 3000-psi pressure ranges. Other capacities and pressure ranges are available on special order.

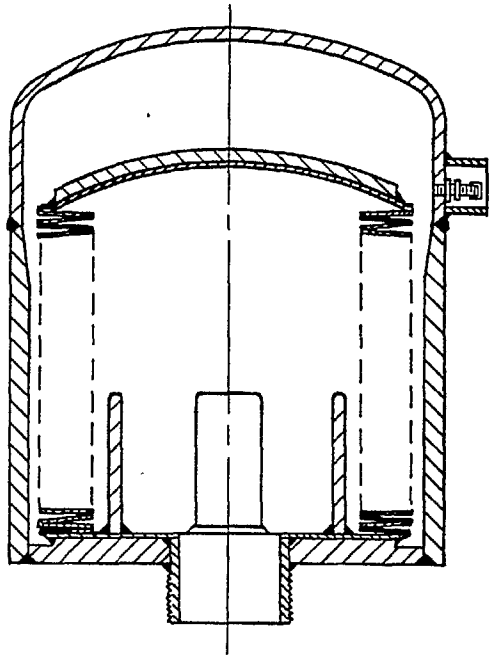

Fig. 23. Stainless Steel–Trimmed MagniLastic Hydropad

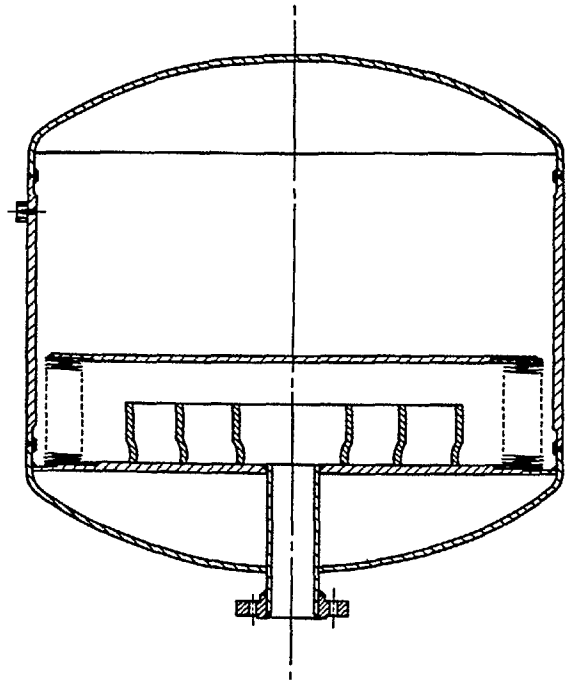

Fig. 24. 55,006 Cubic Inch Hydropad

Six units of the construction shown in Fig. 24 were installed at the terminus of a five-mile fuel transfer line to limit maximum surge pressures to 250 psi. This installation is shown in Plate V. Each

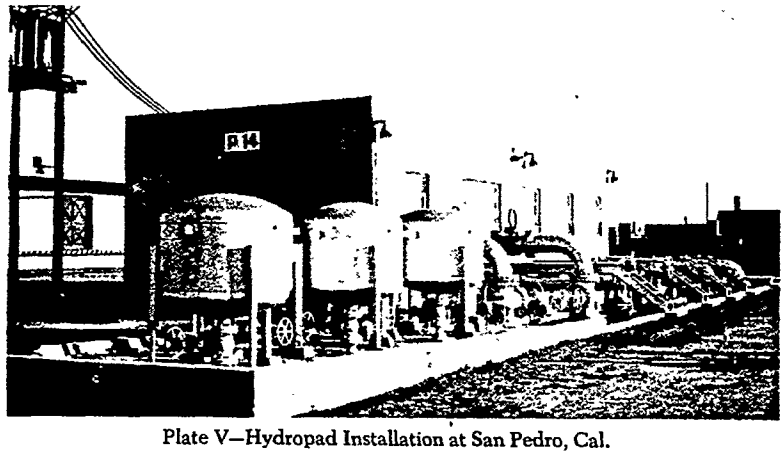

Plate V–Hydropad Installation at San Pedro, Cal.

suppressor has a capacity of 55,006 cubic inches and is precharged with nitrogen gas to the line pressure of 100 psi.

Welding of the lowest bellows diaphragm to the bottom plate provides an effective seal against loss of the pressurizing gas with resultant reduction of maintenance requirements and complete elimination of hazard to operating personnel. All welding is performed in MagniLastic shops by government-certified welders. Design, fabrication and testing are conducted in accordance with the applicable requirements of the API-ASME Code for Unfired Pressure Vessels.

Tests made under laboratory controlled conditions have demonstrated the life expectancy of stainless steel MagniLastic bellows assemblies to be in excess of 10,000,000 cycles of completely reversed stress at full travel produced by the alternate application and release of fluid pressures.

By extensive test a value of 1.015 for n has been determined for MagniLastic Hydropad construction. Close approach to isothermal conditions results in high over-all efficiency.

Evaluation of Storage-Type Suppressors

The following Code is used in the table below: 1—Excellent, 2—Very Good, 3—Good, 4—Fair, 5—Poor, 6—Very Poor.

TYPE	*Air Chamber*	*Floating Piston*	*Bladder*	*Diaphragm*	*Sleeve*	*Bellows*
Cost vs. capacity	1	3	3	3	2	3
Weight vs. capacity	1	3	2	2	2	3
Maintenance frequency	6	5	4	4	3	1
Gas leakage or absorption	6	5	1	3	3	1
Liquid leakage	1	5	1	2	2	1
Life expectancy	1	4	4	4	3	1
Operational safety	2	2	3	3	2	1
Efficiency	1	5	3	4	3	1
Operating temperature range	6	5	4	4	4	1
Range of liquids handled	6	5	4	4	4	1
Averages	3.1	4.2	2.9	3.3	2.8	1.4

PART III—SURGE CONTROL WITH PNEUMATIC SUPPRESSORS

Effect of Suppressors

When the velocity of a fluid flowing in a conduit is decelerated from v_0 to any new value, v_1, with a resultant increase in pressure, an adequately sized pneumatic suppressor will displace a quantity of fluid over a sufficiently long period of time to establish a new value of N, the time parameter, such that $N>1$ and the resultant pressure rise is correspondingly reduced from the maximum value found by applying Equation (22). In this respect inclusion of a suppressor produces a smooth and gradual pressure build-up similar to that of slow valve closure. A tracing of three simultaneous pressure-time records sensed in different locations in a fuel transferring system will be found in Fig. 25. Similarity of these pressure records to those made during slow valve closure (Fig. 10) should be noted.

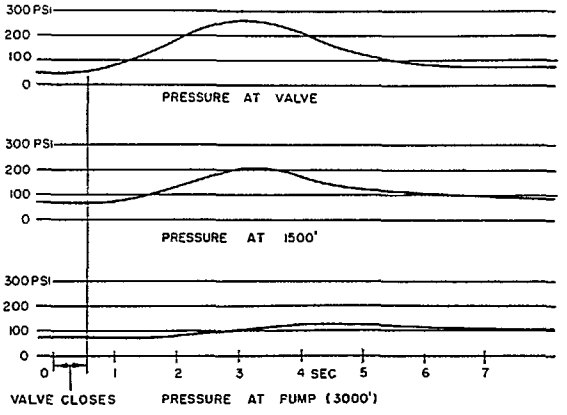

Fig. 25. Simultaneous Pressure-Time Record for a System Incorporating Suppressors

With respect to Fig. 25, the following test conditions were recorded:

Pipe length, $L = 3078$ ft

Flow area of pipe, $A = 0.192$ sq ft

Test fluid: JP-4; Specific weight, $w = 50$ lb/ft^3

Velocity of pressure wave, $a = 2930$ fps

Flowing velocity of fluid, $v_0 = 7.1$ fps
Total suppressor capacity, $C_0 = 2.836$ cu ft
Effective value of polytropic exponent, $n = 1.125$
Initial pressure at valve, $p_0 = 50$ psig
Maximum pressure at valve, $p_m = 260$ psig
Suppressor precharge pressure, $p_0 = 50$ psig

It should be noted that at the instant of maximum surge pressure at the valve, pressures within the system diminish toward the pump. The average pressure rise, $(p_m - p_0)/2$ should be used in calculating the amount of pressure energy at that instant in the expanded pipe and compressed liquid.

The Energy Balance

Assuming valve closure in less than $2L/a$ seconds, the method of balancing the energy present in the system before and during valve closure against the energy stored in the compressed gas within the suppressors and the elastic energy stored in the expanded pipe and compressed fluid at the instant of maximum pressure is the most satisfactory theoretical approach to the sizing of pneumatic suppressors. Sufficient empirical data were recorded during controlled tests to prove the validity of the formulas and their applicability to all ranges of pipe diameter, length, flow velocity, suppressor capacity, and any other variables.

The following brief discussion presents a simplified form of the hydraulic energy balance. Its derivation and a more detailed theoretical analysis will be found at the end of Part III. As corroborated by test data, the formulas given here will yield conservative results from zero to five percent.

The energy present in the system before and during valve closure is the sum of the kinetic energy

$$W_0 = wALv_0^2/2g \quad \text{ft-lb} \qquad (12)$$

where w = Specific weight of fluid, lb/cu ft
A = Cross-sectional area of pipe, sq ft
L = Pipe length, ft
v_0 = Initial fluid velocity, fps
g = Acceleration of gravity = 32.2 ft/sec^2

and the flow work γ of the liquid entering the pneumatic suppressor:

$$\gamma = 144 p_0^\circ C_0 \left[1 - \left(\frac{p_0^\circ}{p_m^\circ} \right)^{\frac{1}{n}} \right] \text{ ft lb} \qquad (29)$$

where p_0° = Initial flowing pressure, absolute, psia
C_0 = Capacity of suppressor, cu ft
p_m° = Maximum pressure, absolute, psia
n = Polytropic exponent of gas expansion

It is assumed that the initial charging pressure of the suppressor is equivalent to the flow pressure, p_0.

At the instant of maximum surge the suppressor will have stored a quantity of potential energy, α, such that

$$\alpha = \frac{144 p_0^\circ C_0}{n-1} \left[\left(\frac{p_m^\circ}{p_0^\circ} \right)^{\frac{n-1}{n}} - 1 \right] \text{ ft-lb} \qquad (30)$$

and simultaneously the expanded pipe and compressed liquid will store, with an average pressure rise $(p_m - p_0)/2$, an amount of energy, δ:

$$\delta = \frac{10{,}368 ALg}{a^2 w} \left(\frac{p_m - p_0}{2} \right)^2 \quad \text{ft-lb} \qquad (31)$$

in which a = velocity of pressure wave, fps

p_m and p_0 are gauge values corresponding to p_m° and p_0°

The energy balance is thus given by:

$$W_0 + \gamma = \alpha + \delta \quad \text{ft-lb} \qquad (32)$$

Solution for the required suppressor capacity, C_0, yields

$$C_0 = \frac{W_0 - \delta}{\frac{144 p_0^\circ}{n-1} \left[\left(\frac{p_m^\circ}{p_0^\circ} \right)^{\frac{n-1}{n}} - 1 \right] - 144 p_0^\circ \left[1 - \left(\frac{p_0^\circ}{p_m^\circ} \right)^{\frac{1}{n}} \right]} \text{ cu ft} \qquad (33)$$

At maximum pressure, p_m, the corresponding volume, C_m, occupied by the gas is

$$C_m = C_0 \left(\frac{p_0^\circ}{p_m^\circ} \right)^{\frac{1}{n}} \text{ cu ft} \qquad (34)$$

The volume of liquid, V_d, entering the suppressor is

$$V_d = C_0 - C_m = C_0 \left[1 - \left(\frac{p_0^\circ}{p_m^\circ} \right)^{\frac{1}{n}} \right] \text{ cu ft} \qquad (35)$$

The volume, C_x, occupied by the gas at any pressure p_x° is

$$C_x = C_0 \left(\frac{p_0^\circ}{p_x^\circ} \right)^{\frac{1}{n}} \text{ cu ft} \qquad (36)$$

For non-pressurized suppressors where $p_x^\circ = 14.7$ psia and for suppressors charged to some pressure, p_x°, other than line pressure, p_0°, the required capacity, C_x, will be found from Equation (36), once the value of C_0 is determined.

To simplify the calculation of the pressure ratio factors, Figs. 26 and 27 have been prepared in which the value of n, determined from test data as applicable to suppressors of MagniLastic stainless steel bellows construction, is taken at 1.015.

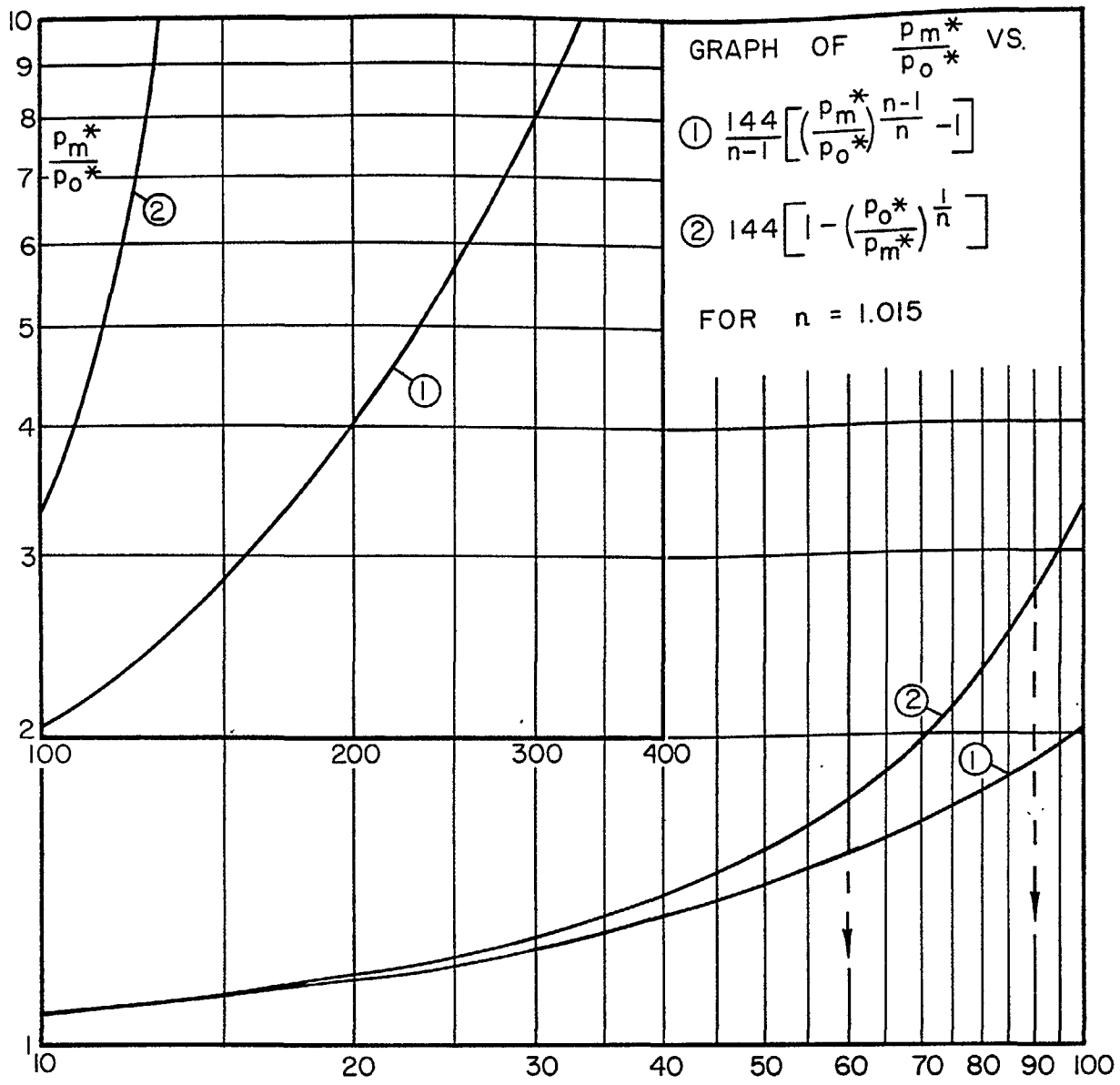

Fig. 26. $\frac{p_m^\circ}{p_n^\circ}$ vs. Capacity Factors

If Equation (33) is put into the form

$$C_0 = \frac{W_0 - \delta}{p_0^\circ Y} \text{ cu ft} \qquad (33a)$$

where $Y = 144\left\{\frac{1}{n-1}\left[\left(\frac{p_m^\circ}{p_0^\circ}\right)^{\frac{n-1}{n}} - 1\right] - \left[1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}\right]\right\}$ (37)

a convenient graphical solution of Y *vs* the ratio p_m°/p_0° for $n = 1.015$ can be used to expedite the computation of C_0. This graph appears as Fig. 28. By its use a close approximation of p_m for a given suppressor capacity C_0 may also be found. The error lies in the predetermination of δ, which may usually be estimated within a few percent once the constants in Equation (31) have been evaluated.

Line friction drop is included in the term γ. Refer to Hydraulic Energy Balance, page 28.

The maximum initial liquid velocity v_0 permitted for a given suppressor capacity, C_0, initially charged to line pressure, p_0, and limiting the maximum surge pressure to p_m is

(38)

$$v_0 = 12\sqrt{\frac{\frac{2gp_0^\circ C_0}{n-1}\left[\left(\frac{p_m^\circ}{p_0^\circ}\right)^{\frac{n-1}{n}} - 1\right] - 2gp_0^\circ C_0\left[1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}} + \frac{144g}{a^2 w}\left(\frac{p_m - p_0}{2}\right)^2\right]}{wAL}} \text{ fps}$$

The following example is given to illustrate the use of the formulas in a simple system.

Given data:

Pipe length, $L = 10{,}000$ ft

Pipe size: 10-inch schedule 40 (See Table V)

Pipe I.D., $d = 10.020$ in.

Pipe wall thickness, $e = 0.365$ in.

Flow area, $A = 0.548$ sq ft

Pipe material, steel

Modulus of elasticity (Young's), $E = 30{,}000{,}000$ psi

Liquid handled: Water

Specific weight of water, $w = 62.4$ lb/cu ft

Bulk modulus, $k = 294{,}000$ psi

Initial flow rate, $Q_0 = 1000$ gpm

Initial pressure at valve, $p_0 = 75$ psig

Atmospheric pressure, $p_a = 14$ psia

Valve closure time, $T = 2$ sec

Polytropic exponent of suppressor gas compression, $n = 1.015$

Acceleration of gravity, $g = 32.2$ ft/sec^2

Allowable maximum surge pressure, $p_m = 200$ psig

Solution:

By (19) the flow velocity,

$$v_0 = \frac{0.4085 Q_0}{d^2} = \frac{0.4085 \times 1000}{10.020^2} = 4.07 \text{ fps}$$

The value of a can be determined from the nomograph, Fig. 4, as 4160 fps. Actual calculation yields

$$a = \frac{12}{\sqrt{\frac{w}{g}\left(\frac{1}{k} + \frac{d}{eE}\right)}} \qquad (14)$$

$$a = \frac{12}{\sqrt{\frac{62.4}{32.2}\left(\frac{1}{294{,}000} + \frac{10.020}{0.365 \times 30{,}000{,}000}\right)}} = 4157 \text{ fps}$$

Valve closure time resulting in $N < 1$ may be regarded as "instantaneous." Solution for N may be made by means of the nomograph, Fig. 6: $N < 1$. By calculation: $N = \frac{aT}{2L}$ (22)

$$N = \frac{4157 \times 2}{2 \times 10{,}000} = 0.4 \qquad N < 1$$

The pressure rise, $p_1 - p_0$, resulting from instantaneous valve closure for a system without suppressors would be

$$p_1 - p_0 = \frac{v_0 w a}{144g} \qquad (15)$$

$$p_1 - p_0 = \frac{4.07 \times 62.4 \times 4157}{144 \times 32.2} = 228 \text{ psi}$$

Under these conditions, and neglecting line drop, the peak pressure, $p_1 = 228 + p_0 = 228 + 75 = 303$ psi. The suppressor must accordingly reduce the maximum pressure by

$$303 - 200 = 103 \text{ psi}$$

The kinetic energy in the system is

$$W_0 = wALv_0^2/2g \qquad (12)$$

$$W_0 = \frac{62.4 \times 0.548 \times 10{,}000 \times 4.07^2}{2 \times 32.2} = 87{,}956 \text{ ft-lb}$$

The strain energy stored in the pipe at the instant of maximum surge, by Equation (31) is

$$\delta = \frac{10{,}368 \times 0.548 \times 10{,}000 \times 32.2}{4157^2 \times 62.4}\left(\frac{200 - 75}{2}\right)^2$$

$$\delta = 6{,}627 \text{ ft-lb}$$

Since $p_0^\circ = p_0 + p_a = 75 + 14 = 89$ psia

and $p_m^\circ = p_m + p_a = 200 + 14 = 214$ psia

and since $n = 1.015$, $1/n = 0.985$ and $(n-1)/n = 0.0148$, the required MagniLastic suppressor capacity can be found directly from Equation (33):

$$C_0 = \frac{87{,}956 - 6{,}627}{\frac{144 \times 89}{1.015 - 1}\left[\left(\frac{214}{89}\right)^{0.0148} - 1\right] - 144 \times 89\left[1 - \left(\frac{89}{214}\right)^{0.985}\right]}$$

$$C_0 = 21.54 \text{ cu ft}$$

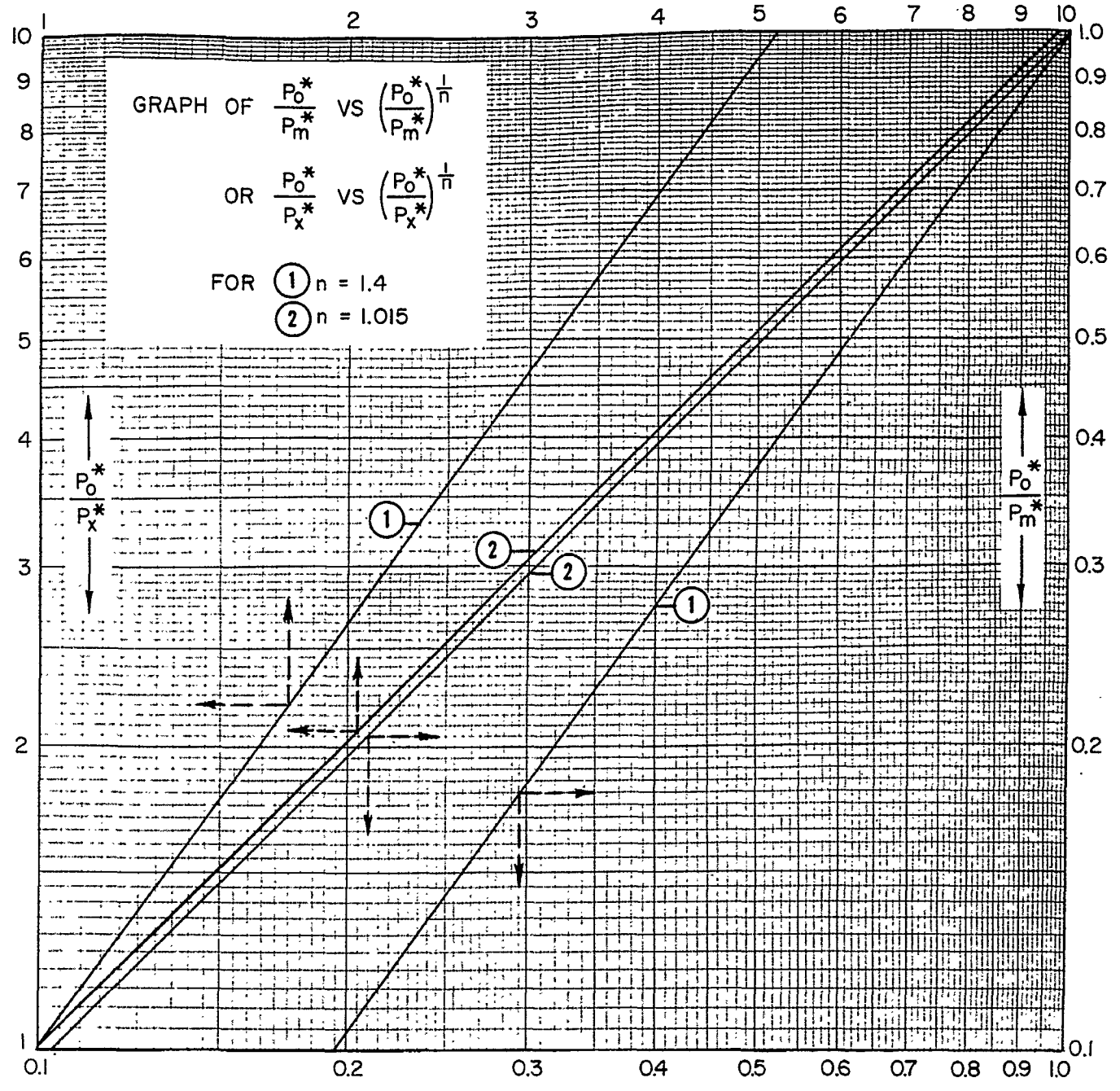

Fig. 27. $\frac{p_o^\circ}{p_m^\circ}$ vs. $\left(\frac{p_o^\circ}{p_m^\circ}\right)^{\frac{1}{n}}$

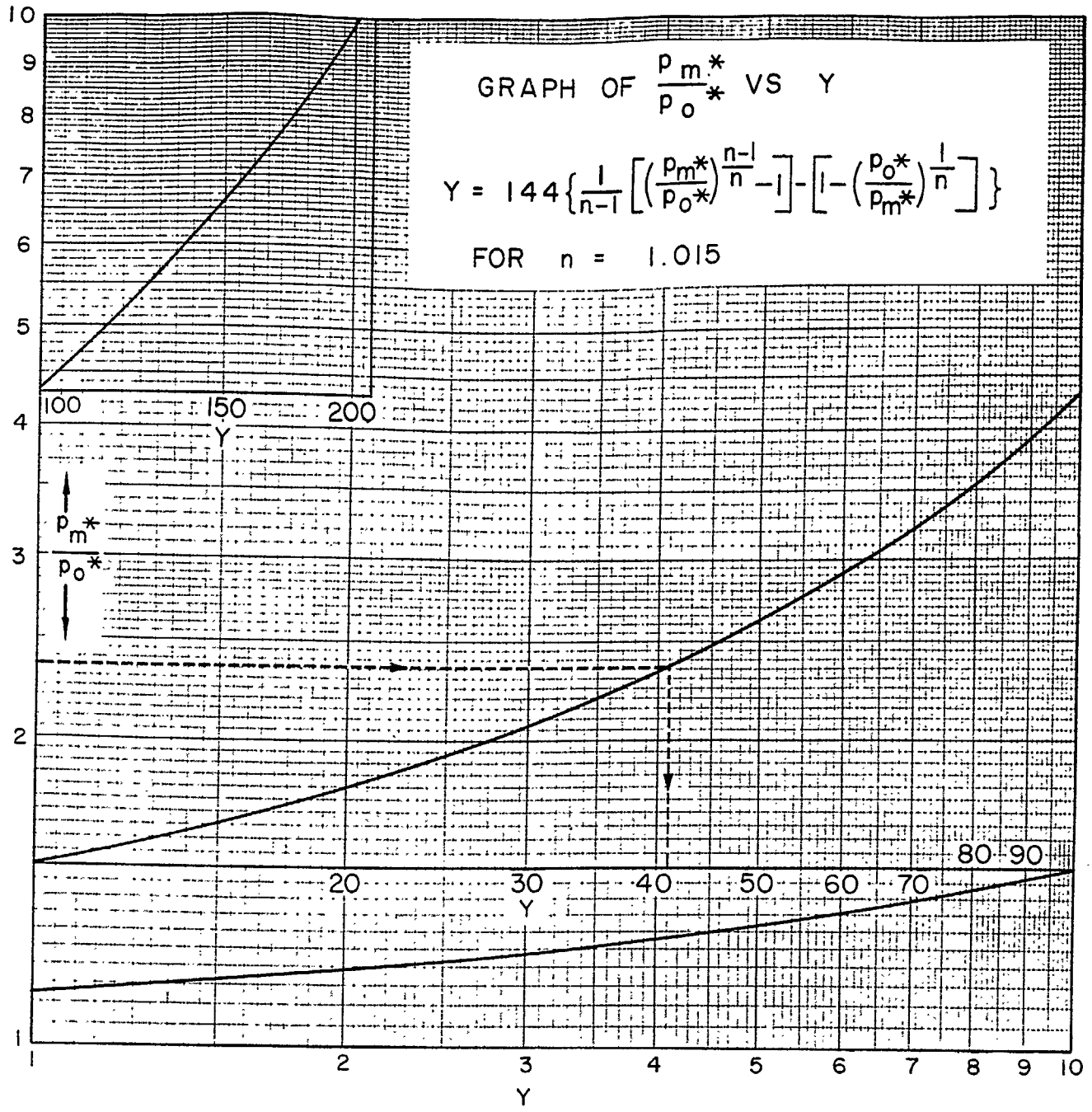

Fig. 28. $\frac{p_m^\circ}{p_0^\circ}$ vs. Y

This computation may be checked by using the simplified method provided by Formula (33a) and Fig. 28:

$p_m^\circ/p_0^\circ = 2.41$

read $Y = 42.5$ from Fig. 28. By (33a):

$$C_0 = \frac{87{,}956 - 6{,}627}{89 \times 42.5} = 21.5 \text{ cu ft}$$

By Equation (35) the volume of water entering the suppressor is

$$V_d = 21.54\left[1 - \left(\frac{89}{214}\right)^{0.985}\right] = 12.47 \text{ cu ft}$$

At the instant of maximum surge the pressurizing gas will occupy a space

$$C_m = C_0 - V_d = 21.54 - 12.47 = 9.07 \text{ cu ft}$$

It will be interesting to examine the actual distribution of energy in the example just given. Solution for α, the energy stored in the gas of the suppressor, by (30) yields:

$$\alpha = \frac{144 \times 89 \times 21.54}{1.015 - 1}\left[\left(\frac{214}{89}\right)^{0.0148} - 1\right]$$

$$\alpha = 241{,}156 \text{ ft-lb}$$

The potential energy or flow work γ of the liquid entering the suppressor, which must ultimately be expelled by part of the energy α, is, from Equation (29):

$$\gamma = 144 \times 89 \times 21.54\left[1 - \left(\frac{89}{214}\right)^{0.985}\right]$$

$$\gamma = 159{,}827 \text{ ft-lb}$$

Summarizing,

$W_0 + \gamma = \alpha + \delta$

$87{,}956 + 159{,}827 = 241{,}156 + 6{,}627$

It is noteworthy that the energy stored in the suppressor is 2.7 times the kinetic energy present in the system before valve closure and that the potential energy of the liquid entering the suppressor makes 66% of the stored energy unavailable for absorption of the kinetic energy.

Complex Systems

Changes in pipe diameter and wall thickness affect the values of W_0 the kinetic energy and δ the pressure energy stored in the pipe. Where subscripts A, $B \ldots N$ refer to individual sections of pipe in a complex system, the modified expression for the total kinetic energy present before valve closure is

$$W_0 = \Sigma w A L v_0^2/2g = (w/2g)\Sigma A L v_0^2$$

$$W_0 = (w/2g)\,(A_A L_A v_A^2 + A_B L_B v_B^2 + \ldots A_N L_N v_N^2) \text{ ft-lb} \qquad (39)$$

With the application of Equation (20) this formula can be simplified to the form:

$$W_0 = (w/2g) V v_e^2 \text{ ft-lb} \qquad (12a)$$

in which v_e is the effective flow velocity and V the total volume.

The pressure energy stored in the complex system at maximum surge is:

$$\delta_1 = \frac{10{,}368 V g}{a_e^2 w}\left(\frac{p_m - p_0}{2}\right)^2 \text{ ft-lb} \qquad (31a)$$

where a_e the effective velocity of the pressure wave will be found from Equation (27).

No changes are required in Formulas (29) or (30) for calculation of γ and α respectively. The energy balance, $W_0 + \gamma = \alpha + \delta$, and Formula (33) and (33a) for C_0 are to be applied without revision except for the new values of W_0 and δ_1, as found from Equations (12a) and (31a) above.

For example consider a complex piping system containing, as in the preceding problem, 10,000 feet of schedule 40 pipe, but of the following diameters and sectional lengths:

Section	*Nominal Diameter*	*Length*
	in.	ft
A (from pump)	14	5000
B	12	2000
C	10	1000
D	8	1500
E (to valve)	6	500

Let the valve closure time be considered instantaneous, and let the other conditions stated in the former example apply. A table can be compiled presenting pipe details.

Dia.	*Wall*	*I.D.*	*Flow Area*	*Length*	*Volume*
in.	in.	in.	sq ft	ft	cu ft
14	0.438	13.124	0.9396	5,000	4,698
12	0.406	11.938	0.7771	2,000	1,554
10	0.365	10.020	0.5479	1,000	548
8	0.322	7.981	0.3472	1,500	521
6	0.280	6.065	0.2007	500	100
				Total volume, cu ft	7,421

Calculation of velocities of propagation of the pressure wave gives, by (14):

$a_A = 4109$ fps
$a_B = 4110$ fps
$a_C = 4157$ fps
$a_D = 4206$ fps
$a_E = 4257$ fps

If no suppressors were installed in this complex system, the pressure rise experienced at the valve—at

the end of the 6-inch line—would be the same as that for a simple system using 6-inch pipe throughout, since $N<1$. With no suppressor capacity the rise, by (15) would be

$$p_1 - p_0 = \frac{v_E w a_E}{144g} \text{ psi}$$

in which v_E and a_E are the fluid velocity and pressure wave velocity, respectively, in the 6-inch pipe.

$$p_1 - p_0 = \frac{11.101 \times 62.4 \times 4257}{144 \times 32.2} = 636 \text{ psi}$$

The effective wave velocity a_e is, by (27)

$$a_e = \frac{5{,}000 + 2{,}000 + 1{,}000 + 1{,}500 + 500}{\frac{5{,}000}{4{,}109} + \frac{2{,}000}{4{,}110} + \frac{1{,}000}{4{,}157} + \frac{1{,}500}{4{,}206} + \frac{500}{4{,}257}}$$

$$a_e = 4{,}136 \text{ fps}$$

Flowing velocities v_0 and kinetic energies W_0, calculated from Equations (19) and (12) respectively, are listed below.

Section	v_0	v_0^2	$w/2g$	V	W_0
A	2.371	5.62	0.969	4,698	25,592
B	2.867	8.22	0.969	1,554	12,377
C	4.066	16.53	0.969	548	8,779
D	6.417	41.18	0.969	521	20,789
E	11.101	123.23	0.969	100	11,941

Total kinetic energy, $W_0 = 79{,}478$ ft-lb

The equivalent effective fluid velocity for the system, by (20):

$$v_e = \sqrt{\frac{5000 \times 2.371 + 2000 \times 2.867 + 1000 \times 4.066 + 1500 \times 6.417 + 500 \times 11.101}{\frac{5000}{2.371} + \frac{2000}{2.867} + \frac{1000}{4.066} + \frac{1500}{6.417} + \frac{500}{11.101}}}$$

$$v_e = 3.3245 \text{ fps}$$

Using this effective value, the total kinetic energy can be checked by Equation (12a):

$$W_0 = 0.969 \times 7421 \times 3.3245^2 = 79{,}477 \text{ ft-lb}$$

From Equation (31a) the energy stored in the pipe at maximum surge is

$$\delta_1 = \frac{10{,}368 \times 7421 \times 32.2}{4136^2 \times 62.4} \left(\frac{200-75}{2}\right)^2$$

$$\delta_1 = 9{,}066 \text{ ft-lb}$$

Since α and γ have been affected only with respect to C_0 by the change in pipe diameters, solution for C_0 can now be made directly. Using Equation (33):

$$C_0 = \frac{79{,}478 - 9{,}066}{\frac{144 \times 89}{1.015 - 1}\left[\left(\frac{214}{89}\right)^{0.0148} - 1\right] - 144 \times 89\left[1 - \left(\frac{89}{214}\right)^{0.985}\right]}$$

$$C_0 = 18.65 \text{ cu ft}$$

Orifice Filling Control

If a flow-restricting orifice is interposed between the main flow line and the liquid chamber of the surge suppressor there results a net loss of kinetic energy from the acceleration and deceleration of the liquid as it passes through the orifice. Because conditions at the orifice are always transient it is not expedient to apply the standard relationships for energy losses across an orifice. Rather, let us examine a pressure-time record of surge pressures measured simultaneously on the inlet and outlet sides of an orifice plate employed in suppressor filling control. These tracings and a schematic diagram of the system will be found in Fig. 29. The valve begins

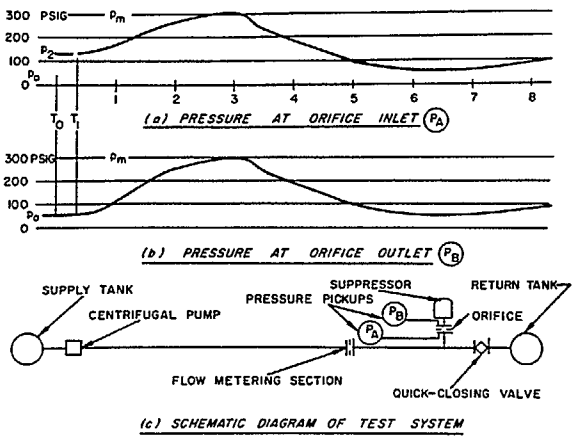

Fig. 29. Pressure-Time Records for a System Incorporating Suppressors and Orifice Filling Control

to close at time T_0. At time T_1 maximum flow is established through the orifice. The pressure in the system rises rapidly to p_2. At this instant the pressure drop across the orifice is $p_2 - p_0$. The sudden pressure rise from p_0 to p_2 has resulted from a reduction of the flow velocity in the system and through the orifice from v_0 to some new value v_1, which may be found by a rearrangement of Equation (23):

$$v_1 = v_0 - \frac{144g}{wa}(p_2 - p_0) \text{ fps} \qquad (23a)$$

At time T_1 the kinetic energy in the system will accordingly be reduced to

$$W_1 = wALv_1^2/2g \text{ ft-lb} \qquad (12b)$$

Selection of the proper sized orifice for any particular application has long been based on conjecture or trial and error methods. Too small an orifice produces the same water hammer conditions that would exist were no suppressors present. Too large

an orifice is little better than none at all. From extensive test data it has been determined that optimum surge suppression results from the ratio:

$$\frac{p_2 - p_0}{p_m - p_0} = 0.68 \qquad (40)$$

The orifice diameter d_2 which will pass q_1 cubic feet per second of a liquid of specific weight w with a differential pressure of h_w is found by applying Formula (98b) from the ASME Report on *Fluid Meters* (Ref. 11):

$$q_1 = 0.0997 \left(\frac{C d_2^2}{\sqrt{1-\beta^4}} \right) \sqrt{\frac{h_w}{w}} \text{ cfs} \qquad (41)$$

in which $\frac{C}{\sqrt{1-\beta^4}}$ = orifice coefficient of discharge including velocity of approach factor, measured at 0.70 for a square-edged orifice under the cited test conditions

d_2 = orifice diameter, inches

h_w = differential pressure across orifice, inches of water

w = specific weight of fluid, lb/cu ft

Using the value of 0.70 for the discharge coefficient, and converting the differential pressure into psi, the equation becomes:

$$q_1 = 0.3674\ d_2^2 \sqrt{\frac{p_2 - p_0}{w}} \text{ cfs}$$

Since $q_1 = Av_1$

$$d_2 = \left(\frac{Av_1}{0.3674\sqrt{\frac{p_2 - p_0}{w}}} \right)^{\frac{1}{2}} \text{ in.} \qquad (42)$$

To illustrate the economies effected by the addition of orifice filling control, the suppressor-sizing examples previously presented will be recalculated to incorporate orifice control. Since the pressures p_0 and p_m are the same for both the simple and complex systems, ratio (40) is applicable to both.

$$p_2 - p_0 = 0.68\ (p_m - p_0)$$

$$p_2 - p_0 = 0.68\ (200 - 75) = 85 \text{ psi}$$

For the simple system, by (23a), the resultant new velocity is:

$$v_1 = 4.07 - \frac{144 \times 32.2}{62.4 \times 4157}\ (85)$$

$$v_1 = 4.07 - 1.52 = 2.55 \text{ fps}$$

The orifice diameter, from Equation (42) is

$$d_2 = \left(\frac{0.548 \times 2.55}{0.3674\sqrt{\frac{85}{62.4}}} \right)^{\frac{1}{2}} = 1.805 \text{ in.}$$

The new value for the remaining kinetic energy in the system at time T_1 is, by (12b):

$W_1 = 0.969 \times 0.548 \times 10{,}000 \times 2.55^2 = 34{,}529$ ft-lb

Except for C_0, other factors in Equation (33) will remain unchanged. The new required capacity is thus:

$$C_0 = \frac{34{,}529 - 6{,}627}{\frac{144 \times 89}{1.015 - 1}\left[\left(\frac{214}{89}\right)^{0.0148} - 1\right] - 144 \times 89\left[1 - \left(\frac{89}{214}\right)^{0.985}\right]}$$

$C_0 = 7.39$ cu ft

With the employment of the orifice the new suppressor capacity is 7.39/21.54 or only 34.3% as large as the former requirement.

For the complex system, considering conditions at the suppressor which is located near the end of section E, the new velocity becomes

$$v_1 = v_E - \frac{144g}{wa_E}(p_2 - p_0) = 11.101 - \frac{144 \times 32.2}{62.4 \times 4257}\ (85)$$

$$v_1 = 11.101 - 1.484 = 9.617 \text{ fps}$$

The new effective velocity v_{e1} for the entire system is thus

$$v_{e1} = \frac{9.617}{11.101} \times 3.3245 = 2.881 \text{ fps}$$

The required orifice diameter, by (42) is:

$$d_2 = \left(\frac{A_E v_1}{0.3674\sqrt{\frac{p_2 - p_0}{w}}} \right)^{\frac{1}{2}}$$

$$d_2 = \left(\frac{0.2007 \times 9.617}{0.3674\sqrt{\frac{85}{62.4}}} \right)^{\frac{1}{2}} = 2.122 \text{ in.}$$

The residual kinetic energy W_1 in the complex system is

$$W_1 = (w/2g) V v_{e1}^2 = 0.969 \times 7{,}421 \times 2.881^2$$

$$W_1 = 59{,}686 \text{ ft-lb}$$

The new required suppressor capacity is determined as before.

$$C_0 = \frac{59{,}686 - 9{,}066}{\frac{144 \times 89}{1.015 - 1}\left[\left(\frac{214}{89}\right)^{0.0148} - 1\right] - 144 \times 89\left[1 - \left(\frac{89}{214}\right)^{0.985}\right]}$$

$$C_0 = 13.41 \text{ cu ft}$$

By the use of the orifice only 13.41/18.65, or 72% of the former capacity is now required.

Series Systems

A piping system composed of the components shown in Fig. 30a and consisting of a centrifugal pump, a length of pipe L_A, a pressure regulating

valve, a length of pipe L_B, and a quick-closing valve may be analyzed as two separate systems separated by the pressure regulating valve. Since the regulating valve will begin to close when the surge pressure has traveled from the quick-closing valve through section L_B the pressure waves will be reflected at the regulating valve and the downstream system L_B may be considered separately. The sizing of a suppressor located at the quick-closing valve is accomplished in the same manner as shown in the previ-

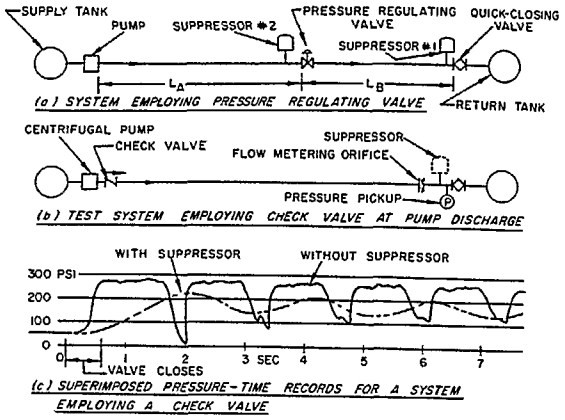

Fig. 30. System Diagrams and Superimposed Pressure-Time Records

ous examples, with only the energies present in section L_B to be accommodated. Similarly section L_A may be considered as an individual system, in which the regulating valve functions like a quick-closing valve. A second suppressor, located at the regulating valve inlet, should be sized to accommodate the fluid energies present in section L_A.

Effect of Check Valves on Surge Characteristics

The inclusion of a check valve at the discharge of the centrifugal pump in a piping system terminating in a quick-closing valve, as shown in Fig. 30b, does not affect the magnitude of the surge pressures produced by fluid deceleration but only their periodicity. Superimposed tracings of pressure-time records for the same system and flow conditions with and without suppressors are presented in Fig. 30c. Comparison of these records with those of Figs. 3 and 25 will show the elimination of the reflected negative wave and the shortening of the basic cycle to $2L/a$.

Suppressor Placement

In systems where fluid deceleration is produced by a quick-closing or pressure regulating valve the ideal location for a surge suppressor is immediately upstream from the valve. Where physical conditions do not permit such optimum placement, tests have shown that the suppressor may be located at any distance L_1 from 0 to 50 feet upstream from the valve without materially affecting the surge reduction.

For distances greater than 50 feet, and for very short valve closure times, T, the maximum distance L_1 permitted for possible surge reduction cannot exceed $aT/2$.

$$\text{If } N_1 = \frac{aT}{2L_1} \text{ intervals} \qquad (22a)$$

under conditions where $L_1 > aT/2$, $N_1 < 1$, and the pressure rise at the instant of complete valve closure will be $v_0 wa/144g$. For $N_1 > 1$ the initial pressure rise at the valve may be determined by applying the parameters K and N_1 to the Allievi charts, Figs. 7, 8, or 9. The value of p_1 thus found will establish the minimum value of p_m. The rest of the system upstream from L_1 will be protected by the suppressor. When the suppressor has reached the maximum surge pressure p_m the section of pipe L_1 will experience the uniform pressure p_m. Upstream the pressure will diminish almost linearly to the pump or reservoir.

In systems where L_1 is one-fourth to one-half the total length of pipe and where the period of the suppressor is nearly in phase with the period of the pipe pressure peaks may be produced on the third or fourth cycle considerably in excess of p_m. The subject of resonance will be given further amplification in a subsequent section.

Uphill Pumping Systems

When the power is cut off from the pumps supplying an elevated tank or rising section of pipe the column of liquid at first decelerates over a sensible period of time, during which its momentum is converted into head and the rotary inertia of the pump is imparted to the fluid and lost in bearing friction. The flow then reverses, closing the check valve at the pump discharge and producing a pressure rise proportional to the velocity of reverse flow.

Two studies of surge phenomena in pumping systems are worth examination. The first, an analysis of transient conditions in the Colorado River Aqueduct Pumping Plant, has been made by R. M. Peabody (Ref. 12). In the second, tests conducted on the Croton Lake Pumping Plant, New York, and reported by S. Logan Kerr in the *Symposium on Water Hammer* (Ref. 4) present corroborating evidence. Generalized curves of flow, pressure, and pump rotation will be found in Figs. 31a and 31b.

From an examination of these data the following simplified analysis can be made. For systems in which the check valve is closed by the reverse flow, when the fluid velocity is zero the instantaneous pressure p_3 at the pump is a minimum of $0.4p_s$, where p_s is the pressure equivalent of the static head. The maximum pressure p_1 recorded for a system using a non-cushioned check valve is 2.1 p_s. The rise $p_r = (p_1 - p_3)$ is thus 1.7 p_s. By Joukowsky's Law, the maximum reverse flow rate

$$v_2 = \frac{144g(p_1 - p_3)}{aw} = \frac{244.8gp_s}{aw} \text{ fps} \qquad (43)$$

In systems containing a check valve which closes before flow reversal takes place, p_3 becomes zero

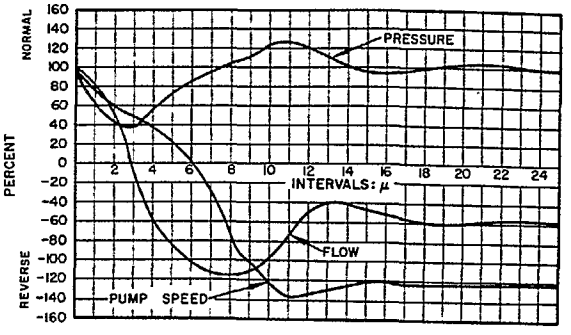

Fig. 31a. Flow, Pressure, and Pump Speed Curves for Power Failure in Uphill Pumping System Without Check Valve

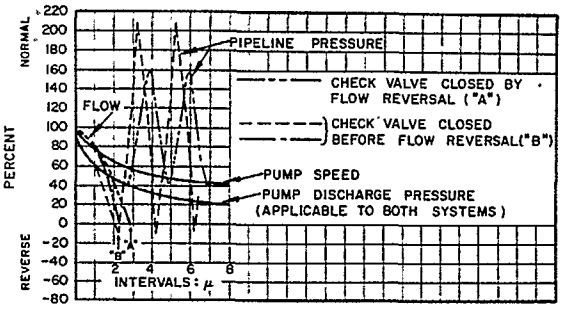

Fig. 31b. Flow, Pressure, and Pump Speed Curves for Power Failure in Two Uphill Pumping Systems Incorporating Check Valves

and $p_1 = 2.1\ p_s$. The rise p_r is thus 2.1 p_s.

An alternate method of determining the velocity of maximum reverse flow has been provided in the study by S. P. Johnson (Ref. 13). Johnson postulated a dimensionless parameter, S, related to the Reynolds' number N_R and the pipe friction factor f by the expression

$$S = N_R \left(\frac{f}{2}\right)^{\frac{1}{2}} \qquad (44)$$

The parameter S, being independent of velocity and defined as

$$S = \frac{(p'D^3\rho)^{\frac{1}{2}}}{\mu_1} \qquad (45)$$

where p' = pressure gradient, lb/sq ft per ft = $144(0.6\ p_s)/L$

D = pipe diameter, feet

$\rho = w/g$ = specific weight of liquid, slugs/cu ft

μ_1 = dynamic viscosity, slugs/ft-sec

may be determined directly. Lines of constant values of the S number will be developed by placing a straightedge between the scales at the top and bottom of the chart, Fig. 32. Intersection of the straightedge with the applicable pipe friction curve will locate the values of N_R and f. The velocity v_2 can be determined without the usual trial and error procedure, since the Reynolds' number equation,

$$N_R = \frac{Dv\rho}{\mu_1} \qquad (46)$$

may be solved for the velocity,

$$v_2 = \frac{N_R\mu_1}{D\rho} \text{ fps} \qquad (46a)$$

in which the units from Equation (45) should be employed.

For a slow-closing check valve ($T > \mu$) the pressure rise may be determined by application of the parameters K and N and the use of the Allievi charts, Figs. 7 through 9, wherein the fluid velocity v_2 will have been previously determined from Equation (43) have been previously determined from Equation (43) or (46a).

In systems where the pipe friction loss p_f is appreciable the initial pumping pressure p_0 will equal the sum of the static head p_s and the friction and throttling losses p_f. Since p_f vanishes at the instant of flow reversal the minimum pressure p_3, as before, is $0.4\ p_s$. The maximum pressure may be found by applying Joukowsky's Law:

$$p_1 = 0.4p_s + \frac{v_2wa}{144g} \text{ psi} \qquad (15b)$$

in which v_2 has been determined from Equation (46a). A conservative value of p_1 will result from the assumption that p_1 cannot exceed 2.1 p_s. This assumption allows the simpler solution provided by Equation (43).

Limitation of the maximum surge pressure to p_m may be effected by the installation of a pneumatic suppressor, precharged to the pressure p_3, and located at the outlet of the check valve. If no orifice control is used the sizing of this suppressor is en-

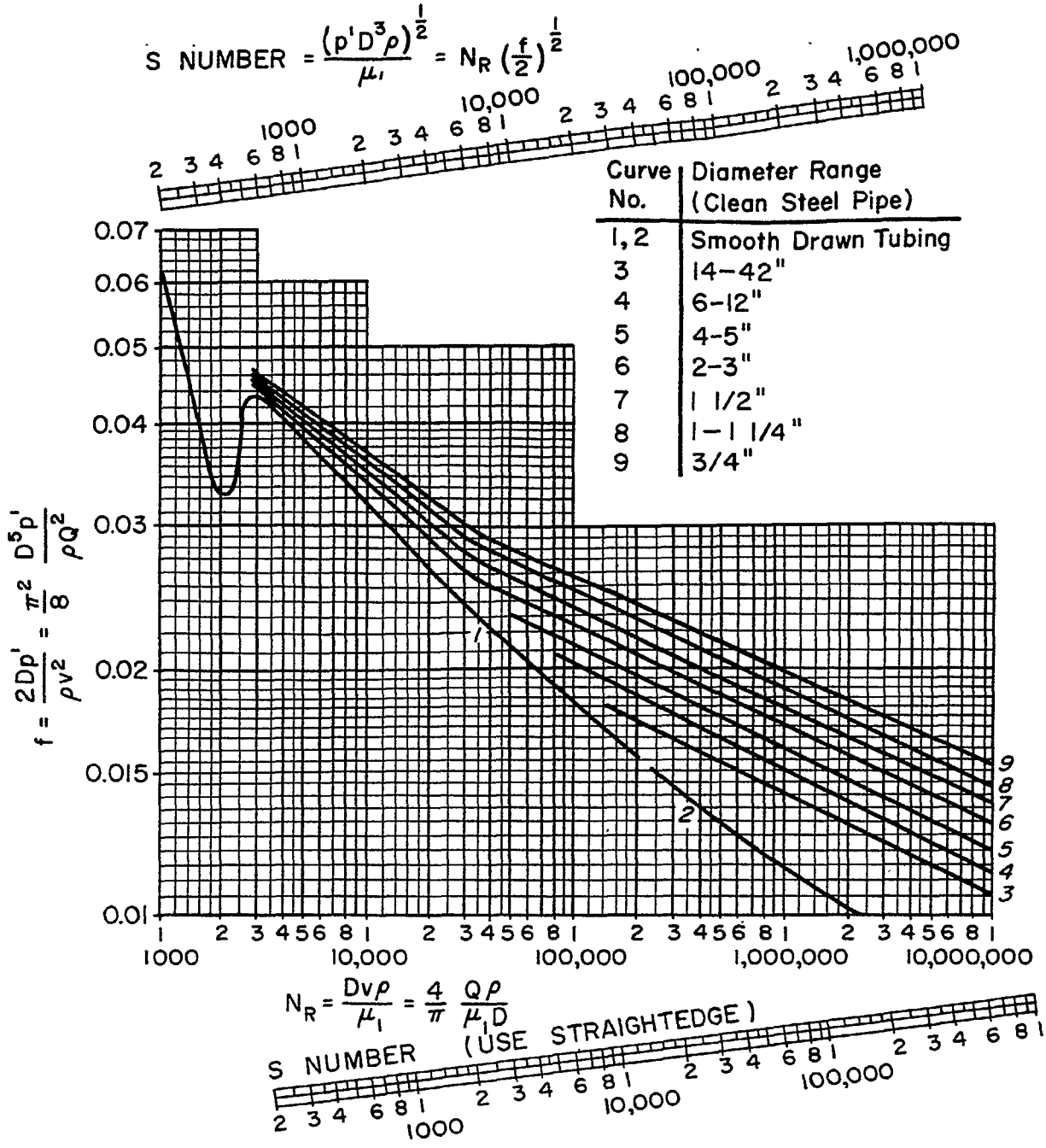

Fig. 32. Relation between friction factor and Reynolds' number for flow in circular pipes, with added S Number coordinates.

tirely straightforward. The basic energy relationship, $W_2 + \gamma = \alpha + \delta_2$, is employed. The slightly modified formulas are given below.

$$W_2 = (w/2g)Vv_2^2 \quad \text{ft-lb} \qquad (12c)$$

$$\delta_2 = \frac{10{,}368Vg}{a^2w}\left(\frac{p_m - p_3}{2}\right)^2 \quad \text{ft-lb} \qquad (31b)$$

$$C_0 = \frac{W_2 - \delta_2}{\frac{144p_3^\circ}{n-1}\left[\left(\frac{p_m^\circ}{p_3^\circ}\right)^{\frac{n-1}{n}} - 1\right] - 144p_3^\circ\left[1 - \left(\frac{p_3^\circ}{p_m^\circ}\right)^{\frac{1}{n}}\right]} \quad \text{cu ft} \qquad (33b)$$

When an orifice filling control is included the ratio $(p_2 - p_3)/(p_m - p_3)$ must be limited to a value such that $p_m - p_2 \geq 0.6p_s$ since the residual static head will inevitably be superimposed upon the orifice-induced system pressure p_2. After setting $p_m - p_2$ equal to $0.6p_s$ solve the ratio $(p_2 - p_3)/(p_m - p_3)$. If it exceeds 0.68 select a lower value of p_2 to produce the optimum ratio of 0.68.

Example: determine the suppressor capacity and orifice diameter required to limit the maximum surge pressure to 250 psig in the event of power failure in the following pumping system:

Pipe length, $L = 11{,}500$ ft

Pipe size: 8-inch, schedule 40: $d = 7.981$ in. (0.665 ft)

Pipe wall thickness, $e = 0.322$ in.

Flow area, $A = 0.347$ sq ft

Young's modulus for pipe material, $E = 30{,}000{,}000$ psi

Static head, $H_s = 498$ ft

Pumping flow rate, $Q_0 = 400$ gpm

Liquid pumped: aviation gasoline, Grade 115-145

Specific gravity of liquid: 0.690 at 60°/60°F (74° API)

Specific weight, $w = 0.69 \times 62.4 = 43.1$ lb/cu ft

Kinematic viscosity, $\nu = 0.640$ centistokes at 60°F

Pressure equivalent of static head, $p_s = 43.1 \times 498/144$; $p_s = 149$ psi

Line friction drop at 400 gpm, $p_f = 11$ psi

Pump discharge pressure at 400 gpm, $p_0 = 149 + 11 = 160$ psi

Maximum allowable surge pressure, $p_m = 250$ psig

Solution: the bulk modulus of aviation gasoline at 60°F will be found from Fig. 35 as 103.000 psi. The velocity of the pressure wave,

$$a = \sqrt{\frac{12}{\frac{43.1}{32.2}\left(\frac{1}{103.000} + \frac{7.981}{0.322 \times 30.000{,}000}\right)}}$$

$$a = 3189 \text{ fps}$$

Assume that the check valve is closed by flow reversal.

$$p_3 = 0.4p_s = 0.4 \times 149 = 60 \text{ psi}$$

The viscosity of the gasoline in slugs/ft-sec (from Tables II and III):

$$\mu_1 = \nu \text{ (centistokes)} \times \text{(specific gravity)} \times 2.088 \times 10^{-5}$$

$$\mu_1 = 0.640 \times 0.690 \times 2.088 \times 10^{-5} = 9.22 \times 10^{-6} \text{ slugs/ft-sec}$$

The pressure gradient,

$$p' = 144(0.6p_s)/L = \frac{144 \times 0.6 \times 149}{11{,}500} = 1.12 \text{ lb/sq ft per ft}$$

The S number,

$$S = \frac{(p'D^3\rho)^{\frac{1}{2}}}{\mu_1} = \frac{(1.12 \times 0.665^3 \times 43.1/32.2)^{\frac{1}{2}}}{9.22 \times 10^{-6}}$$

$$S = 72{,}000$$

From Fig. 32 an S number of 72,000 intersects curve 4 (for 8-inch pipe) at a Reynolds' number of 825,000 and friction factor of 0.0153. By Equation (46a),

$$v_2 = \frac{8.25 \times 10^5 \times 9.22 \times 10^{-6}}{0.665 \times 1.3385} = 8.546 \text{ fps}$$

without a suppressor the maximum pressure would be, by (15b):

$$p_1 = 0.4 \times 149 + \frac{8.546 \times 43.1 \times 3189}{144 \times 32.2}$$

$$p_1 = 60 + 253 = 313 \text{ psi (Note that } p_1 = 2.1p_s)$$

The suppressor must reduce this maximum pressure by $p_1 - p_m = 313 - 250 = 63$ psi.

Unless the resultant orifice ratio is greater than 0.68 the optimum orifice size will be such that $p_m - p_2 = 0.6\ p_s$, or $p_2 = 250 - 0.6 \times 149$; $p_2 = 161$ psi.

The orifice ratio,

$$\frac{p_2 - p_3}{p_m - p_3} = \frac{161 - 60}{250 - 60} = 0.53$$

Since $0.53 < 0.68$ a value of 161 psi for p_2 is acceptable.

The reduced system velocity v_1 is, by Equation (23a)

$$v_1 = v_2 - \frac{144g}{wa}(p_2 - p_3)$$

$$v_1 = 8.546 - \frac{144 \times 32.2}{43.1 \times 3189}(161 - 60)$$

$$= 8.546 - 3.407 = 5.139 \text{ fps}$$

The orifice diameter can now be found from Equation (42).

$$d_2 = \left(\frac{0.347 \times 5.139}{0.3674\sqrt{\frac{161 - 60}{43.1}}}\right)^{\frac{1}{2}} = 1.781 \text{ in.}$$

The kinetic energy corresponding to the reduced velocity v_1 is, by Equation (12b)

$$W_1 = \frac{43.1 \times 0.347 \times 11{,}500 \times 5.139^2}{2 \times 32.2} = 70{,}529 \text{ ft-lb}$$

The energy stored in the pipe at the instant of maximum surge, by Equation (31b), is

$$\delta_2 = \frac{10{,}368 \times 3990.5 \times 32.2}{3189^2 \times 43.1}\left(\frac{250-60}{2}\right)^2 = 27{,}410 \text{ ft-lb}$$

Assuming atmospheric pressure of 14 psia and a value for n of 1.015, solution for the required Magni-Lastic suppressor capacity by Equation (33b) yields:

$$C_0 = \frac{70{,}529 - 27{,}410}{\frac{144 \times 74}{0.015}\left[\left(\frac{264}{74}\right)^{0.014n} - 1\right] - 144 \times 74\left[1 - \left(\frac{74}{264}\right)^{0.985}\right]}$$

$$C_0 = 7.335 \text{ cu ft}$$

The suppressor should be precharged to pressure p_3: 60 psig.

Pulsation Control in Reciprocating Pump Lines

If it is assumed that the velocity and hence the instantaneous quantity of the fluid discharged by a simplex reciprocating pump varies almost sinusoidally with respect to time or angle of crank rotation it is apparent that this instantaneous flow rate is alternately less and greater than the mean flow rate. The mean flow rate may be defined as the product of the total quantity of fluid discharged during a 360° rotation of the cranks and the number of revolutions of the cranks per unit of time. Making the pump double-acting or increasing the number of cylinders results in a less widely fluctuating quantity-time curve since the velocity of one piston will be increasing during the interval that the velocity of another is decreasing. Obviously the flow rate variations would approach zero as a limiting value as the number of equally spaced cranks was infinitely increased. Since there are practical limits to the number of cylinders employed in reciprocating pumps, the variations from the mean flow rate must be removed by other means. An air chamber of adequate capacity located at the discharge of the pump has been proven effective in eliminating volumetric —and hence pressure—fluctuations in countless installations. Diederichs and Pomeroy (Ref. 14) determined from considerable investigation that, "... The intelligent application of air chambers at the pumps absolutely prevented the setting up of surges."

They point out certain limitations inherent in the design and use of air chambers. Among these are the gradual loss of the pressurizing air which is absorbed by the liquid, loss of air in the chamber when the system pressure drops below the design operating pressure, and the formation of an explosive mixture when the air unites with hydrocarbon vapors in a system handling petroleum derivatives.

These shortcomings are completely overcome in the design of the MagniLastic Hydropad, since the diaphragm bellows keep the gas and liquid separated at all times and prevent the loss of the gas when the system pressure is reduced below its normal operating level. Hydropads are precharged with inert nitrogen gas, which will not support combustion.

In order to determine the correct capacity of the suppressor to be used to accommodate the volume of liquid pumped in excess of that flowing through the associated piping system at the mean flow rate, it will first be necessary to examine the flow characteristics of the system. If a constant back pressure p_0 is maintained, the pump, equipped with a surge suppressor, and operating between the pressure limits p_r and p_m, where $(p_r+p_m)/2 = p_0$, will furnish fluid at its rated mean flow Q_0. If the back pressure effective at the pump is maintained by pipe friction with or without flow controlling devices, this pressure p_0 will equal the mean discharge pressure at the pump at the mean flow rate Q_0. Let p_m be the maximum allowable surge pressure. When a portion of the liquid is at pressure p_m its instantaneous flow rate becomes $Q_m > Q_0$. Because over the period of one cycle the pump is delivering fluid at the mean rate Q_0, during a portion of this cycle the instantaneous rate will be reduced to Q_r, where $Q_0 - Q_r = Q_m - Q_0$. If the reduced system pressure which impels a flow of Q_r is p_r, the approximate pressure relationship becomes $p_m - p_0 = p_0 - p_r$. Assuming, then, that p_0 is known or can be calculated and the allowable pressure rise $p_m - p_0$ is specified, the capacity of the required suppressor can be determined from application of Formulas (35) and (36) and a knowledge of excess volume relationships for reciprocating pumps of various types.

Since pressure fluctuations, created by every piston stroke, may occur rapidly—at frequencies of one or more cycles per second—the cushioning gas within the suppressor will be compressed and expanded nearly adiabatically. A value of 1.4 for n should thus be employed for nitrogen-charged suppressors.

The pressure relationship is

$$p_r = 2p_0 - p_m \quad \text{psi} \qquad (47)$$

in which p_r = minimum operating pressure, psig
p_r = suppressor charging pressure

p_0 = design operating pressure, psig
p_m = maximum allowable pressure, psig

Using the starred symbols p_x°, p_0°, p_m° for absolute values of the corresponding gauge pressures p_x, p_0 and p_m, the required capacity C_x is given by

$$C_x = \frac{0.7854 F_1 b^2 s_1 \left(\frac{p_0^\circ}{p_x^\circ}\right)^{\frac{1}{n}}}{1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}} \text{ cu in.} \qquad (48)$$

where F_1 = the conservative pump volumetric factor listed in the table below

b = bore of one cylinder, inches
s_1 = stroke of one piston, inches
n = polytropic exponent of gas expansion
$n = 1.4$ for nitrogen pressurized suppressors.

Numerical values for $\left(\frac{p_0^\circ}{p_x^\circ}\right)^{\frac{1}{n}}$ and $\left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}$ will be found in Fig. 27.

Reciprocating Pump Volumetric Factor Table

Pump Type	*Factor F_1*
Simplex single-acting	0.60
Simplex double-acting	0.25
Duplex single-acting	0.25
Duplex double-acting	0.15
Triplex single-acting	0.03
Triplex double-acting	0.06
Quadruplex single-acting	0.10
Quadruplex double-acting	0.06
Quintuplex single-acting	0.06
Quintuplex double-acting	0.02
Sextuplex	0.06
Septuplex	0.02

The suppressor should be precharged to the pressure p_x. Since the present application of suppressors is based on purely volumetric instead of energy considerations, the use of orifices to limit the flow is not justified.

A second suppressor should be installed in the pump suction line to serve as an accumulator in preventing separation of the liquid column if the pump is operating with a static lift or if the volumetric variation factor F_1 is 0.25 or greater. For static lift applications,

p_0° = design operating suction pressure, psia
p_0° = (atmospheric pressure)-(vapor pressure)-(static lift)-(pump entrance losses)-(pipe friction losses)
p_x° = minimum allowable suction pressure, psia
p_x° = suppressor charging pressure
$p_m^\circ = 2p_0^\circ - p_x^\circ$ psia (47a)

Solution for the required capacity C_x can then be made by the use of Equation (48).

Example: find the capacity of the MagniLastic suppressor required to limit the pressure rise $p_m - p_0$ of a 6×24 duplex double-acting pump to 25 psi for a design operating pressure p_0 of 700 psig.

Solution: assume atmospheric pressure $p_a = 14.5$ psia. The pressures are

$p_0 = 700$ psig
$p_0^\circ = 714.5$ psia
$p_m = p_0 + p_r = 700 + 25 = 725$ psig
$p_m^\circ = 739.5$ psia
$p_x = 2p_0 - p_m = 2 \times 700 - 725 = 675$ psig
$p_x^\circ = 689.5$ psia

From the volumetric factor table above, F_1 for a duplex double-acting pump is 0.15

$b = 6$-inch bore
$s_1 = 24$-inch stroke
$n = 1.4$

By Equation (48) the capacity,

$$C_x = \frac{0.7854 \times 0.15 \times 6^2 \times 24 \left(\frac{714.5}{689.5}\right)^{0.714}}{1 - \left(\frac{714.5}{739.5}\right)^{0.714}}$$

$$C_x = 4315 \text{ cu in.}$$

Resonance

If a series of periodic pressure pulsations traveling the length of a pipe line at approximately the sonic velocity a is reflected from a "closed end" in such a way that the reflected waves are in phase with the incident waves, *stationary* or *standing* waves are formed, and the resulting constructive interference greatly increases the amplitude of particle displacement. Pressure fluctuations are correspondingly magnified. Parts of the medium separated by one-half a wave length experience opposing tendencies which cancel each other's effects, while at points midway between these regions the tendencies toward displacement are reinforced. The former regions are designated *nodes;* the latter *antinodes* or *loops*. The characteristic of a "closed end" in a pipe line may be produced by a partially closed valve, an orifice, a reducer, or a dead-end branch.

Reflections will also occur at an "open end," produced by an enlarged pipe section of at least twice the flow area or by the presence of a component in which the mean flow velocity is less than half that

prevailing in the supply pipe. Reflections from an "open end" undergo a phase shift of 180°. The incident wave is thus reflected with a change of sign.

It was previously stated that the pressure wave traveled at approximately the sonic velocity a. For practical purposes in the frequency range generated by reciprocating pumps the use of a calculated value of a from Equation (14) will be sufficiently accurate without further modification. For a further discussion of wave velocity variation and energy distribution the reader is referred to Thomson's paper (Reference 15).

Fluid temperature is a highly important factor in determinations of the pressure wave velocity. A temperature rise will decrease both the specific weight w and the bulk modulus k of some liquids. The resultant change in a is about -0.088% per degree F, referred to a base temperature of 60° F, and applicable to all hydrocarbons. The negative sign indicates a decrease in the value of a with an increase in temperature.

The wave length λ of the fundamental standing wave produced in a pipe line by a reciprocating pump whose frequency is n_1 pulsations per second and in which the flow velocity is negligible in comparison with the velocity a is, from wave mechanics:

$$\lambda = \frac{a}{n_1} \text{ ft} \qquad (49)$$

From a consideration of resonance in closed and open pipes the conditions conducive to resonance may be stated. The length of a closed pipe—or section of pipe line terminating in a "closed end" as defined above—which will resonate is

$$L_2 = \frac{M_1 \lambda}{4} \text{ ft} \qquad (50)$$

where M_1 is any odd integer.

Similarly the length of an open pipe—or section of pipe line terminating in an "open end"—which will resonate is

$$L_3 = \frac{M\lambda}{2} \text{ ft} \qquad (50a)$$

where M is any integer, even or odd.

It may be shown that the pump represents a particle displacement antinode in the analysis of resonant conditions; a closed end can only permit the formation of a node; an open end will support an antinode. Calculations based on Formulas (50) or (50a) will disclose the tendency of the system to resonate. Maximum resonance and the formation of standing waves will occur if $L = L_2$ or L_3. Beats, or the alternate reinforcement and cancellation of pressure waves may occur if L is from 90% to 110% of L_2 or L_3.

Various attempts to reduce the amplitude of standing waves in reciprocating pump systems have been made from time to time. The problem has been aggravated by the use of higher speed pumps and drivers and the replacement of steam engines by diesel engines, since the advantages of a steam-cushioned stroke are no longer provided. Following the analysis and admirable solution presented by J. W. Squire (Ref. 16) a basic study of pipe line pulsation was initiated at Oklahoma A & M College, Stillwater, Oklahoma. A report (Ref. 17) covering the first phase of the project and presenting a well-formulated electrical analogy for the manipulation of hydraulic parameters has been released by the sponsoring pipe line companies.

R. C. Binder (Ref. 18) develops a method of analysis along similar lines.

MagniLastic suppressors designed for pulsation control have proven effective in a wide variety of applications and over the complete frequency and pressure range required by all pumps in current use. Since each system is unique no further general analysis would be pertinent here. Inquiries regarding suppressor applications for pulsation and resonance control should be addressed to MagniLastic Division, Cook Electric Company, 2700 Southport Avenue, Chicago 14, Illinois.

Accumulators

In hydraulic systems an accumulator converts excess fluid and compressed gas into stored energy, available upon demand. The basic criteria for determining the needed capacity of an accumulator include:

- V_d' the volume of fluid displaced, or usable volume, cu in.
- p_0° the minimum allowable accumulator pressure after the expulsion of the volume V_d'; also the accumulator precharging pressure, psia
- p_m° the maximum allowable accumulator pressure, usually the system design operating pressure, psia

The minimum required capacity c_0 at pressure p_0° can be found directly from a rearrangement of Formula (35):

$$c_0 = \frac{V_d'}{1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}} \text{ cu in.} \qquad (35a)$$

If the accumulator is to be precharged to some absolute pressure $p_x^\circ < p_0^\circ$, the minimum required capacity c_x of the accumulator supplying the volume V_d' of liquid between the pressure limits p_m° and p_0° is

$$c_x = \frac{V_d' \left(\frac{p_0^\circ}{p_x^\circ}\right)^{\frac{1}{n}}}{1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}} \quad \text{cu in.} \qquad (35b)$$

Numerical values for $\left(\frac{p_0^\circ}{p_x^\circ}\right)^{\frac{1}{n}}$ and $\left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}$ will be found in Fig. 27.

MagniLastic accumulators, employing superior stainless steel bellows construction, are designed in strict accordance with government specifications, and are available in all standard capacities.

Thermal Expansion Compensation

If both ends of a pipe line filled with liquid are closed and the line is subjected to a temperature increase from t_0 to t_1 °F, both pipe and liquid will expand volumetrically. Since the coefficient of expansion for most liquids is much higher than that for most conduit materials, and since no liquid can escape from the system the additional volume required will produce a pressure increase $p_1 - p_0$ as given by Equation (18) where V_0 and V_1 are the initial and net expanded volumes, respectively, in cubic feet:

$$p_1 - p_0 = \frac{(V_1 - V_0)a^2 w}{144ALg} \quad \text{psi} \qquad (18)$$

Designating by ε_L the effective value of the cubical expansion of the liquid per °F from the base temperature t_0, and by ε_C the effective coefficient of linear expansion of the conduit material per °F from the base temperature t_0, the volumetric increases resulting from a temperature rise from t_0 to t_1, °F, may now be written. For the liquid:

$$V_L - V_0 = V_0 \varepsilon_L (t_1 - t_0) \quad \text{cu ft} \qquad (51)$$

For the conduit, since the coefficient of cubical expansion very nearly equals $3\varepsilon_C$:

$$V_C - V_0 = 3V_0 \varepsilon_C (t_1 - t_0) \quad \text{cu ft} \qquad (52)$$

The resulting net excess volume is

$$V_L - V_C = V_0 \varepsilon_L (t_1 - t_0) - 3V_0 \varepsilon_C (t_1 - t_0)$$

$$V_L - V_C = V_0 (t_1 - t_0)(\varepsilon_L - 3\varepsilon_C) \quad \text{cu ft} \qquad (53)$$

Accommodation of this excess volume may be provided by the inclusion of a suppressor or accumulator in the system. Since the net excess volume will be distributed between the volume displaced in the suppressor $(V_L - V_C)$ and the increment of volume stored in the pipe V_p and since the gas in the suppressor will be compressed adiabatically ($n = 1.4$) the required suppressor capacity C_0 may be found by combining and rearranging Equations (18) and (35).

$$C_0 = \frac{(V_L - V_C) - V_p}{1 - \left(\frac{p_0^\circ}{p_1^\circ}\right)^{0.714}} \quad \text{cu ft} \qquad (54)$$

where $V_p = 144ALg(p_1 - p_0)/a_t^2 w_t$ cu ft (55)

p_0 and p_0° are the initial gauge and absolute pressures, respectively, psi, at temperature t_0; also suppressor charging pressure

p_1 and p_1° are the gauge and absolute pressures, respectively, psi, to which the system is allowed to rise at temperature t_1

a_t is the velocity of the pressure wave at temperature t_1, fps

w_t is the specific weight of the liquid at temperature t_1, lb/cu ft

w_0 is the specific weight of the liquid at 60°/60°F.

Numerical values of $\left(\frac{p_0^\circ}{p_1^\circ}\right)^{\frac{1}{1.4}} = \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{1.4}}$ will be found in Fig. 27.

For low values of the pressure rise [$(p_1 - p_0) < 50$ psi] the quantity V_p may be neglected. It is obvious that neither p_0° nor p_1° can be less than the vapor pressure of the liquid at the corresponding temperatures t_0 and t_1.

Example: determine the accumulator capacity required to limit the pressure increase from zero psig at 60°F to 50 psig at 120°F, in a closed system consisting of 100 feet of 8-inch schedule 40 steel pipe and containing gasoline with a specific gravity at 60°/60°F of 0.75.

Solution:

$V_0 = AL = 0.347 \times 100 = 34.7$ cu ft

ε_L (from Table III) $= 0.0006$ per °F

ε_C (from Table IV) $= 0.00000583$ per °F

$t_1 - t_0 = 120° - 60° = 60°$

$V_L - V_C = 34.7 \times 60\ (0.0006 - 3 \times 0.00000583)$

$V_L - V_C = 1.213$ cu ft

$$w_t = \frac{w_0}{1 - \varepsilon_L (t_1 - t_0)} = \frac{0.75 \times 62.4}{1 - 0.0006 \times 60}$$

$w_t = 45.2$ lb/cu ft

k_t (from Fig. 35) = 96,000 psi

$$a_t = \frac{12}{\sqrt{\frac{45.2}{32.2}\left(\frac{1}{96{,}000} + \frac{7.981}{0.322 \times 30{,}000{,}000}\right)}}$$

$$a_t = 3026 \text{ fps}$$

$$V_p = \frac{144 \times 34.7 \times 32.2 \times 50}{3026^2 \times 45.2} = 0.019 \text{ cu ft}$$

The required capacity C_0 can now be found from Equation (54):

$$C_0 = \frac{1.213 - 0.019}{1 - \left(\frac{14.7}{64.7}\right)^{0.714}} = 1.83 \text{ cu ft}$$

Hydraulic Energy Balance

From a theoretical standpoint the energy balance related to hydraulic surge consists of equating the energies present in the system plus those added to the system during the time interval from the completion of valve closure until the instant of maximum surge to the energies stored elastically at the instant of maximum surge and the energy dissipated through heat transfer. The energy balance can thus be written

$$W_0 + \gamma_T + 144 p_p \triangle V + E_f = \alpha + \delta + \text{Heat} \qquad (56)$$

To the kinetic energy:

$$W_0 = wALv_0^2/2g \quad \text{ft-lb} \qquad (12)$$

is added the flow work of the liquid entering the system between the time of complete valve closure and the instant of maximum surge. Part of this liquid enters the suppressor. The energy associated with this quantity is γ_T (theoretical):

$$\gamma_T = 144 V_d p_0 \quad \text{ft-lb} \qquad (57)$$

From Equation (35) an expression for V_d is obtained, and

$$\gamma_T = 144 p_0 C_0 \left[1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}\right] \text{ ft-lb} \qquad (58)$$

This formula is identical to (29) except that p_0 is used in lieu of p_0°.

The remainder of the liquid added to the system after valve closure enters the transfer pipe, keeping it full. Its volume is $\triangle V$ cubic feet, and its energy is $144 p_p \triangle V$ ft-lb. The quantity $\triangle V$ can be calculated only if the time interval of fluid deceleration is known. Since this time interval depends upon line length, suppressor capacity and orificing it cannot be predetermined with any degree of accuracy.

The fourth term on the left side of Equation (56) is the energy added from regained friction head. If the maximum recovered pressure is $(p_p - p_0)/\sqrt{2}$—refer to Formula (17)—the average recovered pressure along the pipe, assuming linear pressure distribution, will be $(p_p - p_0)/2\sqrt{2}$. The energy regained is E_f, where V is the total volume of fluid in the pipe (cubic feet)

$$E_f = \frac{144 V (p_p - p_0)}{2\sqrt{2}} \text{ ft-lb} \qquad (59)$$

At the instant of maximum surge some of the energy will be stored in the compressed gas of the suppressor, some will be stored elastically in the compressed liquid and expanded pipe, and some will have been lost by heat transfer.

From thermodynamics the energy α stored in a volume C_0 of a gas between the pressure limits p_0° and p_m° is

$$\alpha = \frac{144 p_0^\circ C_0}{n-1}\left[\left(\frac{p_m^\circ}{p_0^\circ}\right)^{\frac{n-1}{n}} - 1\right] \text{ ft-lb} \qquad (30)$$

If it is assumed that at the instant of maximum surge the pressure along the pipe varies linearly from p_m at the valve to p_p at the pump, the average pressure rise, $\triangle p$ is given by the expression.

$$\triangle p = (p_m + p_p)/2 - (p_p + p_0)/2$$

$$\therefore \triangle p = (p_m - p_0)/2 \quad \text{psi} \qquad (60)$$

From Equation (9b)

$$\delta = \frac{10{,}368 ALg}{a^2 w}\left(\frac{p_m - p_0}{2}\right)^2 \text{ ft-lb} \qquad (31)$$

The heat transferred through the conduit walls will vary with the fluid temperature, ambient temperature, air circulation, surge cycle duration, system friction characteristics and rate of energy conversion. Though small in comparison with the other terms it cannot be neglected.

The inference may be drawn that the terms "$144 p_p \triangle V$" and "Heat" in Equation (56) cannot be predetermined with any degree of accuracy. A very close approximation to the true energy balance is obtained by rewriting Equation (56) in the form

$$W_0 + \gamma_T + (144 p_p \triangle V + E_f - \text{Heat}) = \alpha + \delta \qquad (56a)$$

The terms within the parenthesis are substantially accounted for by the use of p_0° in the term γ to replace p_0 in γ_T

$$\gamma = 144 p_0^\circ C_0 \left[1 - \left(\frac{p_0^\circ}{p_m^\circ}\right)^{\frac{1}{n}}\right] \text{ ft-lb} \qquad (29)$$

This substitution yields results within five percent of measured values for all test runs when employed in the modified energy balance

$$W_0 + \gamma = \alpha + \delta \quad \text{ft-lb} \qquad (32)$$

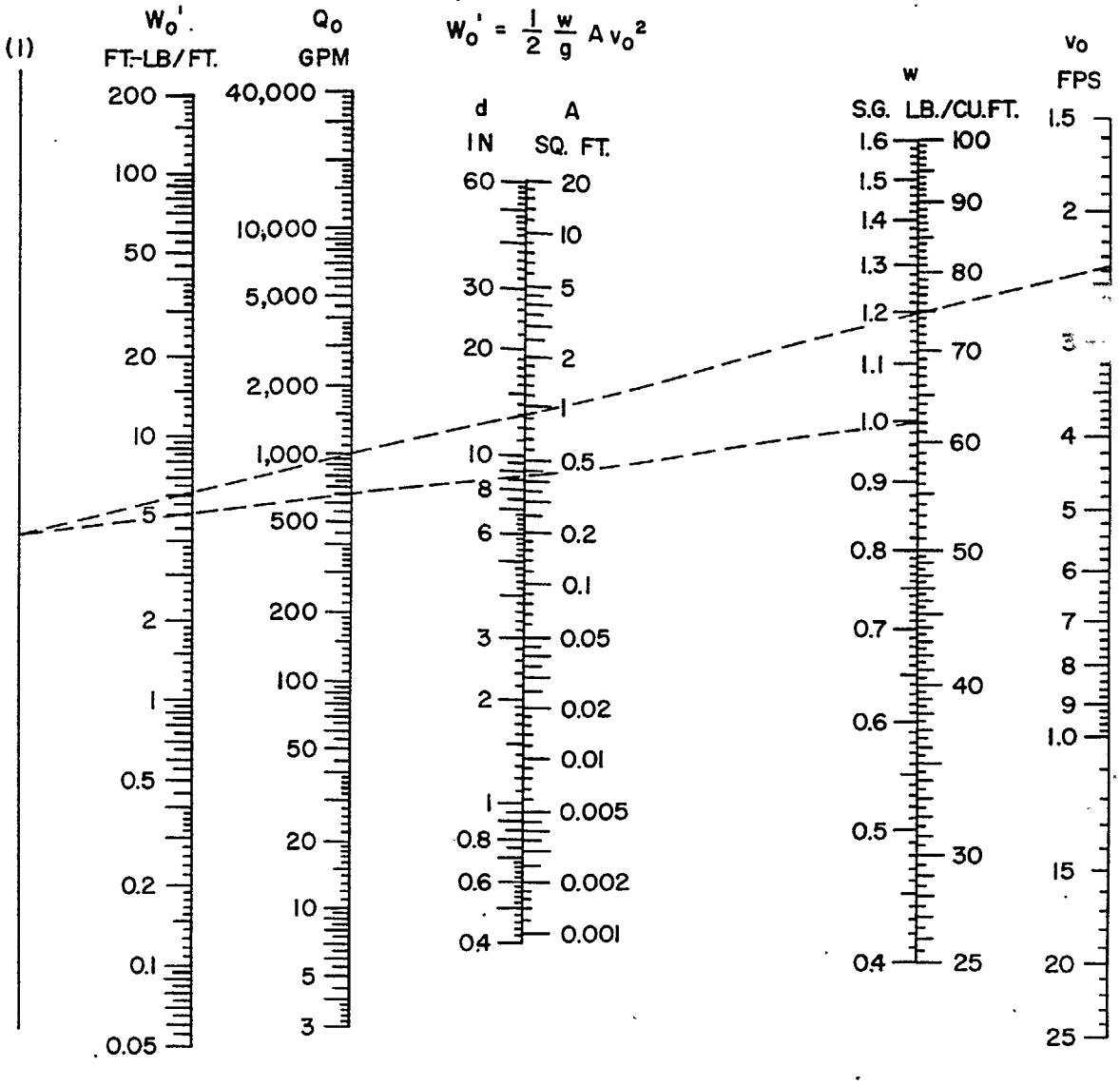

Fig. 33. Nomograph for Kinetic Energy Per Unit Length

Corresponding points on the v_0, d (or A), Q_0 and reference axis (1) will lie in a straight line. A second straight line connects the reference intercept with w and the required solution W_0'. In the example provided (dashed lines) water is flowing at 1000 gpm through 13.124-inch I.D. pipe. Find W_0' the kinetic energy per unit length. Solve by connecting 1000 on Q_0 with 13.124 on d. Extend this line to intersect with reference axis (1) and axis v_0. For reference read 2.37 fps (flowing velocity) on v_0. Connect the intercept on reference axis (1) with 62.4 on w and read 5.1 ft-lb/ft on the W_0' axis.

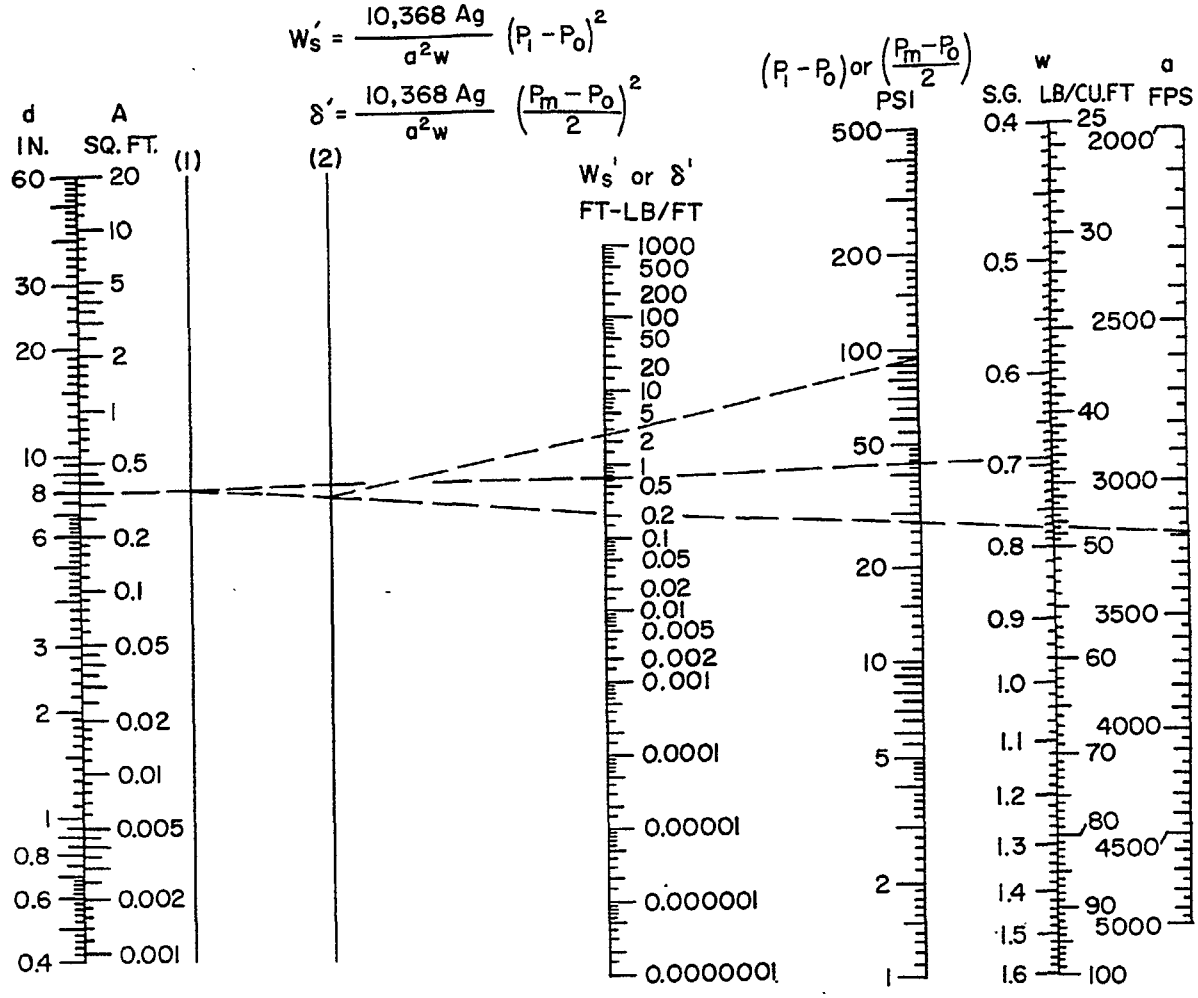

Fig. 34. Nomograph for Elastic Energy Per Unit Length

The nomograph solves the relations for W_S' or δ' depending upon the selection of the pressure factor (p_1-p_0) or $(p_m-p_0)/2$. An example is provided by the dashed lines.

Given data: $a = 3189$ fps; $w = 43.1$ lb/cu ft; $A = 0.347$ sq ft; $p_m = 250$ psig; $p_0 = 60$ psig. Find δ'.

Solution: find the pressure factor, $(p_m-p_0)/2 = (250-60)/2 = 95$ psi. Now lay a straightedge from 0.347 on the A axis to 43.1 on w. Mark the intersection with reference axis (1). Connect this point with 3189 on a and mark a new intercept on reference axis (2). Connect the latter point with 95 on $(p_m-p_0)/2$ and read 2.4 ft-lb/ft on δ'.

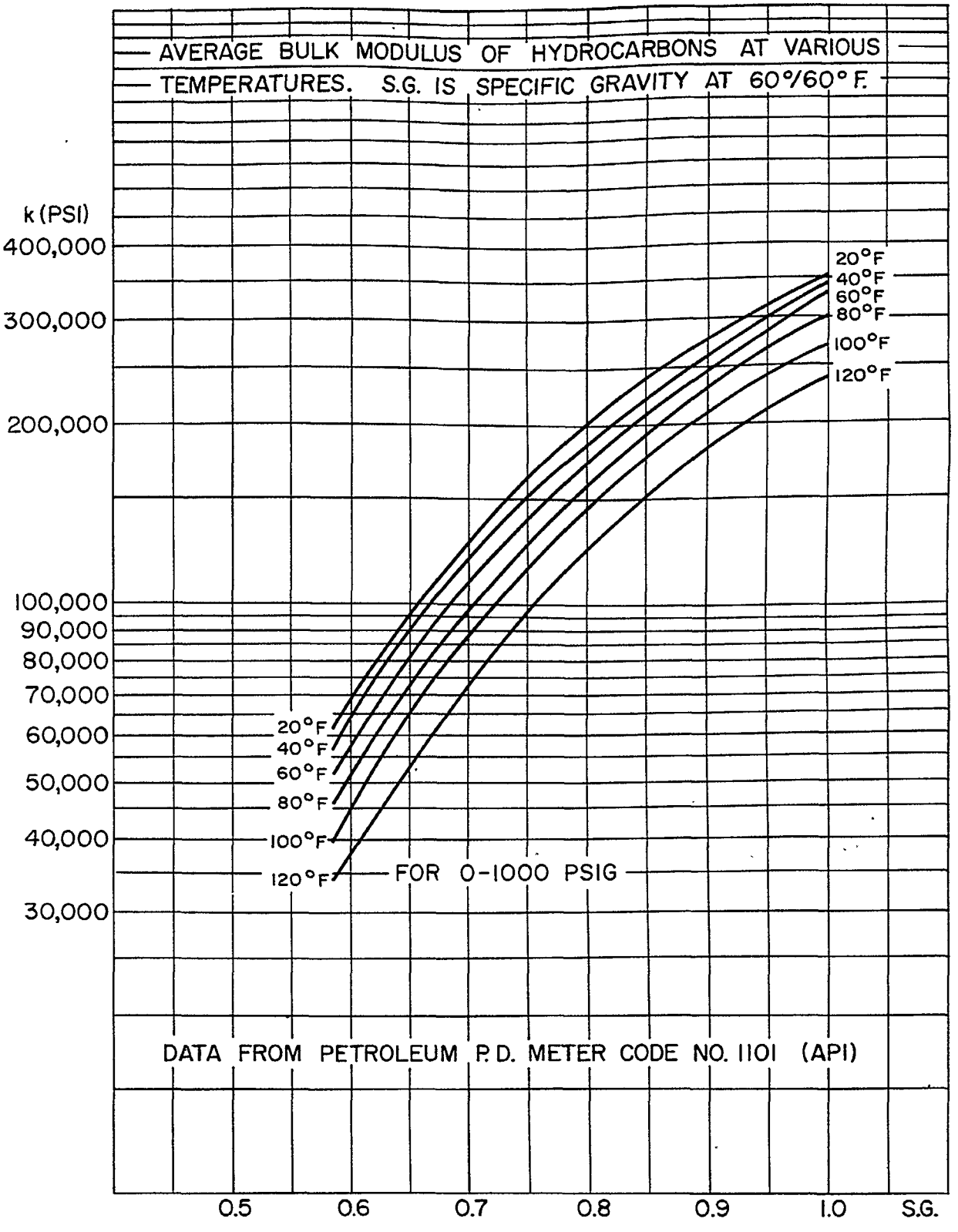

Fig. 35. Bulk Modulus of Hydrocarbons

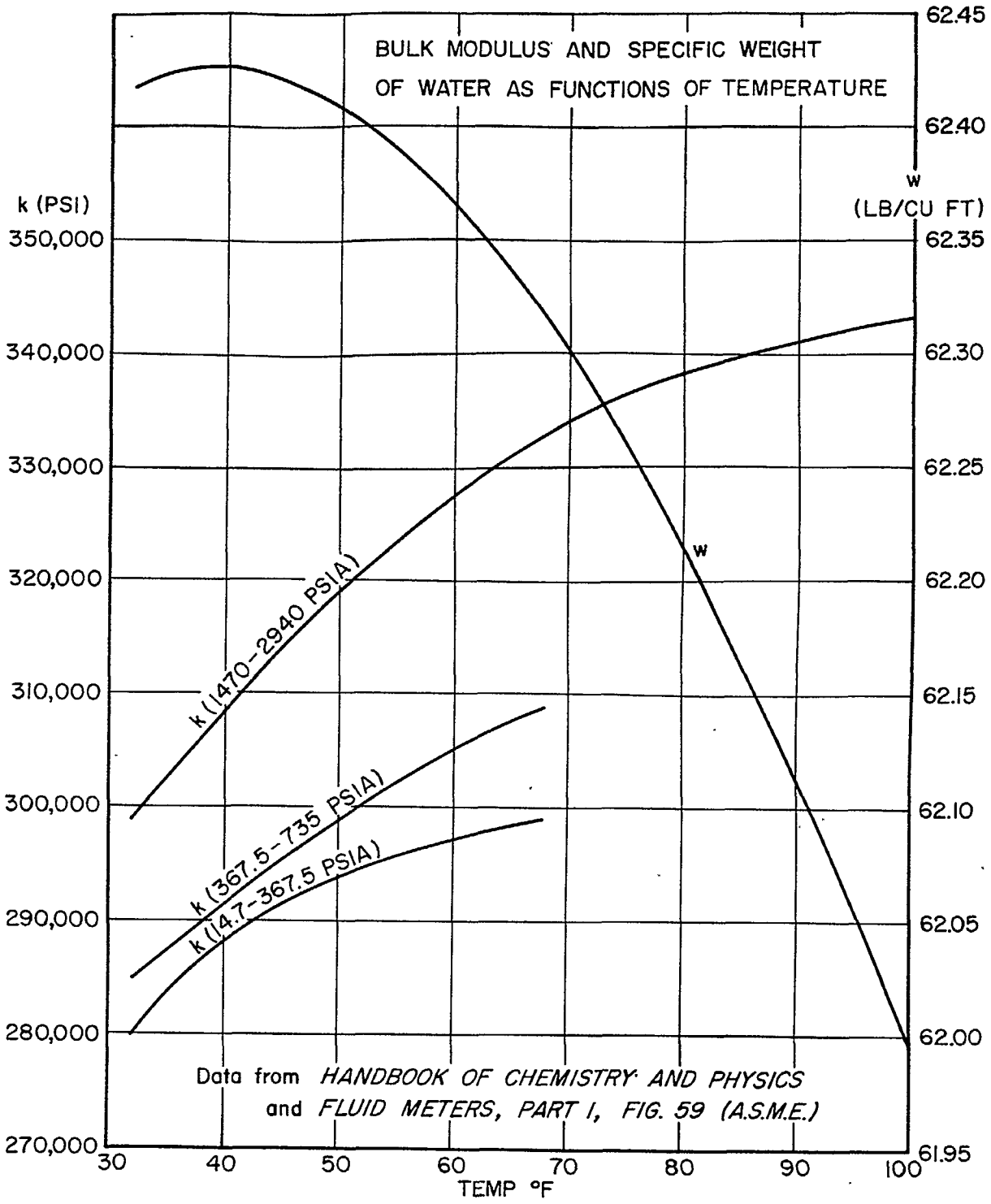

Fig. 36. Bulk Modulus and Specific Weight of Water

TABLE I

Conversion Factors and Formulas

Psi = 0.4331 × (head in feet of liquid) × (specific gravity)

Psi = 0.0361 × (head in inches of liquid) × (specific gravity)

Psi = 0.491 × (head or vacuum inches of mercury)

Psi = 0.01934 × (head or vacuum, mm. of mercury)

Psi = 14.7 × (atmospheres)

Psi = 14.22 × (Metric atmospheres: kg/sq cm)

Gallons = 7.481 × (cubic feet)

Cubic inches = 1728 × (cubic feet) = 231 × (gallons)

Cubic inches = 0.061024 × (cubic centimeters)

Cubic feet = 0.13368 × (gallons)

Barrels = 42 × (gallons)

Cubic feet per second = 0.002228 × (GPM)

Gallons per minute = 449 × (CFS)

Gallons per minute = 0.7 × (barrels per hour)

Gallons per minute = 0.029167 × (barrels per day)

lb/sec = w × (CFS)

lb/sec = 0.002228w × (GPM)

$$\text{Degrees } API = \frac{141.5}{Sp\ Gr\ 60/60F} - 131.5$$

Jet velocity, $v(FPS) = \sqrt{2gh}$, where h = head in feet

Velocity head, $h(ft) = v^2/2g$

Bernouilli's Theorem

$$h_1 + Z_1 + v_1^2/2g = h_2 + Z_2 + v_2^2/2g$$

where Z_1 and Z_2 are elevations (ft) above datum

Friction loss in pipes

$$h_f = fLv^2/2Dg$$

in which h_f = head loss in feet of liquid flowing

f = friction factor, from Fig. 32

L = pipe length, ft

D = inside diameter of pipe, ft

TABLE II

Viscosity Conversion Factors

A. CONVERSION FACTORS FOR DYNAMIC VISCOSITY

Multiply

To Obtain	POISE $\frac{gm\ mass}{cm\text{-}sec}$ or $\frac{dyne\text{-}sec}{sq\ cm}$	CENTIPOISE μ_2 $\frac{gm\ mass}{cm\text{-}sec}$ or $\frac{dyne\text{-}sec}{sq\ cm}$	μ_1 $\frac{slugs}{ft\text{-}sec}$ or $\frac{lb\ force\text{-}sec}{sq\ ft}$	REYNS $\frac{lb\ force\text{-}sec}{sq\ in}$	$\frac{lb\text{-}mass}{ft\text{-}sec}$ or $\frac{pdl\text{-}sec}{sq\ ft}$
	By				
To Obtain POISE	1	0.01	478.8	6.895×10^{4}	14.88
CENTIPOISE μ_2	100	1	47,880	6.895×10^{6}	1,488
μ_1 $\frac{slugs}{ft\text{-}sec}$	2.088×10^{-3}	2.088×10^{-5}	1	144	0.03108
REYNS	1.45×10^{-5}	1.45×10^{-7}	6.95×10^{-3}	1	2.16×10^{-4}
$\frac{lb\ mass}{ft\text{-}sec}$	0.0672	6.72×10^{-4}	32.17	4,637	1

B. RELATIONSHIP BETWEEN DYNAMIC AND KINEMATIC VISCOSITY

Dynamic viscosity = Kinematic viscosity × specific gravity

$$\text{or } \mu_2 = \frac{\nu w}{62.4}$$

where μ_2 = dynamic viscosity, centipoise
ν = kinematic viscosity, centistokes
w = specific weight, lb/cu ft

C. CONVERSION FACTORS FOR KINEMATIC VISCOSITY: FROM VARIOUS UNITS TO CENTISTOKES (ν)

ν = sq ft/sec × 92,900

ν = stokes × 0.01

For viscometer readings of t seconds:

$\nu = 0.220t - 135/t$ for Saybolt Universal when $t > 100$

$\nu = 0.226t - 195/t$ for Saybolt Universal when $32 < t < 100$

$\nu = 2.16t - 60/t$ for Saybolt Furol when $t > 40$

$\nu = 2.24t - 184/t$ for Saybolt Furol when $25 < t < 40$

$\nu = 0.247t - 50/t$ for Redwood No. 1 when $t > 100$

$\nu = 0.260t - 179/t$ for Redwood No. 1 when $34 < t < 100$

$\nu = 2.7t - 2000/t$ for Redwood Admiralty

$\nu = 0.147t - 374/t$ for Engler degrees

TABLE III

Properties of Liquids

(At Base Temperature 60°F)

	w Specific Weight lb/cu ft	k Bulk Modulus psi	ν or μ_z Viscosity centistokes or centipoise	ϵ_L Coefficient of Volumetric Expansion per °F Multiply by 10^{-5}
Ammonia (Saturated) (at −28°F)	42.3	392,000	0.24 cp	
Ethyl Alcohol (100%)	49.4	131,000	1.35 cp	62
Hydrazine (100%)	63.2		1.02 cp	
Mercury	849	3,900,000	1.55 cp	10
Nitric Acid (Red or White Fuming)	96	414,000	1.13 cp	78
Nitrogen (Liquid) (at −321°F)	50	968,000		
Oxygen (Liquid) (at −298°F)	71.35		0.19 cp	
Petroleum Derivatives				
[1]S. G. Range 60°/60°F				
0.6112 − 0.6275	38.1 − 39.2	See Fig. 35	0.30 cs[2]	90
0.6276 − 0.6417	39.2 − 40.0			85
0.6418 − 0.6722	40.0 − 41.9			80
0.6723 − 0.7238	41.9 − 45.2		0.64 cs[4]	70
[2]0.7239 − 0.7753	45.2 − 48.4		0.84 cs[5]	60
0.7754 − 0.8498	48.4 − 53.0		2.6 cs[6]	50
0.8499 − 0.9659	53.0 − 60.3		14.2 cs[7]	40
0.9660 − 1.0760	60.3 − 67.1			35
Fresh Water	See Fig. 36	See Fig. 36	1.1 cp	11.5
Sea Water	63.99	312,000	1.2 cs	23

NOTES:

[1] S.G. $60°/60°F = \dfrac{\text{Wt. of given volume of oil at } 60°F}{\text{Wt. of equal volume of water at } 60°F}$

(Both weights corrected for buoyancy of air)

[2] All blends of gasoline and benzene are considered to fall in this group

[3] Value for 0.6 S.G. butane

[4] Value for 0.69 S.G. gasoline, Grade 115-145

[5] Value for 0.76 S.G. gasoline

[6] Value for 0.815 S.G. kerosene

[7] Value for 0.9 S.G. fuel oil

TABLE IV

Properties of Conduit Materials

(At Base Temperature 60°F)

Material	E Young's Modulus psi (Multiply by 10^6)	εc Coefficient of Linear Expansion per °F (Multiply by 10^{-6})	Density lb/cu in.	Yield Stress psi (Multiply by 10^3)
Aluminum				
2 S-O	10	13.1	0.098	5
3 S-O	10	12.9	0.099	6
61S-T6	10	13.1	0.098	40
63S-T6	10	13.0	0.098	30
Brass (Red)	13	10.4	0.316	14-20
Copper (Deoxidized)	18	9.8	0.323	8-10
Cast Iron	14	6	0.261	
Carbon Steel	28-30	5.83	0.308	12-25
Stainless Steel				
Type 304	28	9.6	0.29	30°
Type 316	28	8.9	0.29	30°
Type 321	28	9.3	0.29	30°
Type 347	28	9.3	0.29	35°
Refueling Hose°°	0.125-0.225			

°Annealed
°°Collapsible and non-collapsible: 3 in. and 4 in. diameter; d/e 6–9

TABLE V

Dimensions of Seamless and Welded Steel Pipe

Nominal Pipe Size-In.	Outside Diam. In.	Schedule No.	Wall Thickness In.	Inside Diam. In.	Flow Area Sq Ft
⅛	0.405	40 ST	0.068	0.269	0.0004
		80 XS	0.095	0.215	0.0003
¼	0.540	40 ST	0.088	0.364	0.0007
		80 XS	0.119	0.302	0.0005
⅜	0.675	40 ST	0.091	0.493	0.0013
		80 XS	0.126	0.423	0.0010
½	0.840	40 ST	0.109	0.622	0.0021
		80 XS	0.147	0.546	0.0016
		160	0.187	0.466	0.0012
		XX	0.294	0.252	0.0003
¾	1.050	40 ST	0.113	0.824	0.0037
		80 XS	0.154	0.742	0.0030
		160	0.218	0.614	0.0020
		XX	0.308	0.434	0.0010
1	1.315	40 ST	0.133	1.049	0.0060
		80 XS	0.179	0.957	0.0050
		160	0.250	0.815	0.0036
		XX	0.358	0.599	0.0020
1¼	1.660	40 ST	0.140	1.380	0.0104
		80 XS	0.191	1.278	0.0089
		160	0.250	1.160	0.0073
		XX	0.382	0.896	0.0044
1½	1.900	40 ST	0.145	1.610	0.0141
		80 XS	0.200	1.500	0.0123
		160	0.281	1.337	0.0097
		XX	0.400	1.100	0.0066
2	2.375	40 ST	0.154	2.067	0.0233
		80 XS	0.218	1.939	0.0205
		160	0.343	1.689	0.0156
		XX	0.436	1.503	0.0123
2½	2.875	40 ST	0.203	2.469	0.033
		80 XS	0.276	2.323	0.029
		160	0.375	2.125	0.025
		XX	0.552	1.771	0.017
3	3.500	40 ST	0.216	3.068	0.051
		80 XS	0.300	2.900	0.046
		160	0.438	2.624	0.038
		XX	0.600	2.300	0.029
3½	4.000	40 ST	0.226	3.548	0.069
		80 XS	0.318	3.364	0.062
		XX	0.636	2.728	0.041
4	4.500	40 ST	0.237	4.026	0.088
		80 XS	0.337	3.826	0.080
		160	0.531	3.438	0.064
		XX	0.674	3.152	0.054
5	5.563	40 ST	0.258	5.047	0.139
		80 XS	0.375	4.813	0.126
		160	0.625	4.313	0.101
		XX	0.750	4.063	0.090
6	6.625	40 ST	0.280	6.065	0.201
		80 XS	0.432	5.761	0.181
		160	0.718	5.189	0.147
		XX	0.864	4.897	0.131
8	8.625	20	0.250	8.125	0.360
		40 ST	0.322	7.981	0.347
		80 XS	0.500	7.625	0.317
		120	0.718	7.189	0.282
		XX	0.875	6.875	0.256
		160	0.906	6.813	0.253
10	10.750	20	0.250	10.250	0.573
		40 ST	0.365	10.020	0.548
		60 XS	0.500	9.750	0.518
		80	0.593	9.564	0.499
		120	0.843	9.064	0.448
		160	1.125	8.500	0.394

TABLE V (continued)

Dimensions of Seamless and Welded Steel Pipe

Nominal Pipe Size-In.	Outside Diam. In.	Schedule No.	Wall Thickness In.	Inside Diam. In.	Flow Area Sq Ft
12	12.750	20	0.250	12.250	0.819
		ST	0.375	12.000	0.785
		40	0.406	11.938	0.777
		80 XS	0.500	11.750	0.753
		120	1.000	10.750	0.631
		160	1.312	10.126	0.559
14	14.000	10	0.250	13.500	0.994
		20	0.312	13.375	0.976
		30 ST	0.375	13.250	0.958
		40	0.438	13.124	0.940
		XS	0.500	13.000	0.921
		80	0.750	12.500	0.852
		100	0.937	12.125	0.802
		160	1.406	11.188	0.683
16	16.000	10	0.250	15.500	1.310
		20	0.312	15.375	1.290
		30 ST	0.375	15.250	1.269
		40 XS	0.500	15.000	1.227
		60	0.656	14.688	1.176
		80	0.843	14.314	1.117
		120	1.218	13.564	1.003
		160	1.593	12.814	0.896
18	18.000	10	0.250	17.500	1.670
		20	0.312	17.375	1.646
		ST	0.375	17.250	1.623
		XS	0.500	17.000	1.576
		40	0.562	16.876	1.553
		60	0.750	16.500	1.485
		100	1.156	15.688	1.342
		160	1.781	14.438	1.137
20	20.000	10	0.250	19.500	2.073
		20 ST	0.375	19.250	2.021
		30 XS	0.500	19.000	1.969
		40	0.593	18.814	1.930
		80	1.031	17.938	1.755
		160	1.968	16.064	1.408
24	24.000	10	0.250	23.500	3.014
		20 ST	0.375	23.250	2.951
		XS	0.500	23.000	2.882
		40	0.687	22.626	2.791
		80	1.218	21.564	2.535
		160	2.343	19.314	2.035
30	30.000	10	0.312	29.376	4.708
		20	0.500	29.000	4.590
		30	0.625	28.750	4.507

TABLE VI

FOUR PLACE LOGARITHMS

N.	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	5	7	9
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7
33	5185	5198	5211	5224	5237	5250	5263	6276	5289	5302	1	3	4	5	6
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	6
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4
N.	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5

FOUR PLACE LOGARITHMS (continued)

N.	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	3	3
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2
N.	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5

LOGARITHMS (continued)

N.		0	1	2	3	4	5	6	7	8	9	d.
1000	000	0000	0434	0869	1303	1737	2171	2605	3039	3473	3907	434
1001		4341	4775	5208	5642	6076	6510	6943	7377	7810	8244	434
1002		8677	9111	9544	9977	*0411	*0844	*1277	*1710	*2143	*2576	433
1003	001	3009	3442	3875	4308	4741	5174	5607	6039	6472	6905	433
1004		7337	7770	8202	8635	9067	9499	9932	*0364	*0796	*1228	432
1005	002	1661	2093	2525	2957	3389	3821	4253	4685	5116	5548	432
1006		5980	6411	6843	7275	7706	8138	8569	9001	9432	9863	431
1007	003	0295	0726	1157	1588	2019	2451	2882	3313	3744	4174	431
1008		4605	5036	5467	5898	6328	6759	7190	7620	8051	8481	431
1009		8912	9342	9772	*0203	*0633	*1063	*1493	*1924	*2354	*2784	430
1010	004	3214	3644	4074	4504	4933	5363	5793	6223	6652	7082	430
1011		7512	7941	8371	8800	9229	9659	*0088	*0517	*0947	*1376	429
1012	005	1805	2234	2663	3092	3521	3950	4379	4808	5237	5666	429
1013		6094	6523	6952	7380	7809	8238	8666	9094	9523	9951	429
1014	006	0380	0808	1236	1664	2092	2521	2949	3377	3805	4233	428
1015		4660	5088	5516	5944	6372	6799	7227	7655	8082	8510	428
1016		8937	9365	9792	*0219	*0647	*1074	*1501	*1928	*2355	*2782	427
1017	007	3210	3637	4064	4490	4917	5344	5771	6198	6624	7051	427
1018		7478	7904	8331	8757	9184	9610	*0037	*0463	*0889	*1316	426
1019	008	1742	2168	2594	3020	3446	3872	4298	4724	5150	5576	426
1020		6002	6427	6853	7279	7704	8130	8556	8981	9407	9832	426
1021	009	0257	0683	1108	1533	1959	2384	2809	3234	3659	4084	425
1022		4509	4934	5359	5784	6208	6633	7058	7483	7907	8332	425
1023		8756	9181	9605	*0030	*0454	*0878	*1303	*1727	*2151	*2575	424
1024	010	3000	3424	3848	4272	4696	5120	5544	5967	6391	6815	424
1025		7239	7662	8086	8510	8933	9357	9780	*0204	*0627	*1050	424
1026	011	1474	1897	2320	2743	3166	3590	4013	4436	4859	5282	423
1027		5704	6127	6550	6973	7396	7818	8241	8664	9086	9509	423
1028		9931	*0354	*0776	*1198	*1621	*2043	*2465	*2887	*3310	*3732	422
1029	012	4154	4576	4998	5420	5842	6264	6685	7107	7529	7951	422
1030		8372	8794	9215	9637	*0059	*0480	*0901	*1323	*1744	*2165	422
1031	013	2587	3008	3429	3850	4271	4692	5113	5534	5955	6376	421
1032		6797	7218	7639	8059	8480	8901	9321	9742	*0162	*0583	421
1033	014	1003	1424	1844	2264	2685	3105	3525	3945	4365	4785	420
1034		5205	5625	6045	6465	6885	7305	7725	8144	8564	8984	420
1035		9403	9823	*0243	*0662	*1082	*1501	*1920	*2340	*2759	*3178	420
1036	015	3598	4017	4436	4855	5274	5693	6112	6531	6950	7369	419
1037		7788	8206	8625	9044	9462	9881	*0300	*0718	*1137	*1555	419
1038	016	1974	2392	2810	3229	3647	4065	4483	4901	5319	5737	418
1039		6155	6573	6991	7409	7827	8245	8663	9080	9498	9916	418
1040	017	0333	0751	1168	1586	2003	2421	2838	3256	3673	4090	417
1041		4507	4924	5342	5759	6176	6593	7010	7427	7844	8260	417
1042		8677	9094	9511	9927	*0344	*0761	*1177	*1594	*2010	*2427	417
1043	018	2843	3259	3676	4092	4508	4925	5341	5757	6173	6589	416
1044		7005	7421	7837	8253	8669	9084	9500	9916	*0332	*0747	416
1045	019	1163	1578	1994	2410	2825	3240	3656	4071	4486	4902	415
1046		5317	5732	6147	6562	6977	7392	7807	8222	8637	9052	415
1047		9467	9882	*0296	*0711	*1126	*1540	*1955	*2369	*2784	*3198	415
1048	020	3613	4027	4442	4856	5270	5684	6099	6513	6927	7341	414
1049		7755	8169	8583	8997	9411	9824	*0238	*0652	*1066	*1479	414
1050	021	1893	2307	2720	3134	3547	3961	4374	4787	5201	5614	413
N.		0	1	2	3	4	5	6	7	8	9	d.

LOGARITHMS (continued)

N.		0	1	2	3	4	5	6	7	8	9	d.
1050	021	1893	2307	2720	3134	3547	3961	4374	4787	5201	5614	413
1051		6027	6440	6854	7267	7680	8093	8506	8919	9332	9745	413
1052	022	0157	0570	0983	1396	1808	2221	2634	3046	3459	3871	413
1053		4284	4696	5109	5521	5933	6345	6758	7170	7582	7994	412
1054		8406	8818	9230	9642	*0054	*0466	*0878	*1289	*1701	*2113	412
1055	023	2525	2936	3348	3759	4171	4582	4994	5405	5817	6228	411
1056		6639	7050	7462	7873	8284	8695	9106	9517	9928	*0339	411
1057	024	0750	1161	1572	1982	2393	2804	3214	3625	4036	4446	411
1058		4857	5267	5678	6088	6498	6909	7319	7729	8139	8549	410
1059		8960	9370	9780	*0190	*0600	*1010	*1419	*1829	*2239	*2649	410
1060	025	3059	3468	3878	4288	4697	5107	5516	5926	6335	6744	410
1061		7154	7563	7972	8382	8791	9200	9609	*0018	*0427	*0836	409
1062	026	1245	1654	2063	2472	2881	3289	3698	4107	4515	4924	409
1063		5333	5741	6150	6558	6967	7375	7783	8192	8600	9008	408
1064		9416	9824	*0233	*0641	*1049	*1457	*1865	*2273	*2680	*3088	408
1065	027	3496	3904	4312	4719	5127	5535	5942	6350	6757	7165	408
1066		7572	7979	8387	8794	9201	9609	*0016	*0423	*0830	*1237	407
1067	028	1644	2051	2458	2865	3272	3679	4086	4492	4899	5306	407
1068		5713	6119	6526	6932	7339	7745	8152	8558	8964	9371	406
1069		9777	*0183	*0590	*0996	*1402	*1808	*2214	*2620	*3026	*3432	406
1070	029	3838	4244	4649	5055	5461	5867	6272	6678	7084	7489	406
1071		7895	8300	8706	9111	9516	9922	*0327	*0732	*1138	*1543	405
1072	030	1948	2353	2758	3163	3568	3973	4378	4783	5188	5592	405
1073		5997	6402	6807	7211	7616	8020	8425	8830	9234	9638	405
1074	031	0043	0447	0851	1256	1660	2064	2468	2872	3277	3681	404
1075		4085	4489	4893	5296	5700	6104	6508	6912	7315	7719	404
1076		8123	8526	8930	9333	9737	*0140	*0544	*0947	*1350	*1754	403
1077	032	2157	2560	2963	3367	3770	4173	4576	4979	5382	5785	403
1078		6188	6590	6993	7396	7799	8201	8604	9007	9409	9812	403
1079	033	0214	0617	1019	1422	1824	2226	2629	3031	3433	3835	402
1080		4238	4640	5042	5444	5846	6248	6650	7052	7453	7855	402
1081		8257	8659	9060	9462	9864	*0265	*0667	*1068	*1470	*1871	402
1082	034	2273	2674	3075	3477	3878	4279	4680	5081	5482	5884	401
1083		6285	6686	7087	7487	7888	8289	8690	9091	9491	9892	401
1084	035	0293	0693	1094	1495	1895	2296	2696	3096	3497	3897	400
1085		4297	4698	5098	5498	5898	6298	6698	7098	7498	7898	400
1086		8298	8698	9098	9498	9898	*0297	*0697	*1097	*1496	*1896	400
1087	036	2295	2695	3094	3494	3893	4293	4692	5091	5491	5890	399
1088		6289	6688	7087	7486	7885	8284	8683	9082	9481	9880	399
1089	037	0279	0678	1076	1475	1874	2272	2671	3070	3468	3867	399
1090		4265	4663	5062	5460	5858	6257	6655	7053	7451	7849	398
1091		8248	8646	9044	9442	9839	*0237	*0635	*1033	*1431	*1829	398
1092	038	2226	2624	3022	3419	3817	4214	4612	5009	5407	5804	398
1093		6202	6599	6996	7393	7791	8188	8585	8982	9379	9776	397
1094	039	0173	0570	0967	1364	1761	2158	2554	2951	3348	3745	397
1095		4141	4538	4934	5331	5727	6124	6520	6917	7313	7709	397
1096		8106	8502	8898	9294	9690	*0086	*0482	*0878	*1274	*1670	396
1097	040	2066	2462	2858	3254	3650	4045	4441	4837	5232	5628	396
1098		6023	6419	6814	7210	7605	8001	8396	8791	9187	9582	395
1099		9977	*0372	*0767	*1162	*1557	*1952	*2347	*2742	*3137	*3532	395
1100	041	3927	4322	4716	5111	5506	5900	6295	6690	7084	7479	395
N.		0	1	2	3	4	5	6	7	8	9	d.

BIBLIOGRAPHY

1. "Water Hammer in Conduits; Study of the Means Used for Diminishing the Effects," by J. Michaud. *Bulletin de la Societe Vaudoise des Ingenieurs et Architects;* Lausanne, 1878.

2. "Water Hammer," by Joukowsky, translated by Miss O. Simin. *Proc. Am. Water Works Assoc.*, 341, 1904.

3. *Theory of Water Hammer*, by Lorenzo Allievi, translated by Eugene Halmos. Riccardo Garroni, Rome, Italy, 1925.

4. *Symposium on Water Hammer*, American Society of Mechanical Engineers, New York, June, 1933.

5. "Pressures in Penstocks Caused by the Gradual Closing of Turbine Gates," by N. R. Gibson. *Trans. A.S.C.E.*, Vol. 83, 1920, p. 707.

6. "Variations of Flow in Water Conduits," by M. L. Bergeron. *Comptes Rendus des Travaux de la Societe Hydrotechnique de France*, 1932.

7. "Methods of Calculating Water Hammer Pressures," by F. M. Dawson and A. A. Kalinske. *J. Am. Water Works Assoc.*, Vol. 31, No. 11, Nov., 1939, p. 1835.

8. "Simple Graphical Solution for Pressure Rise in Pipes and Pump Discharge Lines," by R. W. Angus. *Engineering Journal.* v. 18, n. 2, February, 1935.

9. "The Differential Surge Tank," by R. D. Johnson. *Trans. A.S.C.E.*, Vol. 78, 1915.

10. *Hydraulic Transients*, by G. R. Rich, McGraw-Hill, New York, 1951.

11. *Fluid-Meters*, American Society of Mechanical Engineers, New York, 1937.

12. "Typical Analysis of Water Hammer in a Pumping Plant of the Colorado River Aqueduct," by R. M. Peabody. *Trans. A.S.M.E.*, Vol. 61, 1939, p. 117.

13. "A Survey of Flow Calculation Methods," by S. P. Johnson. *Preprinted Papers and Program—A.S.M.E. Summer Meeting*, June, 1934, p. 98.

14. "The Occurrence and Elimination of Surge or Oscillating Pressures in Discharge Lines From Reciprocating Pumps," by H. Diederichs and W. D. Pomeroy. *Trans. A.S.M.E.*, Vol. 51, Pet. 51-2-9, 1929.

15. "Transmission of Pressure Waves in Liquid Filled Tubes," by W. T. Thomson. *Proc. of the First U.S. Nat'l Congress of Applied Mechanics*, 1951, p. 927.

16. "Pressure Surges and Vibration in Reciprocating Pump Piping," by J. W. Squire. *Trans. A.S.M.E.*, Vol. 71, May, 1949, p. 317.

17. *Pressure Surge Research Project No. 1, Final Report*, by R. L. Flanders, E. J. Waller, *et al.* Oklahoma A & M College, Stillwater, Oklahoma, Feb., 1953.

18. *Advanced Fluid Dynamics and Fluid Machinery*, by R. C. Binder. Prentice-Hall, New York, 1951.

19. *ASME-API Code for Installation, Proving and Operation of Positive Displacement Meters in Liquid Hydrocarbon Service*, No. 1101, American Petroleum Institute, New York, 1952.

20. *Handbook of Chemistry and Physics*, Chemical Rubber Publishing Co., Cleveland.

21. *Engineering Hydraulics*, edited by Hunter Rouse. Wiley. New York, 1950.

The Author

Howard Moore received a degree of B.A. in Mathematics and Physics at the University of Colorado. He taught mathematics and physics in the Colorado public schools for a time. In the field of mechanical and hydraulic engineering he had experience as a research laboratory analyst, draftsman and designer for Douglas Aircraft Company at Long Beach, California where he conducted tests on mechanical, pneumatic, and hydraulic aircraft components and systems; designed flight control and cargo loading mechanisms. He supervised mechanical design and design of fluid handling systems for Aerojet Engineering Corporation, Azusa, California, for the F-84 and B-47 rocket test cell installations, the Rocket-Engine Test Stands at Edwards Air Force Base, California, and the Turbopump Testing Facility, also at Edwards Air Force Base. He directed the design of the Air Transportable Hydrant Refueling System for Aircraft and conducted surge investigations for the American Pipe and Steel Corporation of Alhambra, California. Currently, as a project engineer in the MagniLastic Division of Cook Electric Company he is responsible for the analysis and control of surge in fluid handling systems and the application of surge and pulsation suppressing devices. He is a member of the Society of American Military Engineers.



COPY FORM	INITIALS	DATE
	CSP	10-19-99

BOX NUMBER	SEGMENT NO!	REQUESTING PARTY
04-1913		B&B

NOTES