C.1

AIR CHAMBER DESIGN CHARTS

by

GALATIUK, WILLIAM ROBERT

B.Sc. (Civ. Eng.) The University of Manitoba
Winnipeg, Manitoba, 1964

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF

THE REQUIREMENT FOR THE DEGREE OF

MASTER OF APPLIED SCIENCE

in the Department

of

Civil Engineering

We accept this thesis as conforming to

the required standard

THE UNIVERSITY OF BRITISH COLUMBIA

SEPTEMBER, 1973

In presenting this thesis in partial fulfillment of the requirements for an advanced degree at the University of British Columbia, I agree that the Library shall make it freely available for reference and study. I further agree that permission for extensive copying of this thesis for scholarly purposes may be granted by the Head of my Department or by his representatives. It is understood that copying or publication of this thesis for financial gain shall not be allowed without my written permission.

Department of Civil Engineering

The University of British Columbia
Vancouver 8, Canada

Date September 10, 1973

ABSTRACT

The air chamber has certain advantages over both the open-top surge tank and the valve-type surge suppressor for controlling pressure surges in pump-discharge lines.

The main purpose of this study was to produce charts which can be used for designing or checking the size of an air chamber required for a particular pumping installation.

The characteristics method was used to convert the two partial differential equations of momentum and continuity into four total differential equations. The solution of the equations (finite-difference form) was carried through by digital computer to provide the data required for the preparation of the charts.

Results obtained on the digital computer by the method of characteristics are checked by the graphical method.

Examples demonstrating the use of the charts are included.

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

ACKNOWLEDGEMENT

The author wishes to express his gratitude to his supervisor, Dr. E. Ruus, for his valuable guidance, constructive criticism and suggestions, and to Dr. W.F. Caselton for reviewing the work.

The study was supported by a grant from the National Research Council.

NOTATION

The following symbols are used in this thesis:

A = cross-sectional area of pipe, in ft ^{2}

a = propagation velocity of waterhammer wave, in ft/sec.

C_o = initial volume of air in the air chamber at absolute pressure head H_o*, in ft^3

C_{orf} = orifice loss coefficient

D = inside diameter of pipe, in ft

f = Darcy-Weisbach friction factor

g = gravity acceleration, in ft/sec.^{2}

H = transient-state piezometric pressure head above datum at the beginning of a time interval, in ft

H_o = initial steady-state piezometric pressure head above datum, in ft

H_P = transient-state piezometric pressure head above datum at the end of a time interval, in ft

H^* = transient-state absolute piezometric pressure head above datum, in ft

H_o^* = initial steady-state absolute piezometric pressure head above datum, in ft

H_{orf} = orifice throttling loss corresponding to discharge q, in ft

H_{orfo} = orifice throttling loss corresponding to discharge q_o, in ft

K = coefficient relating total pipe line head loss due to friction to piezometric pressure head above datum

L = length of pipe line, in ft

m = power used in pressure - volume relationship, $H^* v_{air}^m$ = constant, for an air chamber

Q_o = steady state discharge in the pipe line, in ft^3/sec.

q = transient state orifice discharge, in ft^3/sec.

t = time, in seconds

V = transient-state velocity in pipe at the beginning of a time interval, in ft/sec.

V_P = transient-state velocity in pipe at the end of a time interval, in ft/sec.

V_o = initial steady state velocity in pipe, in ft/sec.

v_{air} = transient-state volume of air in air chamber at the beginning of a time interval, in ft^3

v_{oair} = initial steady-state volume of air in air chamber, in ft^3

v_{Pair} = transient-state volume of air in air chamber at the end of a time interval, in ft^3

x = distance along pipe line, from pump, in ft

ρ = pipe line characteristic

ρ^* = pipe line characteristic in terms of absolute pressure head

σ^* = parameter pertaining to a pump-discharge line having an air chamber, in terms of absolute pressure head

θ = angle the pipe makes with the horizontal

θ' = grid mesh ratio, $\frac{\Delta t}{\Delta x}$

Δt = time increment, in seconds

Δx = incremental distance along the pipe line, in ft

INTRODUCTION

Sudden stopping or starting of large centrifugal pumps installed for irrigation, domestic water supply systems, pumped storage hydro-electric plants and other purposes cause transient pressures in the discharge lines. Starting control mechanisms can be designed to delay the starting up time sufficiently to prevent excessive over pressures. But sudden stopping in the event of power failure could result in objectionable waterhammer pressures in the pipe line.

In small installations, no special precautions are taken to avoid high waterhammer effects. Standard pipes and fittings of small diameter have a wall thickness sufficient to withstand appreciable transient pressures. In large pumping installations various pressure-control devices may be used to reduce waterhammer pressures. Some of these devices include:

(1) surge tanks,

(2) air chambers,

(3) surge suppressor valves, and

(4) slow closing check valves.

For controlling pressure surges in pump-discharge lines, the air chamber has certain advantages over both the open-top surge tank and the valve-type surge suppressor. For high head installations where the open surge tank is impractical, a properly designed air chamber provides good surge control. The air chamber can be near the pump whereas the surge tank can not always be so located. The air chamber can be designed to reduce the downsurges in a pump-discharge line, thus preventing collapse of the line and water-column separation; ordinarily, surge-suppressor valves are not suitable for this important function. The main disadvantage of air chambers is that the compressed air is continuously being lost through dissolving in the water and possible leakage. Consequently the air must be replenished periodically.

After it has been decided that certain types of pressure-control devices will meet design requirements, the final choice is usually based on a cost study of the various devices. The cost of an air chamber is determined primarily by its size and inside pressure. In this thesis, charts are presented which provide for the rapid determination of air chamber sizes required to control waterhammer pressures in pump-discharge lines where the transient pressures are caused by rapid pump shut down or by power failure. The charts were prepared using the method of characteristics to convert the two partial differential equations of momentum and continuity into four total differential equations. The solution was done by digital computer.

Examples demonstrating the use of the charts are given in Appendix A.

CHAPTER I

ASSUMPTIONS AND THEORY

1.1 ASSUMPTIONS

For the purposes of this study, the following assumptions were made.

(1) A check valve on the discharge side of the pump closes immediately on power failure. This eliminates the need to consider pump characteristics but introduces an abrupt pressure wave which must be accounted for throughout the computations.

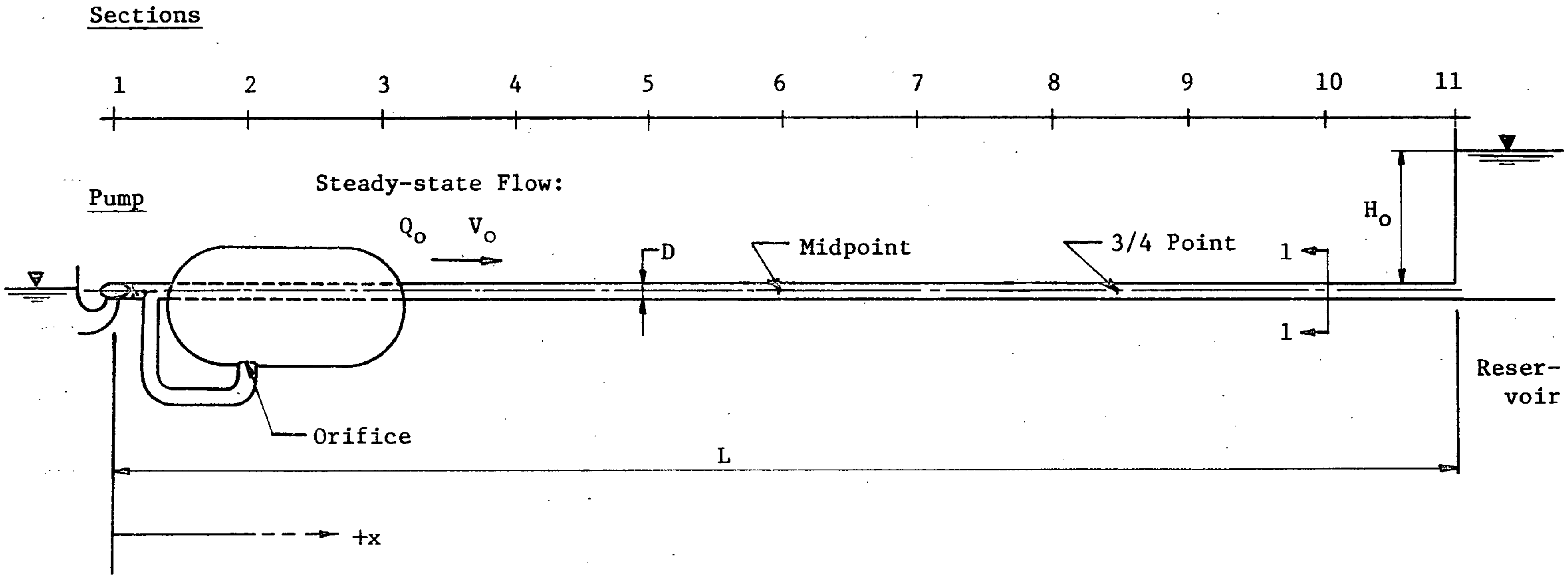
(2) The air chamber is situated near the pump as shown in Fig. 1.1. The steady-state water surface in the chamber has an elevation equal to that of the center line of the pipe (see Fig. 3.1). The transient-state head difference between the chamber water surface and the pipe center line is small and therefore neglected. The head loss through the orifice, if applicable, is taken into account in determining the absolute head, H*, in the tank.

(3) The pressure-volume relationship for the air in the chamber is expressed as:

$$H^* \, v_{air}^{1.2} = \text{a constant.}$$

The power 1.2 is an average of the powers 1.0 and 1.4 for the isothermal and adiabatic expansions respectively.

(4) The head loss, made up of surface friction and loss at the orifice, varies with the square of the velocity. Two types of orifices, one simple and one differential, were considered in the study. The ratio



PIPE LINE WITH AIR CHAMBER

FIG. 1.1

Sec. 1-1
Area of Pipe

of the total head loss for the same flow into and from the air chamber is 2.5:1 for the differential orifice, and 1:1 for the simple orifice.

(5) This study is limited to cases in which no water column separation occurs. This means that water vapour pressure is not reached and the pipe stays full of water at all times.

(6) A reservoir of constant elevation serves as the downstream boundary condition.

1.2 GENERAL THEORY

Normally, with the pump operating, the flow in the pipeline is in the forward direction, toward the reservoir. The check valve closes simultaneously with pump failure. This creates a head differential across the air chamber outlet. The compressed air causes the water in the chamber to discharge into the pipeline to maintain the head and the flow. Water will continue to flow out of the tank until the head in the chamber becomes less than the head in the pipeline at the chamber outlet. At this instant, the water in the discharge line will reverse its direction and flow into the air chamber. During this reverse flow condition, the retardation of the flow into the air chamber causes the pressure in the discharge line to increase to exceed normal operating head and will produce the maximum head for the transient. Resurges in the pipeline will occur with diminishing intensity.

1.3 PARAMETERS

The pressure surges in a pipeline equipped with an air chamber depend on the two parameters, ρ^* and σ^*, when friction is not considered[4]. Because frictional resistance is essential to the efficient use of an air chamber on a pump-discharge line, Evans and Crawford[4] introduced a third variable, K, to account for frictional losses. The variable K is defined so that KH_o* is the total head loss for a reverse flow of Q_o. Q_o is the initial rate of flow in the pipeline, in cubic feet per second (ft^3/sec.).

The pipeline characteristic, ρ, is defined as

$$\rho = \frac{a\ V_o}{2gH_o} \tag{1.1}$$

in which a is the propagation velocity of waterhammer waves in the pipeline, in feet per second (ft/sec); V_o is the steady-state velocity in ft/sec; H_o is the steady-state pressure head, in feet of water (ft); and g is gravity acceleration in feet per second per second (ft/sec^2). The characteristic ρ is dimensionless and is a function of the ratio of the steady-state kinetic energy to the total potential energy in a unit length of conduit. In air chambers, the volume of the air is a function of the absolute pressure to which it is subjected. In terms of absolute pressure, the pipeline characteristic ρ becomes

$$\rho* = \frac{aV_o}{2gH_o*} \tag{1.2}$$

where H_o* is the normal absolute pressure head in the pipeline at the entrance to the air chamber.

The parameter, σ*, that is characteristic to a pump-discharge line having an air chamber is defined[4] as

$$\sigma^* = \frac{2gC_oH_o^*}{ALV_o^2} \tag{1.3}$$

in which C_o is the initial volume of air in the air chamber at absolute pressure head, H_o^*, in cubic feet (ft^3); A is the cross-sectional area of the pipe in square feet (ft^2); and L is the length of the pipe in feet. The parameter σ^* expresses the ratio of the steady-state potential energy of the air in the air chamber to the steady-state kinetic energy of the water in the discharge line.

1.4 RELATIONSHIP BETWEEN σ^* AND ρ^*

From Eqs. (1.2) and (1.3)

$$\sigma^*\rho^* = \frac{C_{oa}}{ALV_o} \tag{1.4}$$

or

$$C_o = \sigma^*\rho^*Q_oL/a\ . \tag{1.5}$$

From Eq. (1.2)

$$2\rho^* = \frac{aV_o}{gH_o^*} \tag{1.6}$$

and the constant for a pipeline having an air chamber will be defined as

$$\rho^*\sigma^* = \frac{C_{oa}}{ALV_o} \tag{1.7}$$

or

$$C_o = \frac{(\rho^*\sigma^*)\ ALV_o}{a}\ . \tag{1.8}$$

CHAPTER II

METHOD OF CHARACTERISTICS

2.1 GENERAL

The characteristics method[9] converts the two partial differential equations of momentum and continuity into four total differential equations. Non-linear friction is retained, as well as the effect of the pipes being non-horizontal. The equations are expressed in finite-difference form, and the solution is carried through by digital computer. Advantages of the method are:

- accuracy of results as non-linear terms are retained
- there is proper inclusion of friction
- it affords ease in handling the boundary conditions and ease in programming complex piping systems
- there is no need for large storage capacity in the computer
- detailed results are completely tabulated.

It is by far the most general and powerful method for handling waterhammer.

2.2 BASIC EQUATIONS FOR UNSTEADY FLOW THROUGH PIPES

The velocity and pressure of moving fluids in pipes are governed by the continuity and momentum equations.

The momentum equation[3] for flow through a pipe which is inclined or horizontal, tapered or straight, slightly or highly deformable, is given by

$$gH_x + V_t + VV_x + \frac{fV|V|}{2D} = 0 , \qquad (2.1)$$

in which g is gravity acceleration, V is fluid velocity, f is the Darcy-Weisbach friction factor, H is the total pressure head above the datum line, D is the inside diameter of the pipe, and $\frac{fV|V|}{2D}$ is the frictional force of the fluid. The absolute sign is introduced to ensure that the frictional force will always be opposite to the direction of velocity.

The subscripts x and t indicate partial differentiation with respect to distance and time. For example,

$$H_x = \frac{\partial H}{\partial x},$$

$$H_t = \frac{\partial H}{\partial t},$$ in which H is the total pressure head in feet of water.

Changes in the density of water may be neglected without introducing significant error. Considering the density as constant, the continuity equation may be stated as

$$\frac{a^2}{g} V_x + H_t + V\left[H_x + \sin\Theta\right] = 0, \qquad (2.2)$$

in which Θ is the angle the center line of the pipe makes with the horizontal axis (measured positive downwards), and a is the velocity of the waterhammer wave.

2.3 GENERAL CHARACTERISTICS METHOD

In this section, a general solution for the continuity and momentum equations is presented. For the complete treatment, see Ref. 9. All of the terms in the equations are retained. The method of specified time intervals which involves linear interpolation is used.

The momentum and continuity equations may be written as

$$L_1 = gH_x + VV_x + V_t + \frac{fV|V|}{2D} = 0 \tag{2.3}$$

and

$$L_2 = H_t + \frac{a^2}{g} V_x + VH_x + V \sin\theta = 0 . \tag{2.4}$$

Multiplying Eq. (2.4) by λ and adding it to Eq. (2.3), one obtains

$$L_1 + \lambda L_2 = \lambda \left[H_x(V + \frac{g}{\lambda}) + H_t \right] + \left[V_x(V + \frac{a^2}{g}\lambda) + V_t \right] + \lambda V\sin\theta + \frac{fV|V|}{2D} = 0 . \tag{2.5}$$

Let $$\frac{dx}{dt} = V + \frac{g}{\lambda} = V + \frac{a^2}{g}\lambda . \tag{2.6}$$

Therefore, $$\lambda = \pm \frac{g}{a} , \tag{2.7}$$

and $$\frac{dx}{dt} = V \pm a . \tag{2.8}$$

Through substitution of Equations (2.6), (2.7), and (2.8), Eq. (2.5) takes the form

$$\lambda \frac{dH}{dt} + \frac{dv}{dt} + \lambda V\sin\theta + \frac{fV|V|}{2D} = 0 . \tag{2.9}$$

It follows froms Eqs. (2.8) and (2.9) that

$$\left.\begin{aligned} &\frac{g}{a}\frac{dH}{dt} + \frac{dV}{dt} + \frac{g}{a} V\sin\theta + \frac{fV|V|}{2D} = 0, && (2.10)\\ &\frac{dx}{dt} = V + a, && (2.11) \end{aligned}\right\} C+$$

$$\left.\begin{aligned} -\frac{g}{a}\frac{dH}{dt} + \frac{dV}{dt} - \frac{g}{a} V\sin\theta + \frac{fV|V|}{2D} &= 0, && (2.12) \\ \text{and} \qquad \frac{dx}{dt} &= V - a\ . && (2.13) \end{aligned}\right\} C-$$

Because $V = V(x,t)$, the characteristic lines C+ and C-, given by Eqs. (2.11) and (2.13), plot as curves on the x-t plane (see Fig. 2.1).

Eqs. (2.10) to (2.13) can be written in the following finite-difference forms:

$$(V_P - V_R) + \frac{g}{a}(H_P - H_R) + \frac{g}{a} V_R\sin\theta\,(t_P - t_R) + \frac{f}{2D} V_R|V_R|\,(t_P - t_R) = 0 \qquad (2.14)$$

$$(x_P - x_R) = (V_R + a)\ (t_P - t_R) \qquad (2.15)$$

$$(V_P - V_S) - \frac{g}{a}(H_P - H_S) - \frac{g}{a} V_S\sin\theta\,(t_P - t_S) + \frac{f}{2D} V_S|V_S|\,(t_P - t_S) = 0 \qquad (2.16)$$

$$(x_P - x_S) = (V_S - a)\ (t_P - t_S) \qquad (2.17)$$

Two techniques are commonly used for obtaining a numerical solution for the finite-difference equations (2.14) to (2.17). These are:

(1) use of a grid of characteristics,

(2) use of specified time intervals.

In single pipe problems as covered by this study, these techniques are identical[9]. The parameters x_P and t_P are assigned definite values throughout the computation leaving only V_P and H_P as unknowns to be determined. In this study the technique of specified time intervals will be used. Since the conditions at points A, B, and C (Fig. 2.1) are known, the conditions at R and S may be evaluated by linear interpolation.

Thus
$$\frac{x_C - x_R}{x_C - x_A} = \frac{V_C - V_R}{V_C - V_A}$$

But
$$x_P = x_C, \quad \text{and} \quad x_C - x_A = \Delta x.$$

Therefore, the above equation takes the form

$$x_P - x_R = \frac{V_C - V_R}{V_C - V_A} \Delta x . \tag{2.18}$$

Since V_R is much smaller than the waterhammer wave velocity a, V_R may be deleted from Eq. (2.15) without incurring any serious loss of accuracy. By combining the modified Eq. (2.15) with Eq. (2.18), one obtains

$$a \, \Delta t = \frac{V_C - V_R}{V_C - V_A} \Delta x \tag{2.19}$$

The grid mesh ratio, θ', is defined as

$$\theta' = \frac{\Delta t}{\Delta x} ;$$

Therefore,
$$a \, \theta' (V_C - V_A) = V_C - V_R,$$

and
$$V_R = V_C - a\theta' (V_C - V_A) . \tag{2.20}$$

Similarly,

$$H_R = H_C - a\theta' (H_C - H_A), \quad (2.21)$$

$$V_S = V_C - a\theta' (V_C - V_B), \quad (2.22)$$

$$H_S = H_C - a\theta' (H_C - H_B). \quad (2.23)$$

Solve Eqs. (2.14) and (2.16) simultaneously to obtain:

$$V_P = 0.5 \left[V_R + V_S + \frac{g}{a} (H_R - H_S) - \frac{g}{a} \Delta t \sin\theta \, (V_R - V_S) - f \frac{\Delta t}{2D} (V_R|V_R| + V_S|V_S|) \right] \quad (2.24)$$

$$H_P = 0.5 \left[H_R + H_S + \frac{a}{g} (V_R - V_S) - \Delta t \sin\theta \, (V_R + V_S) - \frac{a}{g} \frac{f\Delta t}{2D} (V_R|V_R| - V_S|V_S|) \right]. \quad (2.25)$$

At the boundary points (Fig. 2.2), either Eq. (2.14) or Eq. (2.16) or both are used together with the boundary conditions to solve for V and H. Eqs. (2.14) and (2.16) are termed the negative characteristic equation and the positive characteristic equation respectively and may now be written in the following forms:

The negative characteristic equation is

$$V_P = C_1 + C_2 \, H_P, \quad (2.26)$$

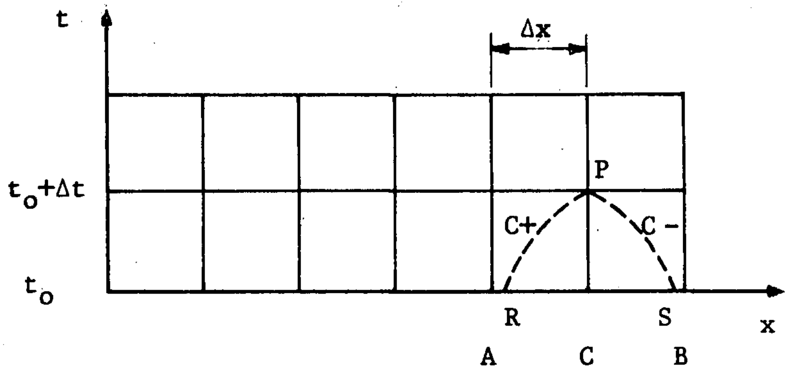
where

$$C_1 = V_S - C_2 H_S + C_2 V_S \sin\theta\Delta t - FF \, V_S| \, V_S| \quad (2.27)$$

$$C_2 = \frac{g}{a}, \quad (2.28)$$

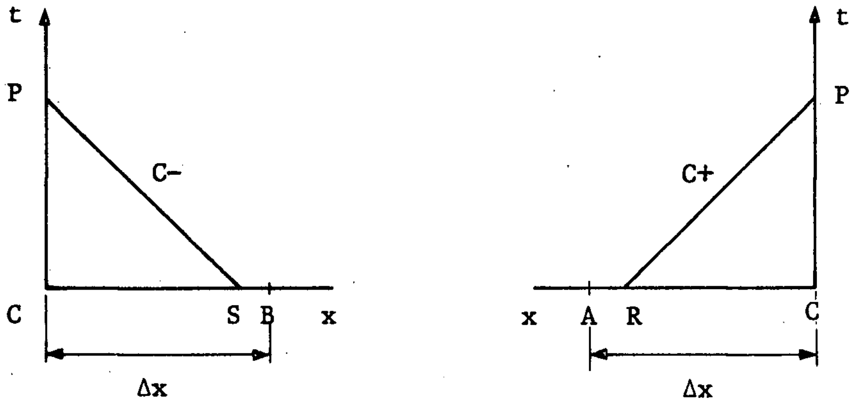
and

$$FF = \frac{f\Delta t}{2D}. \quad (2.29)$$



METHOD OF SPECIFIED TIME INTERVALS

FIG. 2.1



CHARACTERISTICS AT THE BOUNDARIES

FIG. 2.2

The positive characteristic equation is

$$V_P = C_3 - C_2 \; H_P, \tag{2.30}$$

where

$$C_3 = V_R + C_2 \; H_R - C_2 \; V_R \Delta t \sin\theta - FF \; V_R | \; V_R | \; . \tag{2.31}$$

C_2 and FF represent pipe constants. The values of C_1 and C_3 are constant during each time step.

2.4 CONVERGENCE AND STABILITY OF THE METHOD OF FINITE DIFFERENCES

To be assured of stability and/or convergence of the solution[9], it is necessary that $\Delta t \; (V + a) \leqq \Delta x$. Since V is small relative to a, this may be stated as follows:

$$\frac{\Delta t}{\Delta x} \leqq \frac{1}{a} \; .$$

This indicates that it is important to select the grid mesh ratio so that the characteristics through P, C^+ and C^- will not fall outside the line segment AB (Fig. 2.1). The most accurate solutions are obtained[3] when

$$\Delta x = a \Delta t \; .$$

CHAPTER III

BOUNDARY CONDITIONS

3.1 THE AIR CHAMBER (Fig. 3.1)

Because of the assumption that the check valve closes simultaneously with the pump failure, all the flow in the discharge pipe is either from or into the chamber. This assumption eliminates the pump characteristics from the waterhammer computations.

The pressure and volume of air in the chamber follow the gas law[8]

$$H^* \; v_{air}^{\,m} = \text{constant}, \qquad (3.1)$$

where H^* and v_{air} are the absolute pressure head and volume of air in the chamber and m is the power 1.0 for isothermal expansion and 1.4 for adiabatic expansion. The orifice in the chamber may be simple or of the differential type. The differential type of orifice throttles the reverse flow of water from the discharge pipe into the chamber while there is very little throttling of the flow out of the chamber. If there is no orifice in the chamber, the throttling loss is taken equal to zero.

Flow out of the chamber is considered positive.

For the transient condition, Eq. (3.1) may be written:

$$\left[H_P + 34 + H_{orf}\right] \; v_{Pair}^{\,m} = C_{10}, \qquad (3.2)$$

in which H_P is the transient pressure head (in ft) in the pipe at the entrance to the chamber, H_{orf} is the orifice resistance (in ft) corresponding to a discharge of q (ft^3/sec.) and v_{Pair} is the transient volume of air in the chamber (ft^3). C_{10} is a constant given by:

$$C_{10} = H_o^* \quad v_{oair}^{\quad m}, \tag{3.3}$$

in which H_o^* and v_{oair} denote the initial steady-state absolute pressure head and volume of air in the chamber.

For the transient state conditions at the junction of the chamber and the discharge pipe, the following equations can be written:

The continuity equation:

$$VA \, \Delta \, t = v_{Pair} - v_{air} \tag{3.4}$$

where V is the velocity of flow in the pipe (in ft/sec.), A is the cross-sectional area of the pipe (in ft^2), Δt is the length of the time interval under consideration (in secs), v_{Pair} is the volume of air in the chamber (in ft^3) at the end of the time interval and v_{air} is the volume of air in the chamber at the beginning of the time interval.

Rearranging the terms, one gets:

$$v_{Pair} = v_{air} + C_{11} \, \Delta t, \tag{3.5}$$

in which

$$C_{11} = VA.$$

The negative characteristic equation for the pipe is:

$$V_P(1) = C_1 + C_2 \, H_P(1), \tag{3.6}$$

where (1) designates section 1 on the pipe, i.e. at the air chamber. The orifice friction loss is given by:

$$H_{orf} = C_{orf} \frac{H_{orfo}}{q_o^{\,2}} \, q| \, q| \tag{3.7}$$

in which C_{orf} is the orifice coefficient and H_{orfo} is the head loss in the orifice (in ft) corresponding to a discharge of q_o. The absolute value of q ensures the correct sign on the head loss for changes in direction of flow through the orifice. For a simple orifice, C_{orf} = 1.0 for flow in either direction. For a differential orifice, C_{orf} = 1.0 when water flows out of the chamber, i.e. when V is positive, and $C_{orf} = k_1$ when water flows into the chamber, i.e. when V is negative. The value of k_1 depends on the amount of throttling provided by the orifice.

Substituting for q in Eq. (3.7), one obtains:

$$H_{orf} = C_{orf} \frac{H_{orfo}}{q_o^2} VA|\ VA|$$

or

$$H_{orf} = C_{orf}\ C_f\ C_{11}|\ C_{11}| \tag{3.8}$$

in which

$$C_f = \frac{H_{orfo}}{q_o^2} .$$

Substitution of the values of v_{Pair} and H_{orf} into Eq. (3.2) gives:

$$H_P + 34 + C_{orf}\ C_f\ C_{11}|\ C_{11}|\ (v_{air} + C_{11}\ \Delta t)^m = C_{10}$$

or

$$H_P = \frac{C_{10}}{(v_{air} + C_{11}\ \Delta t)^m} - 34 - C_{orf}\ C_f\ C_{11}|\ C_{11}| .$$

Letting $C_{air} = v_{air} + C_{11}\ \Delta t$, one obtains:

$$H_P = \frac{C_{10}}{C_{air}{}^m} - 34 - C_{orf}\ C_f\ C_{11}|\ C_{11}| . \tag{3.9}$$

For each time increment, H_P can be determined from Eq. (3.9), V_P from Eq. (3.6) and V_{Pair} from Eq. (3.5).

3.2 RESERVOIR OF CONSTANT WATER LEVEL AT THE DOWNSTREAM END (Fig. 3.2)

At the junction of the pipe and the reservoir,

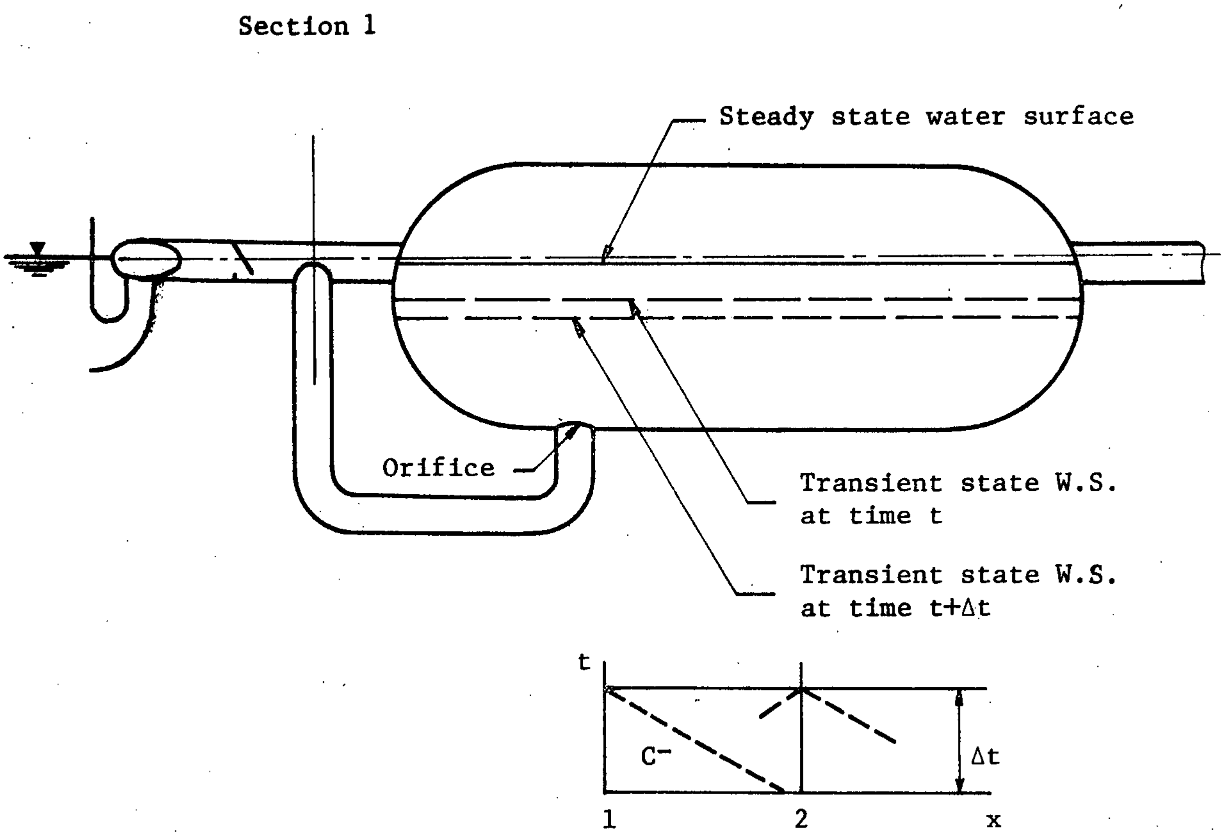
$$H_P(11) = H_{res} .$$

The positive characteristic equation for section 11 is given by:

$$V_P(11) = C_3 - C_2 H_P(11). \tag{3.10}$$

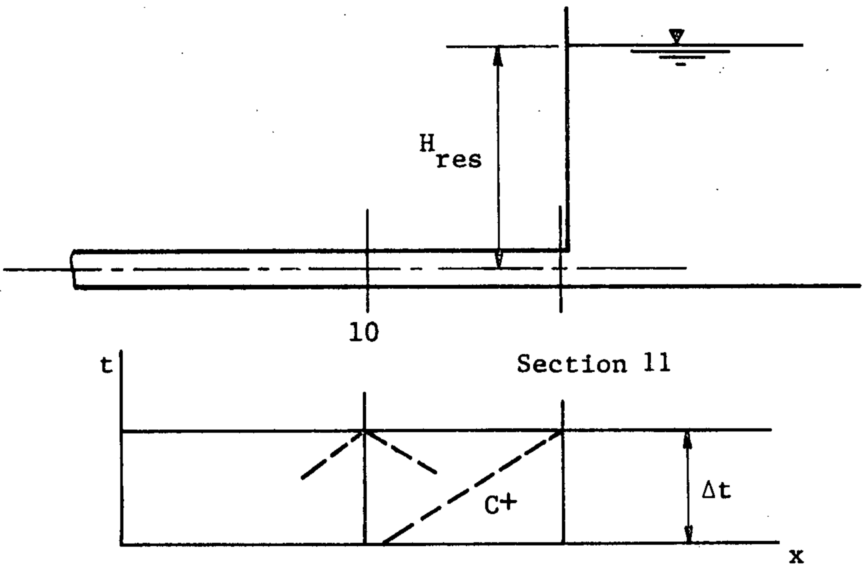
From the above two equations, it follows that:

$$V_P(11) = C_3 - C_2 H_{res} . \tag{3.11}$$



AIR CHAMBER

FIG. 3.1



RESERVOIR AT DOWNSTREAM END

FIG. 3.2

CHAPTER IV

THE PROGRAM

4.1 GENERAL

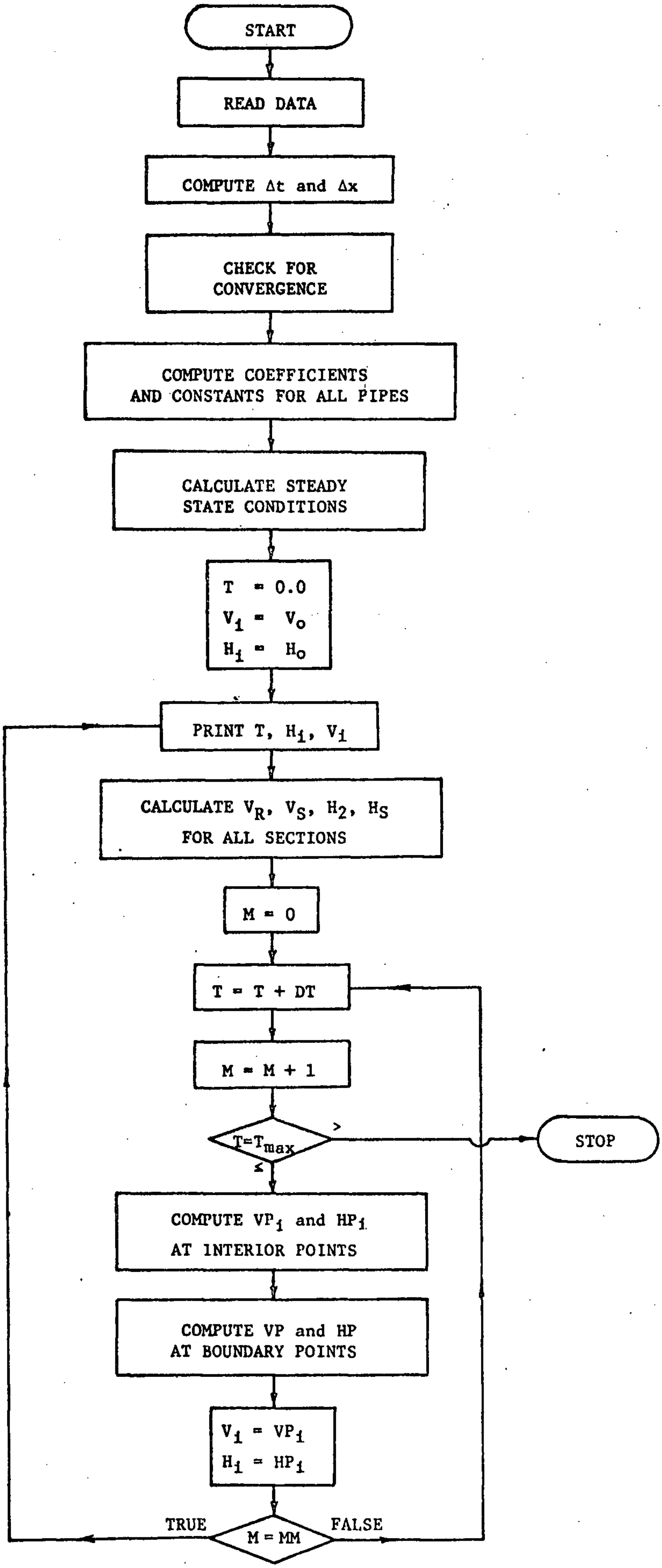
The program for this study designates to the computer all of the operations which must be performed to compute the maximum upsurges and downsurges for the transient phenomena. The flow chart for the program is given in Fig. 4.1 and the entire programs for the entire head loss concentrated at the orifice and entire head loss attributable to distributed friction are reproduced in Appendix C.

4.2 CHECK ON THE PROGRAM

Prior to proceeding with the actual study, the writer checked the validity of the program with several graphical analyses. These checks, presented in Appendix B, indicate that the program gives results which compare well with graphical solutions made by others.

The graphical check for the total head loss concentrated at the orifice[5] shows that the program for this case gives valid results. See Fig. B-1b.

The graphical check using several orifices to approximate head loss due to pipe wall friction[2] (graphical solution by E. Ruus) indicates that the program for distributed friction is also valid. See Fig. B-2b.



PROGRAM FLOW CHART

FIG. 4.1

4.3 DESCRIPTION OF THE PROGRAM

The main functions of the program are as follows:

i) Specification of the storage locations for the subscripted variables, the Dimension statement.

ii) Submission of data to the computer.

iii) Computation of the time increment.

iv) Check for convergence.

v) Computation of constants.

vi) Computation of steady-state values.

vii) Computation of transient-state conditions.

viii) Check for maximum upsurges and downsurges.

ix) Printout.

The variables which the program reads in are:

PLC --- the pipe line constant, $2\rho^*$

TMAX -- the length of time for which the transients are to be calculated, in seconds

CPLAC - the constant for a pipe line with an air chamber adjacent to the pump, $2\ \rho^*\ \sigma^*$.

The remaining parameters are set in the Data statement. For any group, the only parameter which changes in the Data statement is the total head loss coefficient, CK.

The programs are relatively efficient with a typical calculation taking approximately 12 to 13 seconds of computer use time.

The programs for the four basic groups of charts, as listed in Section 5.1, vary only slightly from each other.

Group I - No head loss

For frictionless flow the program Data statement sets CK and the orifice loss equal to zero. The program automatically computes the friction factor, F, to be zero.

Group II - Entire head loss concentrated at the orifice

The Data statement sets the friction factor, F, equal to zero, CK to some value between 0.1 and 1.0, and the orifice inflow coefficient, CORFIN, to 1.0 or 2.5 depending on whether the orifice is simple or differential.

Group III - Entire head loss attributable to distributed wall friction

The Data statement sets the orifice loss, HORF, equal to zero. The program calculates the friction head loss, HF, and the friction factor, F, for the designated values of CK.

Group IV - Head loss equally divided between uniformly distributed friction and orifice loss

The Data statement sets the orifice inflow (CORFIN) and the total head loss (CK) coefficients. The program computes the friction factor, F, the total steady-state friction loss and the total orifice loss for a flow of Q_o into the chamber.

The steady-state friction factor is used to calculate the friction head loss during the transient phase.

4.4 APPROXIMATION OF VELOCITY OF FLOW OUT OF THE CHAMBER

Initially, the average velocity out of the chamber, VAVAPP, after the time interval was incremented, was set equal to the velocity in the pipe at Section (1) (Fig. 3.1) for the previous time interval. The computation was then followed through to the point where the actual average velocity of flow from the chamber was calculated.

i.e.

$$VAV = \frac{V(1) + VP(1)}{2}$$

If the difference between the initial assumed average velocity, VAVAPP, and the calculated average velocity, VAV, was less than or equal to 0.0001, the program continued the transient state computation. If the difference was greater than 0.0001, the values of HP(1) and VP(1) were recalculated using VAV as the new approximation for the velocity of flow out of the chamber. This iteration continued until the error criterion was met.

The writer found that if VAVAPP was set equal to V(1) from the previous time interval, the program would not converge to a solution, but in fact, the pressure surges would magnify increasingly causing the computer to terminate the program with an error message.

CHAPTER V

THE CHARTS

5.1 GROUPS OF CHARTS

Four basic combinations of conditions were investigated in this study. These four combinations include:

(1) No head loss, K = 0.0, (no wall friction, no orifice loss)

There is only one chart in this group.

(2) Entire head loss concentrated at the orifice, (no wall friction)

There are ten charts in this group with K varying from 0.1 to 1.0 in increments of 0.1. Two orifices, one differential and one simple, were investigated in this group. The differential orifice had an inflow to outflow head loss ratio of 2.5:1. That is, for the simple orifice, the orifice resistance is the same for inflow or outflow whereas for the differential orifice the inflow resistance is 2.5 times the outflow resistance.

Note that values of K = 0.7 to 1.0 are not practical but are included for the sake of completeness. Because of the great resistance to flow from the chamber for K = 0.7 to 1.0, large air chambers are needed to control the downsurges whereas the upsurges are not greatly reduced.

(3) Entire head loss attributable to distributed friction, (no orifice loss)

K varies from 0.1 to 1.0 in increments of 0.1.

(4) Head loss equally divided between uniformly distributed wall friction and orifice loss

K varies from 0.1 to 1.0 in increments of 0.1. The orifice considered was a differential orifice with inflow to outflow loss ratio of 2.5:1.

Under the conditions imposed by the assumptions, the entire transient following power interruption is completely described by the variables K, $2\rho^*$ and $2\rho^* \sigma^*$. In the charts, the maximum upsurges and downsurges have been plotted in terms of these variables. Maximum upsurges and downsurges at the pump, the midlength and the three-quarter point of the discharge line are plotted as percentages of H_o^* for various values of these parameters. The normal range[1] of ρ^* is from 0.25 to 2.0 and that of σ^* is from 2 to 30. This range is covered in the charts.

To use the individual charts, one must first determine the parameters K, $2\rho^*$ and $2\rho^* \sigma^*$ for the particular problem. With these known, one determines maximum upsurge by going upwards on the $2\rho^* \sigma^*$ ordinate from the zero surge abscissa to the intersection with the $2\rho^*$ curve. Similarly, the maximum downsurge is found by going downwards on the $2\rho^* \sigma^*$ ordinate from the zero surge abscissa to the $2\rho^*$ curve.

To illustrate:

Known: $K = 0.1$, $2\rho^* \sigma^* = 10$, $2\rho^* = 4$

No wall friction, Differential orifice 2.5:1

Required: Maximum upsurge and downsurge at midpoint.

Solution: Maximum upsurge = $0.771\ H_o^*$

Maximum downsurge = $0.358\ H_o^*$

5.2 NO HEAD LOSS, FRICTIONLESS FLOW

The single chart in this category compares well with the chart for frictionless flow published by Evans and Crawford (Appendix A, Fig. A-1). Since frictionless flow would not occur in reality, this chart would be used for purposes of analysis but not for design problems.

5.3 ENTIRE HEAD LOSS CONCENTRATED AT THE ORIFICE

Differential orifice - inflow to outflow head loss ratio 2.5:1

The graphs for K = 0.3, 0.5 and 0.7 compare well with the corresponding graphs published by Evans and Crawford as shown in Appendix A, Figures A-2, A-3, and A-4. The curves are generally well defined except for the lower values of $2\rho^*$ and $2\rho^* \sigma^*$ for the upsurge region. This is in the range of very low velocities. The $2\rho^* = 0.5$ curves were eliminated for K = 0.8 to K = 1.0 inclusive because the program would not converge to a solution.

Two additional charts for K = 0.5 were included in this group. These were for powers of 1.0 and 1.4, the powers being the values of m in the equation $H^* v_{air}^{m}$ = constant. The intent was to check the possible variation of results caused by using the power m as 1.0, 1.2 and 1.4.

A comparison of the charts and a partial listing of the results as shown in Table 5.1 indicate that the power 1.2 gives an approximate average for the upsurges and downsurges. The charts also indicate that one must accurately determine whether the system is isothermal or adiabatic when using the powers 1.0 and 1.4 because the resultant

TABLE 5.1

Comparison of Results Obtained for the Powers 1.0, 1.2 and 1.4

2ρ*	2ρ*σ*	Point	m = 1.0		m = 1.2		m = 1.4	
			U_p	D_n	U_p	D_n	U_p	D_n
1	2	P	.705	.572	.732	.615	.793	.649
		M	.435	.458	.527	.498	.669	.532
		3/4	.235	.342	.290	.372	.343	.399
	4	P	.413	.452	.475	.499	.542	.532
		M	.254	.355	.313	.386	.331	.414
		3/4	.132	.264	.151	.283	.178	.302
	10	P	.173	.324	.208	.352	.240	.378
		M	.120	.250	.134	.270	.157	.287
		3/4	.058	.200	.065	.210	.073	.219
	30	P	.061	.220	.073	.234	.085	.247
		M	.050	.185	.056	.194	.063	.201
		3/4	.022	.165	.024	.169	.028	.172
4	8	P	.782	.535	.902	.583	1.012	.623
		M	.435	.375	.504	.409	.575	.439
		3/4	.211	.272	.249	.290	.278	.308
	20	P	.322	.385	.375	.421	.427	.454
		M	.191	.270	.220	.290	.248	.310
		3/4	.089	.201	.104	.227	.118	.235
	40	P	.169	.286	.198	.313	.227	.339
		M	.102	.222	.121	.232	.137	.243
		3/4	.049	.201	.056	.205	.064	.209
	80	P	.090	.225	.105	.234	.121	.249
		M	.056	.204	.065	.208	.075	.212
		3 4	.025	.192	.031	.194	.035	.196

upsurges vary by as much as 50% and the downsurges vary by as much as 40%. The greater variation occurs generally for small $2\rho^* \sigma^*$ values.

Simple orifice - inflow to outflow head loss ratio 1:1

The curves in this group are well defined except for some scatter in the range of low $2\rho^*$ and $2\rho^* \sigma^*$ values for upsurge only. The $2\rho^*$ = 0.5 curves were eliminated for the range K = 0.7 to K = 1.0 inclusive because the program would not converge to a solution. Note that for the higher values of K, $2\rho^*$, and $2\rho^* \sigma^*$, the upsurges at the mid-point of the line become higher than the upsurges at the pump.

Comparison of upsurges and downsurges for 2.5:1 and 1:1 orifices

The friction factor, K, is based on inflow to the air chamber. To compare the upsurges and downsurges for the two orifices, one differential with a 2.5:1 inflow to outflow head loss ratio and the other simple, assume that the inflow losses are equal. Therefore, the outflow loss for the simple orifice will be 2.5 times greater than the outflow loss for the differential orifice. It follows that the downsurges will be equal for the following friction factors:

(1) Differential, K = 0.5; Simple, K = 0.2; and

(2) Differential, K = 1.0; Simple, K = 0.4.

Table 5.2 does in fact verify this, except for isolated instances.

5.4 ENTIRE HEAD LOSS ATTRIBUTABLE TO DISTRIBUTED FRICTION

As the total head loss increases, the distributed friction significantly reduces the upsurges, and, to a lesser extent, the downsurges. The downsurges are affected to a greater degree away from the

TABLE 5.2

UPSURGES AND DOWNSURGES FOR 1:1 AND 2.5:1 ORIFICES

		1 : 1												2.5 : 1											
		K = 0.2						K = 0.4						K = 0.5						K = 1.0					
		UP			DN			UP			DN			UP			DN			UP			DN		
2ρ*	2ρ*σ*	P	M	3/4	P	M	3/4	P	M	3/4	P	M	3/4	P	M	3/4	P	M	3/4	P	M	3/4	P	M	3/4
0.5	1	.688	.587	.330	.505	.442	.360	.608	.568	.271	.487	.437	.360	.593	.528	.250	.486	.442	.360						
	2	.561	.314	.184	.442	.371	.284	.488	.326	.153	.437	.370	.296	.478	.308	.146	.442	.371	.284						
	3	.460	.311	.198	.403	.325	.244	.376	.258	.165	.399	.330	.264	.365	.243	.147	.403	.325	.244						
	4	.377	.320	.162	.371	.293	.220	.298	.254	.128	.369	.303	.246	.289	.217	.110	.371	.293	.220						
	6	.295	.213	.093	.325	.252	.192	.224	.182	.083	.329	.270	.224	.215	.145	.071	.325	.252	.192						
	8	.236	.160	.080	.293	.227	.175	.177	.141	.061	.302	.251	.213	.162	.111	.052	.293	.227	.175						
	10	.193	.148	.067	.269	.209	.165	.139	.116	.050	.284	.244	.206	.127	.092	.040	.269	.209	.165						
0.5	15	.137	.103	.048	.231	.183	.150	.092	.085	.035	.254	.241	.202	.083	.064	.027	.231	.183	.150						
1.0	2	.834	.710	.405	.614	.498	.372	.725	.641	.314	.625	.522	.415	.729	.528	.290	.614	.498	.372	.571	.398	.183	.625	.522	.415
	3	.787	.459	.243	.546	.431	.317	.622	.359	.184	.564	.463	.369	.607	.349	.188	.546	.431	.317	.400	.277	.131	.564	.463	.369
	4	.651	.456	.233	.497	.385	.283	.494	.362	.177	.521	.425	.342	.476	.323	.151	.497	.385	.283	.299	.202	.100	.521	.425	.342
	6	.489	.276	.142	.430	.328	.245	.349	.247	.107	.462	.378	.311	.336	.199	.101	.430	.328	.245	.203	.149	.061	.462	.378	.311
	10	.325	.213	.099	.352	.270	.210	.221	.174	.076	.397	.330	.283	.208	.134	.065	.352	.270	.210	.121	.099	.041	.397	.330	.283
	15	.232	.164	.073	.300	.234	.190	.153	.131	.058	.354	.302	.269	.141	.096	.045	.300	.234	.190	.081	.087	.032	.354	.302	.269
	20	.181	.135	.057	.269	.215	.180	.116	.105	.046	.329	.298	.266	.108	.076	.035	.269	.215	.180	.063	.079	.026	.329	.298	.266
1.0	30	.128	.101	.045	.234	.194	.169	.081	.081	.033	.302	.295	.262	.073	.056	.024	.234	.194	.169	.043	.064	.022	.302	.295	.262
2.0	4	1.191	.701	.365	.599	.461	.335	.887	.510	.263	.665	.543	.434	.878	.470	.250	.599	.461	.335	.551	.338	.171	.665	.543	.434
	6	.867	.502	.257	.516	.391	.288	.621	.417	.184	.591	.481	.393	.611	.327	.166	.516	.391	.288	.369	.235	.113	.591	.481	.393
	10	.572	.353	.174	.429	.319	.245	.393	.278	.126	.506	.419	.356	.379	.235	.109	.429	.319	.245	.222	.151	.070	.506	.419	.356
	15	.404	.259	.123	.364	.275	.221	.271	.208	.092	.451	.382	.336	.261	.162	.077	.364	.275	.221	.150	.114	.048	.451	.382	.336
	20	.318	.206	.097	.322	.251	.208	.210	.166	.073	.418	.362	.326	.200	.128	.060	.322	.251	.208	.113	.090	.036	.418	.362	.326
	30	.227	.155	.071	.275	.226	.195	.146	.122	.054	.381	.340	.318	.137	.091	.043	.275	.226	.195	.077	.073	.027	.381	.340	.318
	40	.178	.126	.057	.251	.212	.189	.112	.098	.043	.361	.338	.316	.105	.070	.033	.251	.212	.189	.058	.065	.023	.361	.338	.316
2.0	60	.126	.093	.040	.225	.198	.182	.078	.074	.031	.340	.337	.314	.072	.050	.023	.225	.198	.182	.039	.059	.019	.340	.337	.314
4.0	8	1.303	.718	.363	.583	.409	.290	.914	.557	.260	.636	.519	.430	.902	.504	.249	.583	.409	.290	.541	.331	.158	.636	.519	.430
	10	1.061	.581	.293	.543	.376	.270	.737	.466	.218	.594	.489	.413	.725	.409	.199	.543	.376	.270	.432	.269	.128	.594	.489	.413
	15	.737	.413	.204	.470	.323	.241	.504	.332	.153	.528	.445	.389	.491	.286	.136	.470	.323	.241	.289	.185	.087	.528	.445	.389
	20	.571	.335	.165	.421	.290	.227	.387	.365	.120	.488	.420	.376	.375	.220	.104	.421	.290	.227	.219	.142	.067	.488	.420	.376
	30	.404	.244	.119	.355	.253	.213	.268	.191	.084	.444	.394	.363	.258	.155	.072	.355	.253	.213	.148	.100	.046	.444	.394	.363
	40	.317	.194	.095	.313	.232	.205	.207	.152	.066	.420	.380	.357	.198	.121	.056	.313	.232	.205	.112	.079	.035	.420	.380	.357
	80	.177	.113	.055	.234	.208	.194	.112	.086	.038	.380	.365	.351	.115	.065	.031	.234	.208	.194	.058	.052	.019	.380	.365	.351
	100	.147	.095	.045	.222	.203	.192	.092	.072	.031	.371	.364	.350	.086	.053	.025	.222	.203	.192	.046	.048	.017	.371	.364	.350

pump. This is logical because the distributed friction is in effect over a longer distance. For K = 0.4 and above, upsurges have been eliminated while the downsurges are divided into three distinct groups, at the pump, the mid-point, and the three-quarter point. For K values above 0.6, the downsurges for the various values of $2\rho^*$ in each group become so closely spaced as to almost merge.

5.5 HEAD LOSS EQUALLY DIVIDED BETWEEN UNIFORMLY DISTRIBUTED WALL FRICTION AND ORIFICE LOSS

For K values of 0.7 to 1.0 inclusive, the upsurges disappear completely and the downsurges are segregated into three distinct groups, i.e. at the pump, the mid-point, and the three-quarter point.

5.6 USE OF THE CHARTS

The downsurge charts produced by Evans and Crawford are based on the minimum head in the pipeline. For this reason they stated that the charts were for preliminary design purposes only. Since this program was derived to give the actual absolute pressure in the air chamber, the charts can be used for final design as well as preliminary design and checking purposes.

Usually when an air chamber is being designed for a pump - discharge line, the values of L, a, V_o, Q_o, A, H_o^* and g will be known. From these values, $2\rho^*$ can be computed. The allowable maximum surge values may be dictated by specifications, operating conditions, or the profile of the discharge line. For the computed value of $2\rho^*$ and the specified maximum allowable surges, values of K and $2\rho^* \sigma^*$ can be chosen from the charts such that the surge limitations are met. If the allowable surge conditions can not be satisfied by data from the charts, probably some means other than an air chamber should be used to control the surges.

When $2\rho^* \sigma^*$ has been determined, C_o can be computed from Eq. (1.5) (i.e.):

$$C_o = \rho^* \sigma^* Q_o \frac{L}{a} .$$

Numerical examples demonstrating the use of the charts are given in Appendix A.

Figure 5.1 shows the configuration of the pump, air chamber, pipeline and reservoir.

The Charts begin on page 45.

PIPELINE WITH AIR CHAMBER

FIG. 5.1

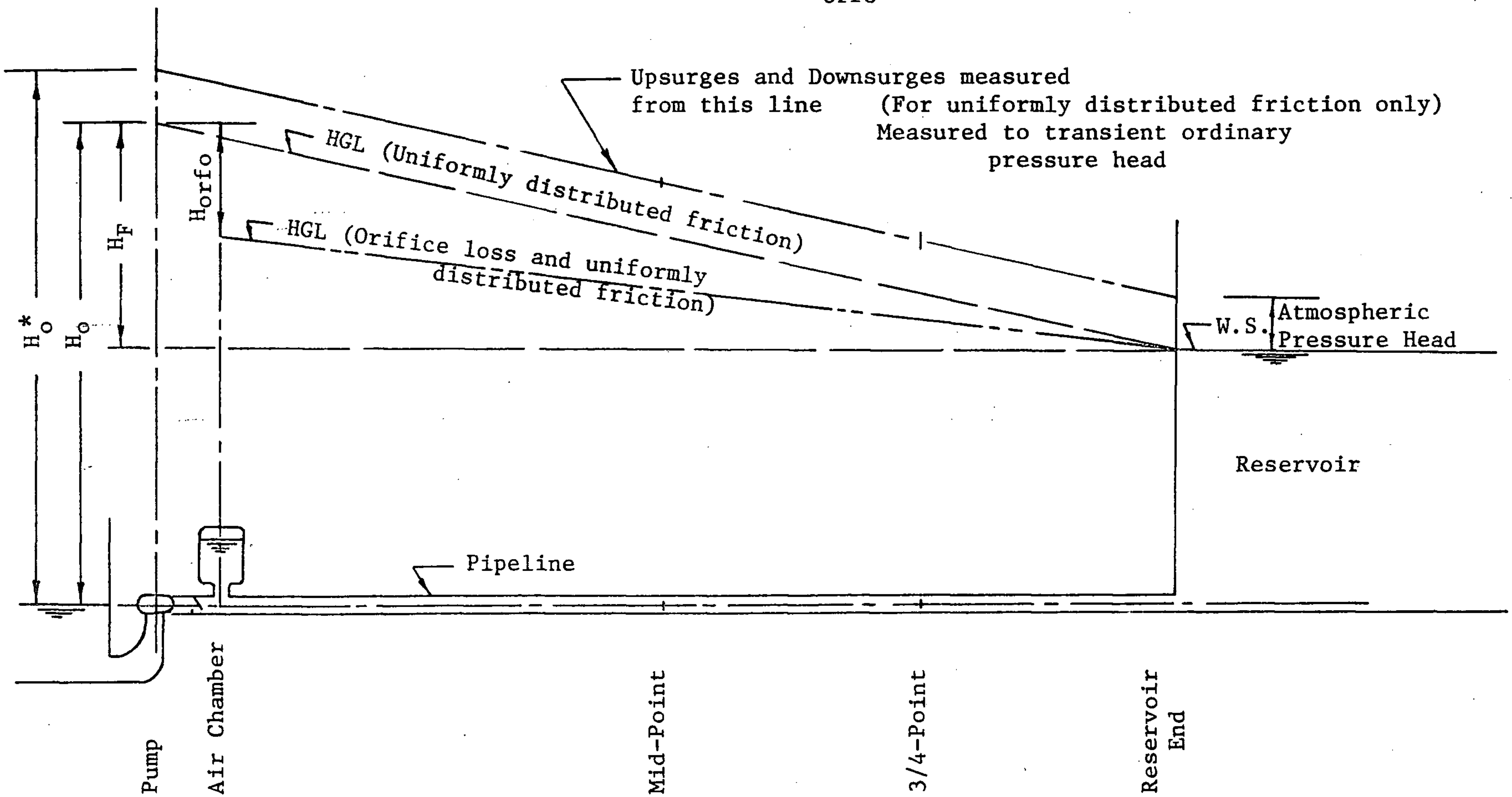
Pipe Wall Friction

H_o^* - Steady-state absolute pressure head at pump

H_o - Steady-state pressure head at pump

H_F - Total head loss

H_{orfo} - Head loss due to orifice resistance



CHAPTER VI

DISCUSSION

6.1 VOLUME OF AIR IN THE CHAMBER

Since σ^*, the parameter pertaining to a pump discharge line having an air chamber, is directly proportional to C_o, the initial volume of air in the chamber, the initial volumes of air and water in the tank must be maintained within certain limits to ensure proper operation of the chamber. The compressed air which dissolves in the water or is lost through leakage must be continually replaced. Some means of automatic shut down of the pump or pumps must be provided should the proper water level in the tanks not be maintained. The minimum controls required are shown schematically in Fig. 6.1.

The following items[4] should be considered when fixing the compressor "on" and "off" levels:

(a) capacity of the compressor,

(b) size of the air chamber,

(c) frequency of starting and stopping of the compressor,

(d) daily temperature variations that might actuate the controls,

and (e) how quickly the system is to be put back into operation after a prolonged shutdown.

The emergency levels can be at nominal distances above and below the compressor operating levels on installations having only one pump or that provide manual starting or stopping for individual pumps on the same line. If automatic starting and stopping of the individual pumps on the same line are required, the emergency levels should be sufficiently removed from the compressor operating levels to contain the surges produced by starting or stopping the largest of the pumps under the most critical initial conditions.

The charts can be usefully employed to check the locations of the emergency levels.

6.2 TOTAL VOLUME OF THE AIR CHAMBER

Once $2\rho^* \sigma^*$ has been determined from the charts, C_o can be calculated by using Eq. 1.8. The volume of the air chamber is then determined by considering that the chamber must contain adequate air above the upper emergency level to control the surges to desirable limits, and enough water below the lower emergency level to prevent unwatering. With allowance for the volume between the upper and lower emergency levels, the total required volume of the air chamber can be computed.

The minimum volume of air that must be maintained in the chamber to control the pressure surges is the volume of the chamber above the upper emergency level. This volume can be designated C' which is numerically equal to the volume C_o. By adding to this quantity the volume of the chamber between the upper and lower emergency levels, one determines the initial volume of air in the chamber that will result in the lowest water-surface level following pump shut down. This new volume of air becomes C'' equal to C' plus the volume of air between the upper and lower emergency levels.

The downsurge at the pump with this initial volume of air can be determined from the curves by computing a new value of $2\rho^* \sigma^*$ based on C" instead of C'. Assuming that this expansion is isothermal[4], the total volume of the air chamber becomes

$$\frac{C'' H_o^*}{H_o^* - \text{downsurge at pump}}$$

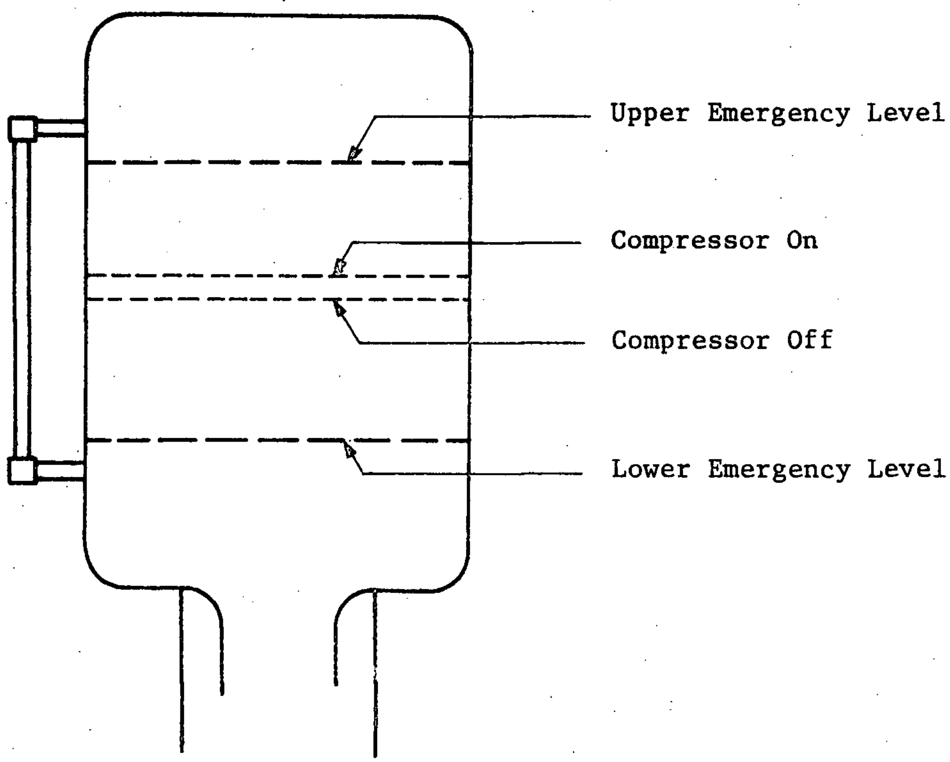
Under favorable conditions, the air tank volume is about one to two percent of the conduit volume. A conservative approximation of tank size would be two to four percent of the conduit volume. Favourable conditions would be interpreted as long pipe lines with high friction losses and no high points of topography.

The initial air volume is generally about 40 percent of the tank volume.

6.3 ORIFICE DESIGN

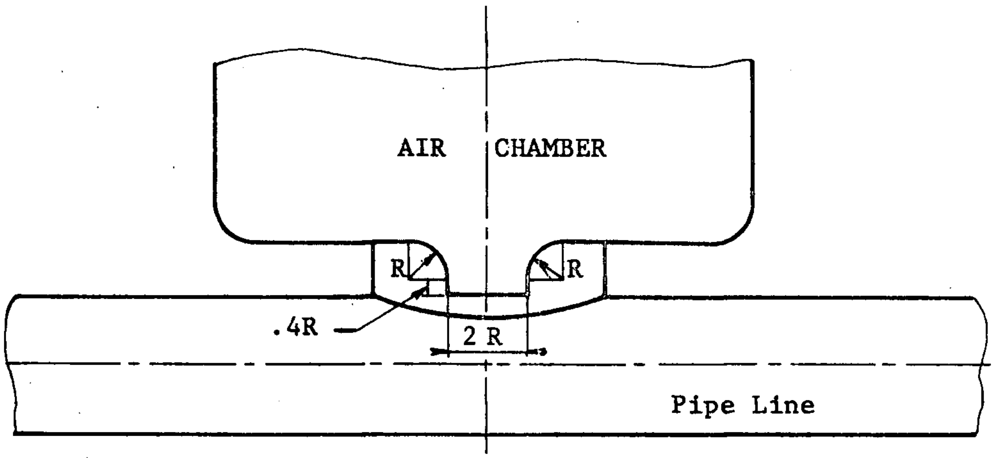
Since the function of an air chamber is to decrease both the upsurges and the downsurges on pump failure, it is necessary to throttle the reverse flow of water from the discharge line into the chamber while providing little throttling for flow out of the chamber.

An effective device for producing a high head loss for inflow while keeping the exit head loss at a minimum is a differential orifice as shown in Fig. 6.2. The design is essentially a bellmouth for flow from the chamber and a re-entrant tube for flow into the chamber. This design will give discharge coefficients of 1.0 and 0.5 for outflow and inflow respectively. The inflow head loss for a specified rate of flow would be approximately four times as great as the outflow headloss. However, this head loss ratio of 4:1 is difficult to obtain in practice.



AIR CHAMBER CONTROL LEVELS

FIG. 6.1



DIFFERENTIAL ORIFICE

FIG. 6.2

A ratio of 2.5:1 is more realistic.

If head loss in the pipeline due to wall friction is considered as concentrated at the orifice, the orifice design should allow for this assumption. For example, design a differential orifice for an inflow loss of 60 per cent of H_o* and outflow loss of 30 per cent of H_o* for an inflow and outflow of Q_o. For a flow of Q_o, the pipeline surface-friction loss is 10 per cent of H_o*. The orifice should be designed for a head loss of 50 per cent of H_o* for an inflow of Q_o and a head loss of 20 per cent of H_o* for an outflow of Q_o. The actual orifice design, head loss ratio of inward flow to outward flow should be 2.5:1.

An orifice may be designed to give a maximum initial head loss through the orifice equal to the maximum downsurge. This condition is known as normal throttling[6]. Greater or smaller throttling losses may be said to give over-throttling or under-throttling, respectively. The minimum head in the pipe will correspond to the maximum air expansion in the chamber for the conditions of normal throttling and under-throttling. For over-throttling the minimum head in the pipe can not be used to determine the maximum air expansion in the chamber. For the condition of over-throttling, the minimum pressure in the tank must be known in order to determine the maximum air expansion.

Large computational errors result when friction is ignored. The inclusion of distributed wall friction increases the accuracy of the maximum and minimum pressures and corresponding maximum air expansion in the chamber. Thus the charts including distributed wall friction give highly accurate results.

6.4 WATER-COLUMN SEPARATION IN PUMP DISCHARGE LINES

Water - column separation[7] is the first phase in the development of one of the most destructive types of waterhammer surge in pump-discharge pipe lines. Following pump failure, the sudden pressure drop downstream might be severe enough to bring about a temporary vapour pressure condition, and possibly the formation of a void in the pipe line. The subsequent closure of this void often results in violent local surges well above any possible transient pressure rises in a continuous water column. The extent of pressure rise is proportional to the fluid velocity destroyed at the instant of vacuous space closure.

The four major factors[7] influencing water - column separation are:

(1) rate of flow stoppage,

(2) length of system,

(3) normal operating pressure at critical points,

(4) velocity of flow.

(1) For pumps that have small rotational inertias the result is complete pump stoppage from within a fraction of a second to a very few seconds after pump failure. This very much aggravates the downsurge problem.

(2) The length of the system determines the length of time the pressure will continue to fall before positive pressure waves reflected from the far end of the line counteract the pressure drop. A long line with a pump having a small rotational inertia very often will experience water - column separation on pump failure.

(3) Points of low pressure are critical. At points of low pressure such as the crests of hills over which a pipe line passes, a

slight interruption of flow may result in a drop to vapour pressure and resulting column separation.

(4) The fourth major element in water-column separation is the velocity of water in the pipe line preceding the cause of perturbation. As the steady state velocity increases, the size of the vacuous space, the reverse flow velocity, and the final surges following the void collapse all become greater.

All of these elements are inter-related. For example, extensive water-column separation may occur even with a very low velocity if the pipe line is long enough and the steady state pressure head is low.

An air chamber is one means of preventing or controlling water-column separation for medium to high-head systems. An example is given in Appendix A indicating the manner in which the charts can be used to determine the possibility of water-column separation.

Although these charts cannot be used to analyze the water-column separation condition, the high degree of accuracy does enhance the ability of being able to predict if water-column separation will occur.

Further studies could be carried out to attempt to determine maximum and minimum pressures occurring for the water-column separation phase of waterhammer.

CHAPTER VII

CONCLUSIONS

(1) Since the program was evolved from the basic differential equations for momentum and continuity to give the absolute pressure in the air chamber, nonlinear terms are retained and friction is included, the charts can be used for final design purposes*.

(2) The validity of the charts is demonstrated by comparing the results obtained by the method of characteristics with those obtained by the graphical method.

(3) It is important to analyze the system properly and to use the group of charts which most closely approximate the system in order to get valid results. In some cases it might be advantageous to interpolate between graphs within the same group. For example, K might be in the range 0.0 to 0.1.

(4) It is important to determine whether the expansion and compression of air in the chamber is adiabatic or isothermal because the results vary significantly for the powers m = 1.0 and m = 1.4, where m is the power in the equation $H^* v_{air}^m$ = constant. For example, for $2\rho^* = 4$ and $2\rho^* \sigma^* = 8$, the upsurge at the pump for m = 1.0 is 0.782 H^*_o and for m = 1.4 is 1.012 H^*_o, the downsurge at the pump for m = 1.0 is 0.535 H^*_o and for m = 1.4 is 0.623 H^*_o.

---

* The charts produced by Evans and Crawford are to be used only for preliminary design purposes. The authors stress that "for final design of an installation having an air chamber, individual solutions similar to that shown by Mr. Angus ('Air Chambers and Valves in Relation to Water-Hammer', Transactions, ASME, Vol. 59, 1937, p.661) should be made to ensure that the air chamber will fulfill design requirements".

The power m = 1.2 gives an approximate average for the upsurges and downsurges. For the same pipeline constants as above, for m = 1.2, the upsurge at the pump is 0.902 H^*_o and the downsurge at the pump is 0.584 H^*_o.

(5) The charts produced by Evans and Crawford are quite accurate as shown by the computer check on these charts using the method of characteristics. The accuracy of the charts produced in this work is enhanced by the inclusion of friction and nonlinear terms.

The charts presented in this thesis cover a much wider range of variables than those published by Evans and Crawford. For each group the following charts are presented: No Line Friction, Line Friction Only - No Orifice Loss, and Friction Loss Equally Distributed between Orifice Loss and wall Friction, the range of K is from 0.1 to 1.0.

(6) Bergeron's method of graphical analysis considering line friction concentrated at five points is quite accurate as demonstrated by the computer check using the method of characteristics.

(7) The number of sections used in analyzing the pipe system is important. If N is too large, excessive computer time will be required; if N is too small, the program will not converge to a solution. In this program, for instance, N was set equal to ten and gave good results. For N equals five, the program would not always converge to a solution.

(8) For high values of K, $2\rho^*$, and $2\rho^* \sigma^*$ the upsurges at the mid-point can be higher than those at the pump.

BIBLIOGRAPHY

1. Allievi, L., "Air chambers for Discharge Pipes", Trans. ASME, Vol. 59, Paper Hyd-59-7, November 1937, pp. 651-659.

2. Bergeron, L., Water Hammer in Hydraulics and Wave Surges in Electricity, John Wiley and Sons, Inc. New York, Copyright 1961 by the ASME.

3. Chaudhry, M.H., Boundary Conditions for Analysis of Water Hammer in Pipe Systems, A thesis submitted in partial fulfillment of the requirements for the degree of Master of Applied Science at the University of British Columbia, 1968.

4. Evans, W.E., and Crawford, C.C., "Design Charts for Air Chambers on Pump Lines", Trans. ASCE, September 1954, pp. 1025-1036.

5. Parmakian, J., Waterhammer Analysis, Dover Publications, Inc. New York, 1963.

6. Paynter, H.M., Discussion of "Design Charts for Air Chambers on Pump Lines", Trans. ASCE, September 1954, pp. 1039-1045.

7. Richards, R.T., "Water-Column Separation in Pump Discharge Lines", Trans. ASME, Paper No. 55-A-74, 1955, pp. 1297-1304.

8. Ruus, E., and Chaudhry, M.H., "Boundary Conditions for Air Chambers and Surge Tanks", Trans. EIC, November 1969, EIC-69-HYDEL 22, Vol. 12, No. C-6.

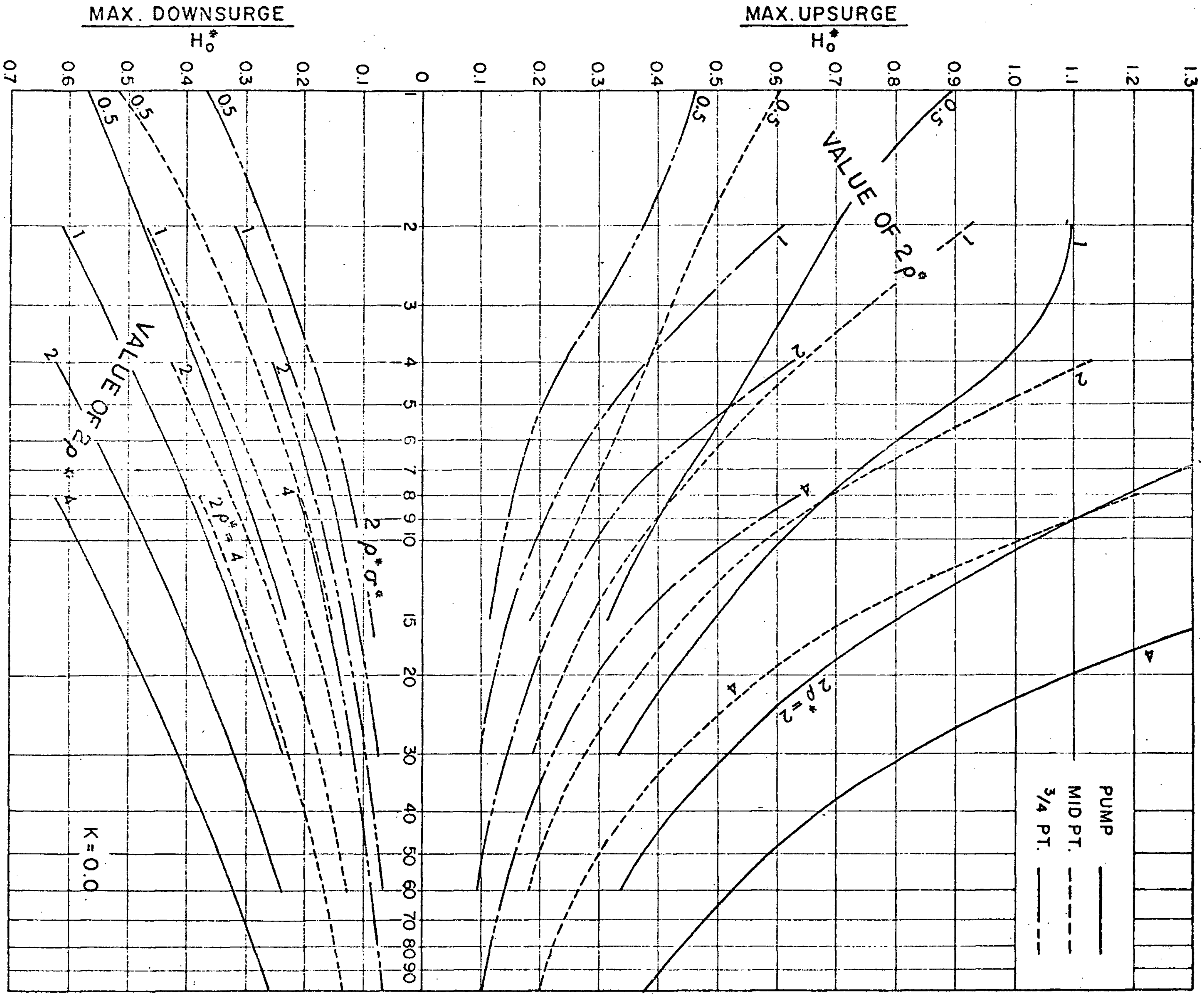
9. Streeter, V.L., and Wylie, E.B., Hydraulic Transients, McGraw-Hill Book Company, New York, 1967.

THE CHARTS

GROUP I

NO HEAD LOSS, FRICTIONLESS FLOW

(No wall friction, no orifice loss)


MAX. DOWNSURGE
H*o
MAX. UPSURGE
H*o
VALUE OF 2ρ*
2ρ*σ*
PUMP
MID PT.
3/4 PT.
K=0.0

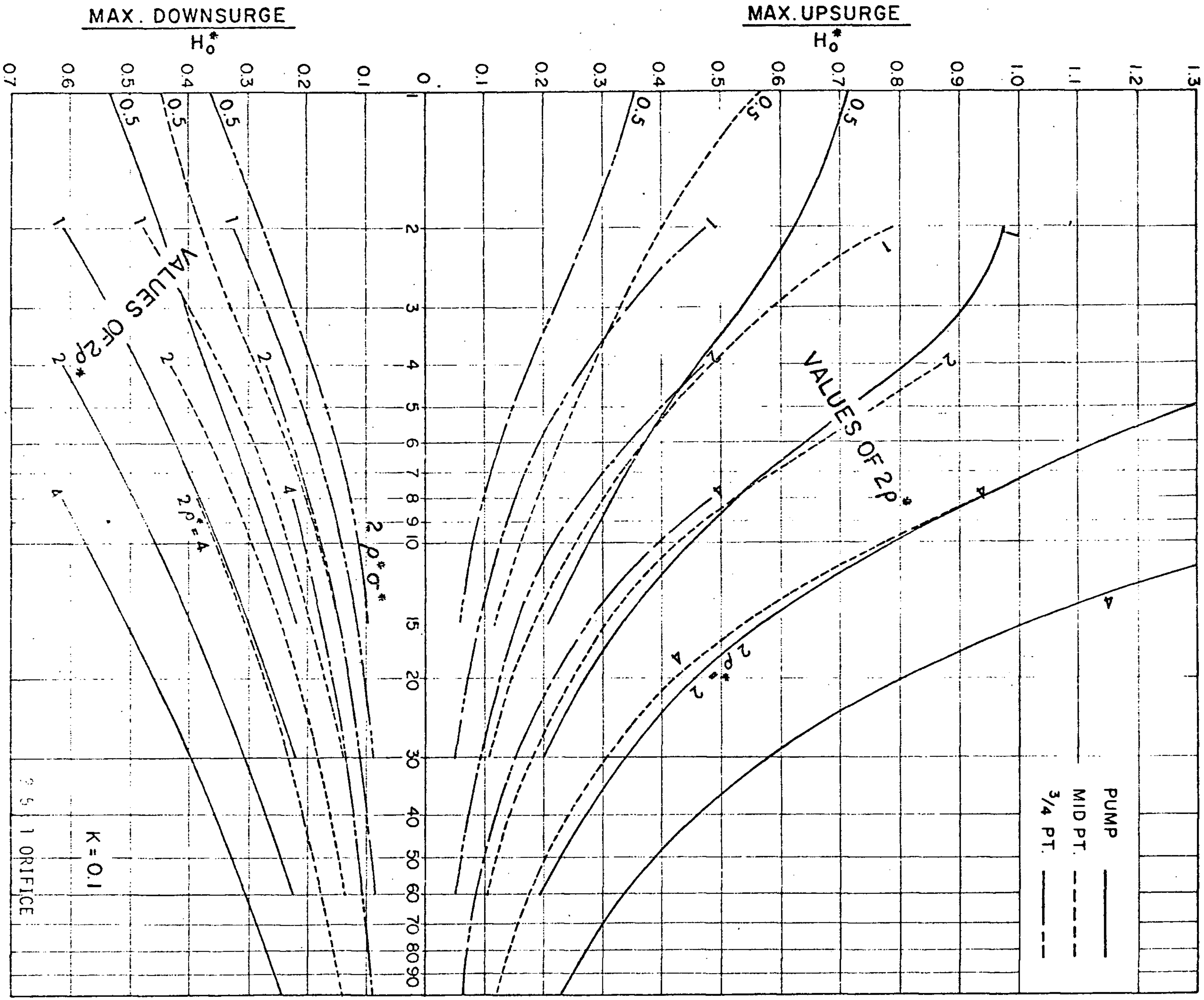
GROUP II

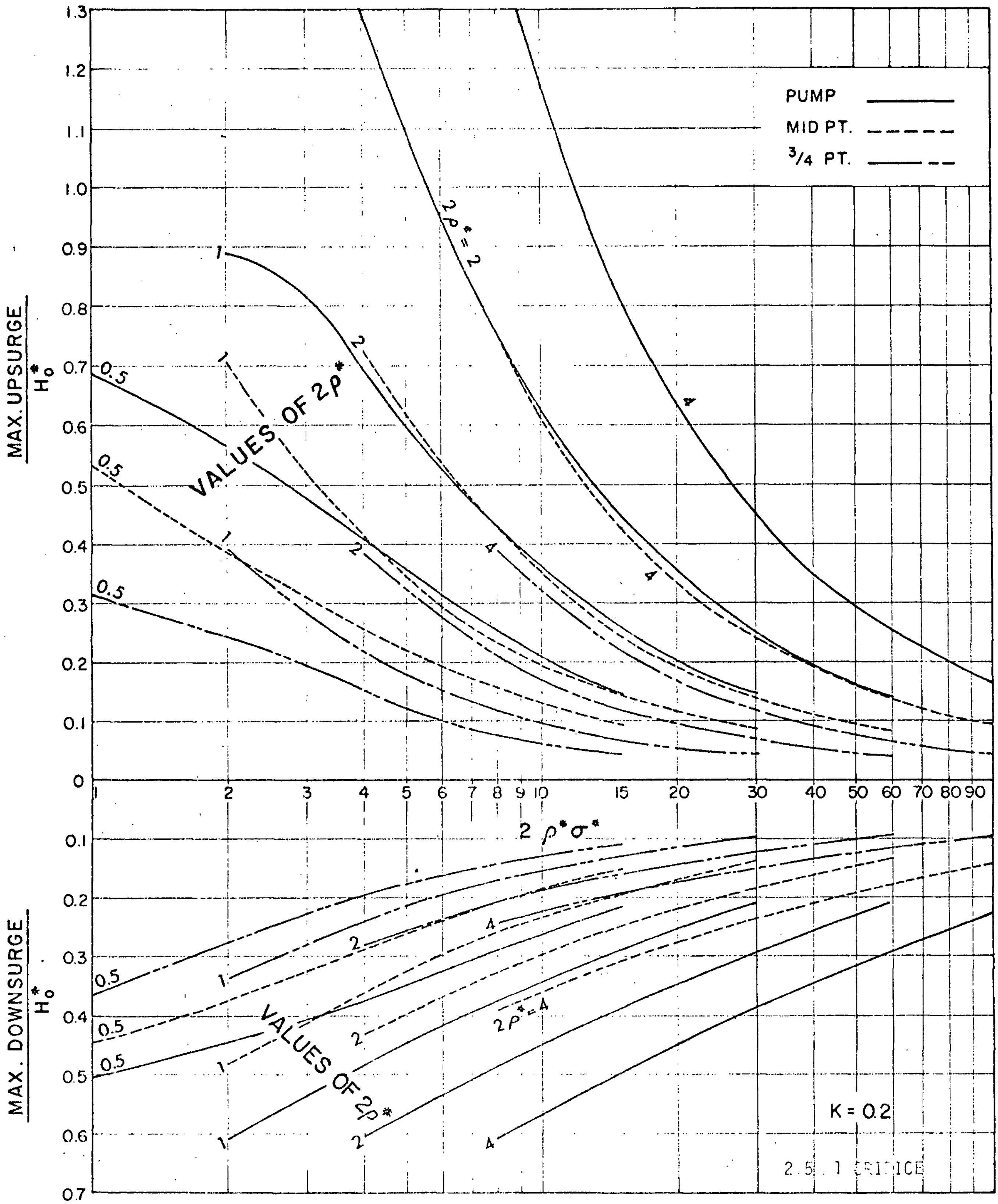
ENTIRE HEAD LOSS CONCENTRATED AT THE ORIFICE

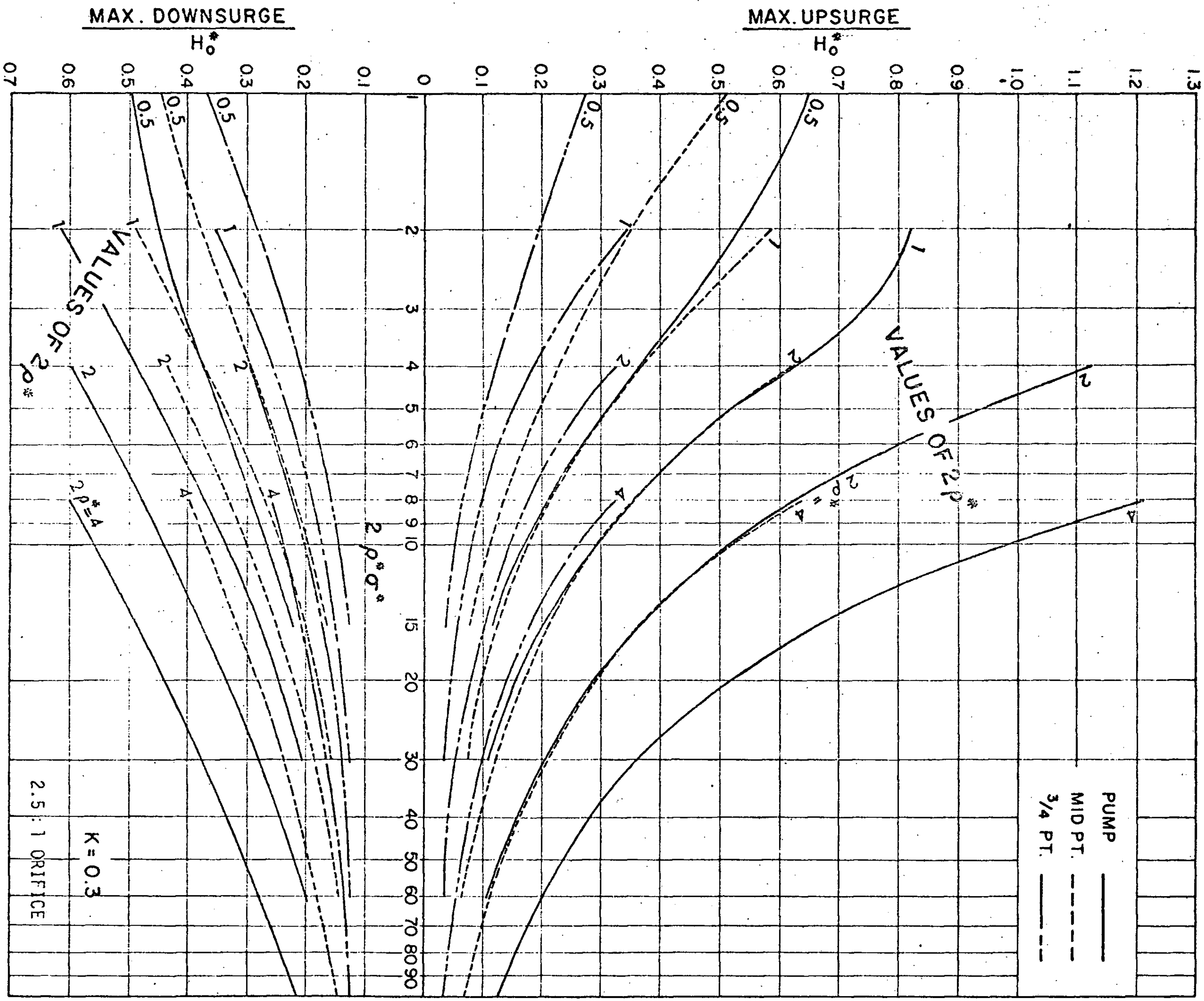
(no wall friction)

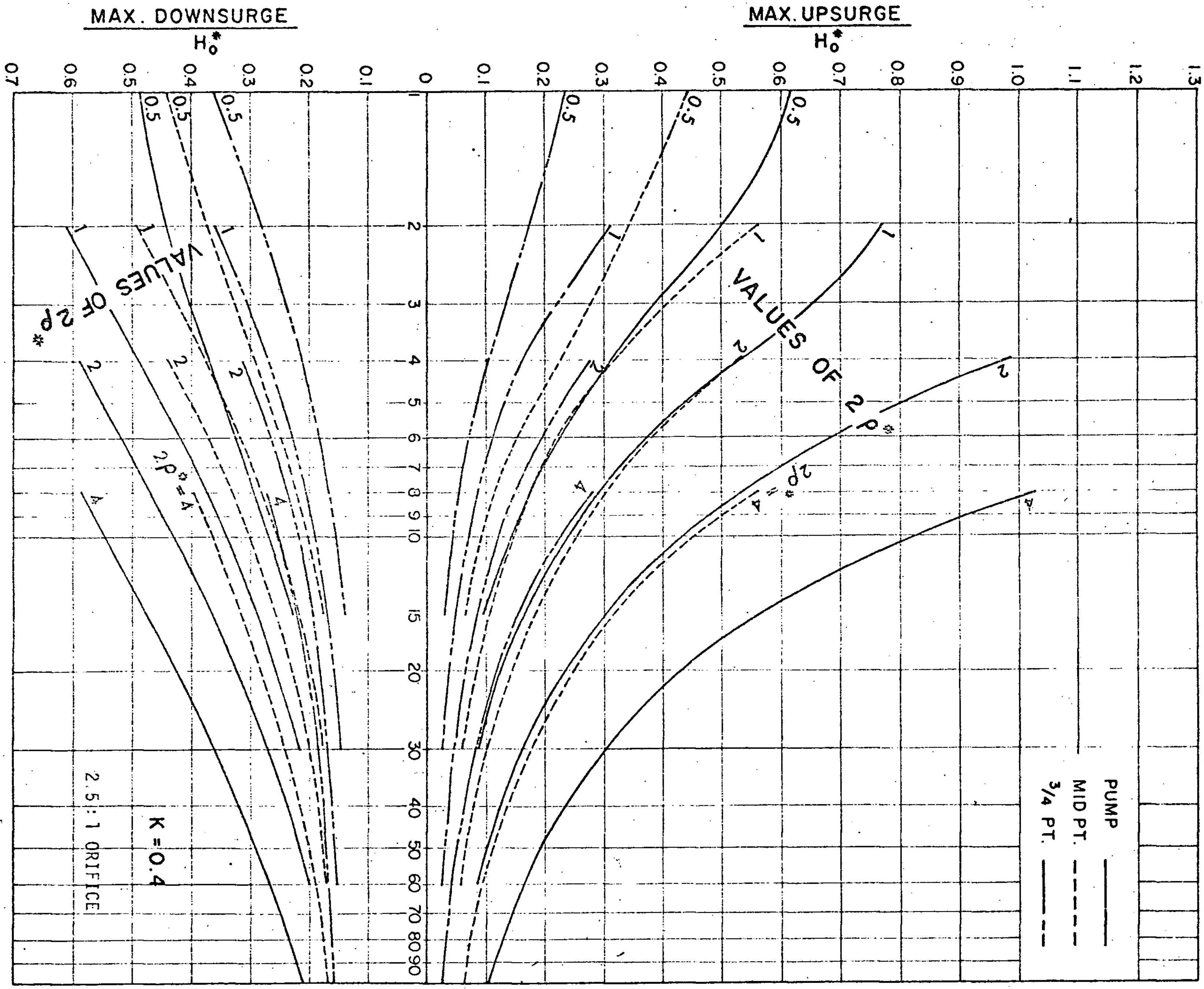
A. DIFFERENTIAL ORIFICE, RATIO 2.5:1

B. SIMPLE ORIFICE, RATIO 1:1

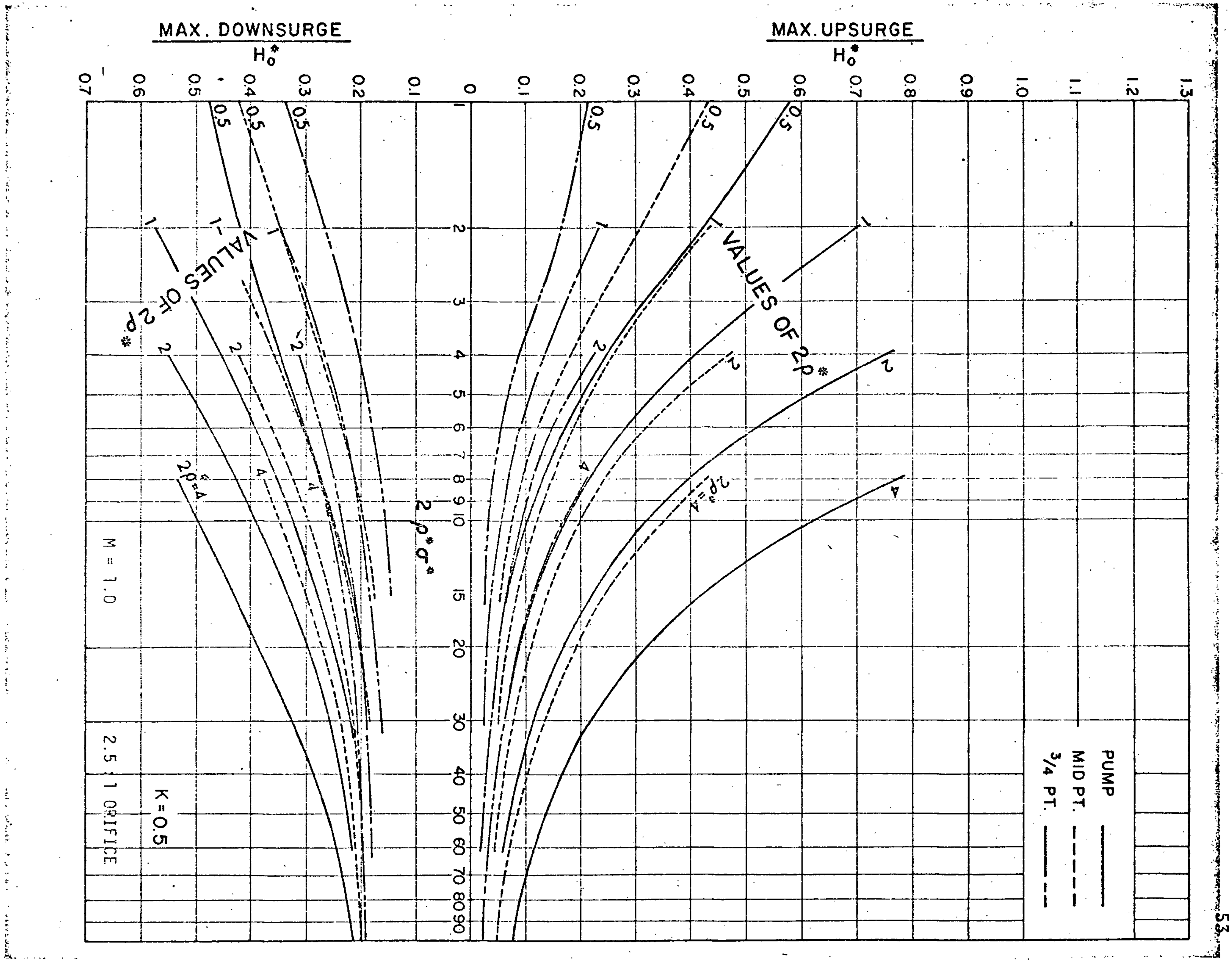

MAX. DOWNSURGE
H*o
MAX. UPSURGE
H*o
VALUES OF 2ρ*
2ρ*σ*
PUMP
MID PT.
3/4 PT.
K = 0.1
1 ORIFICE

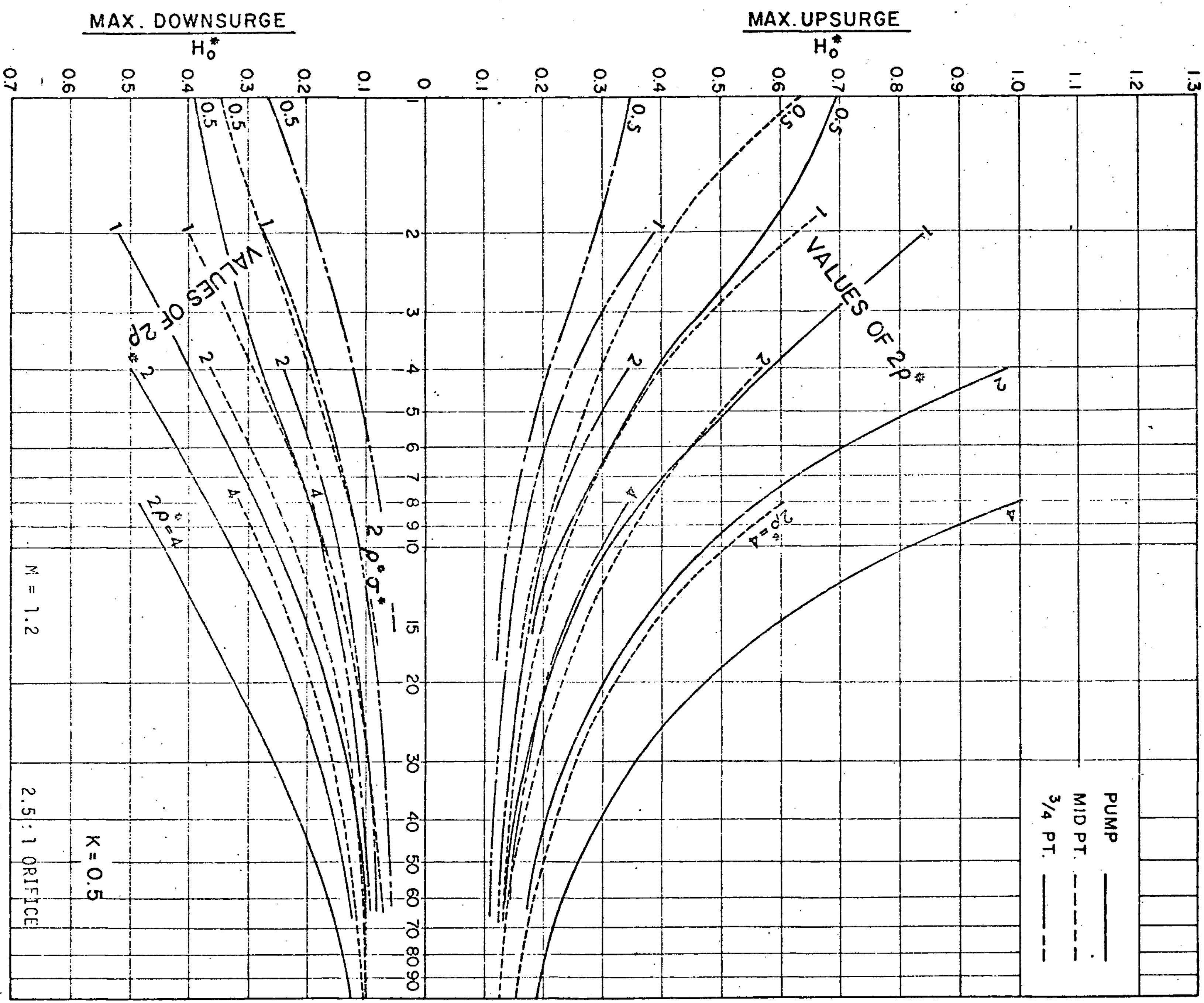

MAX. UPSURGE
H_o^*
MAX. DOWNSURGE
H_o^*
PUMP
MID PT.
3/4 PT.
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^*\sigma^*$
K = 0.2

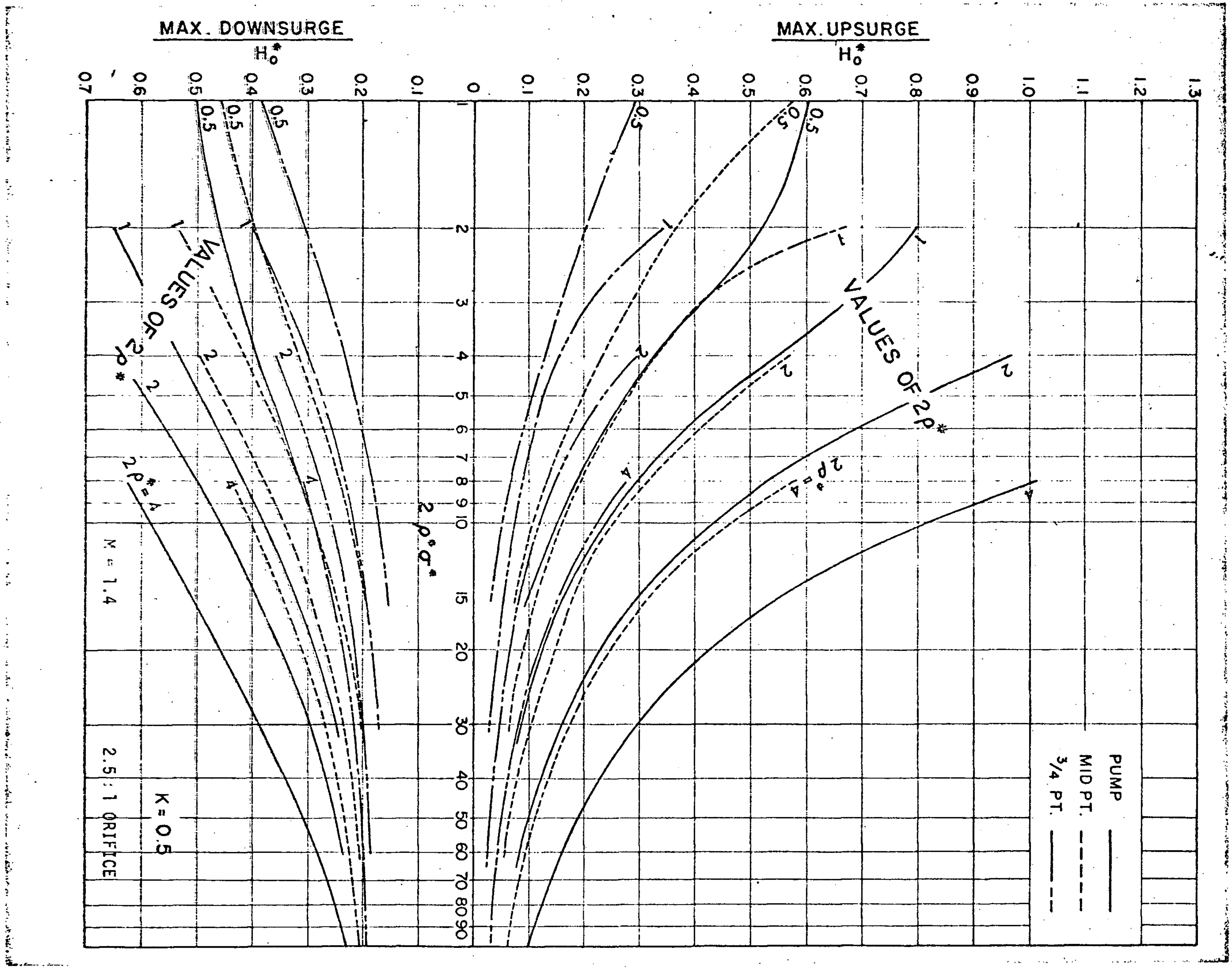

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
$2\rho^*=4$
PUMP
MID PT.
3/4 PT.
K = 0.3
2.5 : 1 ORIFICE

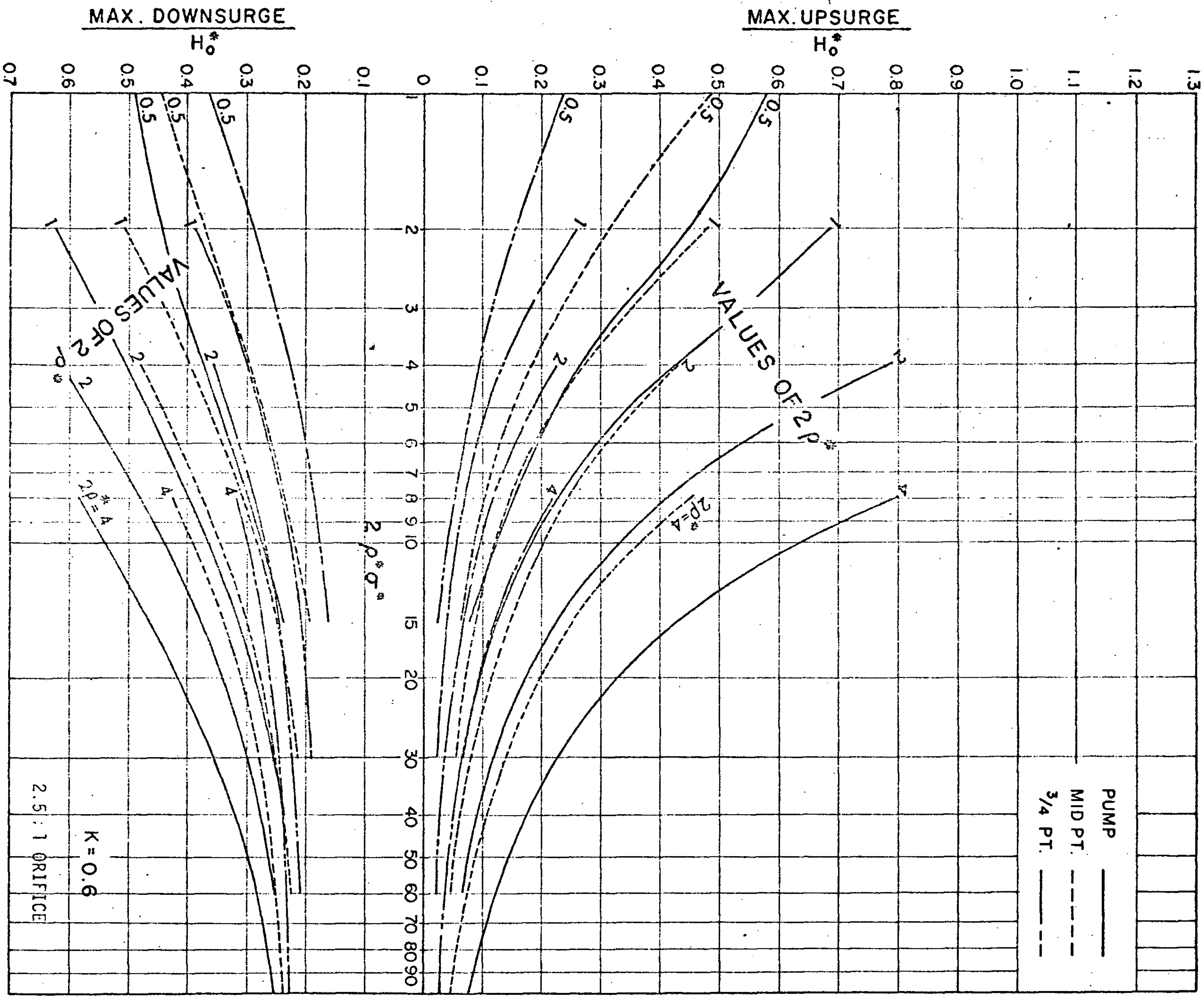


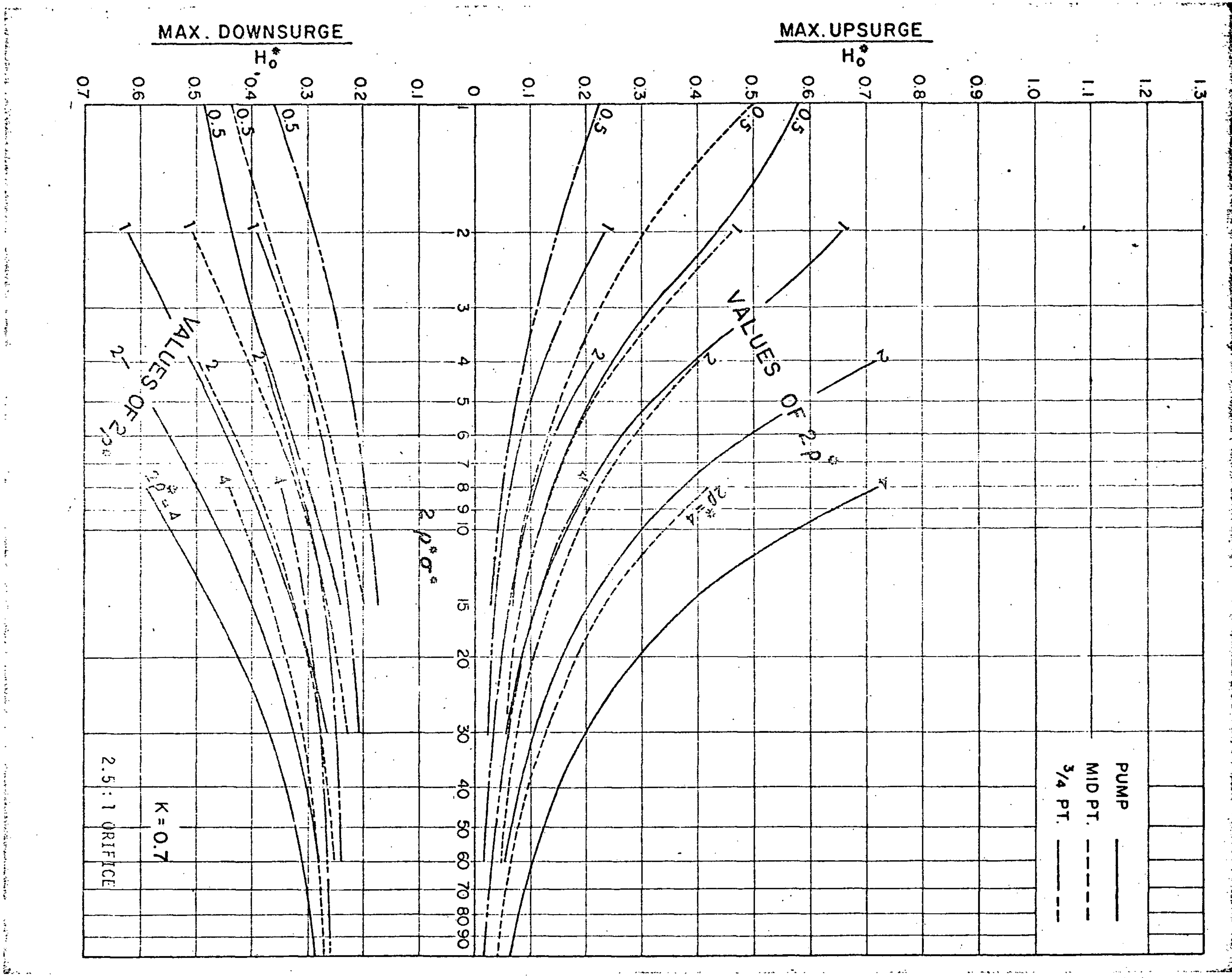
MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
VALUES OF 2ρ*
VALUES OF 2ρ*
2ρ* = 4
2ρ* = 4
0.5
1
2
4
2
3
4
5
6
7
8
9
10
15
20
30
40
50
60
70
80
90
PUMP
MID PT.
3/4 PT.
K = 0.4
2.5 : 1 ORIFICE


MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
$2\rho^*=4$
$2\rho^*=4$
M = 1.0
2.5 : 1 ORIFICE
K=0.5
PUMP
MID PT.
3/4 PT.

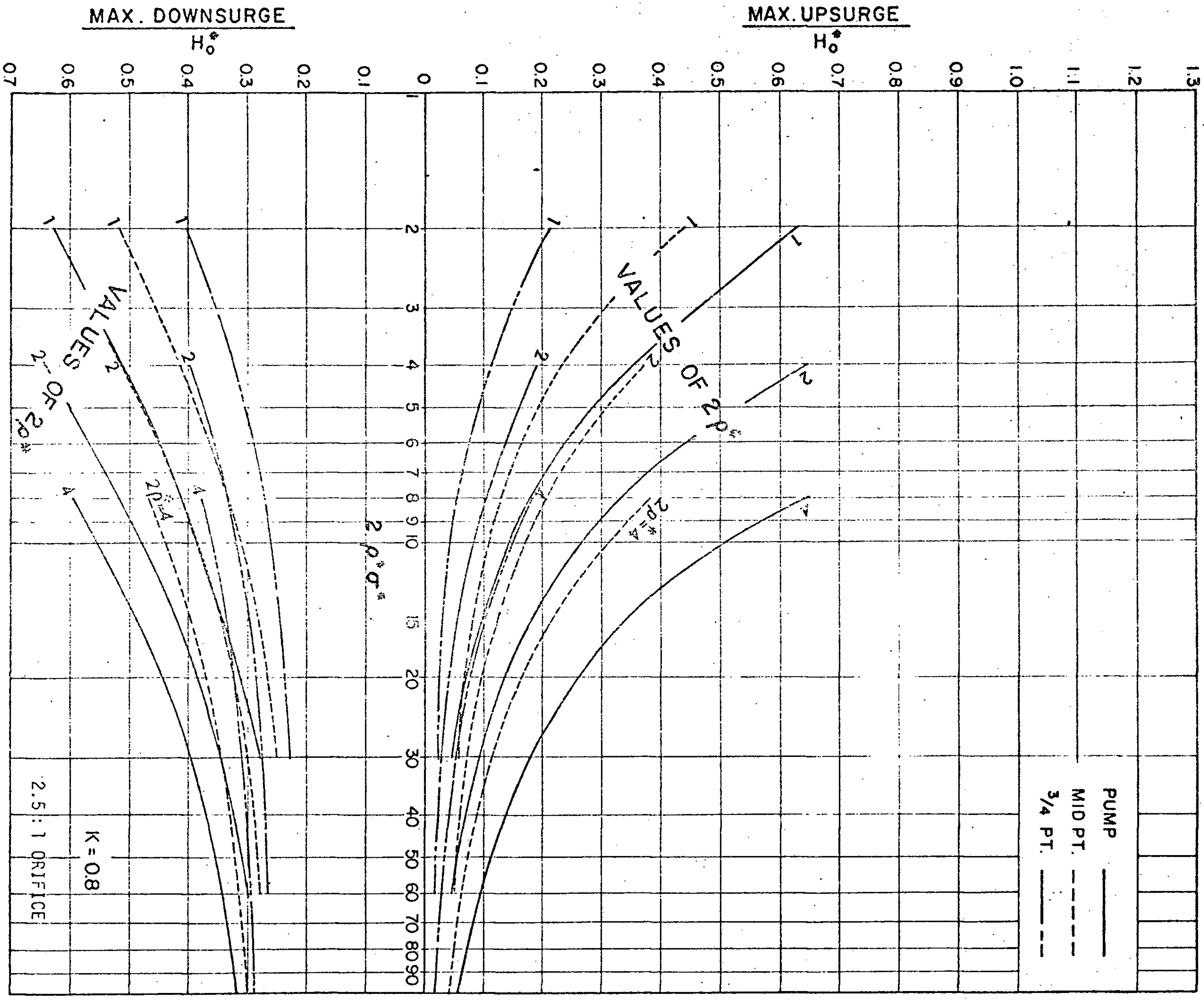

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^* = 4$
$2\rho^* = 4$
$2\rho^* \sigma^*$
M = 1.2
K = 0.5
2.5 : 1 ORIFICE
PUMP
MID PT.
3/4 PT.

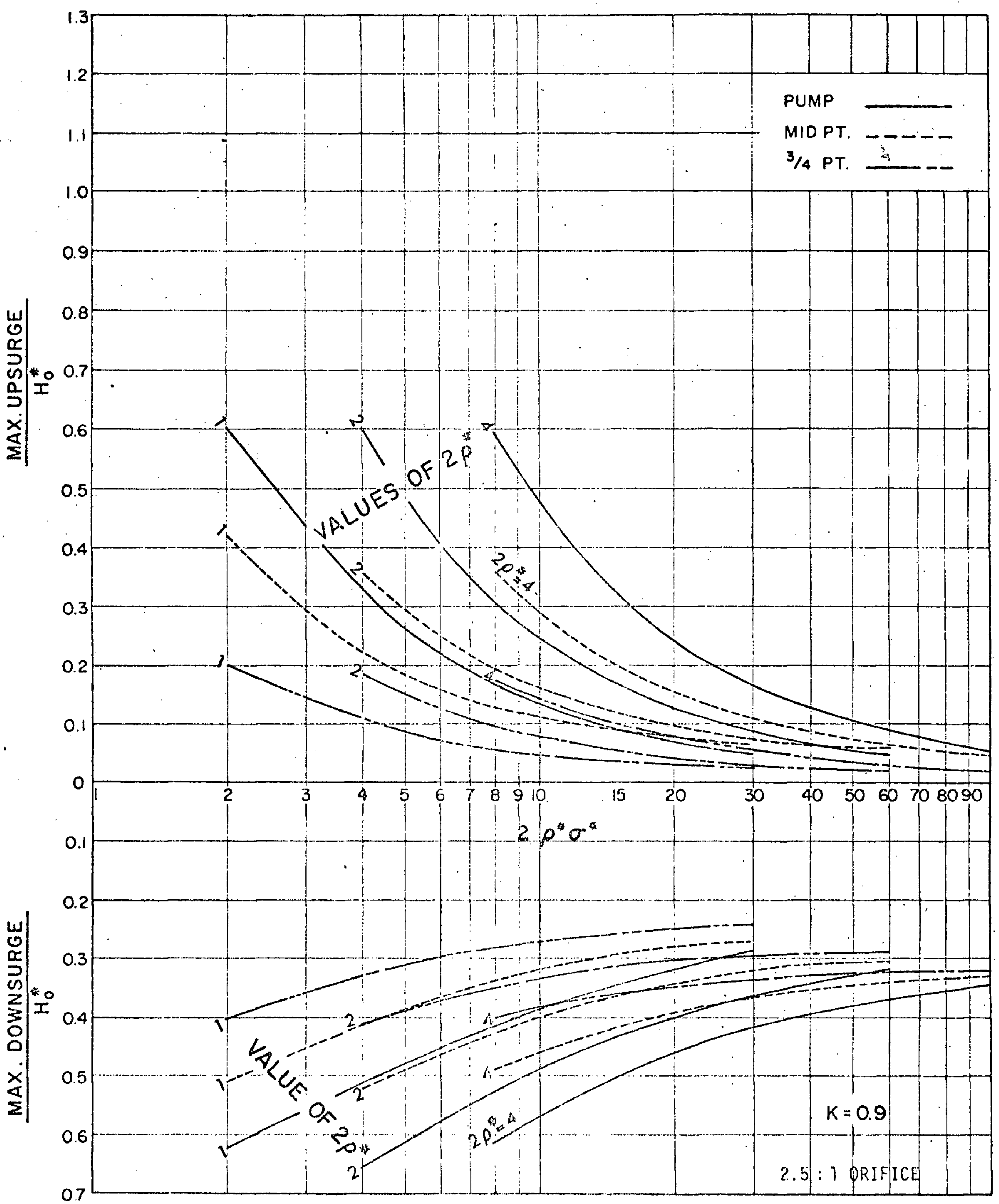

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
M = 1.4
K = 0.5
2.5 : 1 ORIFICE

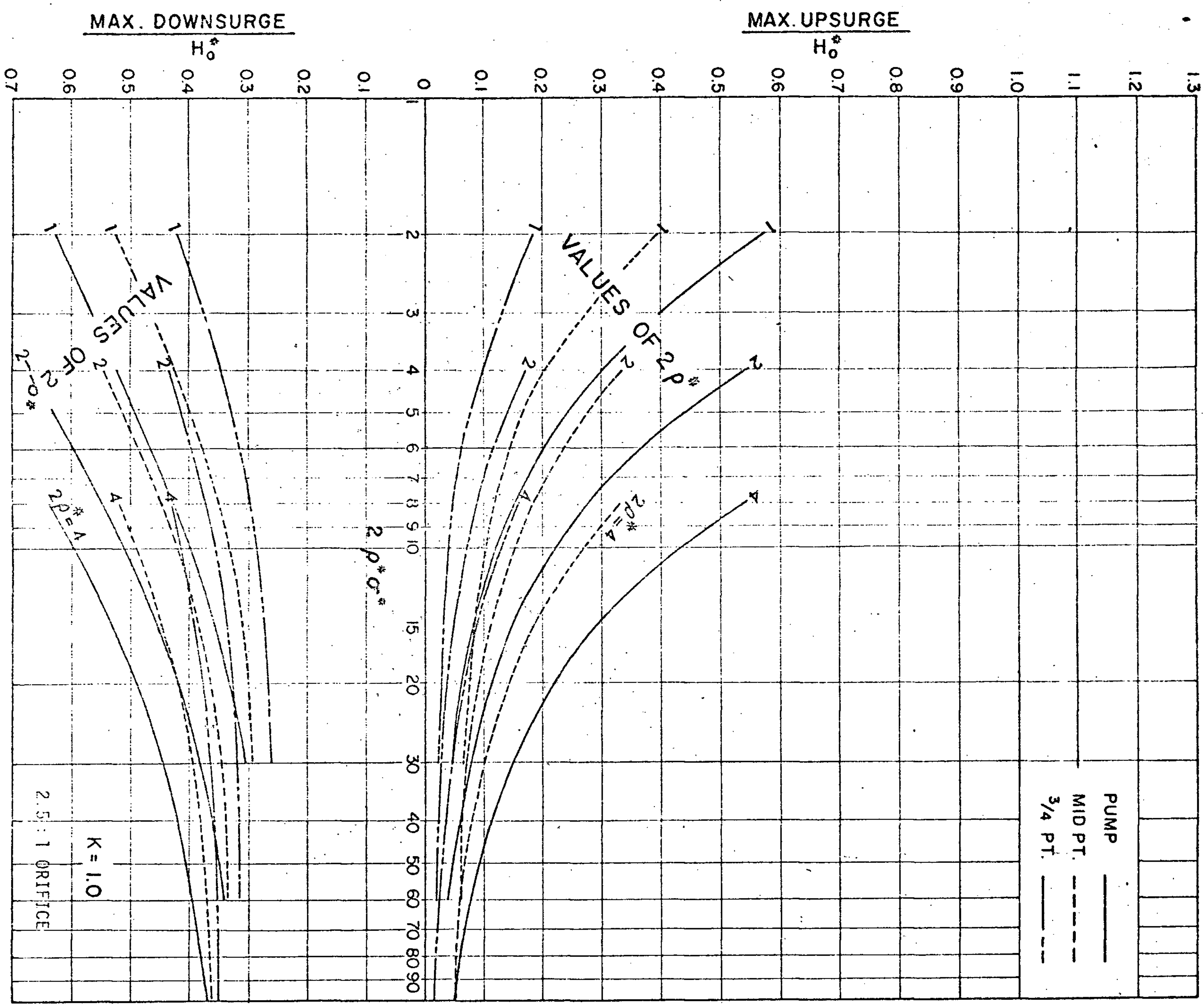

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^* = 4$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 0.6
2.5 : 1 ORIFICE

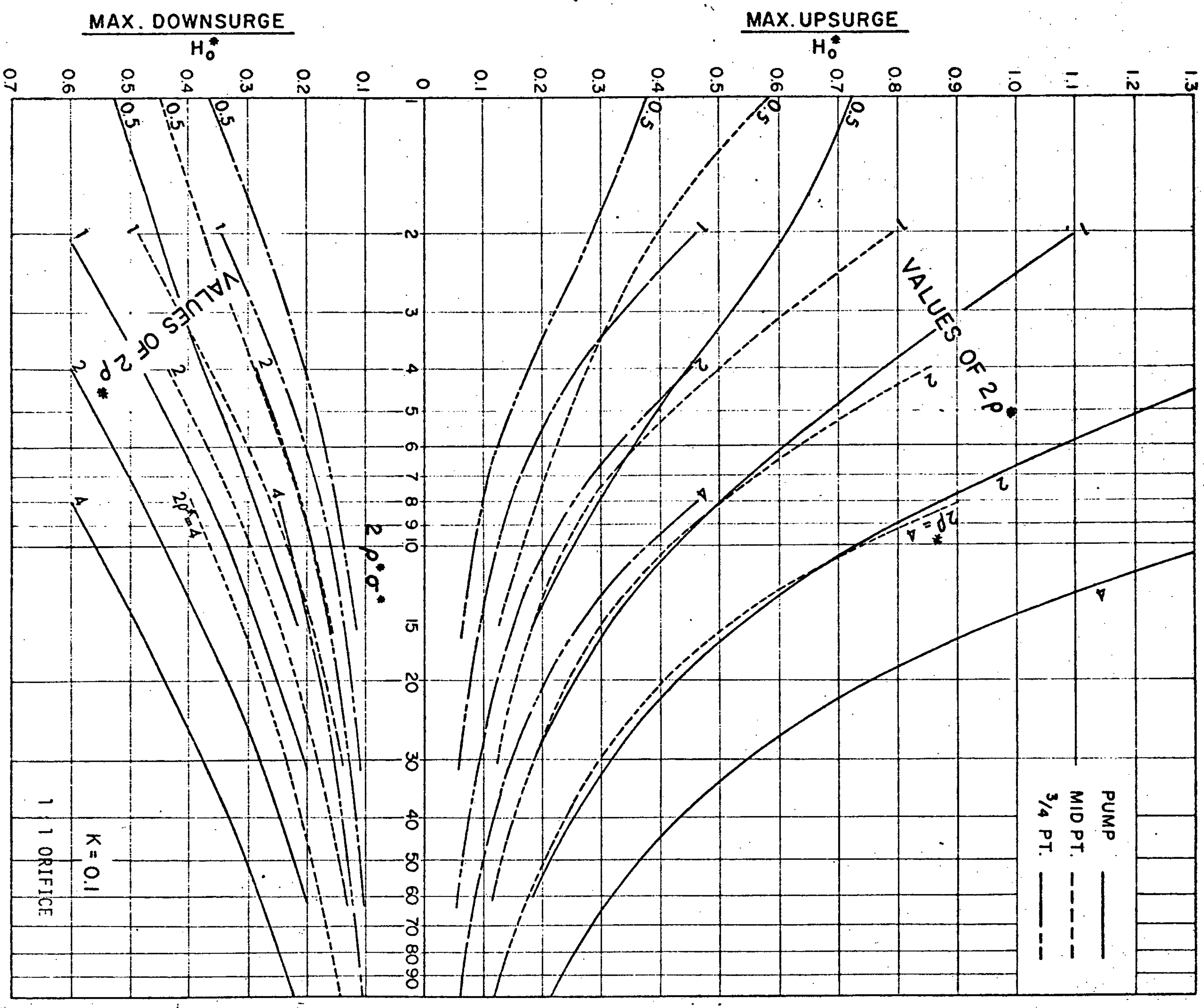


MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 0.7
2.5 : 1 ORIFICE

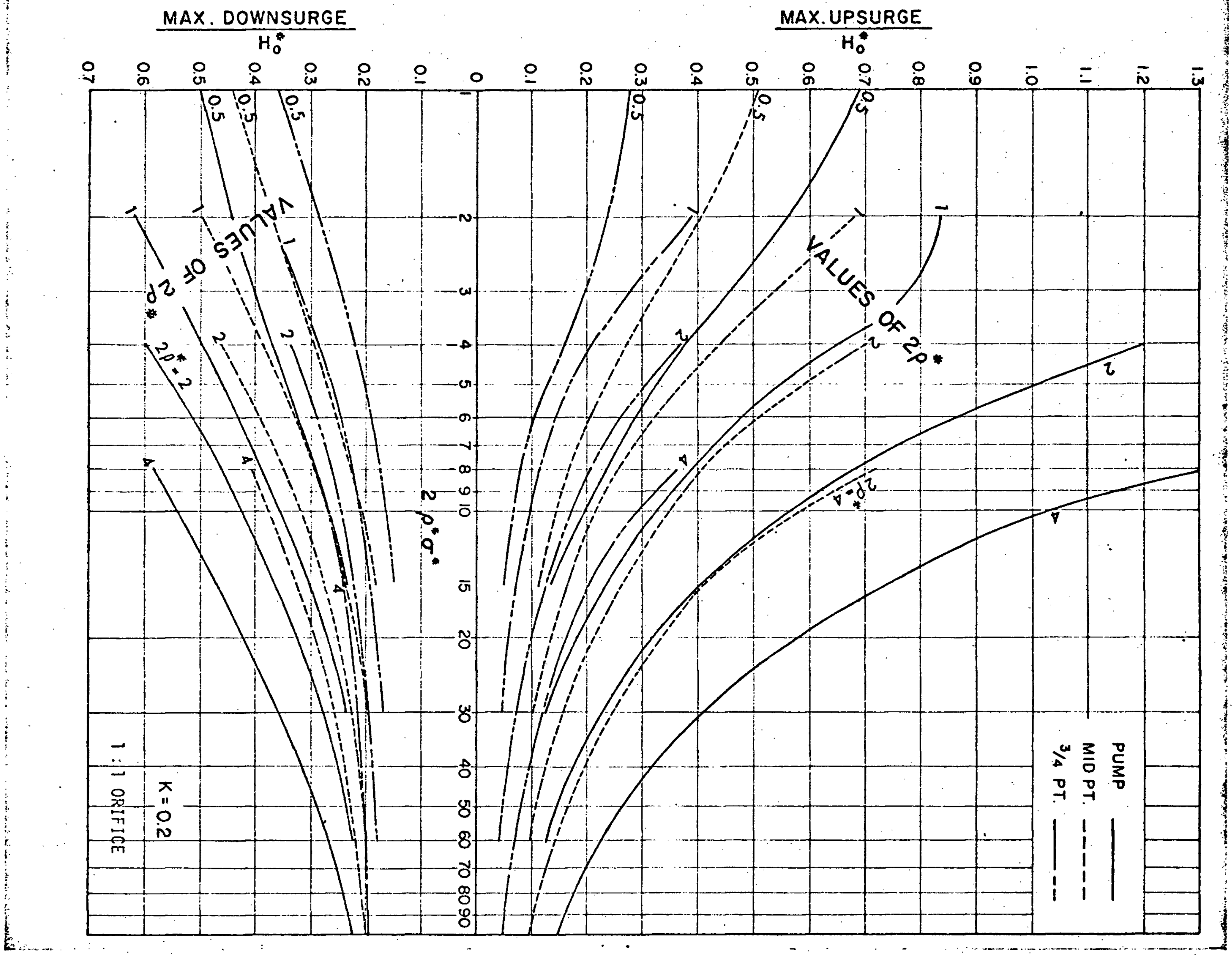

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 0.8
2.5 : 1 ORIFICE


PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H*o
VALUES OF 2ρ*
2ρ*=4
2ρ*σ*
MAX. DOWNSURGE / H*o
VALUE OF 2ρ*
2ρ*=4
K = 0.9
2.5 : 1 ORIFICE

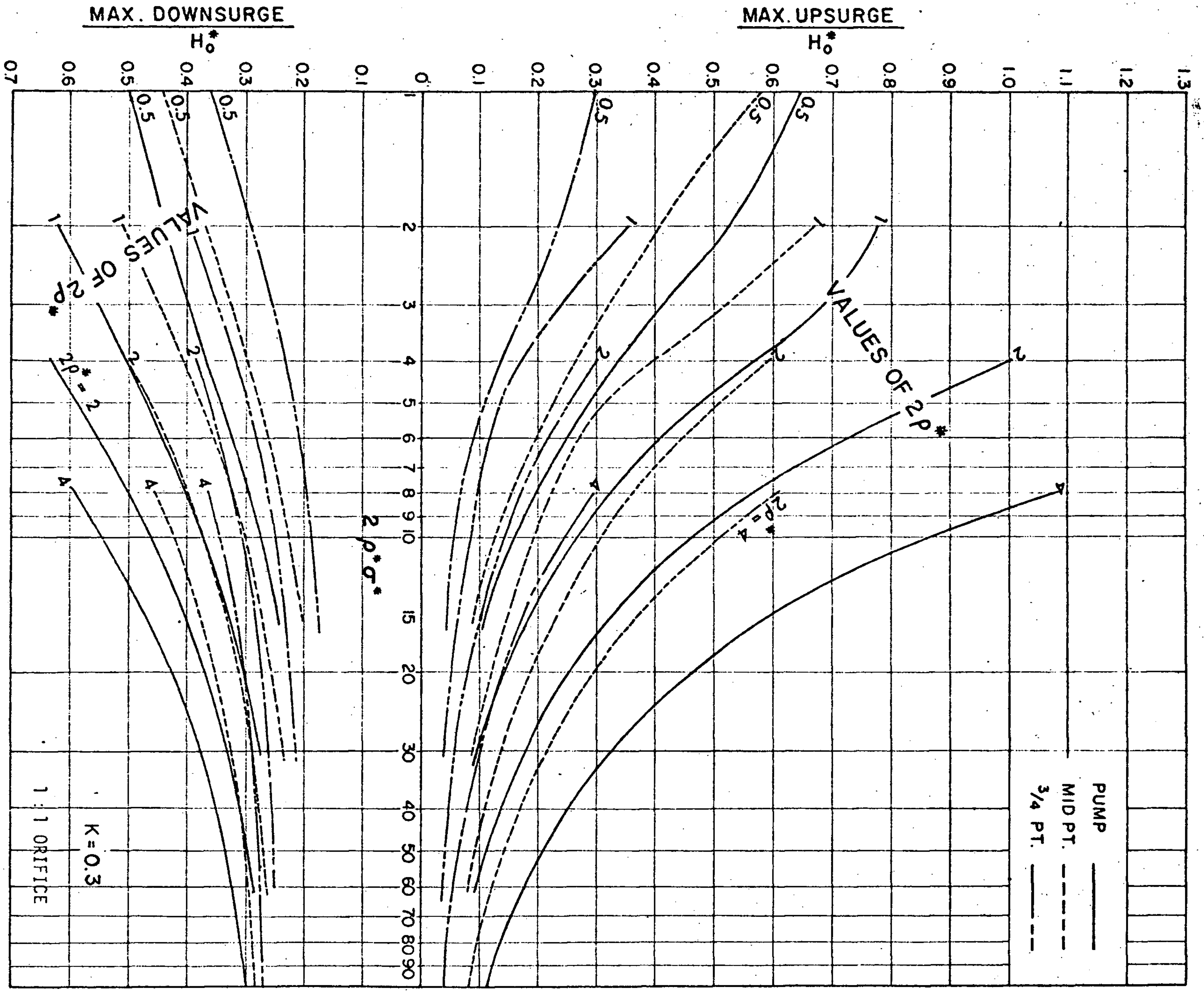

MAX. DOWNSURGE
H*o
MAX. UPSURGE
H*o
VALUES OF 2ρ*
2ρ*σ*
PUMP
MID PT.
3/4 PT.
K = 1.0
2.5:1 ORIFICE



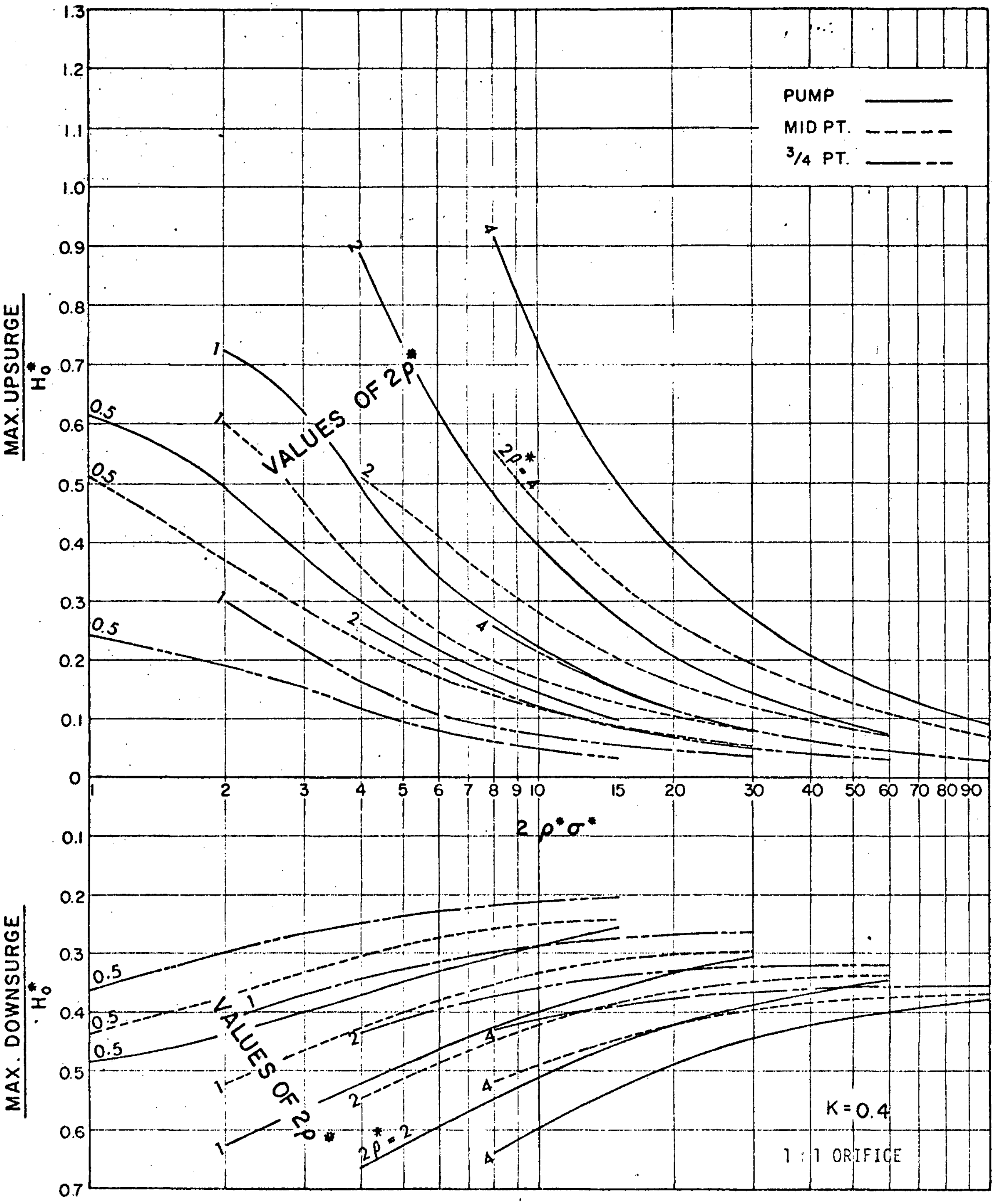
MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
1
2
3
4
5
6
7
8
9
10
15
20
30
40
50
60
70
80
90
$2\rho^*\sigma^*$
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^* = 4$
PUMP
MID PT.
3/4 PT.
K = 0.1
1 : 1 ORIFICE

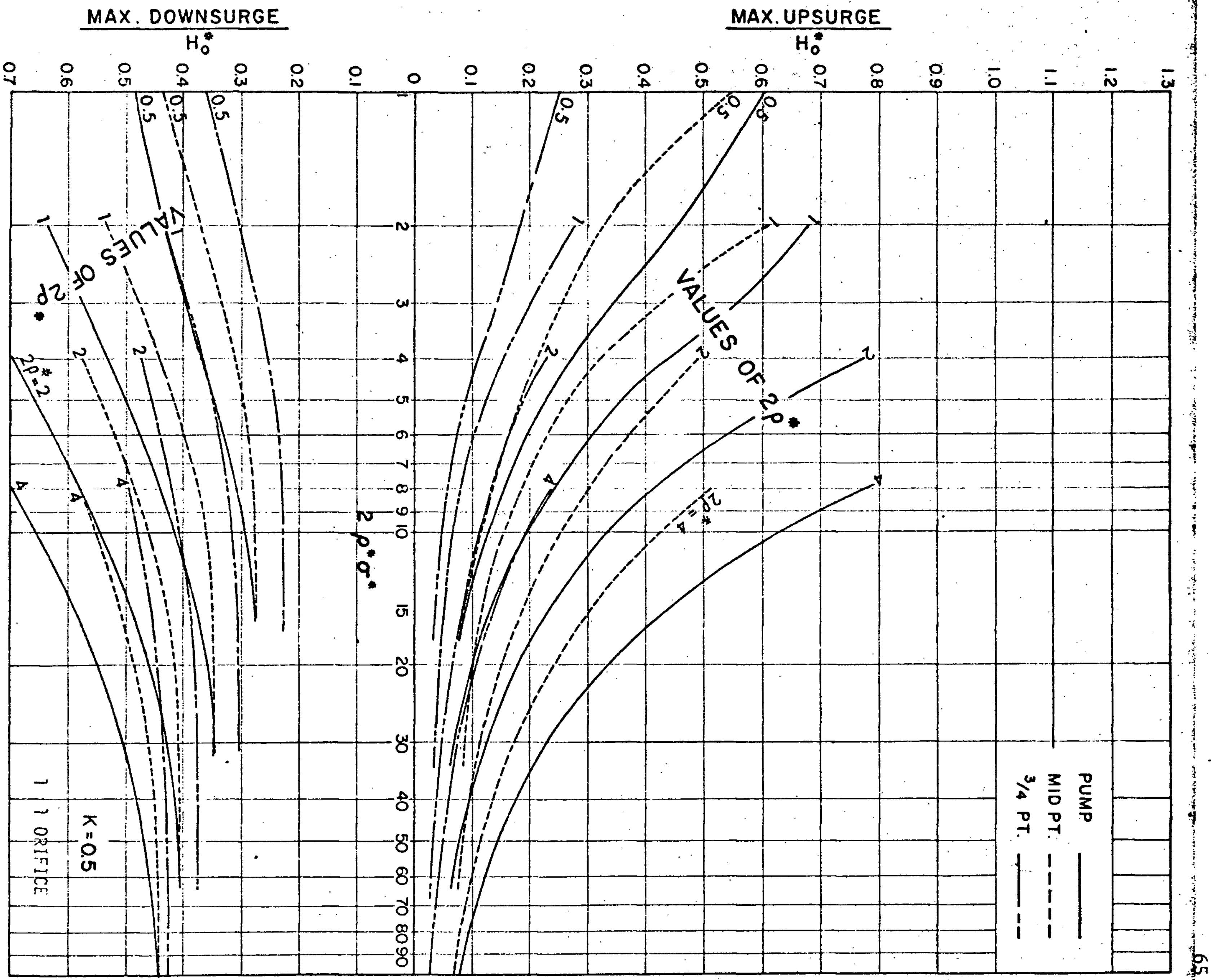


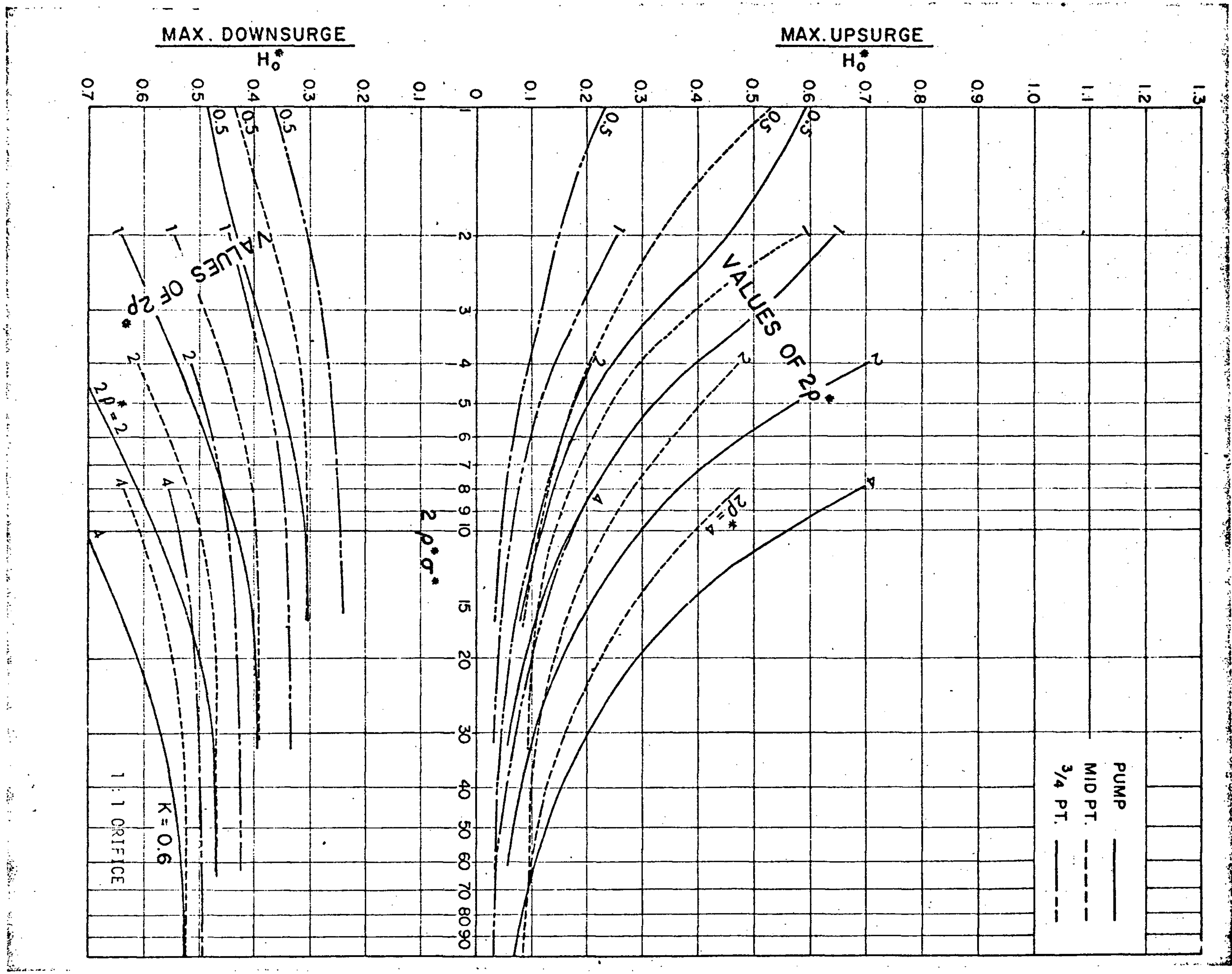
MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 0.2
1 : 1 ORIFICE

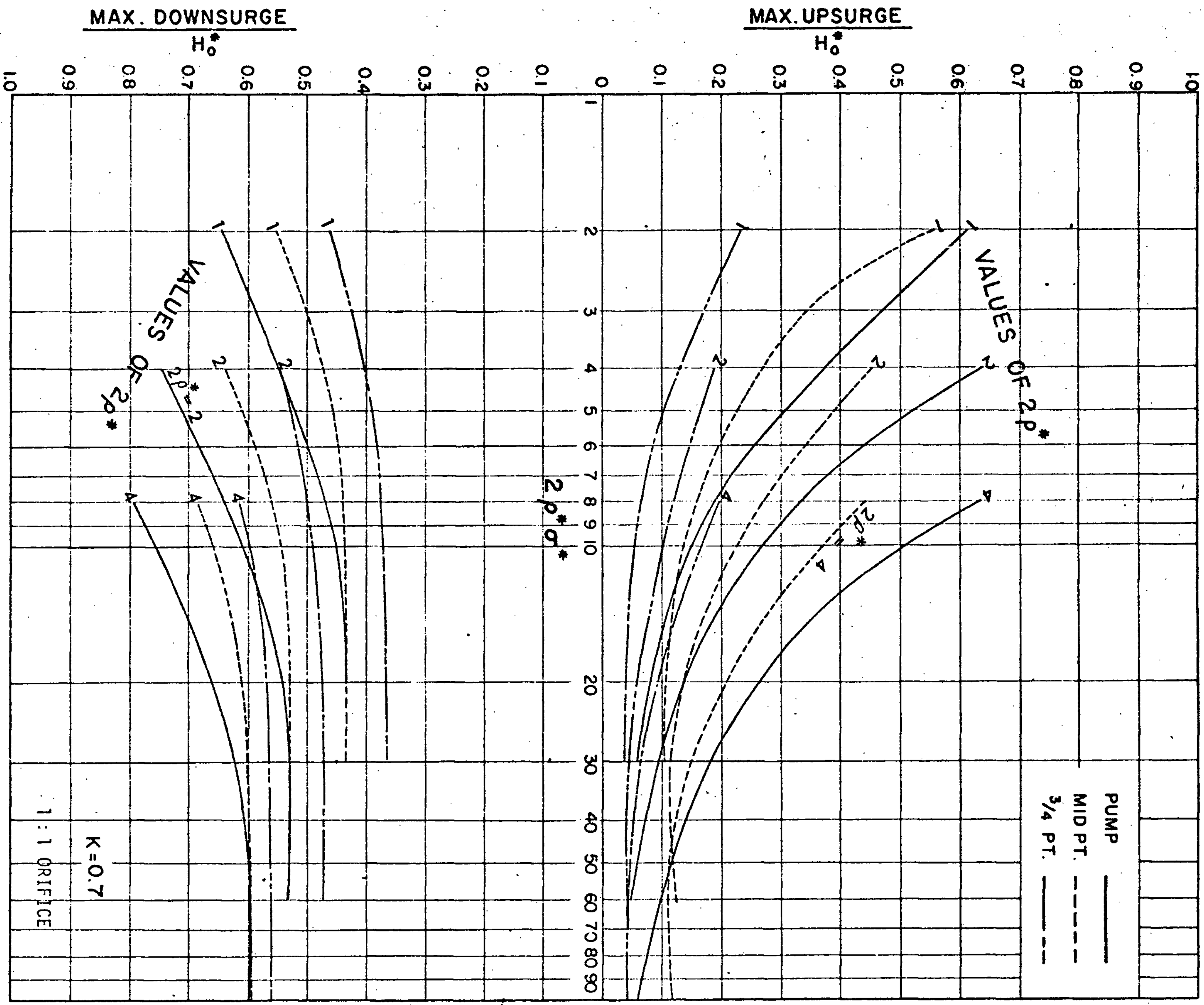


MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 0.3
1 : 1 ORIFICE

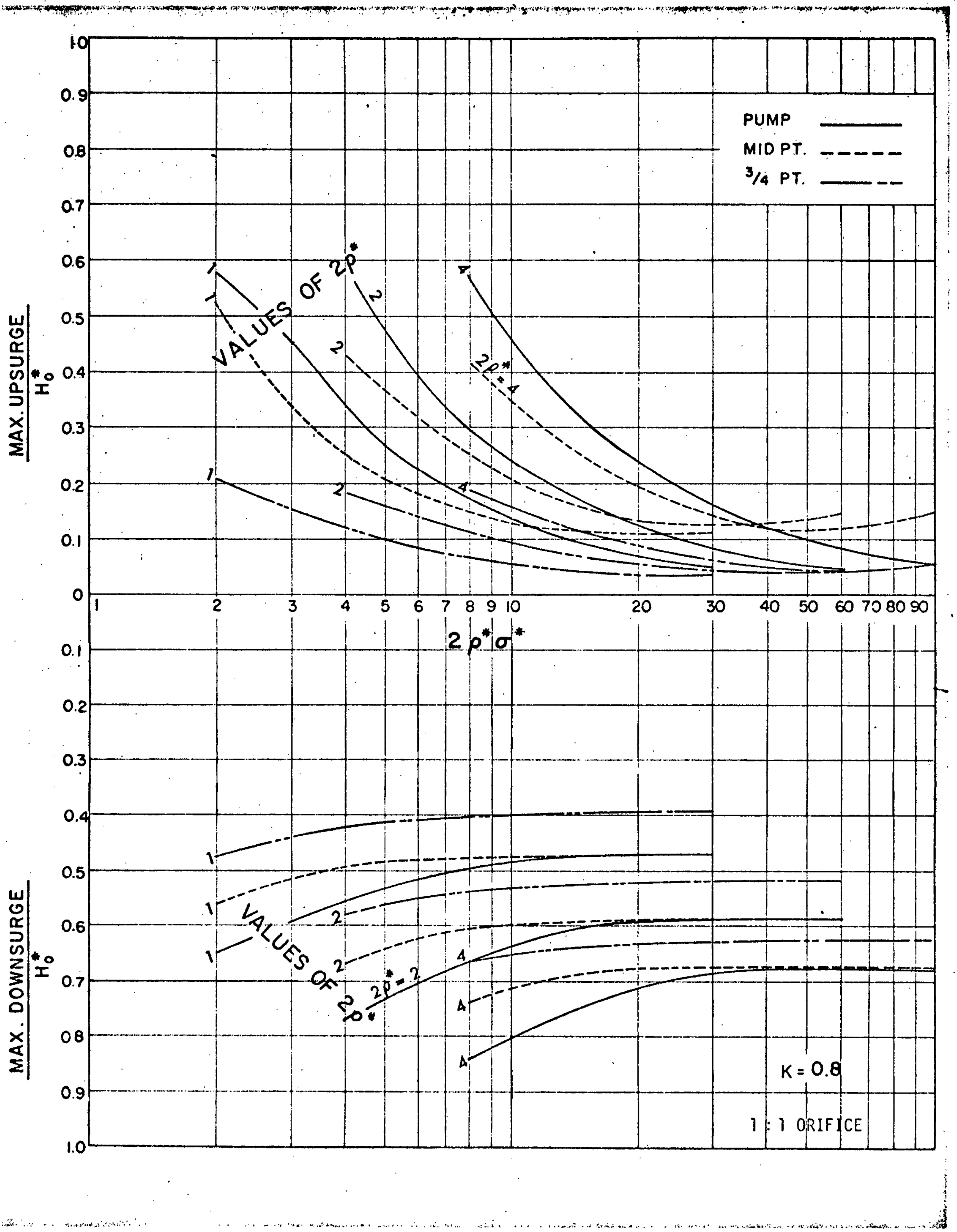

PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^*=4$
$2\rho^*\sigma^*$
MAX. DOWNSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^*=2$
K = 0.4
1 : 1 ORIFICE


MAX. DOWNSURGE
H*o
MAX. UPSURGE
H*o
VALUES OF 2ρ*
2ρ*σ*
PUMP
MID PT.
3/4 PT.
K=0.5
1 1 ORIFICE

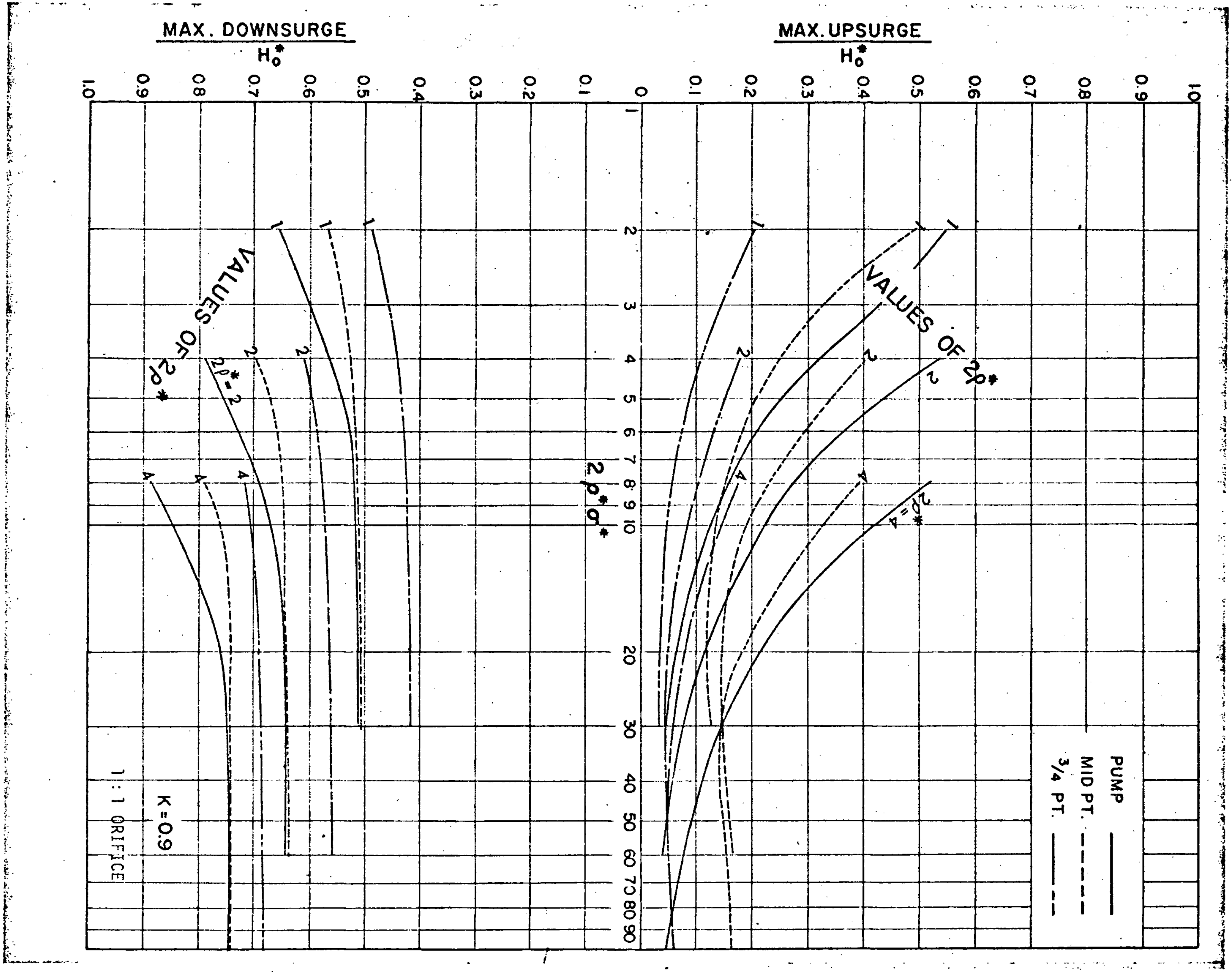

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^*\sigma^*$
K = 0.6
1 : 1 ORIFICE
PUMP
MID PT.
3/4 PT.

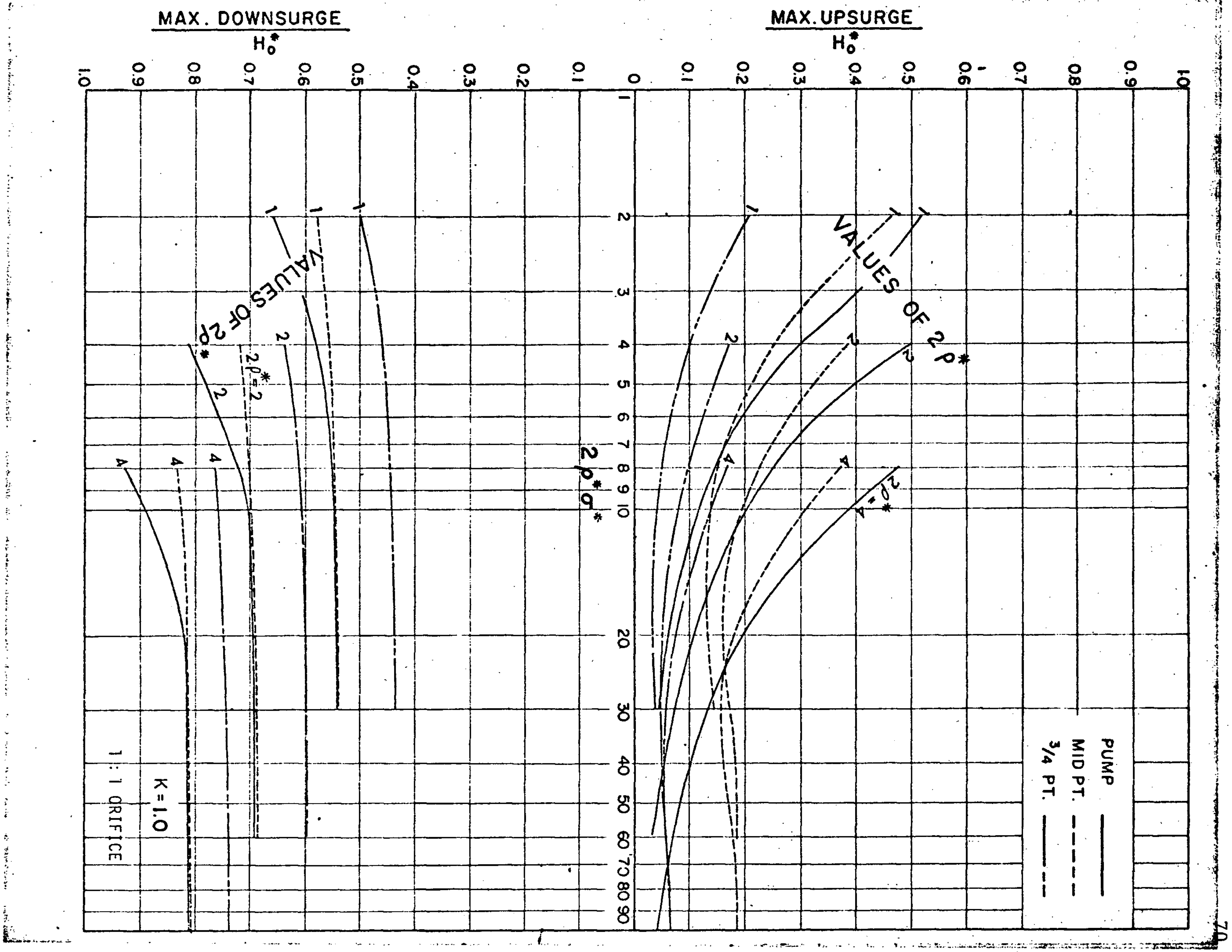


MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K=0.7
1:1 ORIFICE



PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 4$
$2\rho^* = 2$
$2\rho^*\sigma^*$
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10
20
30
40
50
60
70
80
90
K = 0.8
1 : 1 ORIFICE


MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K=0.9
1:1 ORIFICE

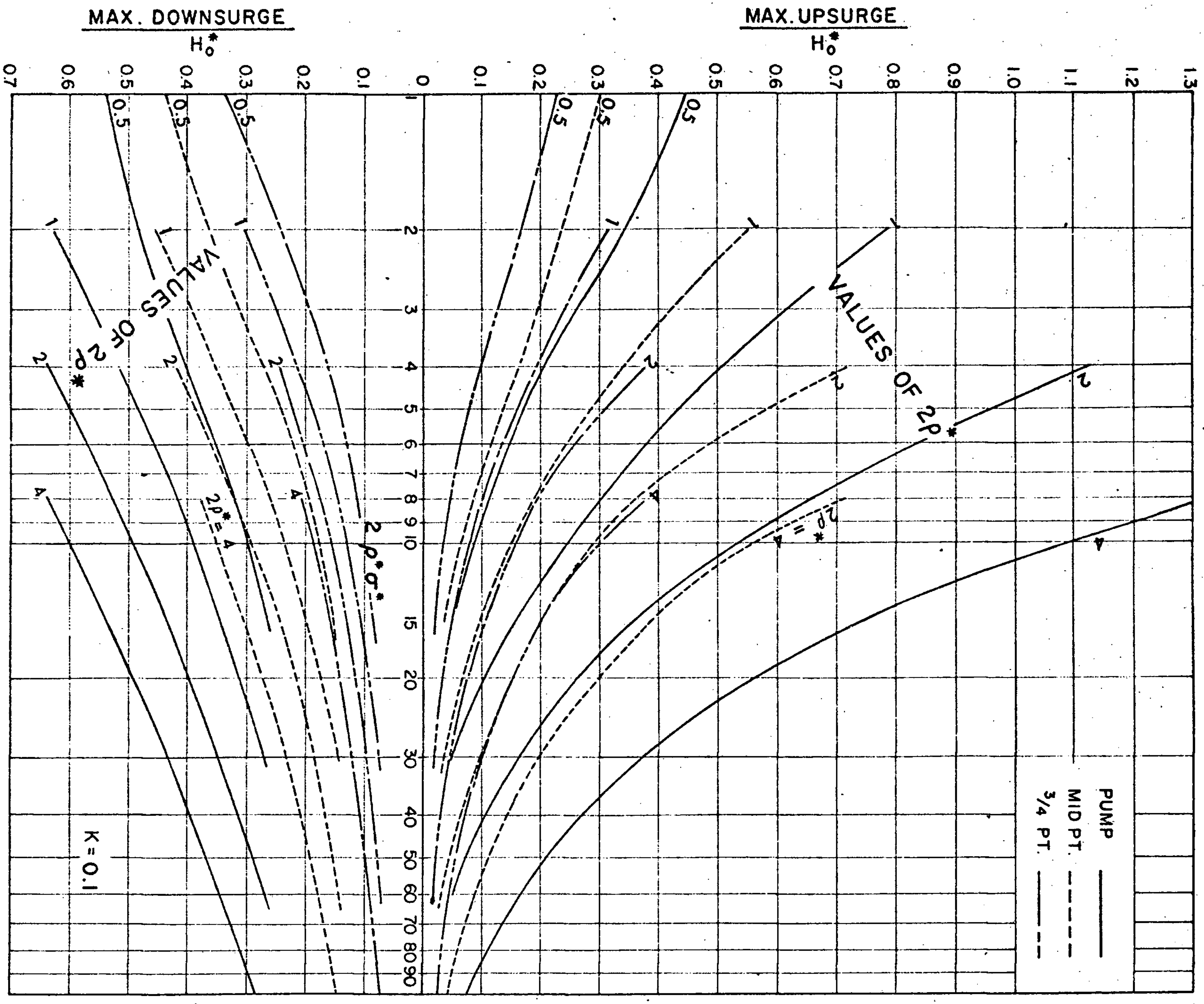


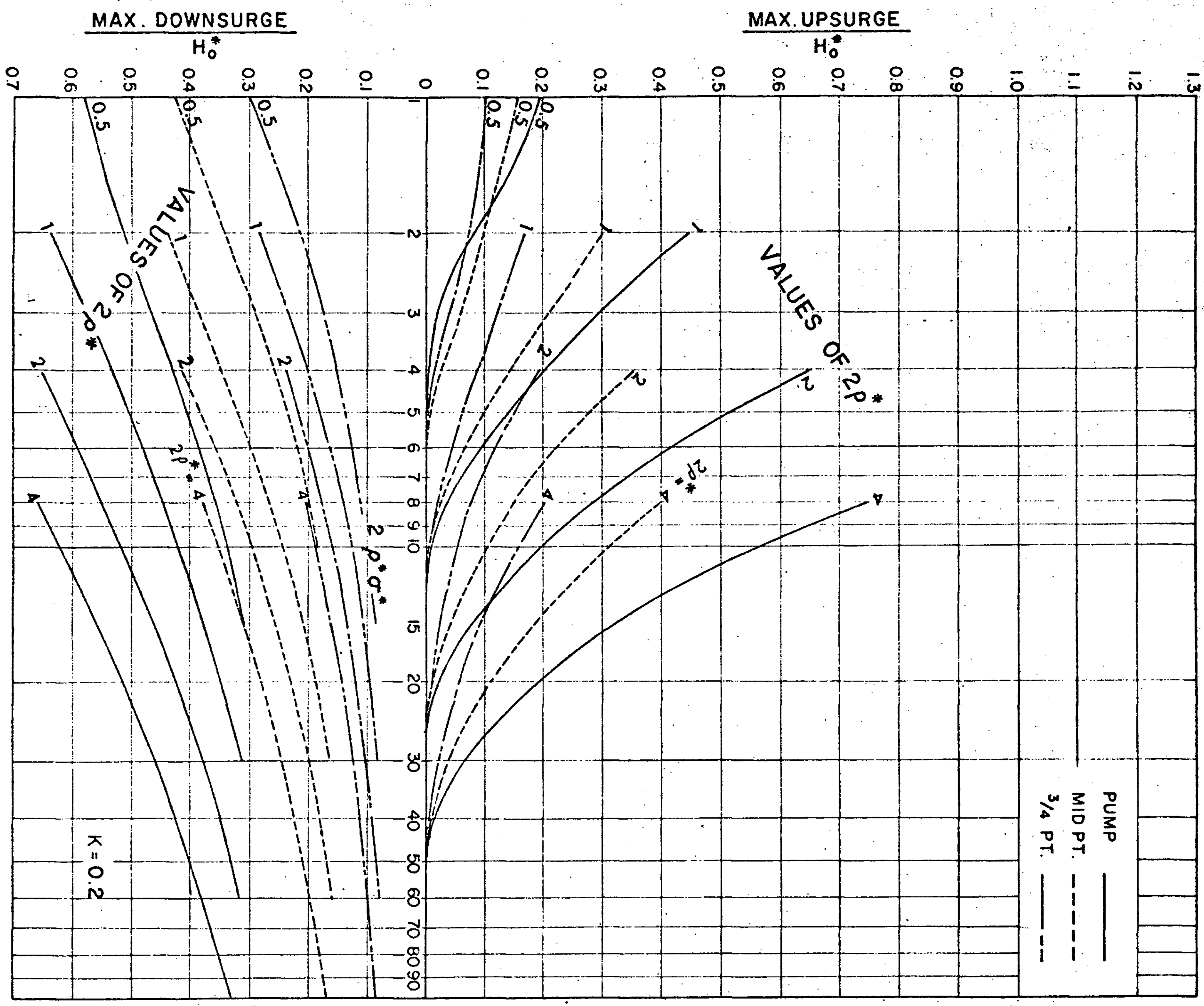
MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 1.0
1:1 ORIFICE

GROUP III

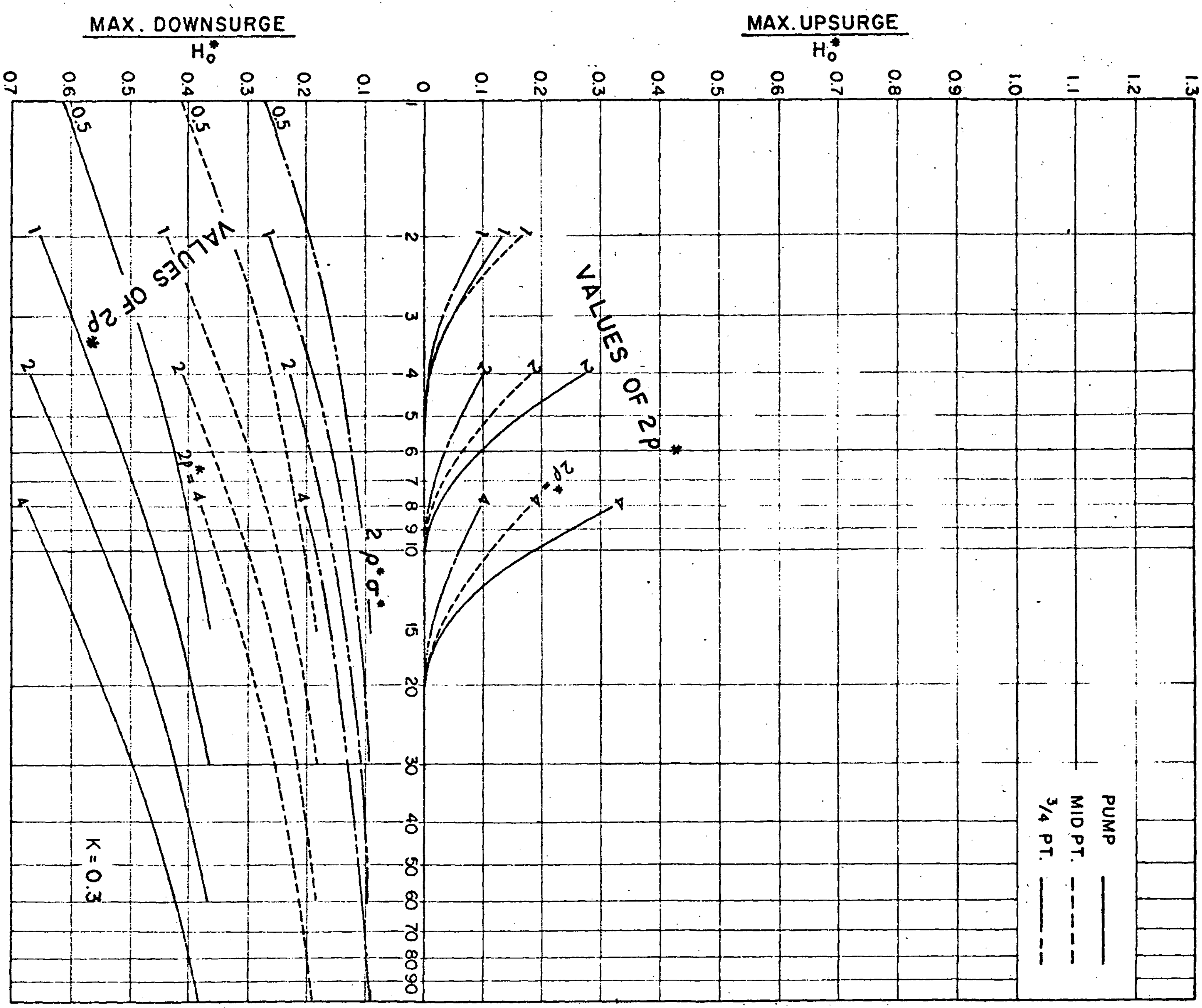
ENTIRE HEAD LOSS ATTRIBUTABLE TO DISTRIBUTED FRICTION

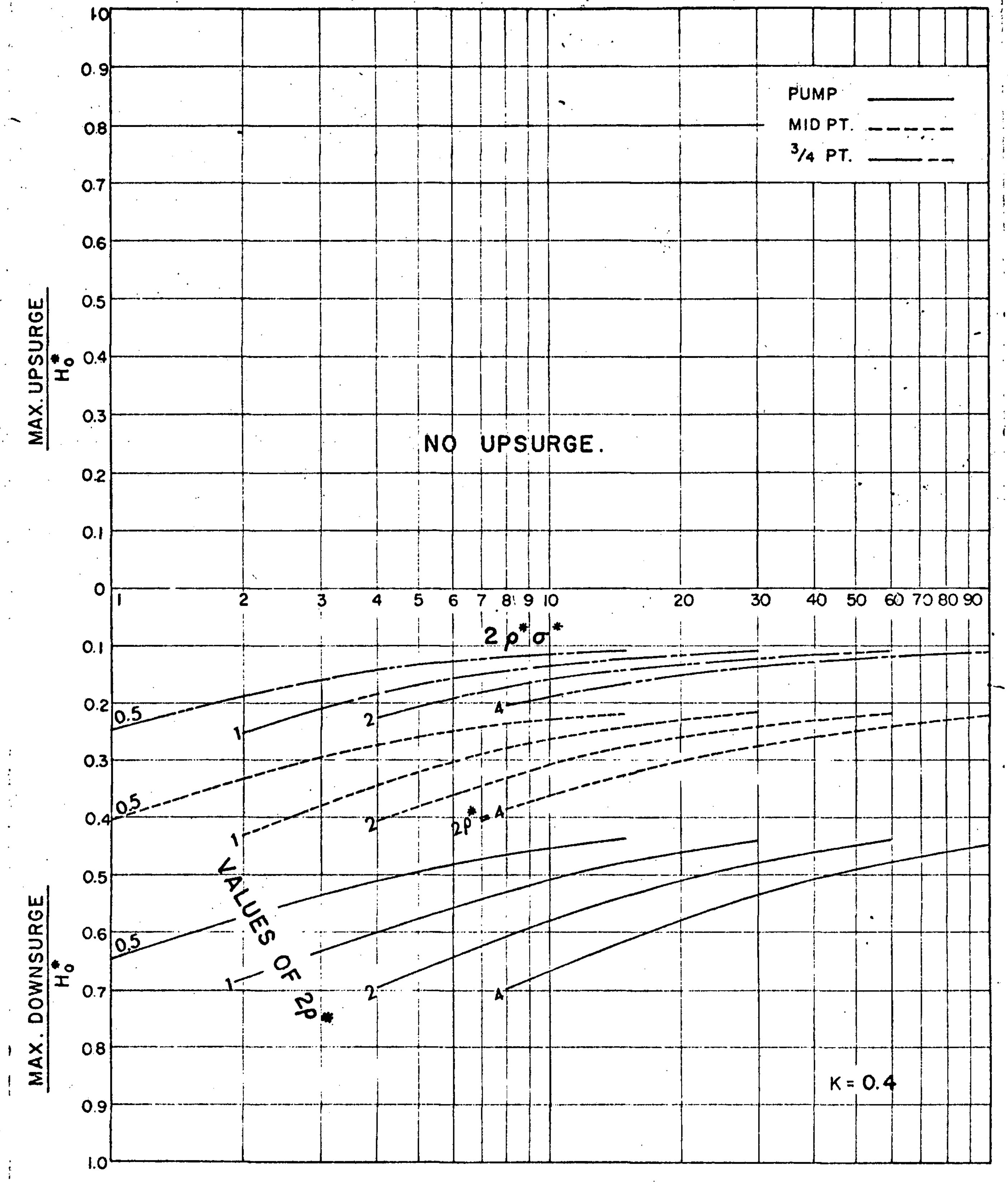
(no orifice loss)

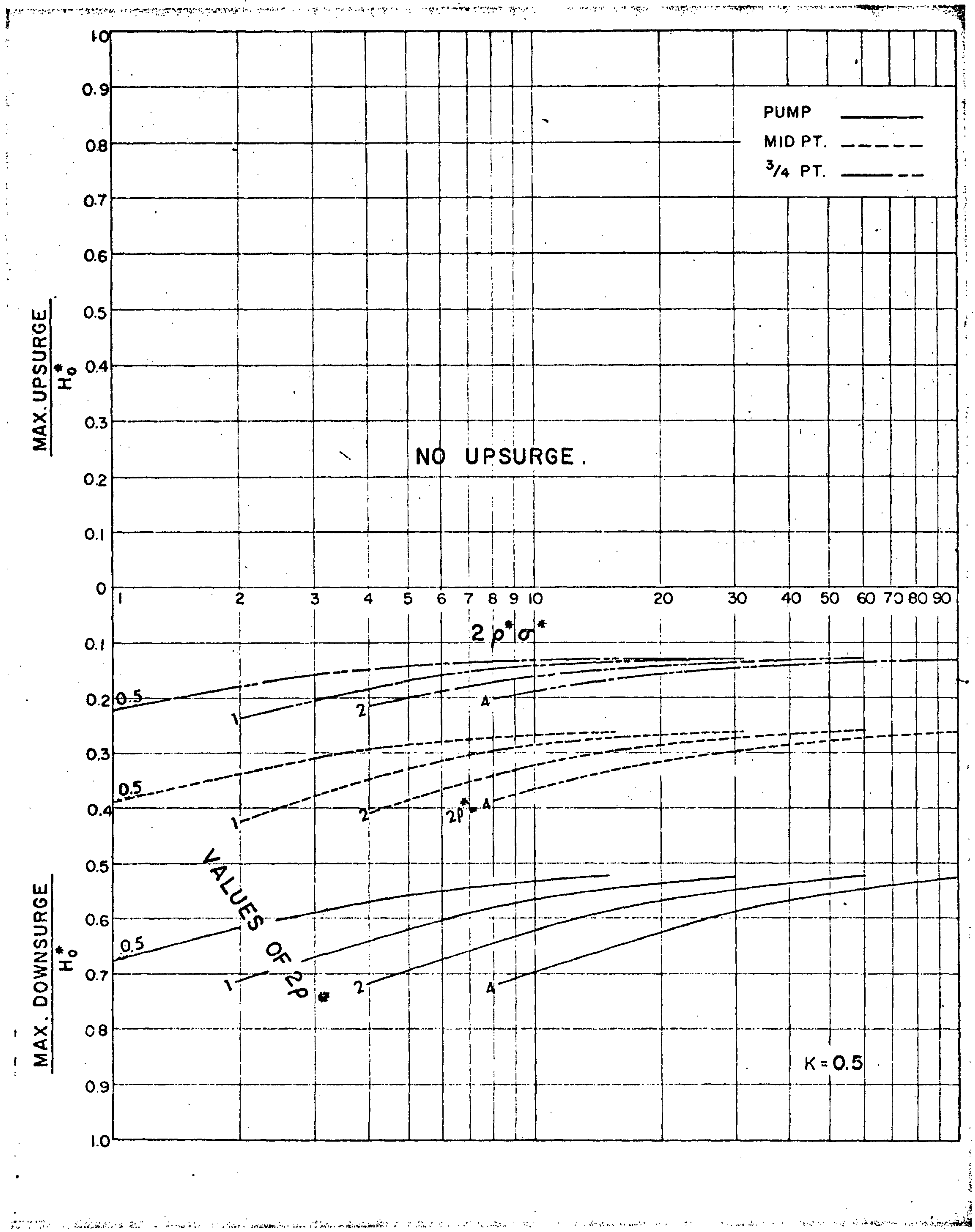

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF 2ρ*
VALUES OF 2ρ*
2ρ*σ*
2ρ* = 4
K = 0.1
PUMP
MID PT.
3/4 PT.



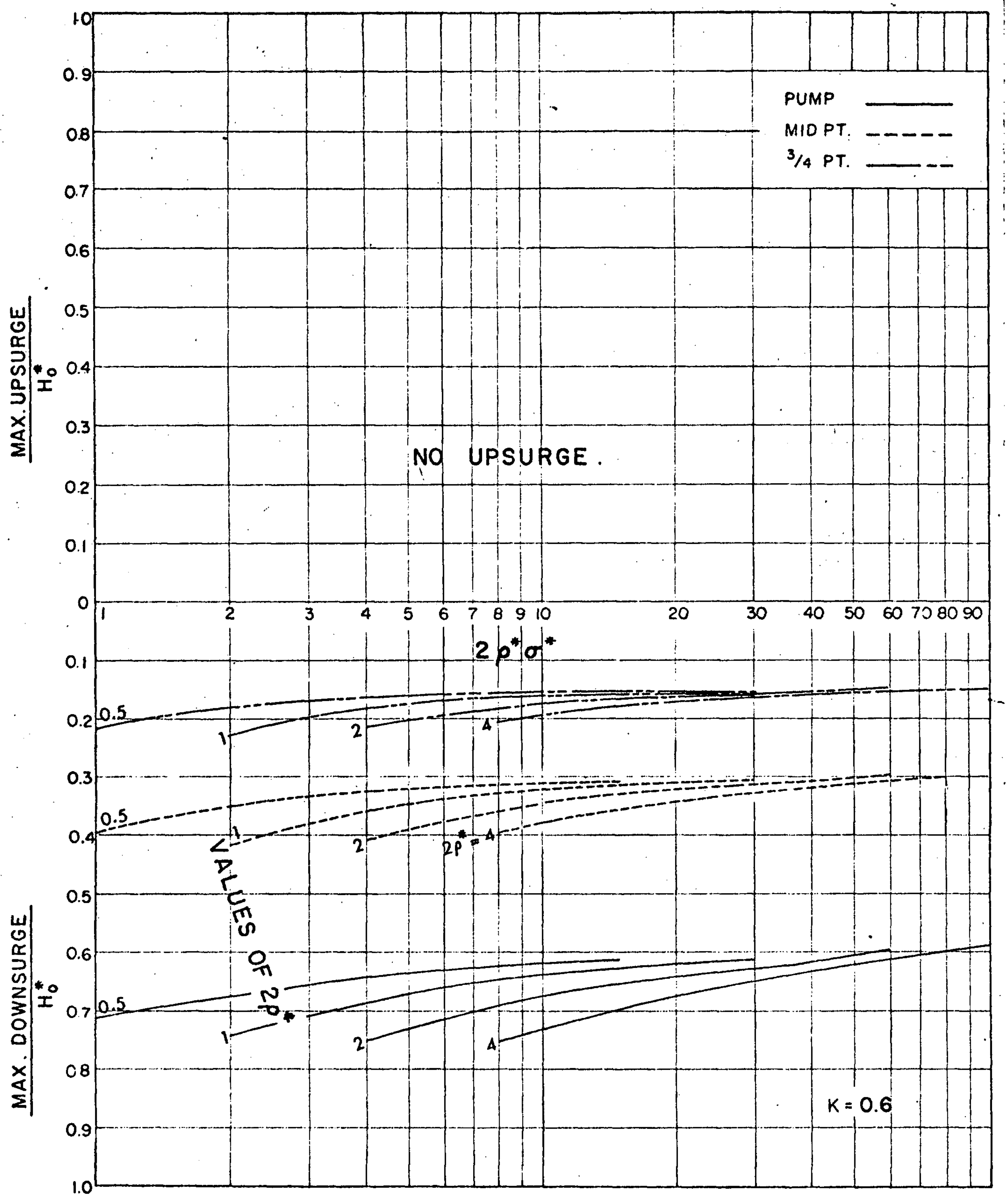
MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
K=0.2
PUMP
MID PT.
3/4 PT.


MAX. DOWNSURGE
H*o
MAX. UPSURGE
H*o
VALUES OF 2ρ*
2ρ* = 4
2ρ*σ*
K = 0.3
PUMP
MID PT.
3/4 PT.

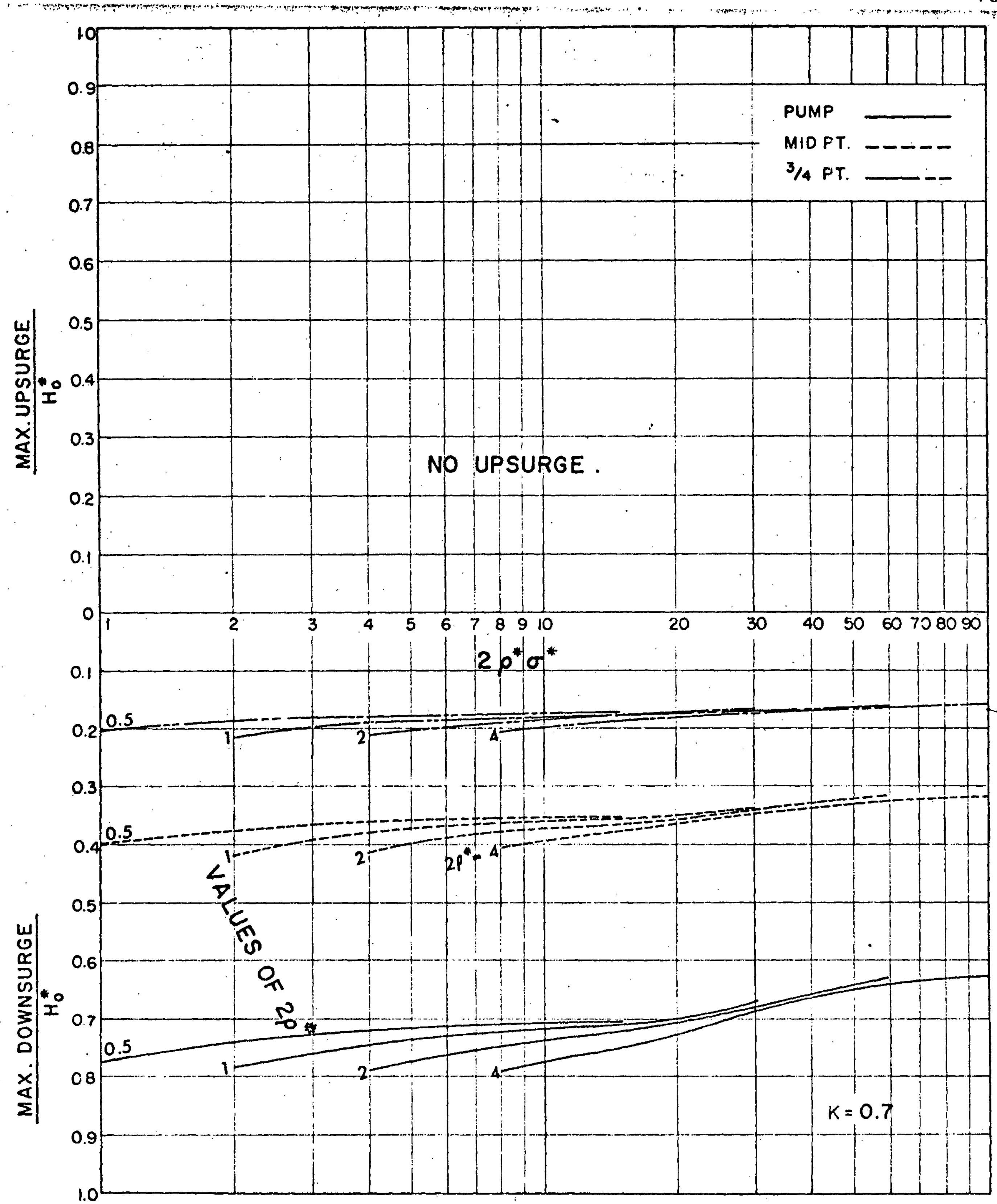

PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
NO UPSURGE.
$2\rho^*\sigma^*$
VALUES OF $2\rho^*$
$2\rho^* = 4$
K = 0.4

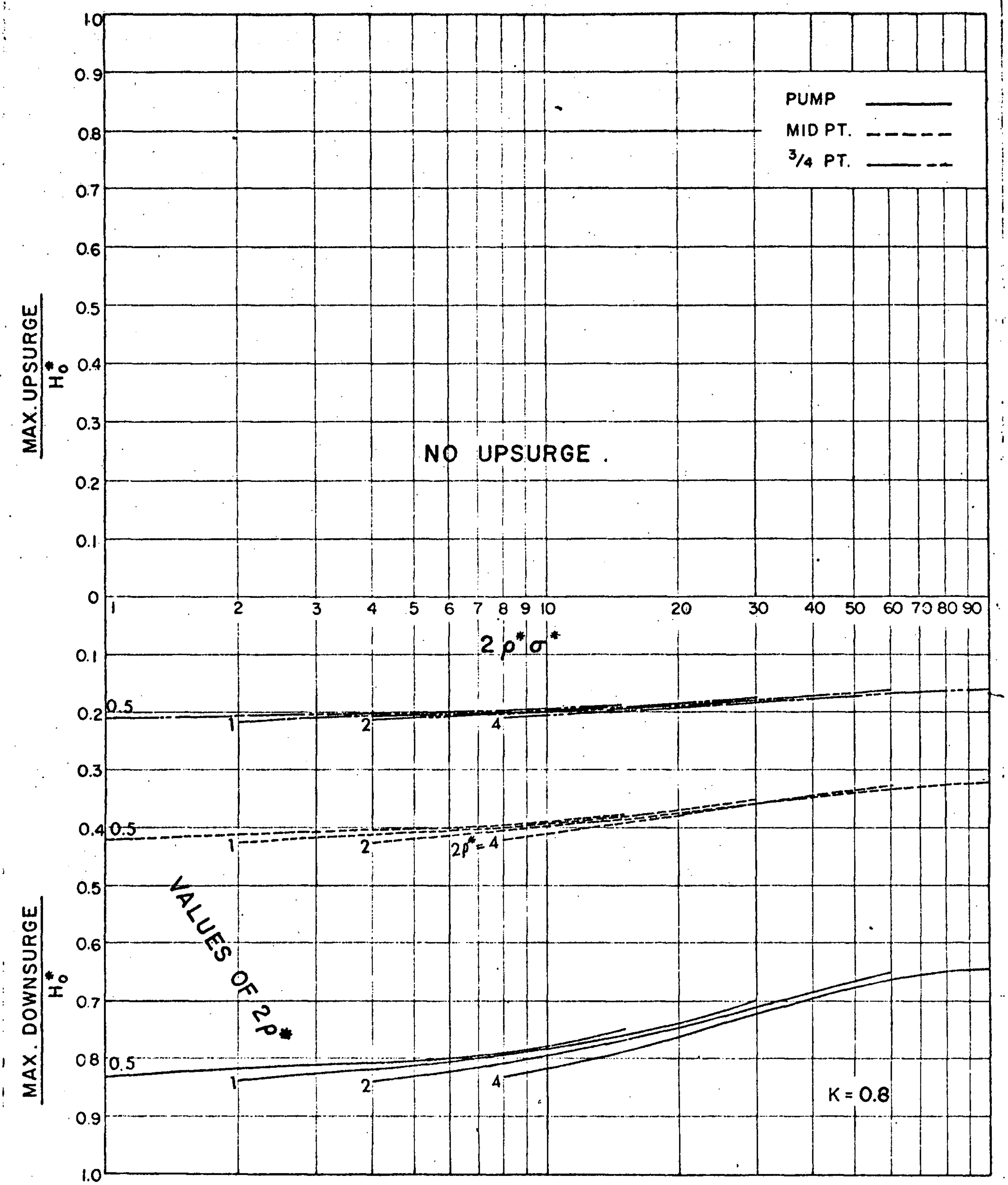


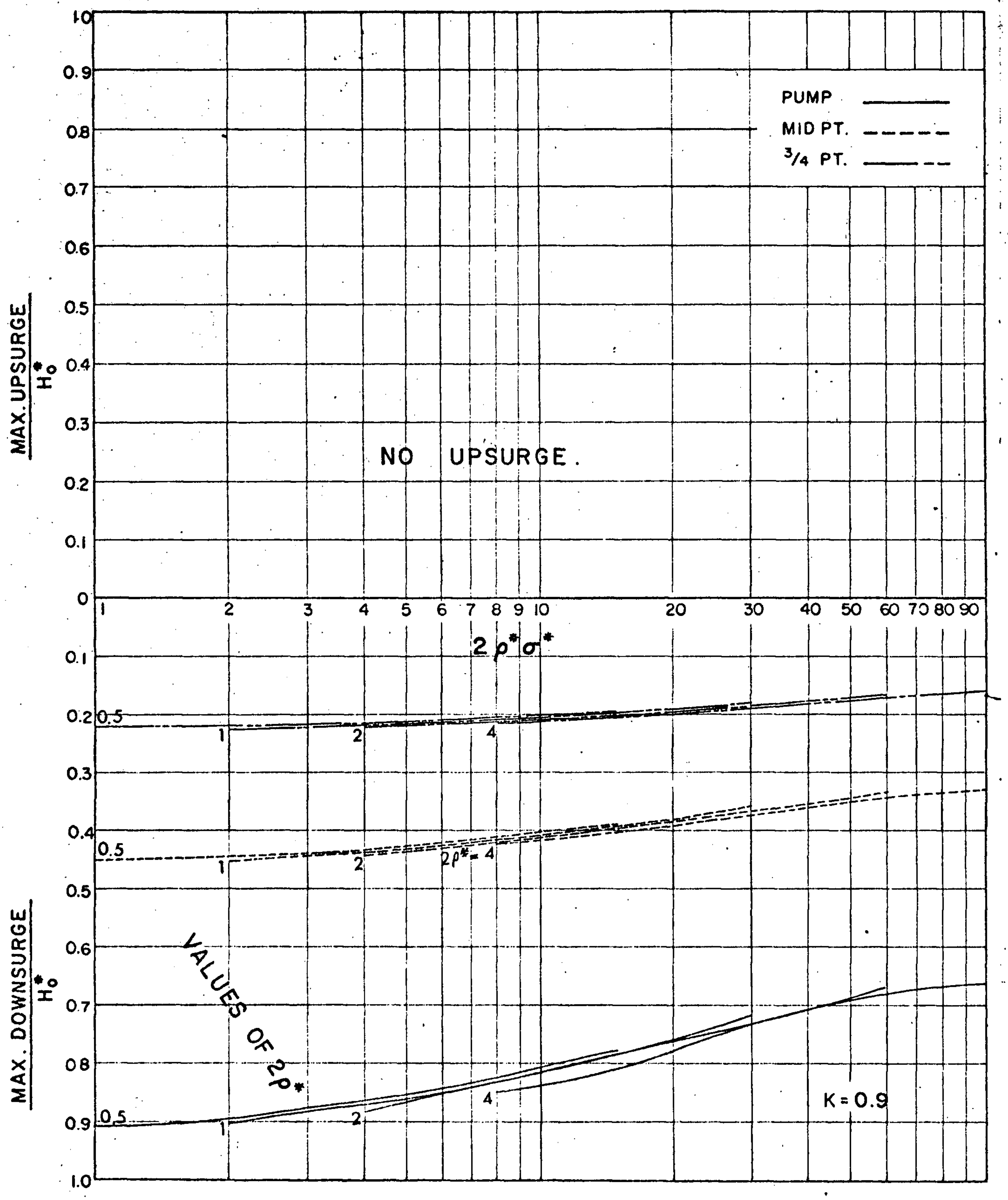
PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
NO UPSURGE.
$2\rho^*\sigma^*$
1 2 3 4 5 6 7 8 9 10 20 30 40 50 60 70 80 90
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0
MAX. DOWNSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 4$
0.5
1
2
4
K = 0.5

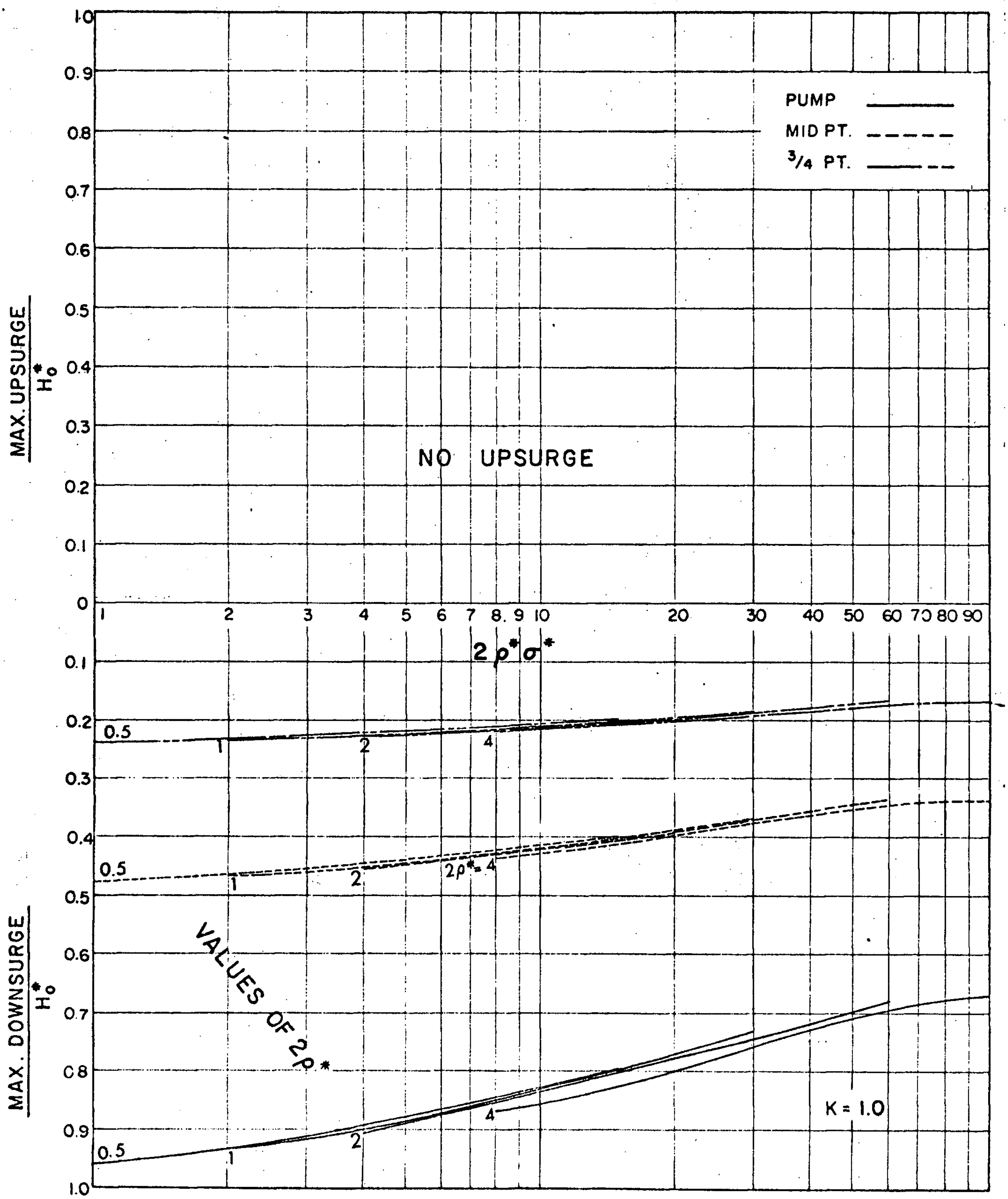


PUMP
MID PT.
3/4 PT.
NO UPSURGE.
MAX. UPSURGE / H*o
MAX. DOWNSURGE / H*o
2ρ*σ*
VALUES OF 2ρ*
2ρ* = 4
K = 0.6


PUMP
MID PT.
3/4 PT.
NO UPSURGE.
MAX. UPSURGE / H*o
MAX. DOWNSURGE / H*o
2ρ*σ*
VALUES OF 2ρ*
2ρ* = 4
K = 0.7
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1 2 3 4 5 6 7 8 9 10 20 30 40 50 60 70 80 90

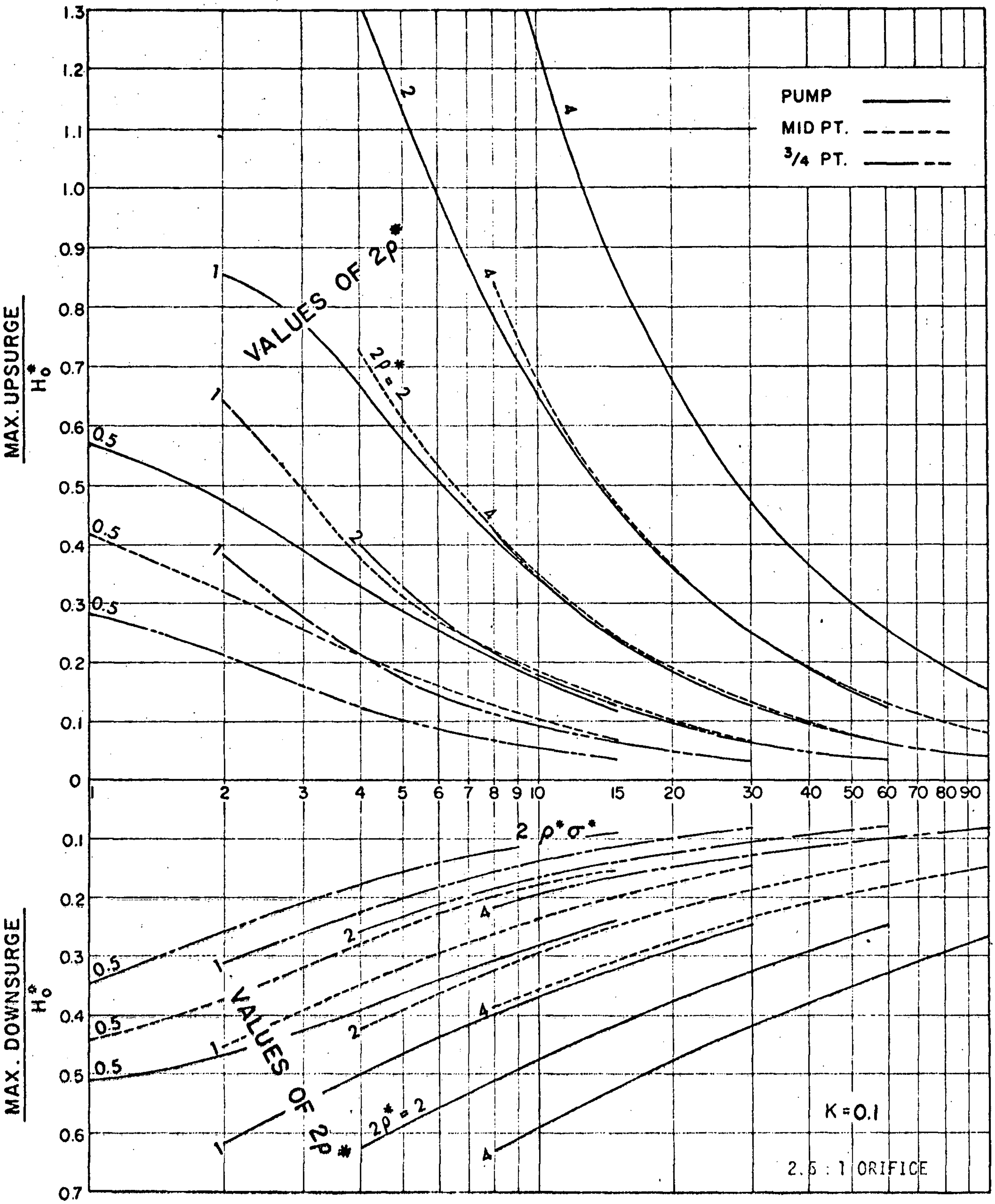

PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
NO UPSURGE .
$2\rho^*\sigma^*$
VALUES OF $2\rho^*$
$2\rho^* = 4$
K = 0.8

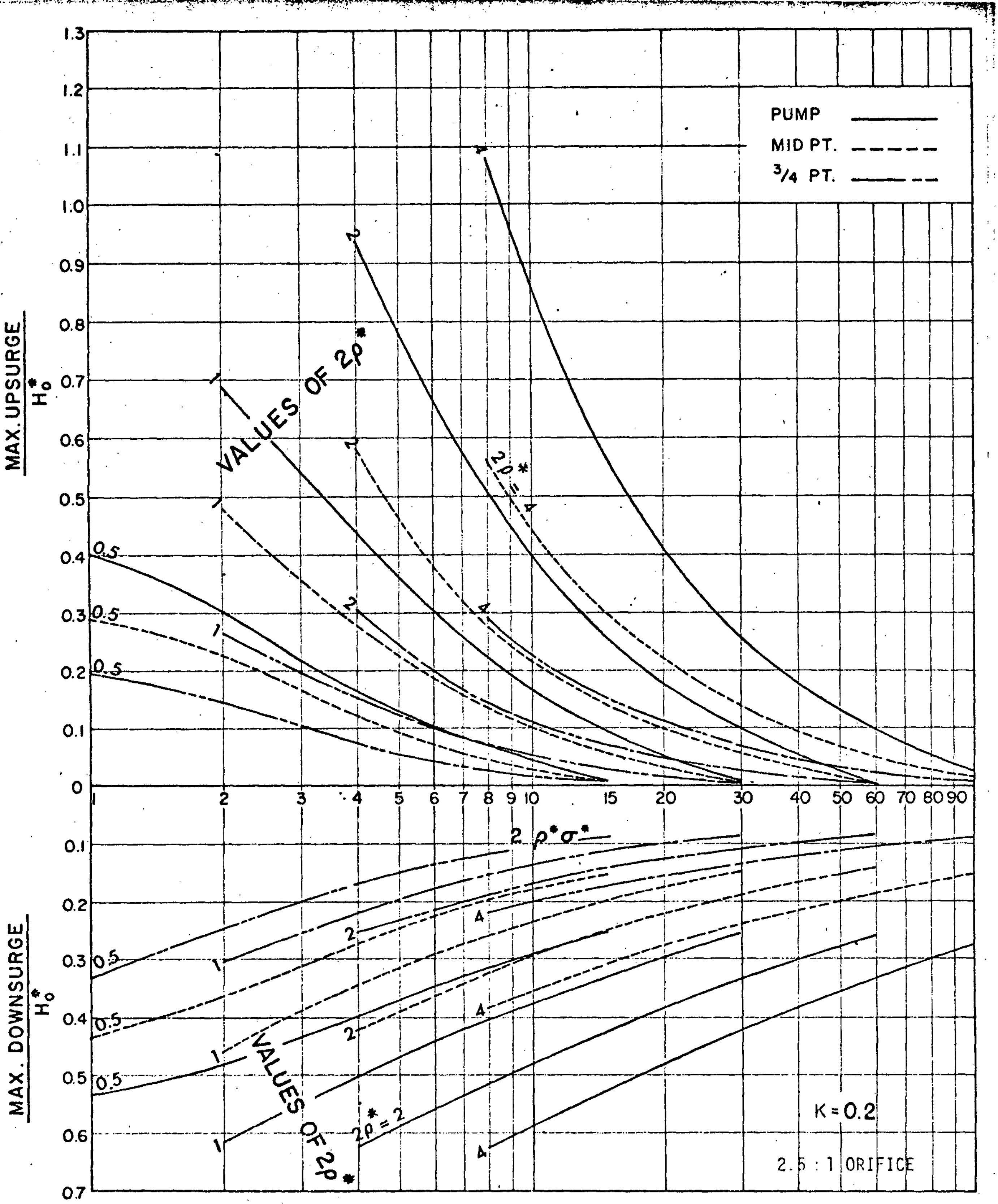

PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H*o
NO UPSURGE.
2ρ*σ*
MAX. DOWNSURGE / H*o
VALUES OF 2ρ*
2ρ* = 4
K = 0.9
0.5
1
2
4
0
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1 2 3 4 5 6 7 8 9 10 20 30 40 50 60 70 80 90

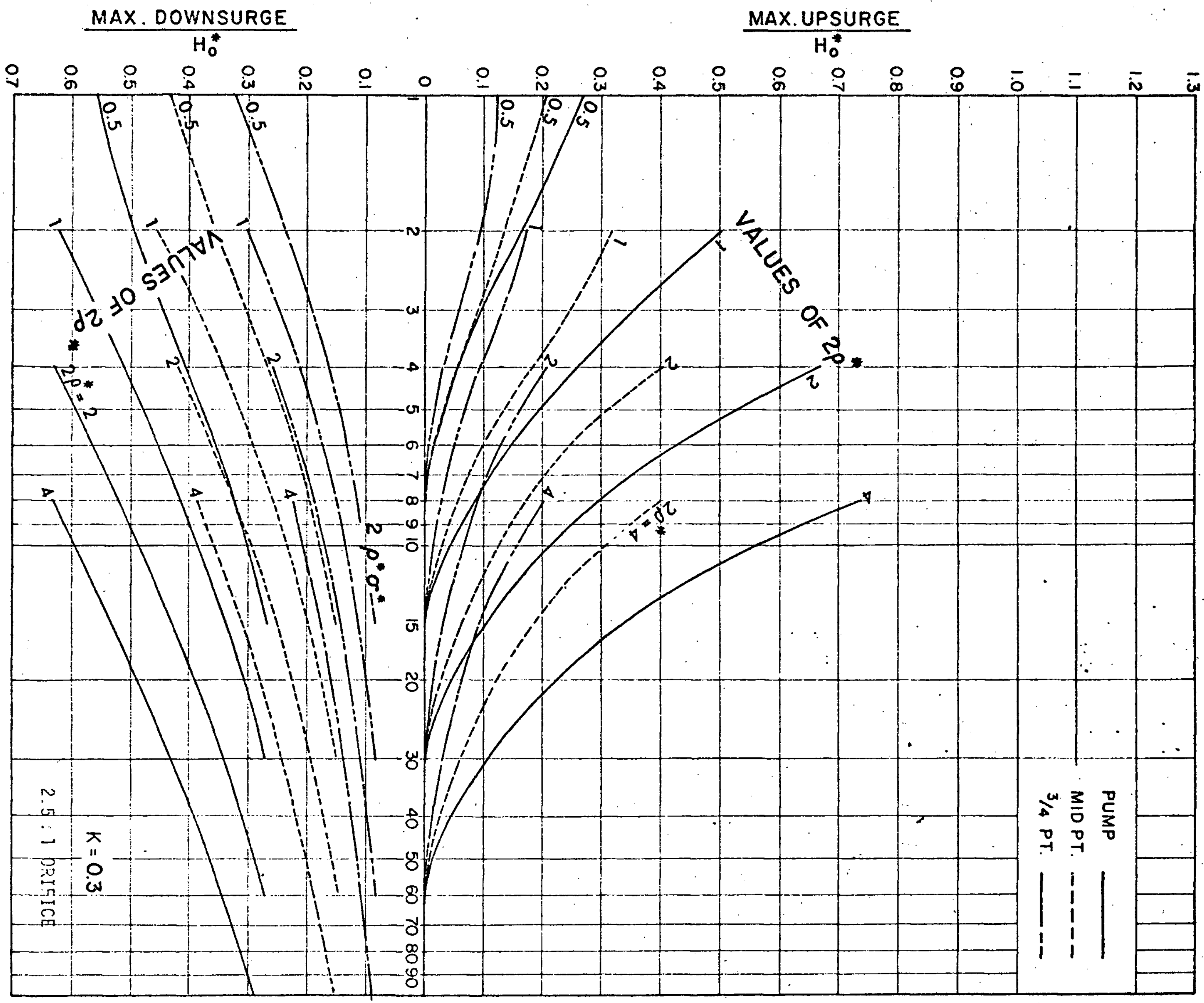

PUMP
MID PT.
3/4 PT.
NO UPSURGE
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
$2\rho^*\sigma^*$
VALUES OF $2\rho^*$
$2\rho^* = 4$
K = 1.0
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10
20
30
40
50
60
70
80
90

GROUP IV

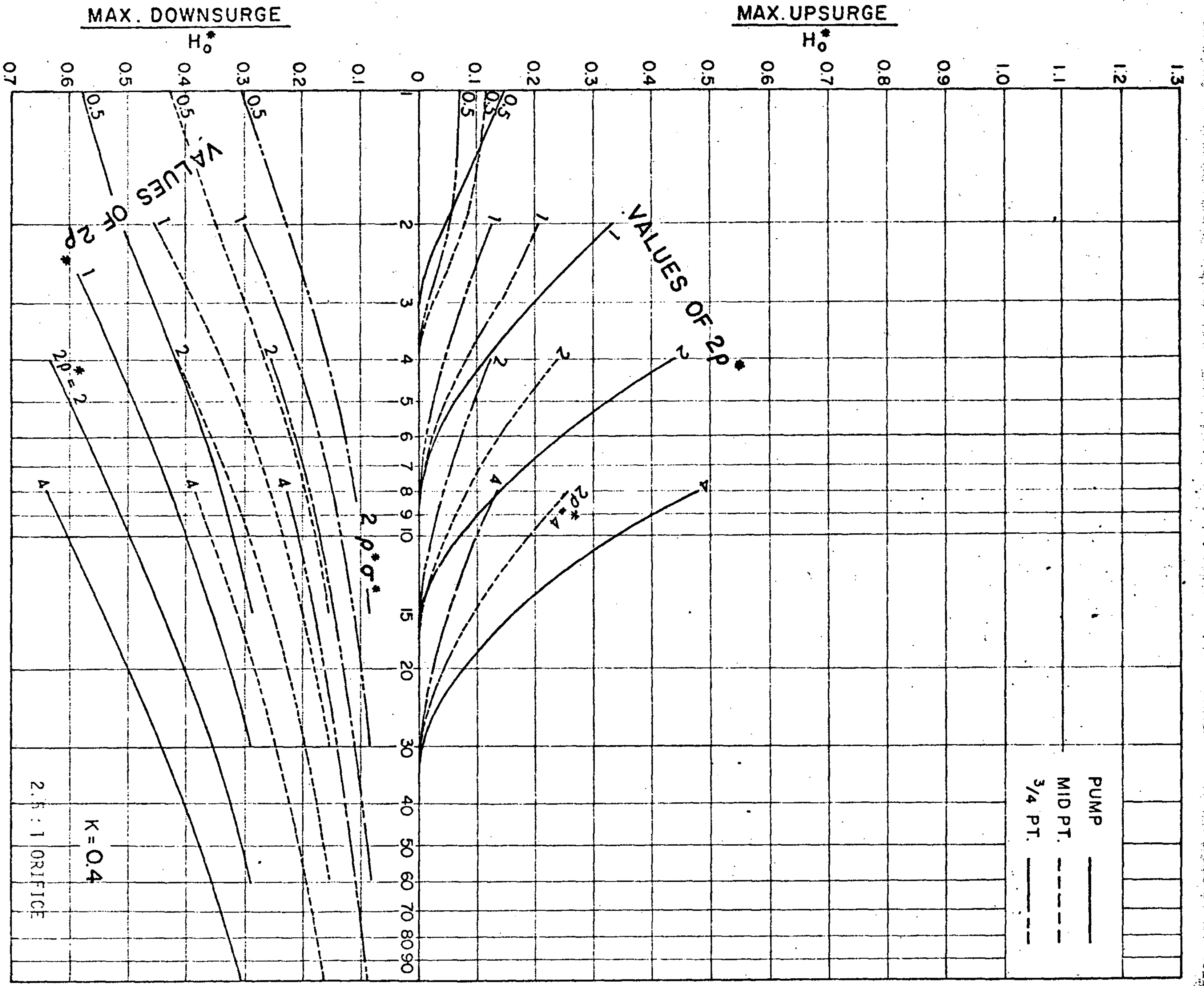
HEAD LOSS EQUALLY DIVIDED BETWEEN UNIFORMLY DISTRIBUTED WALL FRICTION AND ORIFICE LOSS

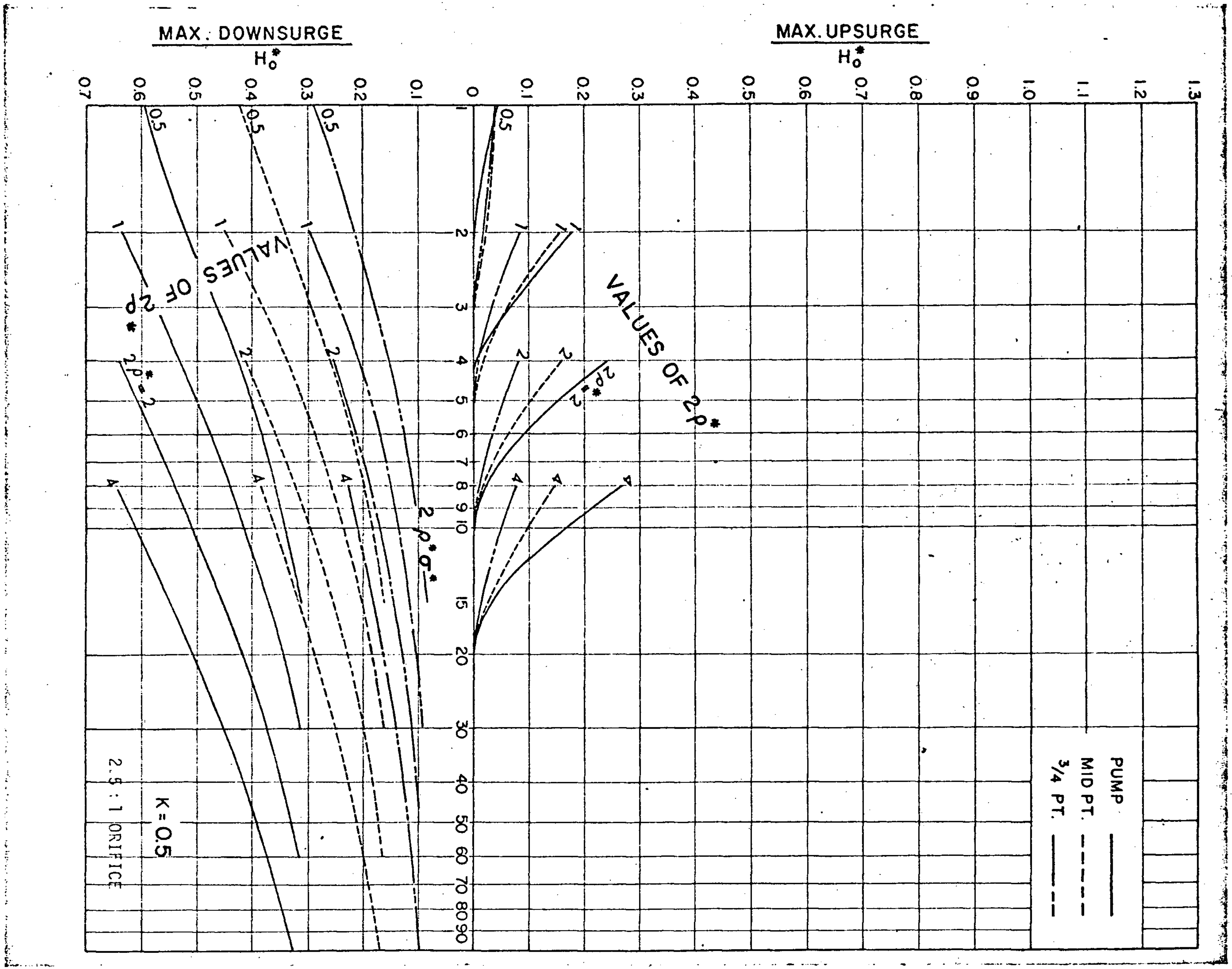

PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^*\sigma^*$
K = 0.1
2.5 : 1 ORIFICE

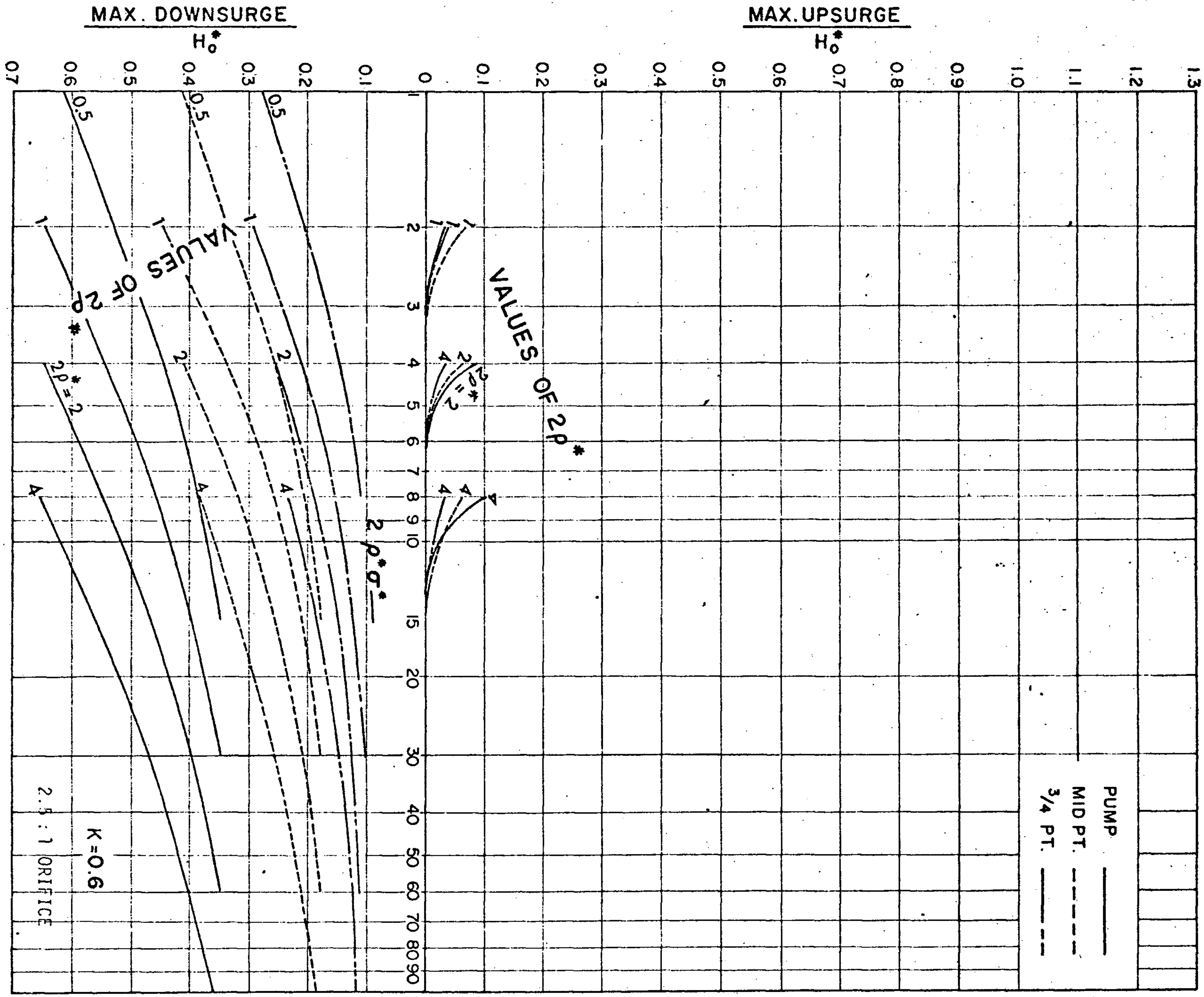

PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^* = 4$
$2\rho^* = 2$
$2\rho^*\sigma^*$
K = 0.2
2.5 : 1 ORIFICE



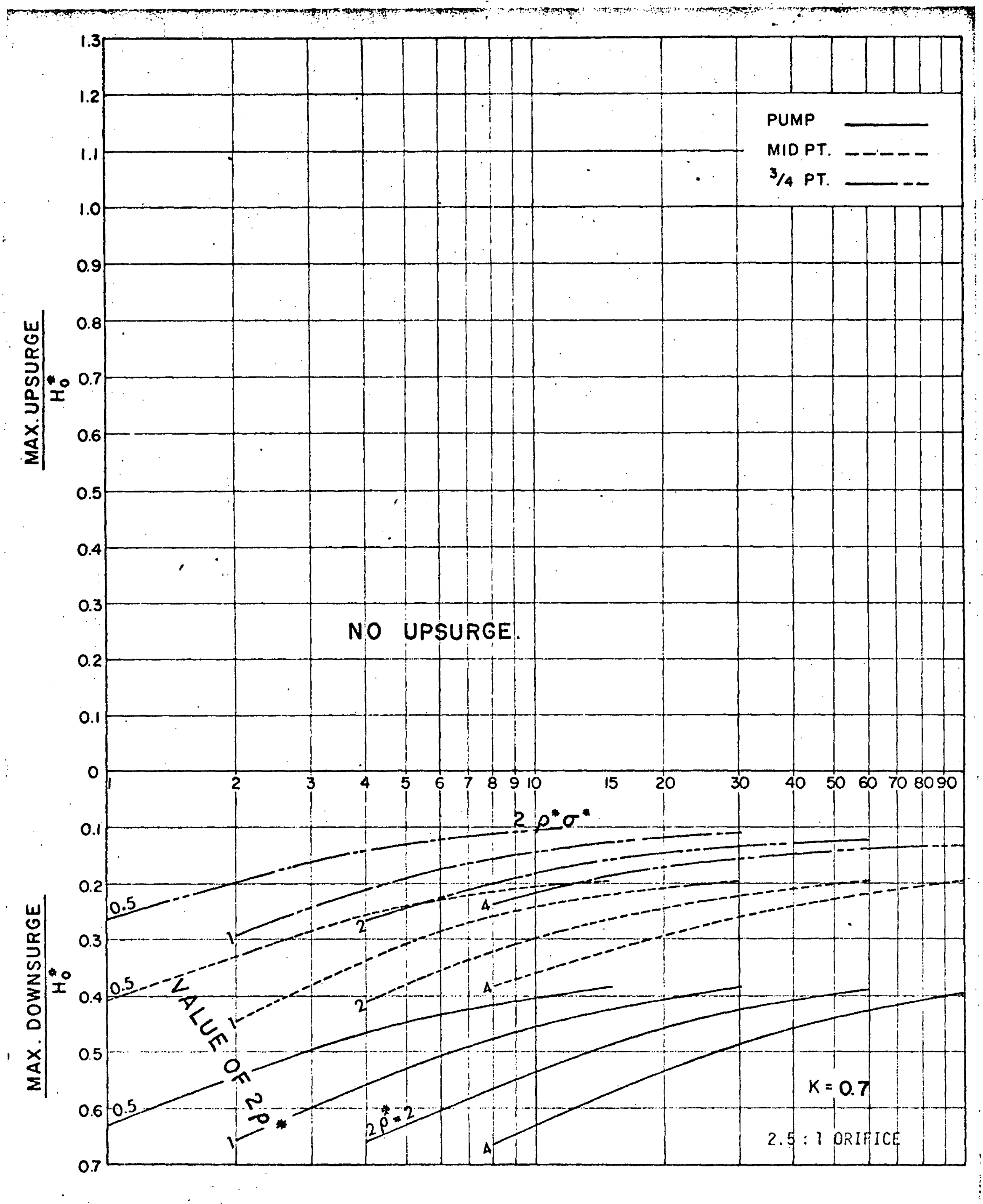
MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^* \sigma^*$
K = 0.3
2.5 : 1 ORIFICE
PUMP
MID PT.
3/4 PT.

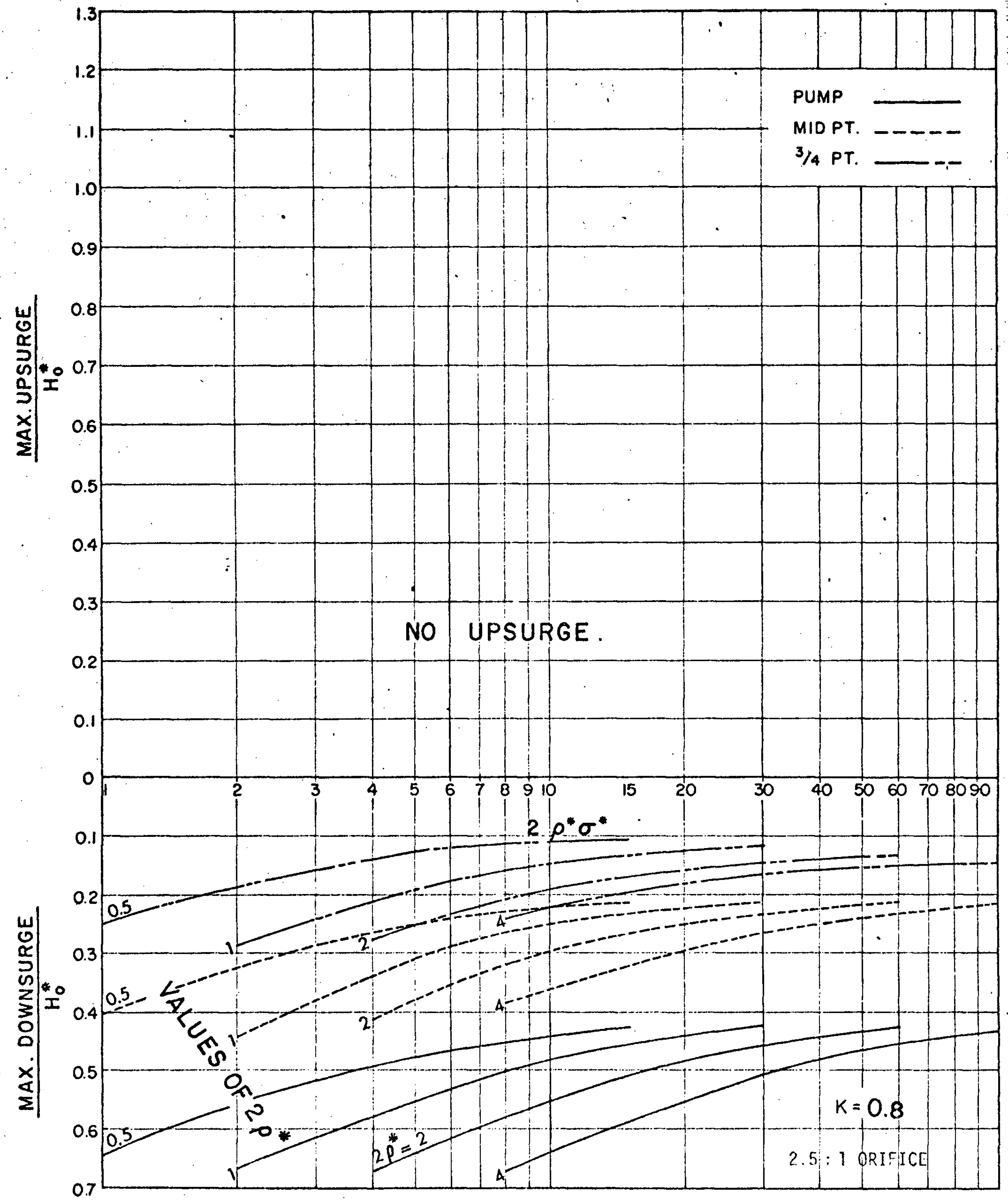

MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^* = 2$
$2\rho^* = 4$
$2\rho^*\sigma^*$
PUMP
MID PT.
3/4 PT.
K = 0.4
2.5 : 1 ORIFICE

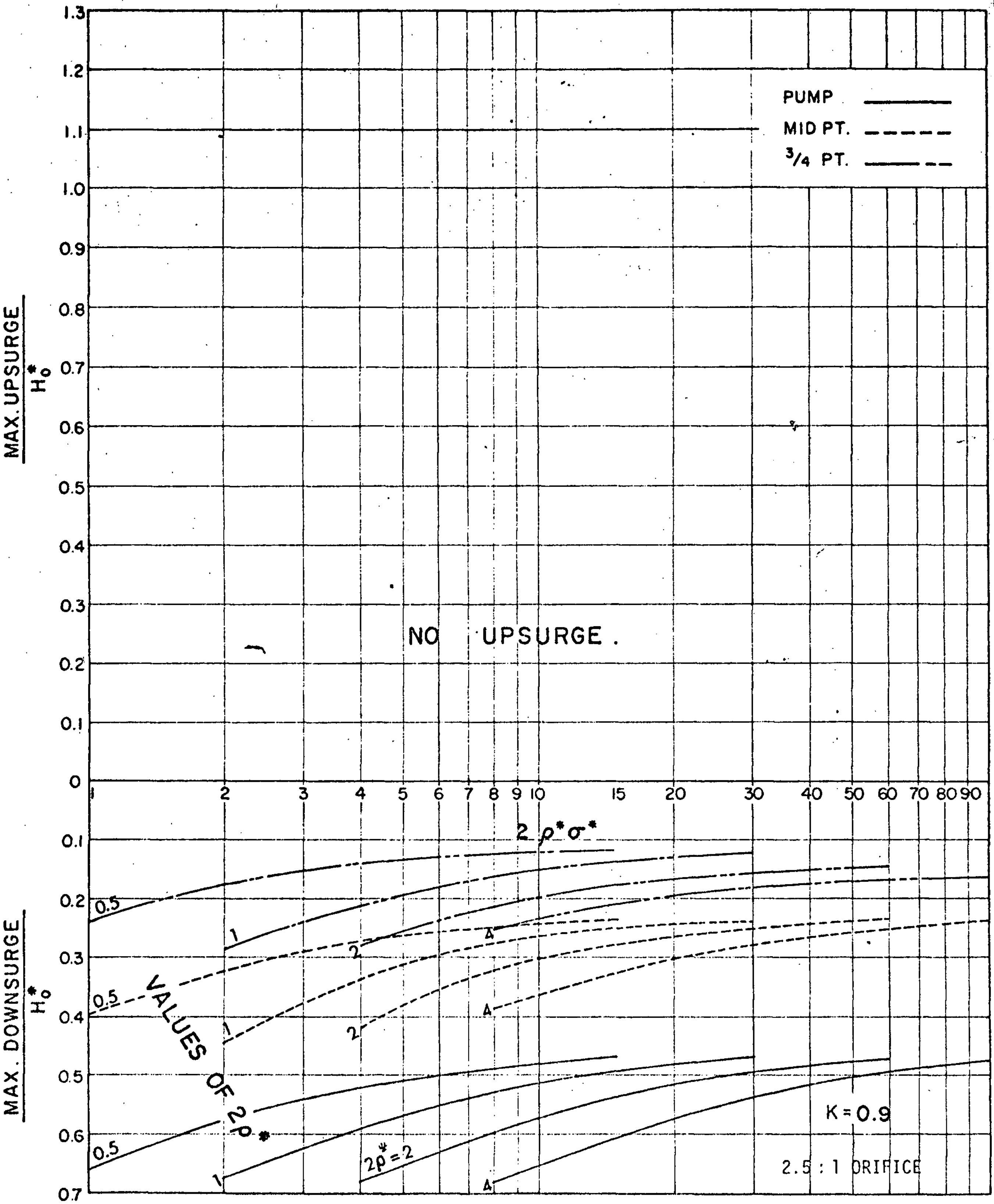

MAX. DOWNSURGE
H*o
MAX. UPSURGE
H*o
VALUES OF 2ρ*
2ρ*σ*
PUMP
MID PT.
3/4 PT.
K=0.5
2.5:1 ORIFICE

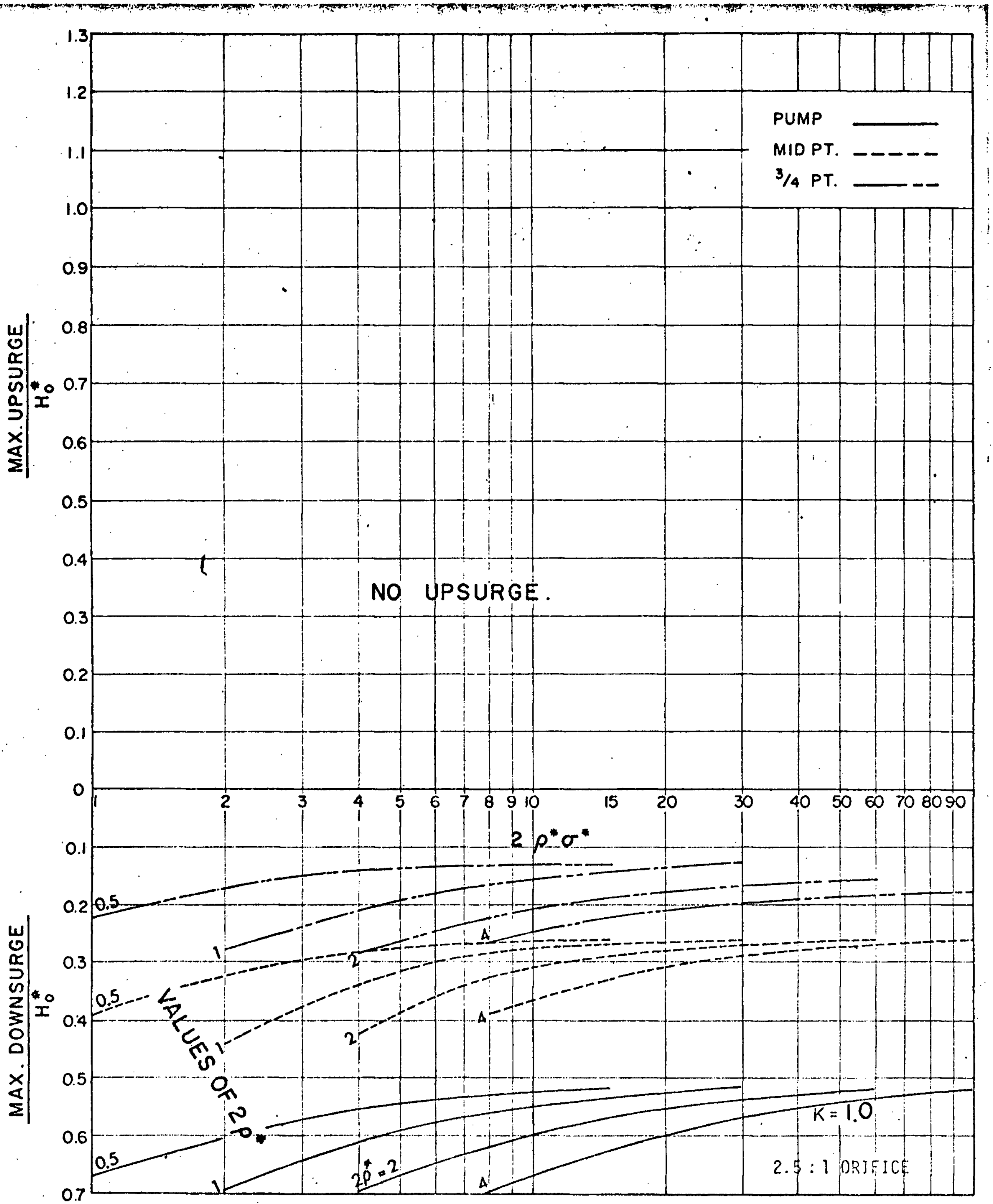


MAX. DOWNSURGE
H_o^*
MAX. UPSURGE
H_o^*
VALUES OF $2\rho^*$
VALUES OF $2\rho^*$
$2\rho^*\sigma^*$
$2\rho^* = 2$
$2\rho^* = 2$
K=0.6
2.5 : 1 ORIFICE
PUMP
MID PT.
3/4 PT.


PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
NO UPSURGE
MAX. DOWNSURGE / H_o^*
$2\rho^*\sigma^*$
VALUE OF $2\rho^*$
$2\rho^* = 2$
K = 0.7
2.5 : 1 ORIFICE


MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
PUMP
MID PT.
3/4 PT.
NO UPSURGE.
$2\rho^*\sigma^*$
VALUES OF $2\rho^*$
$2\rho^* = 2$
K = 0.8
2.5 : 1 ORIFICE


MAX. UPSURGE
H_o^*
MAX. DOWNSURGE
H_o^*
PUMP
MID PT.
3/4 PT.
NO UPSURGE.
$2\rho^*\sigma^*$
VALUES OF $2\rho^*$
$2\rho^*=2$
K=0.9
2.5 : 1 ORIFICE


PUMP
MID PT.
3/4 PT.
MAX. UPSURGE / H_o^*
NO UPSURGE.
$2\rho^*\sigma^*$
MAX. DOWNSURGE / H_o^*
VALUES OF $2\rho^*$
$2\rho^*=2$
K = 1.0
2.5 : 1 ORIFICE

APPENDIX A

COMPARISON OF CHARTS AND NUMERICAL EXAMPLES

A-1 Comparison of the charts derived by the method of characteristics and those produced by Evans and Crawford.

A-2 Example on the design of an air chamber for a short pipeline of large diameter.

A-3 Example on checking the maximum upsurges and downsurges for a long pipeline.

APPENDIX A-1

COMPARISON OF THE CHARTS DERIVED BY THE METHOD OF CHARACTERISTICS AND THOSE PRODUCED BY EVANS AND CRAWFORD

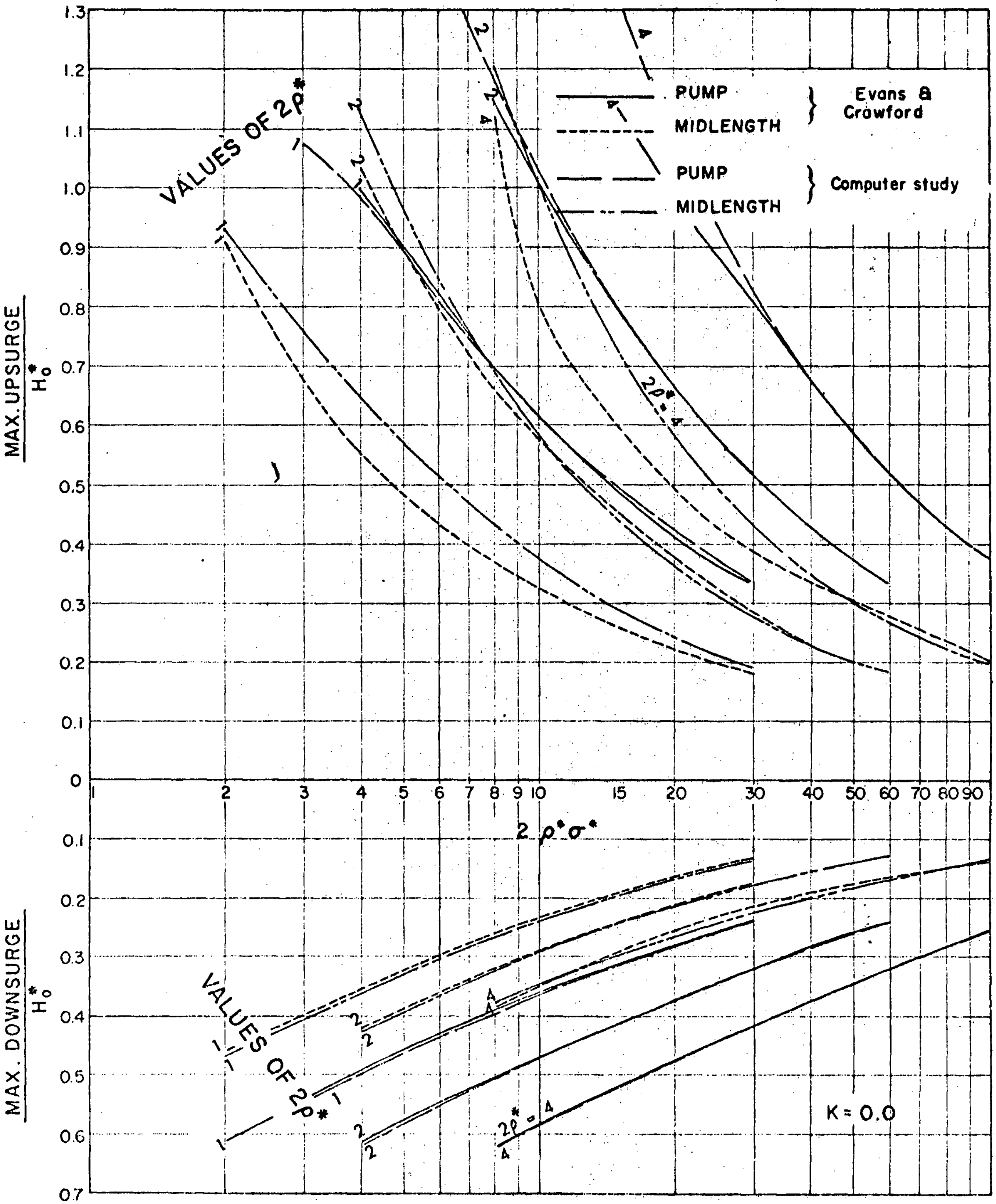

PUMP
MIDLENGTH
Evans & Crawford
PUMP
MIDLENGTH
Computer study
VALUES OF 2ρ*
MAX. UPSURGE / H*o
MAX. DOWNSURGE / H*o
2 ρ*σ*
2ρ* = 4
K = 0.0
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1 2 3 4 5 6 7 8 9 10 15 20 30 40 50 60 70 80 90

FIG. A-1

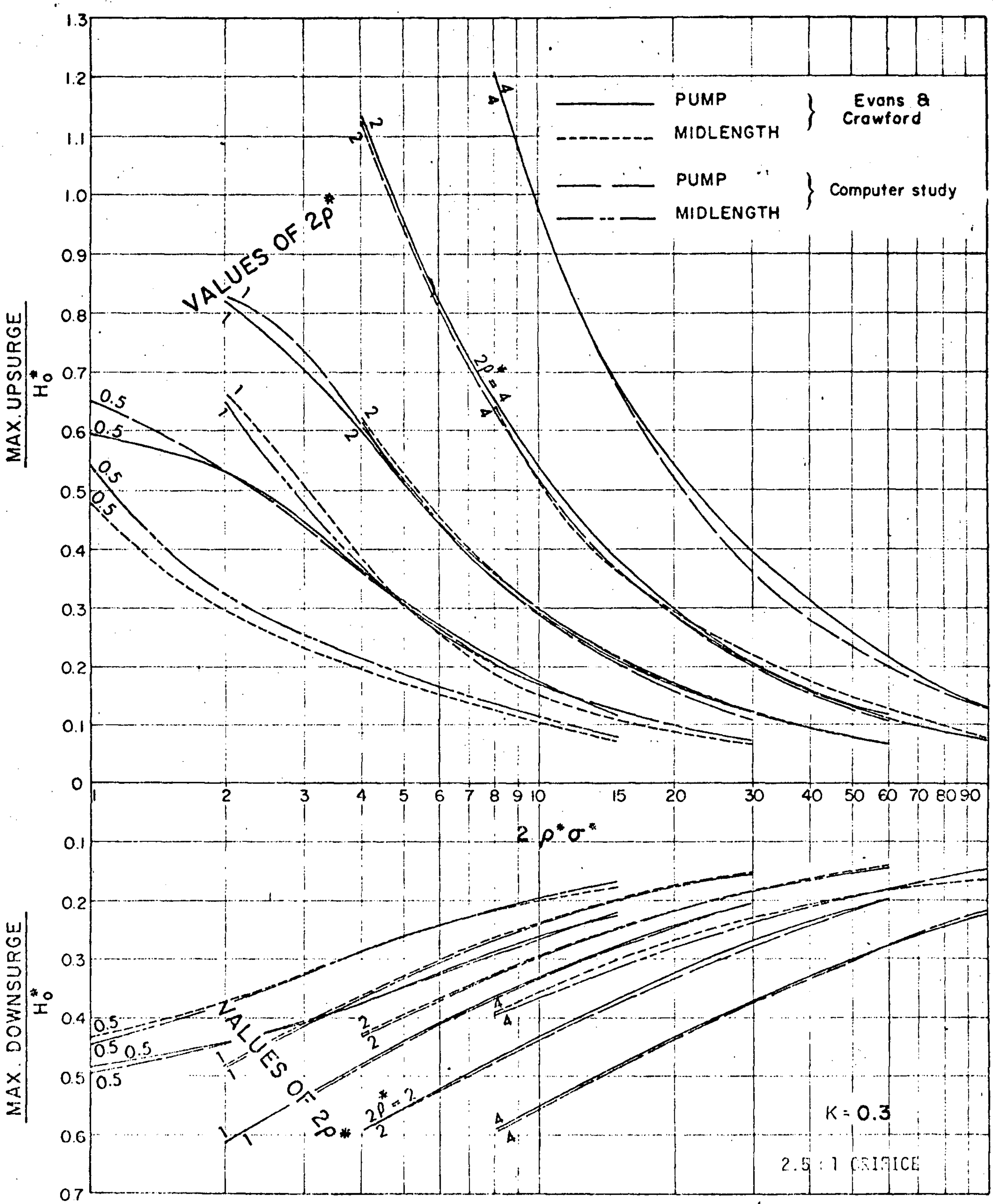
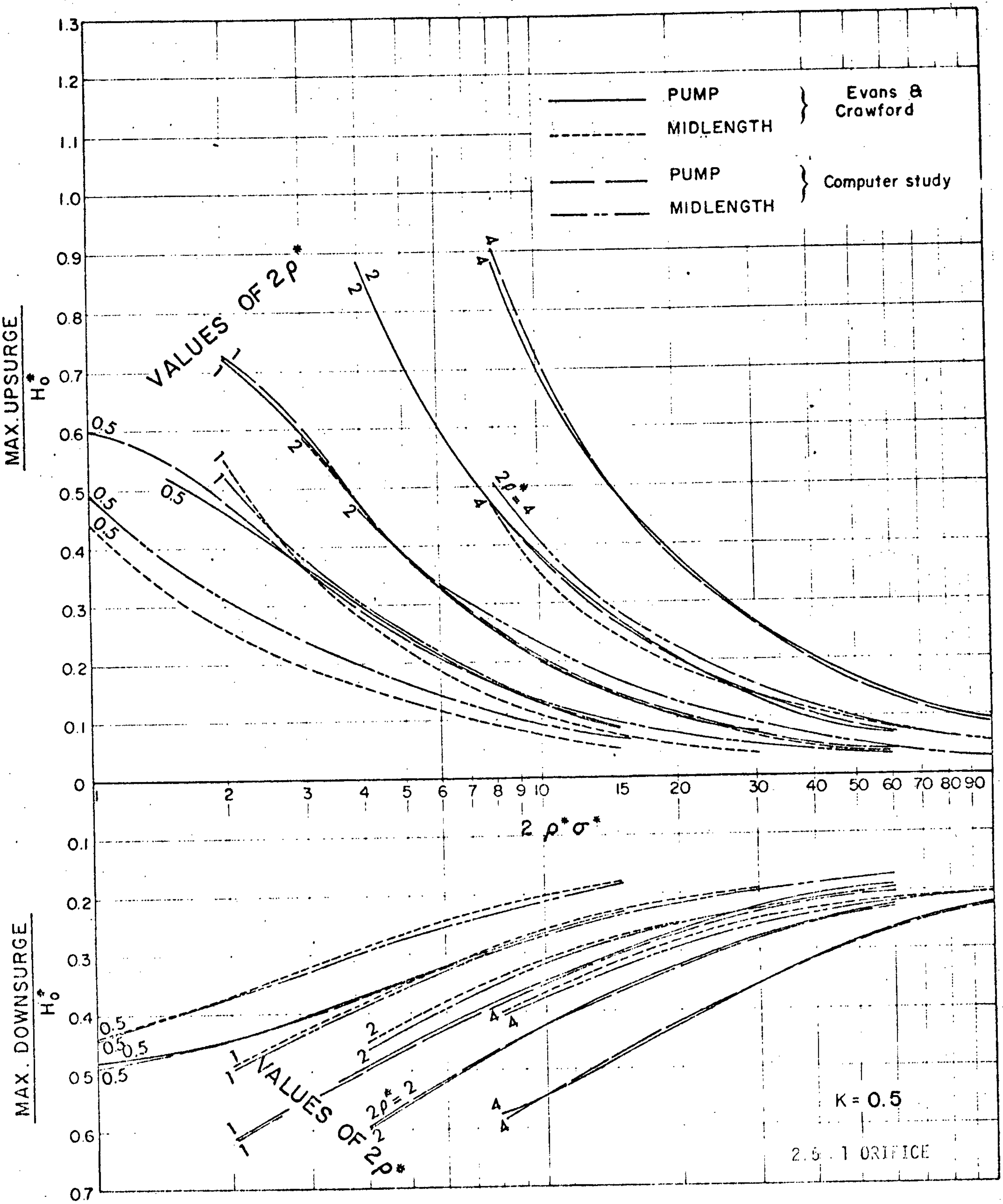

PUMP
MIDLENGTH
Evans & Crawford
PUMP
MIDLENGTH
Computer study
VALUES OF 2ρ*
MAX. UPSURGE / H_o^*
MAX. DOWNSURGE / H_o^*
2 ρ*σ*
K = 0.3
2.5 : 1 ORIFICE

FIG. A-2



PUMP
MIDLENGTH
Evans & Crawford
PUMP
MIDLENGTH
Computer study
VALUES OF 2ρ*
2ρ* = 4
2ρ* = 2
MAX. UPSURGE / H*o
MAX. DOWNSURGE / H*o
2 ρ* σ*
K = 0.5
2.5 : 1 ORIFICE

FIG. A-3

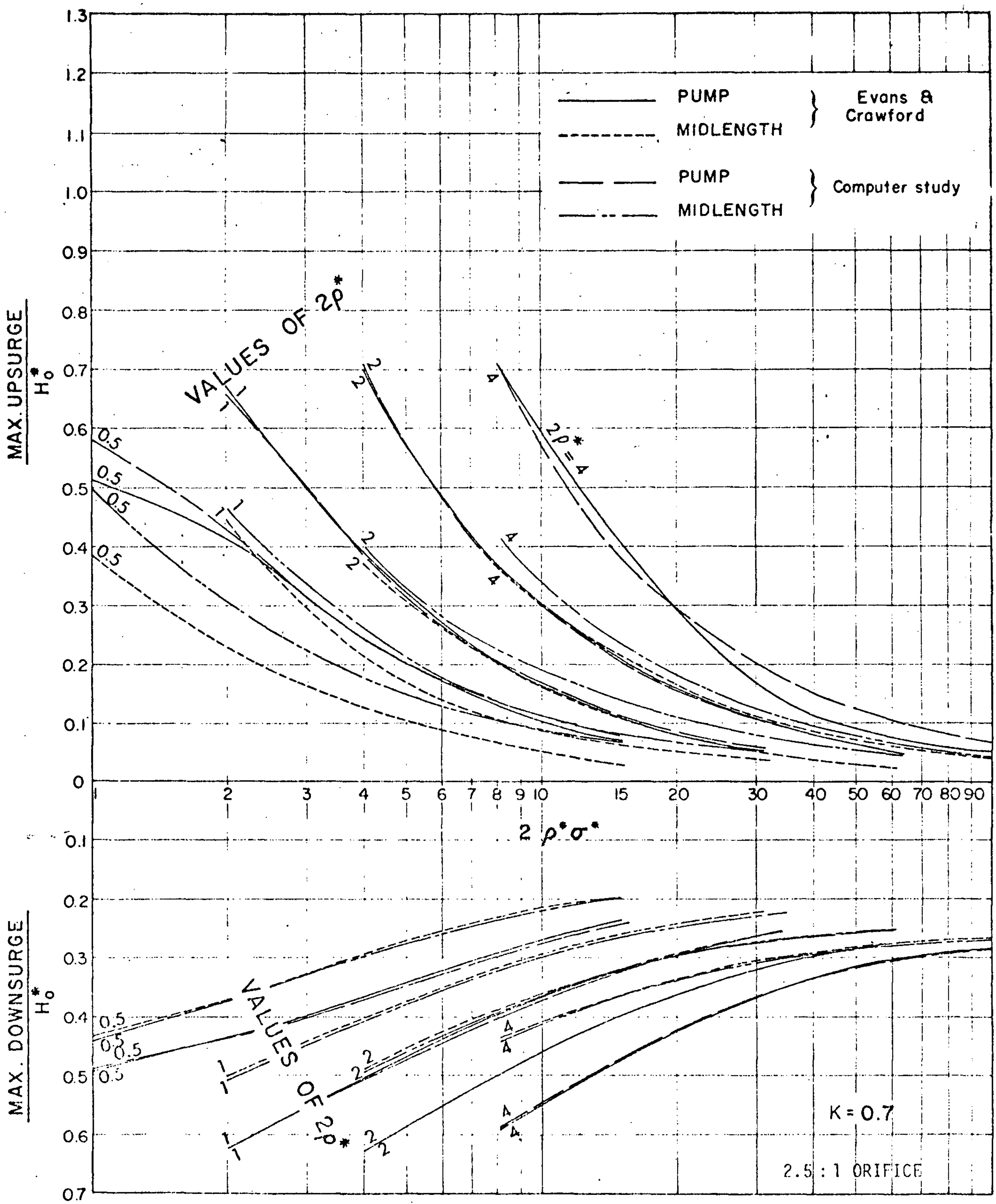

PUMP
MIDLENGTH
Evans & Crawford
PUMP
MIDLENGTH
Computer study
VALUES OF 2ρ*
2ρ* = 4
MAX. UPSURGE / H*o
MAX. DOWNSURGE / H*o
2 ρ* σ*
K = 0.7
2.5 : 1 ORIFICE
0.5
1
2
4
1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10
15
20
30
40
50
60
70
80
90

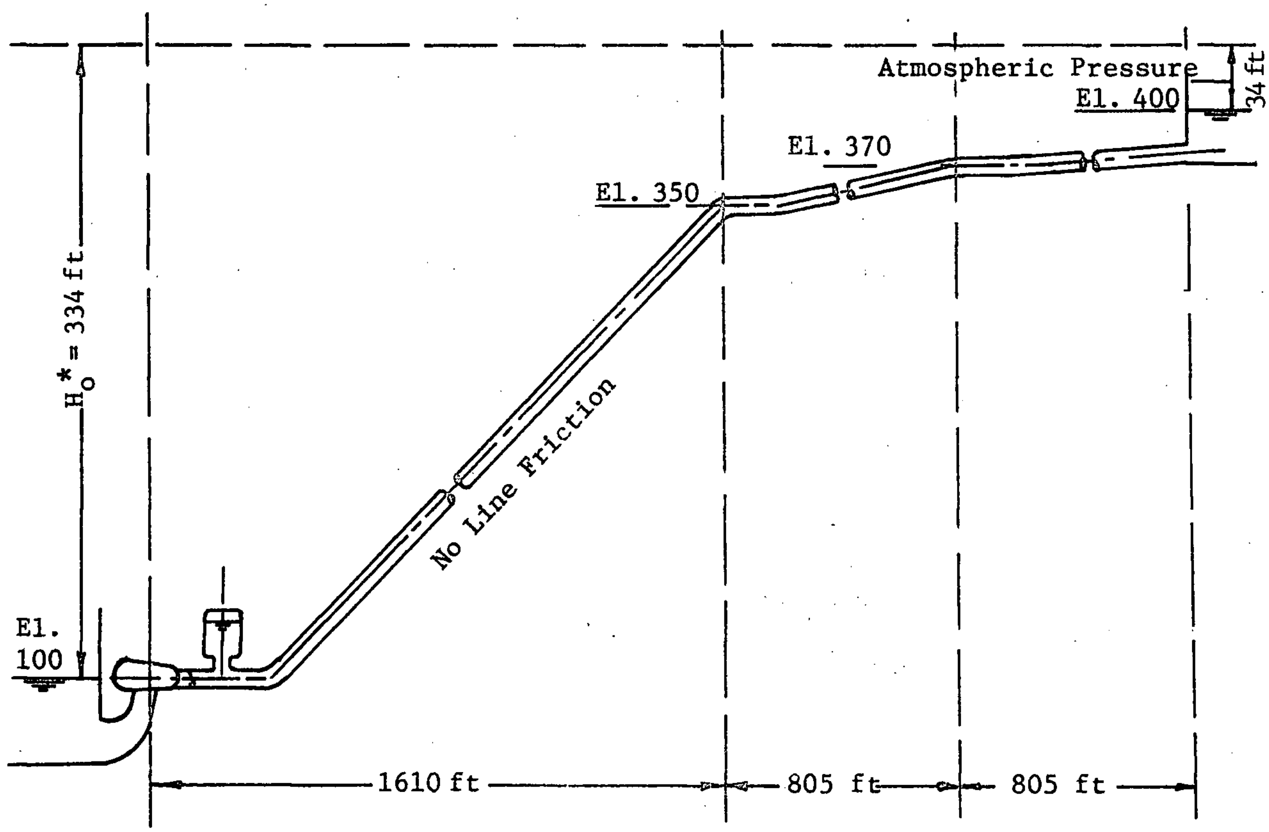
FIG. A-4

APPENDIX A-2

EXAMPLE ON DESIGN OF AN AIR CHAMBER FOR A SHORT PIPELINE OF LARGE DIAMETER

PROBLEM

Given the following data, design the most economical air chamber which will limit the waterhammer surges to the specified limits.



DATA

Check valve closes immediately on pump failure.

Length of pipeline (L) = 3220 feet.

Area of pipe (A) = 3.142 ft^2

Steady-state discharge (Q_o) = 18.5 cu.ft per sec.

Steady-state velocity (V_o) = 5.9 ft per sec.

Steady-state head at pump (H_o) = 300 ft

Water hammer wave velocity (a) = 3660 ft per sec.

Atmospheric pressure = 34.0 ft of water.

Neglect line friction losses.

ALLOWABLE HEADS

Maximum at pump = 400 ft of water.

Maximum negative heads at midlength and three-quarter point = 20 ft. of water (sub-atmospheric).

SOLUTION

The allowable surges are:

At pump - allowable upsurge = 400-300 = 0.30 H_o*

At midlength - allowable downsurge = 400-350+20 = 0.21 H_o*

At three-quarter point - allowable downsurge = 400-370+20 = 0.15 H_o*

$$2\rho^* = \frac{aVo}{gH_o^*} = \frac{(3660)(5.9)}{(32.2)(334)} = 2.0$$

From the charts in Group II, Entire Head Loss Concentrated at the Orifice, Differential Orifice 2.5:1, the surge conditions can be met using the values:

$K = 0.1, \quad 2\rho^* \sigma^* = 35$

$K = 0.2, \quad 2\rho^* \sigma^* = 24$

$K = 0.3, \quad 2\rho^* \sigma^* = 22$

$K = 0.4, \quad 2\rho^* \sigma^* = 60$

The volume of air in the chamber will vary directly as σ^*, so the smallest value of $2\rho^* \sigma^*$ will be used. For the values,

$K = 0.3$

$2\rho^* \sigma^* = 22$

$2\rho^* = 2.0$

At the pump: Maximum upsurge = 0.26 H_o*

Maximum downsurge = 0.32 H_o*

At the midlength: Maximum upsurge = 0.155 H_o*

Maximum downsurge = 0.21 H_o*

At the three-quarter point: Maximum upsurge = 0.07 H_o*

Maximum downsurge = 0.15 H_o*

The differential orifice should be designed to provide a head loss of (0.3)(334) = 100 ft for a flow of 18.5 cu.ft per sec. into the chamber.

From Eq. 1.8,

$$C_o = \frac{(2\rho^* \sigma^*)\ ALV_o}{2a}$$

$$= \frac{(22)\ (3.142)(3220)(5.9)}{(2)(3660)}$$

$$= 179 \text{ cu.ft}$$

$$C' = C_o = 179 \text{ cu.ft}$$

Assume: Volume between upper and lower emergency levels is 20% of C'.

Then, C" = 1.20 C' = 215 cu.ft

and $2\rho^* \sigma^* = 1.2 \times 22 = 26.4$.

The maximum downsurge at the pump becomes 0.295 H_o*.

$$\text{Total air chamber volume} = \frac{C''H_o^*}{H_o^* - \text{downsurge at pump}}$$

$$= \frac{215}{1-.295}$$

$$= \underline{305 \text{ cu.ft}}.$$

REMARKS

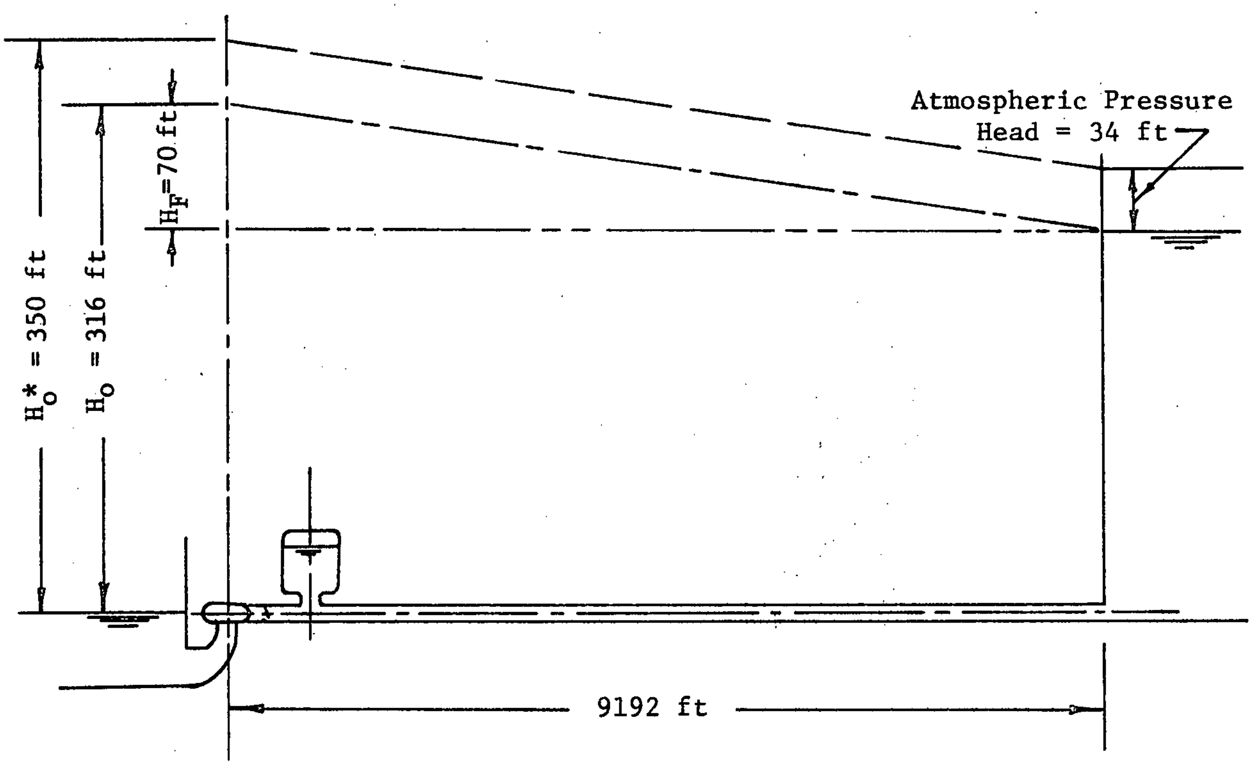
The critical points with respect to water-column separation occur for this example at the midlength and three-quarter point. The design ensures that water-column separation will not occur. If the problem were one of analysis, the maximum downsurge at the critical points would be determined. A pressure of -34 ft or less would indicate the formation of a vacuous space.

APPENDIX A-3

EXAMPLE ON CHECKING THE MAXIMUM UPSURGES
AND DOWNSURGES FOR A LONG PIPELINE

PROBLEM

Given the following data, determine the maximum upsurges and downsurges at the pump, the midlength and the three-quarter point of the pipeline.



DATA

Check valve closes immediately on pump failure.

Length of pipeline (L) = 9192 ft

Area of pipe (A) = 0.79 ft^2

Steady-state discharge (Q_o) = 5.0 ft^3/sec

Steady-state velocity (V_o) = 6.3 ft/sec

Steady-state head at pump (H_o) = 316 ft.

Line friction loss (H_F) = 70 ft.

Waterhammer wave velocity (a) = 3660 ft/sec.

Atmospheric pressure = 34.0 ft. of water.

Initial air volume in chamber (C_o) = 50 ft^3.

SOLUTION

(A) No orifice loss

$$2\rho^* \sigma^* = \frac{2C_o a}{ALV_o} = \frac{2(50)(3660)}{(0.73)(9192)(6.3)} = 8.0$$

$$K = \frac{70}{350} = 0.20$$

$$2\rho^* = \frac{aV_o}{gH_o^*} = \frac{(3660)(6.3)}{(32.2)(350)} = 2.04$$

From the charts in Group III, Entire Head Loss Attributable to Distributed Friction:

(1) At pump: Maximum upsurge = 0.285 H_o^*

Maximum downsurge = 0.55 H_o^*

(2) At midlength: Maximum upsurge = 0.15 H_o^*

Maximum downsurge = 0.32 H_o^*

(3) At three-quarter point: Maximum upsurge = 0.075 H_o^*

Maximum downsurge = 0.175 H_o^*

(B) Total head loss evenly divided between line friction loss and orifice loss (2.5:1 differential orifice).

(1) At pump: Maximum upsurge = 0.50 H_o^*

Maximum downsurge = 0.515 H_o^*

(2) At midlength: Maximum upsurge = 0.28 H_o*

Maximum downsurge = 0.32 H_o*

(3) At three-quarter point: Maximum upsurge = 0.14 H_o*

Maximum downsurge = 0.19 H_o*

REMARKS

It is obvious from the foregoing results that care must be exercised in selecting the charts to best approximate the actual physical condition. The surge results (especially the upsurges) vary considerably for different types of head loss.

APPENDIX - B

GRAPHICAL CHECKS ON PROGRAM

B-1 Check for total head loss concentrated at the orifice.

B-2 Check for total head loss attributable to distributed friction.

APPENDIX B-1

CHECK FOR TOTAL HEAD LOSS CONCENTRATED AT THE ORIFICE

PROBLEM

Determine the transient state pressures and velocities in the pipeline adjacent to the pump at A. The transient conditions are caused by pump failure.

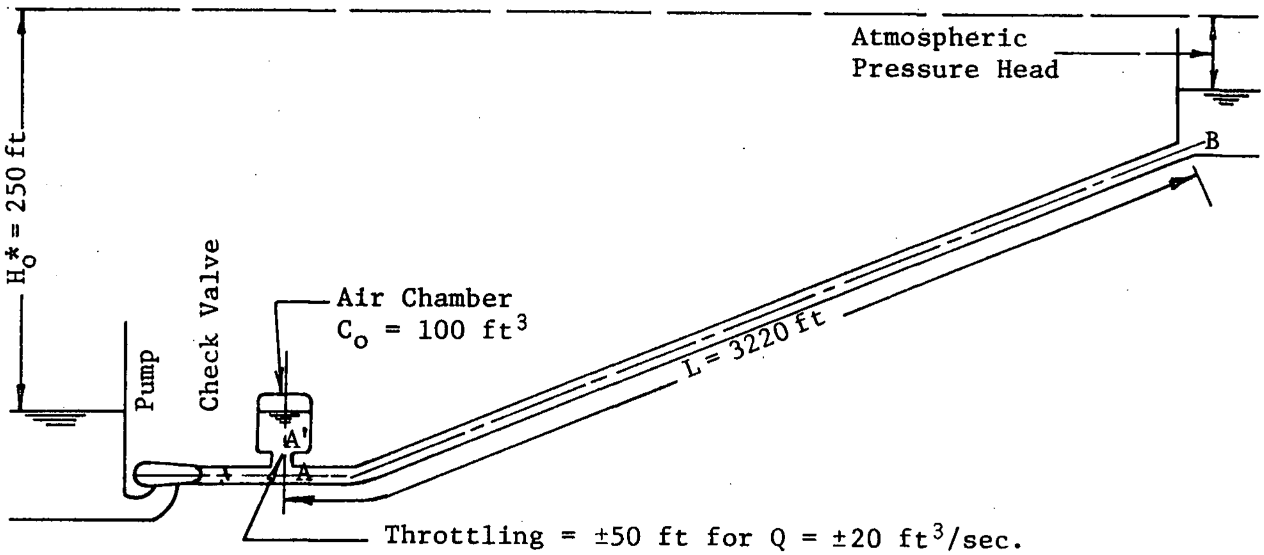


FIG. B-1a

DATA

Check valve closes immediately on pump failure.

Length of pipe line (L) = 3220 ft.

Steady-state discharge (Q_o) = 20.0 ft^3/sec.

Steady-state velocity (V_o) = 5.00 ft/sec.

Waterhammer wave velocity (a) = 3220 ft/sec.

Pipe line constant ($2\rho^*$) = 2.00

Constant for a pipe line having an air chamber

$$(2\rho^* \sigma^* = \frac{2C_o a}{Q_o L}) = 10.0$$

Atmospheric pressure = 34.0 ft of water.

Orifice throttling loss = ± 50 ft for $Q_o = \pm 20$ ft^3/sec.

Air expansion in the chamber is given by

$H^* \, v_{air}^{1.2}$ = a constant, in which H^* and v_{air}

are the absolute pressure and volume of air in the chamber.

Neglect line friction losses.

CHECK

Results obtained on the digital computer using the method of characteristics are close to those obtained by Parmakian[5] (page 135) by the graphical method (see Fig. B-1b).

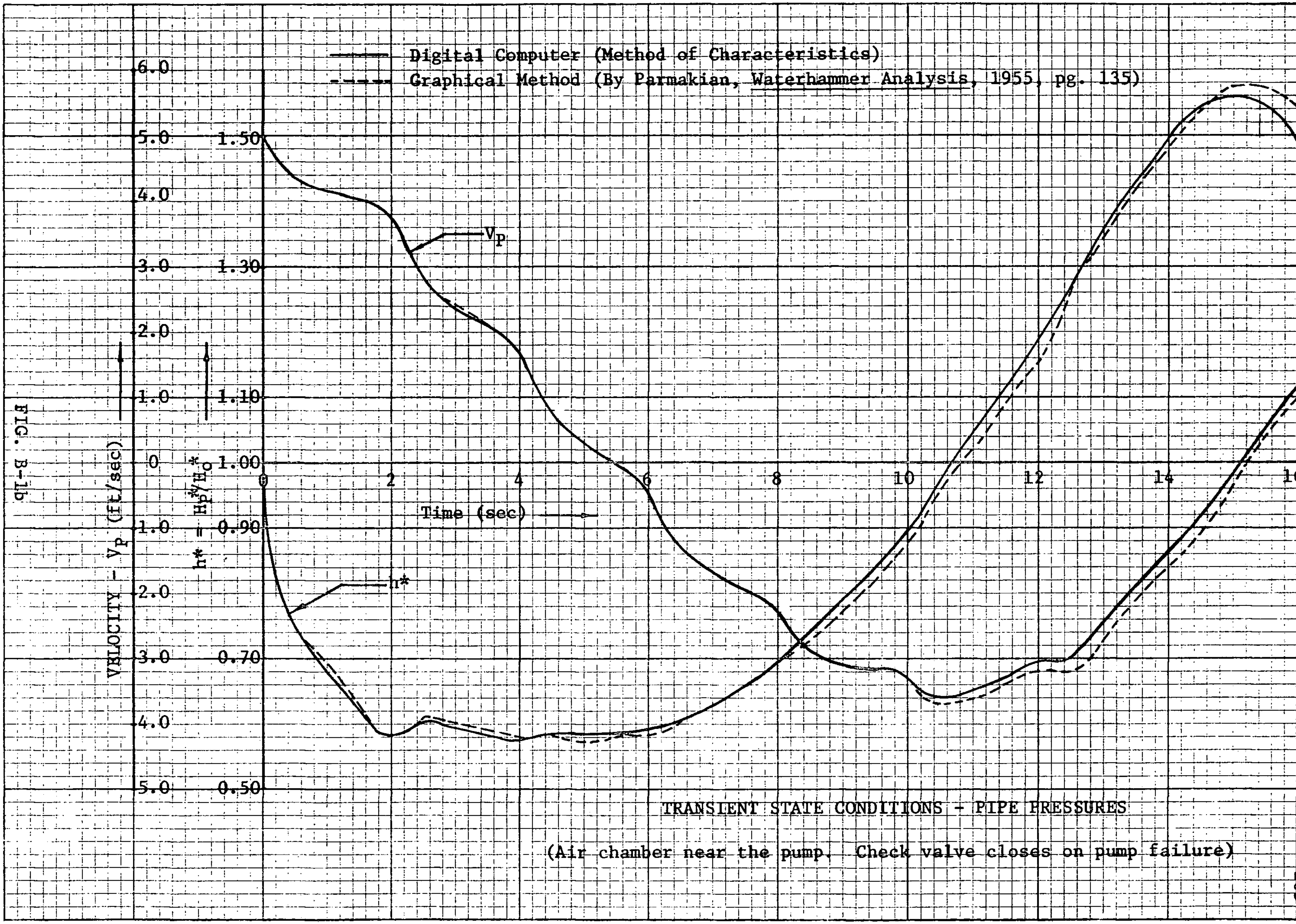

Digital Computer (Method of Characteristics)
Graphical Method (By Parmakian, Waterhammer Analysis, 1955, pg. 135)
VELOCITY - V_P (ft/sec)
$h^* = H_P^*/H_o^*$
Time (sec)
V_P
h*
6.0
5.0
4.0
3.0
2.0
1.0
0
1.0
2.0
3.0
4.0
5.0
1.50
1.30
1.10
1.00
0.90
0.70
0.50
0
2
4
6
8
10
12
14
16
TRANSIENT STATE CONDITIONS - PIPE PRESSURES
(Air chamber near the pump. Check valve closes on pump failure)

FIG. B-1b

APPENDIX B-2

CHECK FOR TOTAL HEAD LOSS ATTRIBUTABLE TO DISTRIBUTED FRICTION

PROBLEM

Determine the transient state pressures and velocities in the pipe line adjacent to the pump at B. The transient conditions are caused by pump failure.

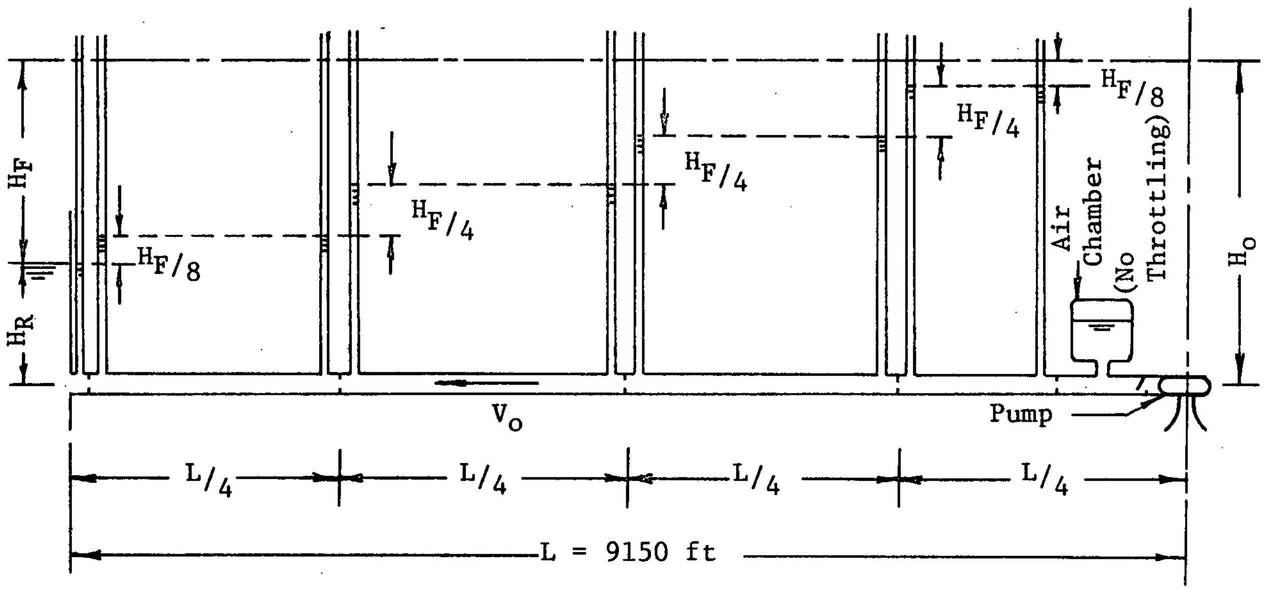


FIG. B-2a

DATA

(Water supply line for city of Trail).

Check valve closes immediately on pump failure.

Length of pipeline (L) = 9150 ft.

Steady-state discharge (Q_o) = 4.0 ft^3/sec.

Steady-state velocity (V_o) = 5.1 ft/sec.

Pipe line constant ($2\rho^*$) = 1.38

Constant for a pipe line having an air chamber ($2\rho^* \sigma^* = \frac{2C_o a}{Q_o L}$) = 5.0

Atmospheric pressure = 33.0 ft. of water

Total friction loss (H_F) = 90.0 ft. of water.

Pressure head at the pump (H_o) = 387.0 ft. of water

Steady-state volume of air in the chamber (C_o) = 25.0 ft^3

Air expansion in the chamber is given by $H^* C^{1.2}$ = a constant, in which H* and C are the absolute pressure and volume of air in the chamber. This may be written as $h^* c^{1.2} = 1$ where $h^* = \frac{H^*}{H_o}$ and $c = \frac{C}{C_o}$.

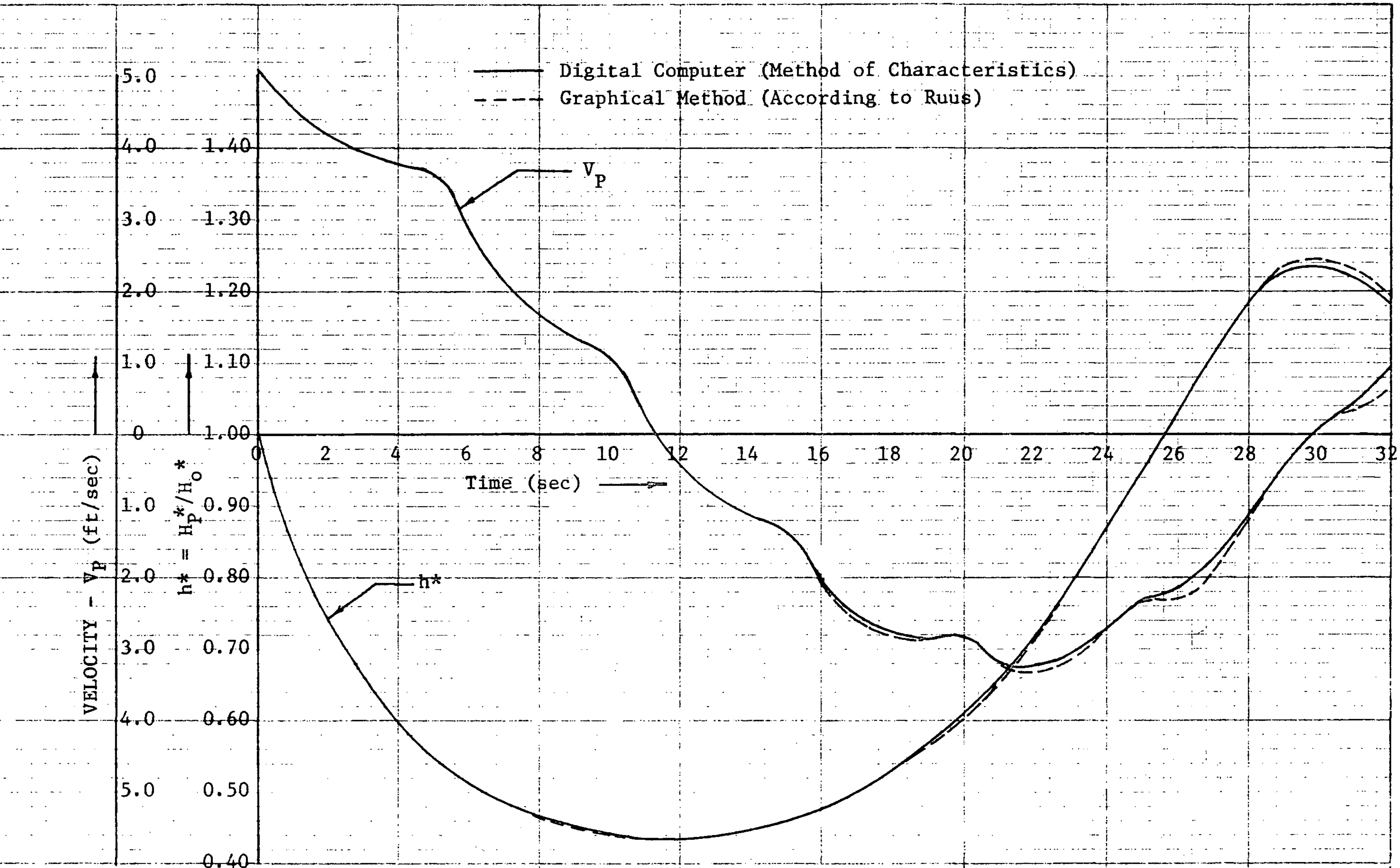
There is no loss for flow into or out of the chamber.

The friction loss in the pipe is considered concentrated at the orifices shown on the diagram.

CHECK

Results calculated on the digital computer using the method of characteristics are close to those (Fig. B-2b) obtained by Eugen Ruus, who analyzed this system by the method of graphical water hammer analysis concentrating the pipe line wall friction at five points as shown in Fig. B-2a.

FIG. B-2b



TRANSIENT STATE CONDITIONS - PIPE PRESSURES

(Air chamber near the pump. Check valve closes on pump failure)

APPENDIX - C

1. PROGRAM FOR THE ENTIRE HEAD LOSS CONCENTRATED AT THE ORIFICE
2. PROGRAM FOR THE ENTIRE HEAD LOSS ATTRIBUTABLE TO DISTRIBUTED FRICTION

1. ENTIRE HEAD LOSS CONCENTRATED AT THE ORIFICE

```
$LIST AIRCHAM10
C     WATERHAMMER PROGRAM. PUMP AT UPSTREAM END WITH AIR CHAMBER ADJACENT
C     TO THE PUMP.  RESERVOIR AT DOWNSTREAM END.
C     CHECK VALVE CLOSES IMMEDIATELY ON PUMP FAILURE.
C     NO LINE FRICTION. HEAD LOSS CONCENTRATED AT ORIFICE.
C     NO MINOR LOSSES.
         DIMENSION V(20),VP(20),H(20),HP(20),VR(20),VS(20),HR(20),HS(20),
        1HMAX(10),HMIN(10),HSS(10),SUMAX(10),SUMIN(10),SUMSS(10),
        2UPSMAX(10),DNSMAX(10),UPSANS(10),DNSANS(10)
         DATA N/10/,VA/3216./,G/32.16/,FL/3216./,PM/1.2/,MM/1/,
        1F/0.0/,CORFIN/2.5/,AP/3./,
        2CK/0.1/,
        3V0/3.5/
         WRITE(6,15) N,VA,G,FL,PM,MM,F,CORFIN,AP,CK,V0
      15 FORMAT(/'  THE PARAMETERS ARE NOW...'/
        1'   N= ',I5,'   VA= ',F8.2,'   G= ',F6.2,'   FL= ',F8.2/
        2'   PM= ',F6.2,'   MM= ',I5,'   F= ',F6.3,'   CORFIN= ',F6.2/
        3'   AP= ',F8.2,'   CK= ',F6.2,'   V0= ',F8.2)
      27 WRITE(6,30)
      30 FORMAT(' PLC   TMAX   CPLAC   UPSANS(1)   UPAN1Q   UPSANS(6)   UPAN3Q'/
        119X,' DNSANS(1)   DNAN1Q   DNSANS(6)   DNAN3Q')
       6 READ(5,10) PLC,TMAX,CPLAC
      10 FORMAT(3F8.3)
         IF(CPLAC.LE.0.0) GO TO 110
C     COMPUTE DT
         DT=FL/((V0+VA)*FLOAT(N))
C     CHECK FOR CONVERGENCE.
         DX=FL/FLOAT(N)
         THETA=DT/DX
         IF(THETA.LE.(1./VA)) GO TO 20
      17 GO TO 110
C     COMPUTE COEFFICIENTS AND CONSTANTS FOR ALL PIPES.
      20 API=3.142
         DP=SQRT(4.*AP/API)
         C2=G/VA
         HF=(F*FL*V0*V0)/(2.*G*DP)
C     H0ABS=H0+HF+34.
         H0ABS=V0/(C2*PLC)
C     H0= HEAD AT RESERVOIR.
         H0=H0ABS-HF-34.
C     HORF0= ORIFICE HEAD LOSS FOR FLOW Q0 FROM TANK.
         HORF0=(CK*H0ABS-HF)/CORFIN
C     V0AIR= INITIAL AIR VOLUME IN TANK.
         V0AIR=(CPLAC*V0*AP*FL)/(2.*VA)
C     Q0= STEADY STATE DISCHARGE.
         Q0=V0*AP
         FF=F*DT/(2.*DP)
         CF=HORF0/(Q0*Q0)
         C10=H0ABS*V0AIR**PM
C     STEADY STATE CALCULATIONS.
         DHF=F*FL*V0*V0/(2.*G*DP*FLOAT(N))
         NN=N+1
         DO 25 I=1,NN
         V(I)=V0
         TEMP=NN-I
         H(I)=H0+TEMP*DHF
```

14


```
      25 CONTINUE
C     INITIALIZATION OF MAX. AND MIN. HEADS.
         DO 26 I=1,6,5
         HMAX(I)=H(I)
         HMIN(I)=H(I)
         HSS(I)=H(I)
      26 CONTINUE
         SUMSS(1)=H(3)+H(4)
         SUMSS(2)=H(8)+H(9)
         SUMAX(1)=H(3)+H(4)
         SUMIN(1)=H(3)+H(4)
         SUMAX(2)=H(8)+H(9)
         SUMIN(2)=H(8)+H(9)
C     TIME INITIALIZATION.
         T=0.0
         VAIR=VOAIR
C     PRINTOUT INTERVAL INITIALIZATION.
         M=0
C     COMPUTATION OF VR,VS,HR,HS FOR ALL SECTICNS.
C     INTERIOR SECTIONS.
      40 DO 50 I=2,N
         VR(I)=V(I)-VA*THETA*(V(I)-V(I-1))
         HR(I)=H(I)-VA*THETA*(H(I)-H(I-1))
         VS(I)=V(I)-VA*THETA*(V(I)-V(I+1))
         HS(I)=H(I)-VA*THETA*(H(I)-H(I+1))
      50 CONTINUE
C     BOUNDARY SECTIONS.
C     RESERVOIR.
         VR(N+1)=V(N+1)-VA*THETA*(V(N+1)-V(N))
         HR(N+1)=H(N+1)-VA*THETA*(H(N+1)-H(N))
         C3=VR(N+1)+C2*HR(N+1)-FF*VR(N+1)*ABS(VR(N+1))
C     AIR CHAMBER.
      54 VS(1)=V(1)-VA*THETA*(V(1)-V(2))
         HS(1)=H(1)-VA*THETA*(H(1)-H(2))
         C1=VS(1)-C2*HS(1)-FF*VS(1)*ABS(VS(1))
      38 T=T+DT
         M=M+1
         IF(T.GE.TMAX) GO TO 107
C     TIME INCREMENTED.  BOUNDARY CONDITIONS.
C     AIR CHAMBER.  HOABS*VOAIR**PM=CONSTANT.
C     LOOP (89,106) TO APPROX. AVE. VELOCITY FROM CHAMBER.
         VAVAPP=V(1)
         GO TO 210
     200 VAVAPP=VAV
     210 C11=VAVAPP*AP
         CAIR=VAIR+C11*DT
         IF(C11) 53,52,51
      51 CORF=1.0
         GO TO 59
      52 CORF=0.0
         GO TO 59
      53 CORF=2.5
      59 HORF=CORF*CF*C11*ABS(C11)
      60 HP(1)=(C10/CAIR**PM)-HORF-34.
C     NEGATIVE CHARACTERISTIC EQUATION.
         VP(1)=C1+C2*HP(1)
         VAV=(V(1)+VP(1))/2.
         VERR=VAVAPP-VAV
         IF(ABS(VERR).LE.0.0001) GO TO 230
     220 GO TO 200
```

KEYSTONE BUSINESS FORMS LTD VANCOUVER

```
  230 VPAIR=VAIR+(AP*DT*VAV)
C     RESERVOIR AT DOWNSTREAM END.
      HP(N+1)=HO
C     POSITIVE CHARACTERISTIC EQUATION.
      VP(N+1)=C3-C2*HP(N+1)
C     COMPUTATION OF INTERIOR POINTS.
      DO 55 I=2,N
      VP(I)=0.5*(VR(I)+VS(I)+C2*(HR(I)-HS(I))-FF*(VR(I)*ABS(VR(I))+
     1VS(I)*ABS(VS(I))))
      HP(I)=0.5*(HR(I)+HS(I)+(VR(I)-VS(I))/C2-FF*(VR(I)*ABS(VR(I))-VS(I)
     1*ABS(VS(I)))/C2)
   55 CONTINUE
C     CONVERT V(I)=VP(I), AND H(I)=HP(I) FOR ALL SECTIONS.
   80 DO 90 I=1,NN
      V(I)=VP(I)
      H(I)=HP(I)
   90 CONTINUE
      VAIR=VPAIR
C     TABULATION OF MAX. AND MIN. HEADS.
      DO 95 I=1,6,5
      IF(H(I).LT.HMIN(I)) GO TO 123
  121 IF(H(I).GT.HMAX(I)) GO TO 125
  122 GO TO 95
  123 HMIN(I)=H(I)
      GO TO 95
  125 HMAX(I)=H(I)
   95 CONTINUE
      IF((H(3)+H(4)).LT.SUMIN(1)) GO TO 135
  131 IF((H(3)+H(4)).GT.SUMAX(1)) GO TO 137
  132 GO TO 140
  135 SUMIN(1)=H(3)+H(4)
  136 GO TO 140
  1370SUMAX(1)=H(3)+H(4)
  140 CONTINUE
      IF((H(8)+H(9)).LT.SUMIN(2)) GO TO 147
  142 IF((H(8)+H(9)).GT.SUMAX(2)) GO TO 150
  144 GO TO 155
  147 SUMIN(2)=H(8)+H(9)
  149 GO TO 155
  150 SUMAX(2)=H(8)+H(9)
  155 CONTINUE
      GO TO 40
C     COMPUTATION OF MAX. UPSURGES AND DOWNSURGES.
  107 DO 170 I=1,6,5
      UPSMAX(I)=HMAX(I)-HSS(I)
      DNSMAX(I)=HSS(I)-HMIN(I)
      UPSANS(I)=UPSMAX(I)/HOABS
      DNSANS(I)=DNSMAX(I)/HOABS
  170 CONTINUE
      HMAX1Q=SUMAX(1)/2.
      HMIN1Q=SUMIN(1)/2.
      HMAX3Q=SUMAX(2)/2.
      HMIN3Q=SUMIN(2)/2.
      HSS1Q=SUMSS(1)/2.
      HSS3Q=SUMSS(2)/2.
      UPMA1Q=HMAX1Q-HSS1Q
      DNMA1Q=HSS1Q-HMIN1Q
      UPMA3Q=HMAX3Q-HSS3Q
      DNMA3Q=HSS3Q-HMIN3Q
      UPAN1Q=UPMA1Q/HOABS
```

```
      DNAN1Q=DNMA1Q/HOABS
      UPAN3Q=UPMA3Q/HOABS
      DNAN3Q=DNMA3Q/HOABS
      WRITE(6,180) PLC,TMAX,CPLAC,UPSANS(1),UPAN1Q,UPSANS(6),UPAN3Q,
     1DNSANS(1),DNAN1Q,DNSANS(6),DNAN3Q
  180 FORMAT(/F4.1,2X,F5.1,2X,F5.1,4X,F6.3,3X,F6.3,4X,F6.3,3X,F6.3/
     122X,F6.3,3X,F6.3,4X,F6.3,3X,F6.3)
      GO TO 6
  110 STOP
      END
END OF FILE
```

```
$COPY *SKIP *SINK*
```

KEYSTONE BUSINESS FORMS LTD VANCOUVER

2. ENTIRE HEAD LOSS ATTRIBUTABLE TO DISTRIBUTED FRICTION

```
$LIST AIRCHAM11
      C   WATERHAMMER PROGRAM. PUMP AT UPSTREAM END WITH AIR CHAMBER ADJACENT
      C   TO THE PUMP.  RESERVOIR AT DOWNSTREAM END.
      C   CHECK VALVE CLOSES IMMEDIATELY ON PUMP FAILURE.
      C   LINE FRICTION ONLY. NO ORIFICE LOSS. NO MINOR LOSSES.
            DIMENSION V(20),VP(20),H(20),HP(20),VR(20),VS(20),HR(20),HS(20),
          1HMAX(10),HMIN(10),HSS(10),SUMAX(10),SUMIN(10),SUMSS(10),
          2UPSMAX(10),DNSMAX(10),UPSANS(10),DNSANS(10)
            DATA N/10/,VA/3216./,G/32.16/,FL/3216./,PM/1.2/,MM/1/,
          1HORF/0.0/,AP/3./,
          2CK/1./,
          3VO/3.5/
            WRITE(6,15) N,VA,G,FL,PM,MM,HORF,AP,CK,VO
       15 FORMAT(/'  THE PARAMETERS ARE NOW...'/
          1'   N= ',I5,'   VA= ',F8.2,'   G= ',F6.2,'   FL= ',F8.2/
          2'   PM= ',F6.2,'   MM= ',I5,'   HORF= ',F6.3/
          3'   AP= ',F8.2,'   CK= ',F6.2,'   VO= ',F8.2)
       27 WRITE(6,30)
       30 FORMAT(' PLC   TMAX  CPLAC  UPSANS(1)  UPAN1Q  UPSANS(6)  UPAN3Q'/
          119X,' DNSANS(1)  DNAN1Q  DNSANS(6)  DNAN3Q')
        6 READ(5,10) PLC,TMAX,CPLAC
       10 FORMAT(3F8.3)
            IF(CPLAC.LE.0.0) GO TO 110
      C   COMPUTE DT
            DT=FL/((VO+VA)*FLOAT(N))
      C   CHECK FOR CONVERGENCE.
            DX=FL/FLOAT(N)
            THETA=DT/DX
            IF(THETA.LE.(1./VA)) GO TO 20
       17 GO TO 110
      C   COMPUTE COEFFICIENTS AND CONSTANTS FOR ALL PIPES.
       20 API=3.142
            DP=SQRT(4.*AP/API)
            C2=G/VA
      C   HOABS=HO+HF+34.
            HOABS=VO/(C2*PLC)
      C   HEAD LOSS FOR FLOW INTO CHAMBER.
            HF=CK*HOABS
      C   HO= HEAD AT RESERVOIR.
            HO=HOABS-HF-34.
      C   F= FRICTION FACTOR.
            F=(HF*2.*G*DP)/(FL*VO*VO)
      C   VOAIR= INITIAL AIR VOLUME IN TANK.
            VOAIR=(CPLAC*VO*AP*FL)/(2.*VA)
            QO=VO*AP
            FF=F*DT/(2.*DP)
            C10=HOABS*VOAIR**PM
      C   STEADY STATE CALCULATIONS.
            DHF=HF/FLOAT(N)
            NN=N+1
            DO 25 I=1,NN
            V(I)=VO
            TEMP=NN-I
            H(I)=HO+TEMP*DHF
       25 CONTINUE
      C   INITIALIZATION OF MAX. AND MIN. HEADS.
```

5/8


```
53          DO 26 I=1,6,5
54          HMAX(I)=H(I)
55          HMIN(I)=H(I)
56          HSS(I)=H(I)
57       26 CONTINUE
58          SUMSS(1)=H(3)+H(4)
59          SUMSS(2)=H(8)+H(9)
60          SUMAX(1)=H(3)+H(4)
61          SUMIN(1)=H(3)+H(4)
62          SUMAX(2)=H(8)+H(9)
63          SUMIN(2)=H(8)+H(9)
63.5  C  TIME INITIALIZATION.
64          T=0.0
65          VAIR=VOAIR
65.5  C  PRINTOUT INTERVAL INITIALIZATION.
66          M=0
67    C  COMPUTATION OF VR,VS,HR,HS FOR ALL SECTIONS.
68    C  INTERIOR SECTIONS.
69       40 DO 50 I=2,N
70          VR(I)=V(I)-VA*THETA*(V(I)-V(I-1))
71          HR(I)=H(I)-VA*THETA*(H(I)-H(I-1))
72          VS(I)=V(I)-VA*THETA*(V(I)-V(I+1))
73          HS(I)=H(I)-VA*THETA*(H(I)-H(I+1))
74       50 CONTINUE
75    C  BOUNDARY SECTIONS.
76    C  RESERVOIR.
77          VR(N+1)=V(N+1)-VA*THETA*(V(N+1)-V(N))
78          HR(N+1)=H(N+1)-VA*THETA*(H(N+1)-H(N))
79          C3=VR(N+1)+C2*HR(N+1)-FF*VR(N+1)*ABS(VR(N+1))
80    C  AIR CHAMBER.
81       54 VS(1)=V(1)-VA*THETA*(V(1)-V(2))
82          HS(1)=H(1)-VA*THETA*(H(1)-H(2))
83          C1=VS(1)-C2*HS(1)-FF*VS(1)*ABS(VS(1))
84       38 T=T+DT
85          M=M+1
86          IF(T.GE.TMAX) GO TO 107
87    C  TIME INCREMENTED.  BOUNDARY CONDITIONS.
88    C  AIR CHAMBER.  HOABS*VOAIR**PM=CONSTANT.
88.5  C  LOOP (89,106) TO APPROX. AVE. VELOCITY FROM CHAMBER.
89          VAVAPP=V(1)
90          GO TO 210
91      200 VAVAPP=VAV
92      210 C11=VAVAPP*AP
93          CAIR=VAIR+C11*DT
101      60 HP(1)=(C10/CAIR**PM)-HORF-34.
101.5 C  NEGATIVE CHARACTERISTIC EQUATION.
102         VP(1)=C1+C2*HP(1)
103         VAV=(V(1)+VP(1))/2.
104         VERR=VAVAPP-VAV
105         IF(ABS(VERR).LE.0.0001) GO TO 230
106     220 GO TO 200
107     230 VPAIR=VAIR+(AP*DT*VAV)
108   C  RESERVOIR AT DOWNSTREAM END.
109         HP(N+1)=HO
109.5 C  POSITIVE CHARACTERISTIC EQUATION.
110         VP(N+1)=C3-C2*HP(N+1)
111   C  COMPUTATION OF INTERIOR POINTS.
112         DO 55 I=2,N
113         VP(I)=0.5*(VR(I)+VS(I)+C2*(HR(I)-HS(I))-FF*(VR(I)*ABS(VR(I))+
114        1VS(I)*ABS(VS(I))))
```

KEYSTONE BUSINESS FORMS LTD VANCOUVER

```
      HP(I)=0.5*(HR(I)+HS(I)+(VR(I)-VS(I))/C2-FF*(VR(I)*ABS(VR(I))-VS(I)
     1*ABS(VS(I)))/C2)
   55 CONTINUE
C     CONVERT V(I)=VP(I), AND H(I)=HP(I) FOR ALL SECTIONS.
   80 DO 90 I=1,NN
      V(I)=VP(I)
      H(I)=HP(I)
   90 CONTINUE
      VAIR=VPAIR
C     TABULATION OF MAX. AND MIN. HEADS.
      DO 95 I=1,6,5
      IF(H(I).LT.HMIN(I)) GO TO 123
  121 IF(H(I).GT.HMAX(I)) GO TO 125
  122 GO TO 95
  123 HMIN(I)=H(I)
      GO TO 95
  125 HMAX(I)=H(I)
   95 CONTINUE
      IF((H(3)+H(4)).LT.SUMIN(1)) GO TO 135
  131 IF((H(3)+H(4)).GT.SUMAX(1)) GO TO 137
  132 GO TO 140
  135 SUMIN(1)=H(3)+H(4)
  136 GO TO 140
  1370SUMAX(1)=H(3)+H(4)
  140 CONTINUE
      IF((H(8)+H(9)).LT.SUMIN(2)) GO TO 147
  142 IF((H(8)+H(9)).GT.SUMAX(2)) GO TO 150
  144 GO TO 155
  147 SUMIN(2)=H(8)+H(9)
  149 GO TO 155
  150 SUMAX(2)=H(8)+H(9)
  155 CONTINUE
      GO TO 40
C     COMPUTATION OF MAX. UPSURGES AND DOWNSURGES.
  107 DO 170 I=1,6,5
      UPSMAX(I)=HMAX(I)-HSS(I)
      DNSMAX(I)=HSS(I)-HMIN(I)
      UPSANS(I)=UPSMAX(I)/HOABS
      DNSANS(I)=DNSMAX(I)/HOABS
  170 CONTINUE
      HMAX1Q=SUMAX(1)/2.
      HMIN1Q=SUMIN(1)/2.
      HMAX3Q=SUMAX(2)/2.
      HMIN3Q=SUMIN(2)/2.
      HSS1Q=SUMSS(1)/2.
      HSS3Q=SUMSS(2)/2.
      UPMA1Q=HMAX1Q-HSS1Q
      DNMA1Q=HSS1Q-HMIN1Q
      UPMA3Q=HMAX3Q-HSS3Q
      DNMA3Q=HSS3Q-HMIN3Q
      UPAN1Q=UPMA1Q/HOABS
      DNAN1Q=DNMA1Q/HOABS
      UPAN3Q=UPMA3Q/HOABS
      DNAN3Q=DNMA3Q/HOABS
      WRITE(6,180) PLC,TMAX,CPLAC,UPSANS(1),UPAN1Q,UPSANS(6),UPAN3Q,
     1DNSANS(1),DNAN1Q,DNSANS(6),DNAN3Q
  180 FORMAT(/F4.1,2X,F5.1,2X,F5.1,4X,F6.3,3X,F6.3,4X,F6.3,3X,F6.3/
     122X,F6.3,3X,F6.3,4X,F6.3,3X,F6.3)
      GO TO 6
  110 STOP
```

```
      173              END
END OF FILE


$CCPY *SKIP *SINK*
```

KEYSTONE BUSINESS FORMS LTD VANCOUVER