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AN EXPERIMENTAL INVESTIGATION AND DESIGN OF HYDRAULIC RAM PUMPS.

by

PATRICK O. KAHANGIRE

A thesis
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ABSTRACT

Hydraulic ram pumps (hydrams) have been in use for a long time but they have not been as widely used as they should.Their simplicity of design, high efficiency,reliability and operation without fuels makes hydrams ideal for small scale water supply schemes to serve individual households and isolated settlements.

Ignorance of the hydram technology, particularly in developing countries, lack of understanding of their operating characteristics and basic hydraulic principles are obstacles to their widespread use.

The objective of this thesis is to increase the understanding of hydram operating characteristics and to try to identify the major factors in hydram design and installation.This will,hopefully,encourage increased use of hydrams.

Five different hydrams were tested and their operating characteristics identified and compared.Design modifications of a locally-made hydram were done and a step-by-step identification of the effect of the various hydram components on hydram design and operation identified.An improved locally-made pump was made and tested and its operating characteristics identified and compared with the other hydrams.

A simple theoretical model of the hydraulic ram pump was developed and together with two of the existing models was tested against experimental data from two different hydrams.The results were compared for the three models and showed that the major operating characteristics of the hydram could be predicted within acceptable errors.It was also shown that the new simple model was superior to the existing simple model but not as good as the detailed model.

Finally, it was demonstrated that the developed model could be utilized for hydram design,especially in the investigation of proposed hydram installations.

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DEDICATION

To my family, for their understanding.

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LIST OF SYMBOLS

		dimensions
A	cross-sectional area of the drive pipe	[L^2]
A_v	Area of the waste valve	[L^2]
A_0	Area of the orifice of the waste valve	[L^2]
c	speed or celerity of an acoustic wave in water	[L/T]
C_D	dimensionless drag force coefficient	---
D	diameter of the drive pipe	[L]
d	diameter of the delivery pipe	[L]
E	modulus of elasticity of the pipe wall material	[M/LT^2]
e	mechanical efficiency of the pump	---
f	friction factor of the drive pipe	---
F_D	total drag force on the waste valve	[ML/T^2]
FR	flow ratio = Q/Q_w	---
Fr	Froude number	---
g	acceleration due to gravity	[L/T^2]
H	supply or drive head, above the waste valve	[L]
ΔH	change in pressure head in the drive pipe due to waterhammer	[L]
Hd	total delivery head above the waste valve	[L]
HR	head ratio = h/H	---
HVD	friction head loss factor of the delivery valve alone	---

h	pump head (Hd-H)	[L]
Σh_L	total minor losses in the drive pipe	---
hr	head loss through the delivery valve during the pumping period	[L]
h_{max}	the maximum pressure head the pump can develop	[L]
h_{min}	the minimum pressure head at which the pump fails	[L]
h_v	hydrostatic force on the waste valve	[L]
K	bulk modulus of elasticity of water	[M/LT^2]
K_c	composite bulk modulus of elasticity of water and pipe material	[M/LT^2]
L	length of the drive pipe	[L]
L_{max}	maximum length of the drive pipe practicable	[L]
l	length of the delivery pipe	[L]
Mc	mach number	---
M	total head loss factor of the drive pipe and waste during the acceleration period	---
σ	Poisson's ratio	---
m	velocity ratio = V_0 /V_s	---
N	total head loss factor of the drive pipe and delivery valve during the pumping period	---
π	Pi =3.142	---
P	power delivered by the pump	[ML^2/T^3]
Δp	change in pressure due to waterhammer	[M/LT^2]
Q	delivery or pumped flow rate	[L^3/T]
Q_s	total supply flow rate to the pump	[L^3/T]
Q_w	drive or waste flow rate	[L^3/T]

RS	head loss factor of the pump and waste valve alone	---
Re	Reynold's number	---
r	radius of the drive pipe	[L]
S	stroke length of the waste valve	[L]
t	time in general	[T]
T	total duration of a pumping cycle	[T]
t_p	thickness of the drive pipe	[L]
VB	valve beat frequency	---
V	velocity of water in the drive pipe in general	[L/T]
V_0	velocity of water in the drive pipe when the waste valve begins to close	[L/T]
V_s	steady state flow velocity in drive pipe	[L/T]
V	velocity in the delivery pipe	[L/T]
V_{max}	the maximum flow velocity in the drive pipe	[L/T]
V_m	velocity in the drive pipe just before complete waste valve closure	[L/T]
ΔV	change in velocity due to waste valve closure	[L/T]
$\forall$	flow volume per period of the cycle	[L^3]
$\forall_w$	wasted flow volume per cycle	[L^3]
W	weight or spring tension of the waste valve	[ML/T^2]
W_{max}	maximum weight of the waste valve or spring tension	[ML/T^2]
XM	friction loss factor of the drive pipe alone	---
γ	specific weight of water	[M/L^2T^2]
ρ	density of water	[M/L^3]

Chapter I

INTRODUCTION

1.1 HISTORICAL DEVELOPMENT OF HYDRAULIC RAM PUMPS.

A hydraulic ram (also called hydram) is a pump that uses energy from a falling quantity of water to pump some of it to an elevation much higher than the original level at the source.No other energy is required and as long as there is a continuous flow of water,the pump will work continuously and automatically.

The first hydraulic ram pump was invented by an Englishman,John Whitehurst in 1775.His hydram was not automatic but the operation was controlled manually by opening and closing a stop-cock (fig 1). Whitehurst installed a few of these machines but in general the apparatus never made much headway and was soon forgotten.

The automatic hydraulic ram pump was invented by a Frenchman, Joseph M. Montgolfier,in 1797 in France(fig. 2 and 3).He called the machine,'le belier hydraulique' from which the term hydraulic ram is derived.The waste valve,W,opened and closed automatically.Although small hydrams worked,attemps to make bigger hydrams failed due to the absorption of air in the air chamber.Montgolfier designed and installed several hydrams,carried out experimental studies and even presented formulae of the hydram operation [19],[50].

In general hydraulic ram technology after Montgolfier can be divided into two major categories; improvement in mechanical design and research into the principles of its operation.

1.1.1 Improvement of and Variation in Mechanical Design

Since Montgolfier's time the ram has been improved by various designers. Improvement in mechanical design can be subdivided into the following developments; a) Development of the ordinary hydraulic ram pump based on Montgolfier's invention and (b) Design of other machines based on the hydram principles.

Early improvements on the ordinary hydraulic ram were by Montgolfier's son, Pierre Francois Montgolfier, who, among other things added the snifter valve that enabled the design of big hydrams and made it possible to pump to higher delivery heads. He is also credited with the development of the double-acting (also called dirty water) hydram that could use dirty water for drive power to pump clean water for areas where the clean water quantities were small and dirty water abundant [19].

During the early nineteenth century, many people got interested in hydraulic ram pumps and many different designs were produced and several patents were issued. Notable early hydram designers include Boulton,Mellington,Easton and Anderson.However,almost two centuries later the commercial hydraulic ram has not changed significantly (figure 4).This was because most of the designers were concerned with minor improvements and,therefore,the basic design remained the same [32]

At the end of the last century,a few large hydram installations were made.One of these was designed and installed by Professor Mead for the West Dundee village,U.S.A in 1896 and is one of the largest hydrams ever built.The 10 inch(254 mm) hydram was 20 ft (6.1 m) high with an air chamber 12 ft (3.66 m) high and 22 inches (559 mm) in diameter.The cast-iron drive pipe was 2200 ft (671 m) long.The waste valve was spring-loaded and the delivery valve specially designed with multiple valves.The pump was made from pipe fittings and used 580 gallons per minute (2198 litres/minute) and a fall of 37 ft (11.3 m) to pump 230 gallons (872 litres) per minute to a level 84 ft (25.6 m) higher with an efficiency (Rankine) of 84 % [38] [40].Another large installation at Seattle gave very high efficiency.The two Sterling hydrams of 12 inch (305 mm) each,used a fall of 50ft(15.2 m) to pump 720,000 and 1,300,000 gallons (272,8800 and 4,927,000 litres) per day against a head of 140 ft (42.7 m) with record efficiencies (D'Aubuisson) of 91 % and 85 % respectively [14] [38].

Apart from these and possibly a few special installations, commercial hydrams are usually of small size, mainly 0.5 to 4 inches (13mm -102 mm) inlet diameters. Very few exceed 3 inch (76 mm) in diameter. These sizes also apply to the waste valve diameters. Hydrams of these sizes are considered durable and efficient, while the larger ones are not and often fail [50].

The trend, particularly in the U.S.A ,has been to develop small and cheaper hydrams. A unique design by H.H Strawbridge U.S.A in 1843 was the first American made hydram. The hydram was entirely made of wood and it exploded [21]. H.R Fleming in 1980 introduced hydrams made from plastic materials (PVC) which are much cheaper and very efficient. Other commercial hydrams are usually made from cast-iron.

Another notable development in hydram design since 1948 has been the design and use of hydrams made entirely from ordinary steel or cast- iron pipe fittings and simple valves [27],[34],[56], [64]. Many individuals and institutions showed interest in the design and installation of the cheap and simple models [27],[42],[56],[58],[64], including the University of Ottawa[53],[54]. Another variation in design was by the Technische Hogeschool Endhoven, Holland with the concrete water ram made of concrete and bamboo designed for use in Indonesia. No practical results are available but test results showed normal operation (Attwood, personal commuication). Improvements were also done on the double-acting ram and commercial models are availble with comparable efficiency ranges as the single-acting hydrams [3],[4],[24],[52].

Various machines based on the hydram technology were designed and although they differ in design and operation they are traditionally classified as hydraulic rams. These include the suction hydram by Decoeurs [18] and the siphon hydram by Leblanc [9]. To minimize or eliminate energy losses in hydraulic rams due to the escape of water after the waste valve begins to close, Sommellier used a series of valves that closed successively by the effect of water rising in the air or compression chamber [50]. Sommellier later adapted his hydram for use as an aircompressor and it was actually used on the Mt. Cenis tunnel in France [23]. The hydram's highest development is due to the invention of the hydraulic engine by Pearsall, which was an extension of Sommellier's work. It used an auxilliary motor of air compressed by the pump itself to operate the waste valve. The innovation reduced the loss of energy due to excessive shock and escape of water after the waste valve begins to close [9],[48]. The Pearsall and Sommellier designs could not be applied to hydrams smaller than 305 mm in diameter [50]. Due to the size and cost, the Pearsall engine soon fell into disuse. The Pearsall development had made the ram noiseless and very efficient. Pearsall also adapted his hydram to work as an air compressor by enlarging the air or compression chamber.One of his machines was still in use by 1922 ,supplying rock drills in Devonshire, England [3].Richards [50] reported that two Pearsall machines had been brought into the U.S.A.They were 40 inches (762 mm) and 24 inches (609.6 mm) in diameter.He said that no further information about them was available.

Another massive 40-ton ram,specially built in England for use in Canada.,is worth mentioning.The horizontal treble ram consisted of three independent self-contained pumps driven from a single central shaft and connected to a common delivery main.The 20 inch(508 mm) diameter hydrams with a stroke length of 24 inches(610 mm),delivered 3,000,0000 gallons (11,370,000 litres) per day against a pressure head of 120 psi (827 kPa) at a valve beat frequency of 28 beats per minute [61].

J.Montgolfier had earlier mentioned some of the above innovations as possible applications of the hydraulic ram [19].In most cases these machines were not commercially produced and the commercial hydram remained small and noisy. In a recent development,an attempt was made to develop fluidic hydrams using vortex-switched devices [62].No practical test results have yet been reported.

1.1.2 Research into the principles of hydram operation

The investigation into the principles of hydram operation can be divided into three groups;empirical,theoretical and rational methods [32].

The empirical methods are based on experimental tests with the results not being correlated with or supported by theory.The approach led to several 'rules of thumb',some of which were misleading. The leading empirical reasearchers include Eytelwein,D'Aubuisson, Morin,Clarke and Richards [32][50]. Empirical formulas were insufficient because the operation of the hydraulic ram depends on many variables,most of which are neglected by the empirical formulations.

The theoretical methods were based purely on the rules of hydraulics. An attempt was made to ascertain the rate of change of the variable velocity of water in the drive pipe during each phase of the cycle.The quantities wasted and delivered were determined together with the duration of the cycle and the valve beat frequency.The approach was not successful because several parameters relating to the operation of hydraulic rams are best obtained experimentally. Each hydram design and installation has several unique parameters such as loss of head by friction and turbulence through the waste valve, head losses in the drive pipe and others which are best obtained experimentally.The leading researchers using this approach include Bergeron [7] and Iversen [28].

The first rational method of research based on theoretical analysis and verified by experiments was by Harza in 1908 [32],[44]. Further rational studies were done by Gosline and O'Brien [44], Lansford and Dugan [37],Krol [32],Calvert [11].

After Montgolfier's invention, very litle hydram development was done between 1796 and 1886. Most mechanical improvements and modifications were done during the period 1886 and 1916,mainly by American and British Engineers and manufacturers [32],[33].Most theoretical and rational investigations were done between the period 1916-1946,with some work continuing to the present day.There has also been the increased manufacture of hydrams made from pipe fittings and plastic materials.

Krol [32],[33] reports that a lot of developments in hydram technology have taken place in the U.S.S.R.Since 1946 several Russian institutions assumed a leading role in hydram technology.

For centuries the manufacture of hydrams has been a monopoly of a few companies whose designs are based on past experience [50]. Theoretical analyses have not always been utilised in the design of these pumps.The search for an inexpensive,simple method to make hydrams from pipe fittings and easily obtainable commercial or locally-made valves may increase the gap between theory and practice even more.

1.1.3 Hydram Utilization

The hydraulic ram pump is one of the simplest, most durable and efficient water raising machines, given that conditions of continuous flow of water and fall in land relief exist. There are many such areas in the world where the hydrams can be utilized. A well-designed and properly installed hydram can work trouble-free for many years. J.R Easton [10] reported a case of a hydram that had worked continuously for over 100 years. He also reported a hydram that had worked for 27 years without replacing the valves. The pump was still working but at reduced capacity and efficiency. Although these were rare or exceptional cases hydrams do, in general, last a long time.

In spite of all the obvious advantages of simplicity of mechanical design and high efficiency, the hydraulic ram has not been utilized widely. There are many countries today, mainly developing countries, where the hydram is unknown. The hydrams have mainly been used in France, Britain, U.S.S.R and the U.S.A. [32], [33], [50].

The hydraulic ram pumps have not been widely used because of four main reasons:-

1. It is only applicable in specific areas where there is continuous flow of water and a fall in the relief to create a supply head and drainage conditions.
2. Electrification programs like those in the U.S.A in the 1930's and development of affordable centrifugal

pumps made the hydram use less attractive, limiting their use only to isolated cottages and farm houses. This led, among other things, to the neglect of the development of large hydrams. The noisy operation and vibrations makes them unsuitable for use in or near homes.

3. Many people are not aware of the existence and capabilities of the hydraulic ram pump technology. In modern times, publicity about hydrams has been sparse.
4. Although the pumps are structurally simple, the detailed mechanics of its operation are not well understood, thus making significant design improvements difficult.

Increased fuel costs, concern for environmental protection and the search for energy-saving technologies, have led to renewed interest in hydram pumps in the U.S.A and Britain.

Even with the availability of easily made, cheap non-commercial models the use of hydrams in many parts of the world is still very low. At present, the only country seriously engaged in widespread technological development and utilization of the hydraulic ram pumps is the U.S.S.R where a lot of work has been done to identify possible sites for hydram installations [33].

1.2 THE AUTOMATIC SINGLE-ACTING HYDRAULIC RAM PUMP.

Before presenting the details of hydram operation,it is necessary to describe the various parts of the hydram.

The automatic hydraulic ram pump is structurally simple consisting of two moving parts;the waste valve(impulse or impetus valve) and the check valve (delivery or discharge valve).The other components are the air chamber and the impact box or main body of the ram(fig. 4) and in most models,the snifter or air valve.Since a hydram is usually considered as a combination of pump and motor,the drive pipe is considered as part of the pump assembly.

Waste or Impulse valve. Is an important component of the hydram operation.The valve can be of spring type,weighted-impulse type or spring-loaded weighted-impulse type.The proper design and adjustment of the weight or spring tension of the waste valve affects both hydram capacity and efficiency.

Delivery or check valve. This is located below the air chamber and prevents the back-flow during recoil.It is usually designed to offer as little resistance to the flow as possible and to withstand the back pressure from the delivery pipe and air chamber during recoil at the end of a pumping cycle.

Air chamber. Contains both air and water under pressure and performs two important roles:

(a) Absorbs the pressure surges due to waterhammer so that they don't reach the delivery pipe.

(b) Changes the pulsating flow in the pump and drive pipe into a uniform continuous discharge in the delivery pipe.

Snifter(or air)valve. This is required in most hydrams to replenish the air in the air chamber.Under the high pressure and mixing during pumping,the air in the air chamber slowly dissolves into the water and has to be replaced for continuous and smooth operation of the hydram.

In some hydrams,the air chamber is equipped with a bladder under pressure or a diaphragm and a bladder so that the air valve is not necessary.The snifter or air valve can be a simple one-way valve or a small hole.

Drive pipe.

This is an important integral part of the hydram.It must be able to sustain a high waterhammer pressure caused by the closing of the waste valve.Its diameter is determined by the size of the hydram used,strength requirements,cost considerations and availability of pipe materials. The drive pipe should not be too short or too long and there is a range of L/D ratio for good hydram operation.Calvert[12] suggested this ratio to be between the limits

$$150 \leq \frac{L}{D} \leq 1000 \qquad (1)$$

where D is the drive pipe diameter and L its length. European and American practice favours the drive pipe length to be between 6H and 12H such that

$$6H \leq L \leq 12H \tag{2}$$

where H is the static supply head [33].The Russian researchers developed an empirical relationship for drive pipe length [33] of the form

$$L = \left[\frac{900}{N^2}\right]\left[\frac{H}{D}\right] \tag{3}$$

where N is the number of ~~valve~~ beats/minute

Due to the high pressures involved,drive pipes are usually made of steel or cast iron. Other materials like concrete,copper and PVC can be used but will not give as good results as steel in terms of the delivery pressures that the pump will develop.The pipes must,also,have sufficiently thick walls as indicated in figure 5.

1.3 TYPICAL INSTALLATION AND HYDRAM OPERATION.

A typical hydram installation consists of the water source (supply), drive pipe,the pump itself,the delivery (or discharge) pipe and the service or header tank (fig.6).

At the start of the cycle of operation,the waste valve is open and a quantity of water flows down the drive (or supply) pipe from the source and escapes through the waste valve.The water accelerates under the influence of the supply head,H,until a sufficient velocity is attained in the drive pipe and the total dynamic and drag forces on the valve overcomes its weight and the valve begins to close. The valve closes rapidly and then due to the water hammer effect,the pressure in the pump and drive pipe rises in excess of the pressure in the air chamber and delivery pipe.The check valve is then forced open and a quantity of water discharges into the air chamber.The air chamber prevents the occurrence of water hammer or pressure surges in the delivery pipe and ensures a continuous flow of water in the delivery pipe to the service or header tank.

The discharge through the delivery pipe continues until the momentum of the column of water in the drive pipe is exhausted,the disharge velocity becomes zero and the delivery valve closes.The closure of the delivery valve forces the water column to recoil and flow towards the supply reservoir.The recoil,temporarily,creates a negative pressure in the hydram causing a small amount of air to be admitted into the ram through the snifter or air valve.The pressure on the underside of the waste valve which had remained closed up to this instant is also reduced.The waste valve then opens due to the reduced pressure and its own weight and the cycle is repeated continuously. The air admitted into the hydram is carried by the flow during the next cycle into the air chamber to replace that absorbed by the water due to the high pressure and mixing in the air chamber.

The cycle consists of two main periods;the period of acceleration during which kinetic energy is created in the drive pipe and the period of retardation during which the pumping takes place.The cycle is repeated at a rate (frequency) of a few beats (or pulsations) to over 300 beats per minute.

For given hydram design and installation conditions,the hydram can be adjusted or tuned by varying the waste valve weight or tension and the stroke length to achieve optimum performance.

1.4 THE NEED AND SCOPE OF THE THESIS

Several hydraulic ram models have been designed,built and used with varying degree of success.In general,theoretical models have not been utilized in the design of different hydrams.

The boom in the making of less-costly,locally-made hydrams based either on trial and error or past experience with little or no regard for the fundamental principles of hydram operation will widen the gap between theory and practice.This will lead to uneconomical designs and less efficient installations.

Many researchers have,with varying degrees of simplifications, assumptions and relevant empirical parameters,shown that the principles of fluid mechanics can be used to predict the performance of hydram installations with reasonable accuracy. Such models if utilised could lead to better and more economical hydram designs and more efficient installations. Models that predict hydram performance reliably are quite involved and complicated.This has minimised their usefulness for designers and practical engineers.

There is,therefore a need for an experimental study of different hydram designs so that a broad understanding of hydram operating characteristics can be achieved .Previous studies and conclusions were based on particular single designs.One benefit of a broadly based study would be to indicate how the cheaper and easy-to-make hydrams compare with the more expensive commercial hydrams.

There is also a need for a scientific study of the hydraulic ram so that a rational model which is easy to understand but fairly accurate could be developed.A simple model would be easy to use for rational and economic design and installation of hydraulic ram pumps.

The goal of this thesis was to build from the work of earlier researchers and designers and do the following:-

(a) Experimentally test and compare small commercial and non-commercial hydrams.Working characteristics,within the limits of the experiments, were to be developed and compared.

(b) With the information from (a) make and test an improved design of a locally made hydraulic ram pump to establish its operating characteristics and demonstrate the effect of the proportions and sizes of the various components on its operating characteristics.

(c) Develop a simple model for the prediction of hydram performance which is easy to apply and can be of use to practical hydram designers and users with a reasonable degree of accuracy.

It is hoped that the results will, to a small extent, contribute to the understanding of the operation of hydrams and give an indication as to their advantages and capabilities. Finally, as a result of these studies it is hoped that interest in hydram use will be revived, especially in developing countries.

Chapter II

LITERATURE REVIEW.

Investigation into the operation and design of hydraulic ram pumps has attracted the interest of both practicing and theoretical engineers. Although the basic components in the hydram operation are simple, how they relate with each other is not well understood. Most of the investigations have resulted in empirical rules based only on the observed or measured data and not on the computed theoretical results. Such rules are of limited applicability and sometimes only valid for a particular hydram design. As a result even the scanty literature on the subject of hydrams is not much used by the practical designers of hydrams.

One of the early investigators, Richards[50], experimented with a commercial ram for five years and concluded that,

> after hundreds of alterations and adjustments, the only clue to the efficiency of performance is by sight and sound

He observed that the weight of the waste valve affected its period of closing but not its opening as most people assumed. He, also, noted that the velocity in the drive pipe never reaches the theoretical maximum steady flow velocity, V_{max}

$$V_{max} = \sqrt{2gH} \tag{4}$$

where

H is the static supply head above the waste valve.

g is the acceleration due to gravity The author, concluded that the main energy losses in the hydram were from pipe friction, resistance in the delivery valve and concussion (shock) of the high pressure wave. However, he noted that the main loss comes from the escape of water after the waste valve begins to close. He estimated this loss to be between 15-25%, depending on ram adjustment, valve type and length of stroke. To minimize the losses, the waste valve weight should be adjusted in relation to the supply head and valve stroke.

Weisbach and Herrmann in 1897[65], presented a simple approximate theory of the hydraulic ram. They believed its accuracy increased with the flow in the drive pipe and waste valve beat frequency. They said that the complete theory was complicated and insufficient for the determination of the operating characteristics of the ram without using empirical parameters. The authors neglected all forms of friction head losses and waterhammer effects. The equation for pump efficiency was derived as

$$e = 1 - \left(\frac{h}{H}\right)^2 \left(\frac{t_2}{t}\right)^2 \qquad (5)$$

where

e - pump efficiency (%).

h - pump head (m).

H - supply head (m).

t_2 - pumping or discharge period(secs).

t - acceleration time (secs).

They deduced that if the pumping period is short compared to the acceleration period and the head ratio (h/H) is also small,the efficiency tends to unity.No experimental verification was done,and it is doubtful that such an oversimplified theory would give reasonable comparison.

From an experimental study,with a 4-inch (102 mm) hydram, Anderson [3] concluded that with few exceptions,better pump efficiency was obtained with light waste valves,especially with long strokes 0.75-1.0 inch (19.0-25.4mm). The minimum head ratio at which the pump operated could be increased if the waste valve was assisted by a spring.With some exceptions, increasing waste valve stroke increased the delivery flow but reduced pump efficiency and vice-versa.

Regarding the delivery valve,Anderson recommended an allowance of one square inch(6.45 square cms) through the delivery valve for every one gallon(4.5 liters) of water delivered per minute. He noted that if the waste valve stroke was kept constant,a long drive pipe was needed for high head ratios,while shorter pipes could be used for small head ratios.He recommended drive pipe lengths between 6H and 10H,where H was the static supply head.

In 1928 Bergeron [7] proposed a comprehensive theoretical model of the hydraulic ram pump.He divided the cycle into four parts; namely the periods of acceleration of water in drive pipe,closing of waste valve,pumping and opening of waste valve.The author showed that the theoretical maximum velocity,equation (4),is never attained in the drive pipes.He included details of waste valve closure and parameters for the delivery valve.He indicated that for better hydram design the orifice of the waste valve should equal or exceed area of the drive pipe to avoid 'choking' the flow.This feature is common in most of the commercial hydram designs.From the analysis,the author concluded that better pump efficiency was obtained with low velocities in the drive pipe during waste valve closure such that

$$V_0 \leq 0.40 \sqrt{2gH} \tag{6}$$

where

V_0 is the flow velocity in the drive pipe when waste valve begins to close.

The above could be achieved with long drive pipes and waste valves that were large,light in weight with short strokes so that they closed fast.He theoretically derived equations for the determination of minimum length of the drive pipe as a function of head ratio, time for waste valve closure and maximum (theoretical) velocity in the drive pipe.

$$L \geq \frac{h}{H} T \sqrt{2gH} \tag{7}$$

and

$$L \geq 9 T \sqrt{\frac{2gH}{m}} \tag{8}$$

where

m = dimensionless ratio = $V_0 / \sqrt{2gH}$

L = drive pipe length.

T = time for linear waste valve closure(secs)

H = supply head above the waste valve.

Most researchers found T one of the most difficult parameters to determine reliably.The expressions for minimum drive pipe are,therefore, of little practical value because they depend on the parameter T. Bergeron concluded that since the flow escaping through the waste valve after it begins to close is wasted,the waste valve should be light and or of short stroke to minimize the losses.

Pump efficiency and power were derived as functions of head ratio, friction head losses,length of drive pipe,stroke,characteristics of the delivery valve and other factors,thus making them lengthy and complicated with parameters that are difficult to estimate.The analysis was lengthy and demonstrated the effect of various parameters on hydram performance.The author concluded that no other pumping machine worked as efficiently as the hydraulic ram.

Gosline and O'Brien [44] carried out a rational study of the hydraulic ram pump supplementing the earlier work by Harza with special emphasis on the period of retardation (discharge) and its relation with the period of acceleration. They used a 1.0 inch (25.4mm) commercial hydram with a rigid waste valve. The air chamber was recharged by a compressor. They divided the cycle of the hydram operation into four main periods which were later modified by considering the details of recoil.

The authors observed that for a waste valve of weight, W, the hydrostatic force necessary to hold it closed is

$$h_v = \frac{W}{A_v \gamma} \tag{10}$$

where

W is the weight or spring tension of the waste valve.

h_v is the hydrostatic force on the valve.

γ is the specific weight of water.

A_v is the cross-sectional area of the waste valve

The pressure must therefore, drop below this value for the pump to operate continuously. From comparison of theoretical model results and experimental data, they observed that the model generally underestimated the wasted flow and the duration of the cycle. The errors were attributed to the assumption that friction head losses did not vary with velocity in the drive pipe but were constant, possible friction in the valve guide plus other assumptions on which the whole theoretical analysis was based.Details of waste valve closure were treated in an approximate way since the complete equation could not be solved.The experiments also revealed that retardation takes place in a period less than

$$\frac{2L}{c} \tag{11}$$

where c is the speed of wave propagation or wave celerity. They showed that the maximum pressure generated when the waste valve closes is approximately equal to the maximum pressure in the air chamber if the pump operates with the delivery valve completely closed. The computed value was 2-4 % higher than the observed.They investigated the minimum discharge or delivery head (h_{min}) at which the pump will operate for different waste valve strokes and the results agreed closely with the theoretical value of the hydrostatic force on the valve (h_v). For short strokes,h_{min} was higher than for long strokes.They concluded that for continuous pump operation the delivery head must be at least twice or even three times the supply head.

From continuous records of the pressure in the drive pipe,it was observed that discharge did not take place in one single pressure surge but consisted of several surges with progressive reduction in velocity in the drive pipe due to recoil action after each surge.The number of surges increased with the reduction in delivery head.The analysis indicated that the discharge volumes computed based on one to eleven (1 - 11) surges agreed within 5% so that for practical purposes assuming a single surge during the discharge period could be sufficient.

The authors demonstrated that retardation affected subsequent acceleration and duration of the next cycle,and that the recoil velocity (negative) was a periodic function of the head ratio. Efficiency increased with the decrease of valve stroke.Regarding drive pipe length,they recommended L to be between 5H and 7H

$$5H \leq L \leq 7H \tag{12}$$

depending on friction factor and pipe diameter for long drive pipes in which the friction head losses were dominant.

The authors concluded that with basic hydraulic constants measured under steady flow conditions,complete operating characteristics of the ram could be predicted with less than 15% error.The acceleration period should be short for better water hammer effects and development of high delivery heads in the installation.The recoil at the end of a cycle greatly affected quantities wasted(Q_w)and number of beats per minute at high discharge heads.

The authors observed that the use of a mechanically operated waste valve as employed by the earlier researcher,Harza, obscured the effect of recoil and caused a change in the operating characteristics of the hydraulic ram.

Mead [38] in 1933 suggested another equation for hydram effeciency that considered the hydram as a machine exclusive of the drive and delivery pipes.He considered the friction head losses in the drive pipe(f) and the delivery pipe(F). The energy supplied to the pump becomes $(Q+Q_w)(H-f)$ and the energy delivered by the pump becomes $Q(h+F)$. The efficiency is thus given by the equation

$$e(\text{Mead}) = \frac{Q(h+F)}{(Q+Q_w)(H-f)} \qquad (13)$$

This equation gives higher values than either the Rankine formula

$$e\,(\text{Rankine}) = \frac{Qh}{Q_w H} \qquad (14)$$

or the D'Aubuisson formula

$$e(\text{D'Aubuisson}) = \frac{Q(h+H)}{(Q+Q_w)\,H} \qquad (15)$$

Mead's formula has not been used in any analyses on hydrams.

Utahara [63], carried out an experimental study of the pressure variations in the drive pipe, air chamber and the delivery pipe in relation to waste valve movement. He concluded that when the delivery head is low, the pressure in the air chamber has a significant effect on the hydram system and pressure in the waste valve chamber is almost equal to pressure in the drive pipe. For very low delivery heads, the drive pipe pressure (maximum) exceeds the valve box pressure. When the delivery head is high, the valve box pressure generally exceeds drive pipe pressure. The time during which the waste valve is open increased with delivery head but the time for valve closure decreased so that the total time was almost constant. The author noted that the waste valve opened at the first lower pressure wave (recoil) after the delivery valve closed. However, this does not agree with the earlier results of Gosline and O'Brien [44] and the later results of Lansford and Dugan [37].

Lansford and Dugan [37] at the University of Illinois carried out an analytical and experimental study of two hydraulic rams. The study was to complement the earlier work by Gosline and O'Brien [44] in that different types of hydrams with elastic and resilient disc waste valves were used instead of the rigid waste valve used by Gosline and O'Brien. The effects of elasticity of drive pipe and waste valve parts, neglected by the earlier researchers, were also included in the analysis. Pressure time diagrams were obtained with an electrical pressure gauge and oscillograph. The authors divided the cycle of operation into six periods starting at the instant the waste valve began to close.

The authors stated that an exact theoretical analysis of the period of the waste valve closure was impossible. They observed that the velocity at which the waste valve began to close, V_0, was a function of the shape of the valve, valve box, length of stroke and weight of waste valve and these factors were not easily quantified. Therefore, V_0, was determined experimentally.

An analysis of the valve closure included effects of the elasticity of valve disc and valve box which other researchers had neglected. The period covering the delivery valve opening and closure was an extension of Gosline and O'-Brien's work with the additional fact that the elasticity of the valve was included.Discharges were analysed as a series of surges until pumping ceased.They indicated that assuming rigid waste valves as was done by Gosline and O'Brien [44],could lead to an error of less than 5% in computed discharge(Q) values provided the number of surges used equalled or exceeded 5,for the 4 inch(102 mm) hydram.

From the experimental study,they concluded that an increase of delivery head decreased the quantities wasted and pumped per cycle.Decreasing V_0 ,had the same effect on these quantities. Increasing the delivery head decreased duration of the cycle provided the delivery head was below a certain value,at which point the duration of the cycle was a minimum.Beyond this point increasing delivery head tended to increase the duration of the cycle though not at a constant rate.These changes were more marked in the 4 inch (102 mm) hydram than in the 2 inch (51 mm) hydram. A decrease of V_0 ,also decreased the duration of the cycle,thus increasing the valve beat frequency. The authors identified V_0 as the dominant factor controlling the functioning of the hydram.They noted that V_0 depended mainly on the valve stroke.

The quantities pumped and wasted per cycle were predicted within 10% error for delivery heads less than half the maximum pressure the hydram could develop.For higher delivery pressures,the error was much higher,particularly for the small 2-inch (51 mm) hydram. The duration of the cycle was predicted with less than 10% error for the 4-inch (102 mm) hydram and 20% for the 2-inch(51 mm) hydram. The results confirmed that the elasticity of the waste valve should not be neglected.It was also noticed that the delivery head for which maximum efficiency was attained increased as V_0 increased. The maximum efficiency varied very little with the various adjustments of the waste valve except for extremely high values of V_0 ,for which the efficiency was somewhat lower.This was probably due to the fact that most commercial hydrams are designed in such a way that they will operate with almost constant efficiency with minimum adjustment over a wide range of operating conditions.

Krol [32] presented a theoretical analysis for the prediction of hydram performance and verified it on a specially designed hydraulic ram.He went further than the earlier investigators [37], [44] and developed theoretical expressions for efficiency and power for the hydraulic ram.

Krol divided the cycle into six periods,starting with the instant the waste valve is fully open with the water in the drive pipe at rest. He developed lengthy and detailed equations to determine flow rates and the duration of each period of the cycle.The main characteristics of the hydram,namely the head losses in the drive pipe,head losses of the waste(impulse) valve,its drag force coefficient and head losses during discharge were determined empirically.

The author made the assumption that friction losses in the drive pipe remained constant and the flow velocity in the pipe at the moment the valve began to close and that at which it closed were equal.The effect of the air chamber was not considered.From the experiments on his particular hydram,the maximum deviation between observed and theoretical values did not exceed 15%.

He noted that efficiency was not the only important operating criterion. A pump could operate with reasonable efficiency but with a negligible pumping capacity or power. He stressed that the selection of a waste valve load was most important since an increase of valve weight increased the power of the pump and its capacity to work under increased delivery heads.The author noted that increasing valve weight also increased efficiency up to a certain limit after which it decreased,tending towards zero. There is a relationship between valve weight and stroke and in general increasing valve stroke decreased the permissible load on the valve. This was due to the decrease in drag coefficient produced by the increased stroke length.

Krol also provided a solution to the question of desirable drive pipe length and showed that with other things remaining constant, increasing the length generally increased the amount of water delivered and wasted per cycle and the duration of the cycle. Efficiency and power also increased to an optimum value after which they decreased to zero.

It was ascertained experimentally that variation of diameter and grade of hardness of rubber discs used on the delivery valve had minor effect on ram performance. This is at variance with the results of Dugan and Lansford [37]. Increased hardness and rubber thickness may be necessary for increased delivery heads.

Experiments showed that using larger air chambers increased pump efficiency and power. Efficiency increased by approximately 10%, whem air chamber volume was doubled. This was due to the additional potential energy available for pumping formed by water temporarily stored in the air chamber during the period of retardation.

Krol in another paper[33] drew some conclusions based on the theory and data of his earlier research [32]. He examined flow volumes wasted and pumped, valve beat frequency, efficiency and power as functions of the waste valve weight and stroke. He concluded that for each stroke setting, there was a waste valve weight, (W_{max}) for which the valve would not close at all.

The quantity of water delivered per cycle varied in direct proportion to the hydrostatic force on the waste valve. Wasted flow (Q_w) and cycle time (T) tended to infinity as W approached W_{max}. The efficiency and power became zero for W=0 and W = W_{max}. From his experimental data and theoretical formulation, he concluded that there was a relationship

$$S * W_{max} = \text{const.} \tag{16}$$

where

S - waste valve stroke length.

W_{max} - maximum weight of the waste valve or spring tension.

The author pointed out that where water is abundant, efficiency is of secondary importance and power delivered should be the important operating criterion.

Iversen [28] attempted to predict the hydram's complete performance analytically in a simplified way with no details on waterhammer and friction head losses. He assumed the cycle to be composed of the acceleration and discharge periods only. The author equated head losses during acceleration to those during retardation. Pump efficiency, duration of cycle and power became functions of only the head and velocity ratios. He defined velocity ratio as the ratio between the velocity in the drive pipe when valve began to close (V_0) to steady state flow velocity, V_s

$$V_s = \sqrt{\frac{2gH}{M}} \tag{17}$$

where M is the total friction head loss factor of the drive pipe and the waste valve during the acceleration period.

He tested his model on data of earlier researchers, Gosline and O'Brien [44]. The author could not predict the cycle duration, frequency and efficiency with high accuracy. However, the trends were well predicted. The author simply fitted the model to the data by assuming and trying different velocity ratios (V_0 /V_s).

Calvert [11] used dimensional analysis to reduce the independent variables in the hydram operation to five dimensionless parameters, namely; the Reynolds number (Re), Froude number (Fr), Mach number (Mc), head ratio and friction coefficient. The dependant variables were pumped flow (Q), wasted flow (Q_w) and valve beat frequency (VB). Calvert defined Re, Fr and Mc as follows:-

Reynold's number, Re

$$R_e = \frac{V_0 D}{VB} \tag{18}$$

Froude number, Fr

$$Fr = \frac{\sqrt{2gH}}{V_0} \tag{19}$$

Mach number, Mc

$$Mc = \frac{V_0}{c} \tag{20}$$

The author carried out experimental trials to determine trends rather than absolute values from which general conclusions applicable to all hydrams could be drawn. Actual numerical values were expected to vary according to the particular design. From the experiments, he concluded that:-

1. The Froude number(Fr), defines the limits within which the hydram will operate. The hydram will not operate at very low and high Fr values. The Reynold's number(Re), has no significant effect in the common hydram sizes on Q, Q_w and valve beat frequency, and it can be ignored.
2. For satisfactory values of Fr, only the head ratio determines pump efficiency and performance. The efficiency falls at high head ratios due to excessive shock losses and at low head ratios due to resistance of the delivery valve.
3. The Mach number(Mc) has little effect unless unusual materials like rubber or wood are used in the construction. Fluid friction is important and like other investigators, he recommended the drive pipe and ram should be smooth and with no bends or obstructions. His results indicated the wide range of conditions under which Mc, Re, Fr have litle influence on the hydram operation. Calvert found that the best operating conditions occurred with high supply heads and low head ratios.

Further to his earlier work, Calvert investigated the optimum drive pipe length [12]. He observed the effect of the ratio L/D on Q_w, Q, and valve beat. His results were consistent and he concluded that pump output, efficiency and stability were independent of the length of the drive pipe length for L/D ratios between the limits

$$150 \leq \frac{L}{D} \leq 1000 \qquad (1)$$

The pump worked poorly outside these limits. To minimize valve wear, liability to fatigue and noise, a low valve beat frequency is desirable. He, therefore, suggested that drive pipe lengths longer than the minimum should be used.

Inversin [27] did experiments on hydrams from pipe fittings to investigate pump operation under several operating conditions. He concluded that the valve stroke affected pumped and wasted flows and that efficiency increased with valve stroke up to a point then decreased continuously. The valve beat decreased exponentially with an increase in valve stroke. The size of the air chamber had negligible effect on pump performance. The author found that increasing the valve weight had the same effect as imcreasing valve stroke. He also noted that with supply heads higher than 4m the valve frequency was very high so that water delivered per cycle was very small. Increasing waste valve weight reduced the valve beat frequency and increased the water delivered. The resulting increase in inertial forces, however, would destroy the pump quickly unless a spring was used to regulate the waste valve closure. The author experimented with and showed that PVC pipes could be used for drive pipes with reasonable results,if necessary.He found the air valve size was not critical.The hydram efficiency and output were primarily a function of the valve beat frequency other factors being constant.Valve beat could be regulated by either increasing spring tension(valve weight)or valve stroke.

Chapter III

THEORETICAL ANALYSIS OF THE HYDRAULIC RAM PUMP.

3.1 GENERAL APPROACH.

A detailed description of the theory of the hydraulic ram results in many complex equations, some of which are so detailed and complicated as to be of little practical use. The important parameters in the operation of the hydraulic ram are best obtained experimentally. Theory alone is not able, presently, to predict the hydram operation.

The analysis presented here borrows from the work of earlier researchers like Gosline and O'Brien[44], Lansford and Dugan[37], Krol [32] and Iversen[28] and lies between the simplified analysis of Iversen[28], and the detailed analysis of Krol[32]. It is considered to be an improvement of Iversen's analysis.

The analysis differs from that of Iversen in that:-

1. The pumping cycle is divided into four periods instead of just two.
2. The friction losses during acceleration and discharge periods are empirically determined and explicitly used in the analysis.
3. The major effects of waterhammer are included in the analysis for better estimation of discharge velocity and pumping rates.

4. The period of waste valve closure is separated from the acceleration period on the assumption that the water escaping through the waste valve during closure does not do any useful work, but is wasted. The water flowing to waste before the valve begins to close is regarded as the useful drive power essential for hydram operation.

The analysis differs from that of Krol in that:-

1. Only four periods are used instead of seven that Krol used.
2. The concept of physical water separation from the waste valve and negative velocity in the drive pipe during recoil are neglected.
3. The friction losses in the drive pipe are included in the analysis of the pumping period, together with friction losses through the delivery valve. Krol [32] only considers the friction losses through the delivery valve during the pumping period.
4. The details of waste valve closure are simplified.

This results in a simple model based on the main periods of the cycle of hydram operation. The expressions for wasted and pumped flow together with efficiency, power cycle duration are derived. The effects of drive pipe velocity on wasted ~~and~~ pumped flow, efficiency, power and valve beat frequency are also demonstrated. The effects of drive pipe length, weight of waste valve and friction losses in the drive pipe and valves on hydram operation are examined.

The analysis is based on the position of the waste valve and time- velocity variation in the drive pipe.As a result only the velocity in the drive pipe during waste valve closure(V_0)is needed for the computations.

The cycle of operation is divided into 4 main periods as follows.

Period 1

The waste valve is fully open and water starts to flow from the supply tank and escapes through the waste valve,till a certain velocity is attained in the drive pipe and the waste valve begins to close.

Period 2

The waste valve continues to close and is fully closed.

Period 3

The waste valve is fully closed and remains closed.The delivery valve opens and pumping takes place until it ceases.

Period 4

Delivery valve closes,recoil takes place,waste valve opens and pressure in the drive pipe returns to the static supply presssure.

The time-velocity history in the drive pipe for one cycle of hydram operation is given schematically in figure 7.

The formulation for friction head loss through the delivery valve used in the analysis is of the form

$$hr = HVD \frac{V_1^2}{2g} \left(1 - \frac{h}{h_{max}} \right) \qquad (21)$$

where HVD is the friction head loss factor of the delivery valve alone and is equal to N minus the head loss factor of the drive pipe (XM).

3.2 ANALYSIS OF THE PUMPING CYCLE

The Analysis of Period 1.

The waste valve is fully open and water accelerates in the drive pipe and escapes through the waste valve under the effect of the supply head, H. Theoretically, the water in the drive pipe accelerates according to the equation.

$$H - \left(1 + \Sigma h_L + f\frac{L}{D} + RS \right) \frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \qquad (22)$$

$$\left(1 + \Sigma h_L + f\frac{L}{D} + RS \right) = M \qquad (23)$$

putting equation (22) becomes

$$H - M\frac{V^2}{2g} = \frac{L}{g}\frac{dV}{dt} \qquad (24)$$

Σh_L - minor losses in the drive pipe and entrance.

f - friction factor of the drive pipe.

RS - friction head loss factor of the waste valve alone.

V - velocity of water in the drive pipe.

Acceleration continues till a velocity, V_0, is attained and the total force on the waste valve equals its weight, W. Thereafter, the force exceeds W and the valve begins to close.

The force on the valve can be represented as a drag force(F_D) by the equation.

$$F_D = C_D A_v \gamma \frac{V^2}{2g} \tag{25}$$

A_v -cross-sectional area of the waste valve.

γ -specific weight of water.

C_D -drag coefficient of the waste valve.

C_D, the drag coefficient is the ratio of the dynamic and drag forces acting on the valve.

$$C_D = \frac{F_D}{A_v \gamma \frac{V^2}{2g}} \tag{26}$$

At the point of start of valve closure, velocity of the water in the drive pipe, V_0, is obtained from the equation (25) with F_D =W, weight of the waste valve.

$$W = C_D A_v \gamma \frac{V_0^2}{2g} \tag{27}$$

so that $$V_0 = \sqrt{\frac{2gW}{C_D A_v \gamma}} \tag{28}$$

therefore,

$$V_1 \doteq V_0 = \sqrt{\frac{2gW}{C_D A_v \gamma}} \tag{29}$$

V_0 can be obtained experimentally and C_D evaluated for the waste valve weight and stroke setting.

The duration of period 1, t_1, is obtained by integrating equation (24) for V between the limits V=0 and $V = V_1$.

$$\int_0^{t_1} dt = 2L \int_0^{V_1} \frac{dV}{(2gH - MV^2)} \tag{30}$$

$$t_1 = \left(\frac{L^2}{2gHM}\right)^{0.5} \ell n \left[\frac{\left(\sqrt{\frac{2gH}{M}} + V_1\right)}{\left(\sqrt{\frac{2gH}{M}} - V_1\right)}\right] \tag{31}$$

Substituting for V_1

$$t_1 = \left(\frac{L^2}{2gHM}\right)^{0.5} \ell n \left[\frac{1.0 + \sqrt{\frac{WM}{C_D A_v \gamma H}}}{1.0 - \sqrt{\frac{WM}{C_D A_v \gamma H}}}\right] \tag{32}$$

If the acceleration of flow in the drive pipe continues till the average steady flow velocity, V_s, is reached, before the valve begins to close, the equation for V_s is obtained from equation (24) with dV/dt =0

$$V_s = \sqrt{\frac{2gH}{M}} \tag{33}$$

V_0 can be expressed as a fraction of V_s, such that

$V_0 = mV$, where m is a fraction such that $0 < m < 1.0$

$$V_0 = V_1 = m\sqrt{\frac{2gH}{M}} \qquad (34)$$

Equation (32) can therefore be re-written as

$$t_1 = \left(\frac{L^2}{2gHM}\right)^{0.5} \ln \left[\frac{1.0 + \frac{V_1}{V_s}}{1.0 - \frac{V_1}{V_s}}\right] \qquad (35)$$

If the the expressions for V_1 and V_s are substituted in equation (35), the result is, again, equation (32). The ratio m is given by V_0/V_s so that

$$\frac{V_0}{V_s} = \frac{V_1}{V_s} = m = \sqrt{\frac{WM}{C_D A_V \gamma H}} \qquad (36)$$

From equations (32) and (35), it can be seen that if the valve does not close until the steady state flow velocity, V_s, is attained, then $V_0 = V_s$, and the duration of period 1, $t_1 = \infty$. The valve will not close at all . The maximum permissible waste valve weight can be obtained by putting m = 1 in equation (36) so that

$$WM = C_D A_V \gamma H \qquad (37)$$

then

$$W = W_{max} = \frac{C_D A_V \gamma H}{M} \qquad (38)$$

Equation (38) shows that the weight of the waste valve has to be less than $\frac{C_D A_V \gamma H}{M}$ for it to close.

For continuous pump operation, W has to be between the limits

$$0 < W \leq \frac{C_D A_V \gamma H}{M} \tag{39}$$

Krol[32] gives the same equation without the same derivation.

The volume of water, $\forall_1$, flowing to waste during period 1 is given by

$$\int_0^{t_1} AV dt = \forall_1 = \int_0^{V_1} 2AVL \frac{dV}{\left(\frac{2gH}{M} - V^2\right)} \tag{40}$$

$$\forall_1 = \frac{LA}{M} \left[\ln \left(\frac{1.0}{1.0 - \frac{MV_1^2}{2gH}} \right) \right] \tag{41}$$

$$\text{or } \forall_1 = \frac{LA}{M} \left[\ln \left(\frac{1.0}{1.0 - \frac{V_1^2}{V_s^2}} \right) \right] \tag{42}$$

Equation(42), also, shows that if $V_1 = V_s$, $V_3 = \infty$ and the pump will not work.

From equation(39), Krol[32] showed that the question of suitable drive pipe length can be solved as follows;

$$W = \frac{C_D\ A_V\ \gamma\ H}{M} = \frac{C_D\ A_V\ \gamma\ H}{(1+\Sigma h_L+f\frac{L}{D}+ RS)} \tag{43}$$

$$\text{so that } \frac{L}{D} = \frac{C_D A_V \gamma H - W(1+\Sigma h_L+RS)}{W\ f} \tag{44}$$

The drive pipe length should lie within the limits

$$0 < L \leq D\left[\frac{C_D A_V \gamma H - W(\ 1+\Sigma h_L+RS\ )}{W\ f}\right] \tag{45}$$

for the valve to close and the pump to work automatically and continuously.

The kinetic energy in the drive pipe at the end of period 1 is

$$\frac{1}{2}\ \pi\ \frac{D^2 L}{4}\ \frac{\gamma}{g}\ V_1{}^2 = \frac{\pi\ D^2}{4 C_D A_V}\ W\ L \tag{46}$$

<u>The Analysis of Period 2.</u>

The waste valve continues to close. The details of waste valve closure are complicated since both water and the valve are moving. The valve closure will, therefore, depend on several parameters including velocity of water in the drive pipe, weight of valve, shape of valve box, dimensions of valve and the orifice. These parameters are difficult to quantify. Theoretically, the exit velocity through the orifice increases as closure progresses. However, since the valve stroke is usually small, the distance moved by the valve is small and the flow resistance through the valve increases as the valve closes; it is probable that the maximum velocity V_m, in the drive pipe just before the valve closes completely, does not differ significantly from V_1. Supported by evidence from detailed measurements of velocity and pressure variation in drive pipe, earlier researchers Krol[32], Gosline and O'-Brien[44] and Lansford and Dugan[37], assumed $V_m=V_1$. Therefore the Velocity in the drive pipe at instant of valve closure, V_2 is approximately equal to V_1 which is equal to V_m and V_0 and is given by equation (16).

$$V_2 = V_0 = \sqrt{\frac{2Wg}{C_D A_V \gamma}} \tag{47}$$

Since V_1 is assumed to be equal to $V_0 = V_m$, there is no acceleration during this period. The duration of period 2 is estimated by assuming instantaneous valve closure. It is, therefore, assumed

$$t_2 = \frac{2L}{c} \tag{48}$$

Secondly, since the flow out of the waste valve during this period is not directly included in the computations, this approximation is regarded as adequate. Earlier researchers found this period difficult to estimate and used various approximations. It was, however, noted, that valve closure took place in less than t_2 given by equation (48) [44].

Period 2 is separated from period 1 due to the assumption that the quantity of water escaping after the valve starts to close is wasted, and the energy is wasted. The quantity of water flowing to waste during period 1 is considered as the useful drive power for the hydram.

The quantity of water escaping to waste during period 2 is given by the equation,

$$\forall_2 = A\ V_2\ t_2 \tag{49}$$

$$\forall_2 = A\ V_2\ \frac{2L}{c} \tag{50}$$

Analysis of Period 3.

At the end of period 2, there is an abrupt retardation of water as the waste valve closes. The velocity of water in the drive pipe is reduced in proportion to the pressure head (h+hr) generated in excess of the static supply head, H. According to the waterhammer theory, for an instantaneous valve closure in an inelastic pipe, the maximum theoretical pressure rise ΔH is given by the equation

$$\Delta H = - \Delta V \frac{c}{g} \tag{51}$$

The speed of an accoustic wave (or wave celerity), c, in an elastic medium is given by the formula

$$c = \sqrt{\frac{K_c}{\rho}} \tag{52}$$

where Kc, the composite modulus of elasticity of water and pipe material replaces K, the usual bulk modulus of elasticity of water and is given as

$$K_c = \frac{1}{\left(\frac{1}{K} + \frac{D}{E\, t_p} \right)} \tag{53}$$

according to Joukowski [46],

$$K_c = \frac{1}{\left(\frac{1}{K} + \frac{D}{Et_p} \right)\left(5 - \frac{4}{\sigma}\right)} \tag{54}$$

according to Gibson[23].

Krol[32], found that the two equations give close results for ferrous materials. Therefore equation (53) is used in this analysis.

The pressure increase due to waterhammer in excess of that in the air chamber forces the delivery valve open and discharge commences.It is assumed that prior to complete waste valve closure during period 2,the pressure in the hydram is zero or negligible.Experiments by Lansford and Dugan[37] showed that there is a small pressure but very small in comparison to Δp,after valve closure.It can,therefore,be neglected particularly for high delivery heads.The assumption,however,may introduce errors at low delivery heads.

If V_3 is the maximum velocity of water in the drive pipe during period 3,then there is a relationship

$$(V_3 - V_2)\,\frac{c}{g} = -\,(h + hr) \tag{55}$$

Experiments[37],[44]showed that at instant of valve closure, the pressure head rise ΔH exceeds slightly $(hr+h)$ for a short time and that generally equation (55) is approximately correct.The velocity during period 3 is therefore

$$V_3 = V_2 - (\,h + hr\,)\,\frac{g}{c} \tag{56}$$

since $V_2 = V_1 = V_0$,

$$V_3 = V_0 - (\,h + hr\,)\,\frac{g}{c} \tag{57}$$

The maximum,theoretical pressure head that the hydram can develop is obtained from equation (45) for $V_3=0$ and $hr=0$

$$h_{max} = h = \frac{c\ V_2}{g} = \frac{c\ V_0}{g} \qquad (58)$$

Loss of kinetic energy due to the reduction of velocity at the start of period 3 is given as

$$\frac{1}{2}\ \pi\ \frac{D^2}{4}\ L\ \frac{\gamma}{g}\ (\ V_2{}^2 - V_3{}^2) \qquad (59)$$

Discharge continues at the rate governed by the equation

$$-\ h\ -\ \frac{NV^2}{2g}\ =\ \frac{L}{g}\frac{dV}{dt} \qquad (60)$$

where N is the total head loss factor for the drive pipe and delivery valve and h is the delivery or pump head [28].

Discharge continues until the velocity becomes zero and the delivery valve starts to close [28]. The duration of period 3 is obtained by integrating equation (60) for the limits $v = V_3$ and $v = 0$.

$$\int_c^d dt\ =\ -\ 2L \int_{V_3}^{0} \left(\frac{dV}{2gh + NV^2} \right) \qquad (61)$$

so that the pumping time, t_3, is given by the equation

$$t_3 = \left(\frac{2L^2}{ghN} \right)^{0.5} \tan^{-1} \left(\frac{NV_3{}^2}{2gh} \right)^{0.5} \qquad (62)$$

V_3 can be expressed as a dimensionless ratio of V_s, so that t_3 becomes

$$t_3 = \left(\frac{2L^2}{ghN} \right)^{0.5} \tan^{-1} \left(\frac{N}{M}\ \frac{H}{h}\ \frac{V_3{}^2}{V_s{}^2} \right)^{0.5} \qquad (63)$$

The quantity of water discharged during period 3 is

$$\int_c^d A\,V dt = \forall_3 = \int_{V_3}^{0} \frac{AVL}{g} \left(\frac{dV}{h + \frac{NV^2}{2g}}\right) \qquad (64)$$

so that

$$\forall_3 = \frac{AL}{N}\left[\ln\left(\frac{NV_3{}^2}{2gh} + 1.0\right)\right] \qquad (65)$$

$$\text{or} \quad \forall_3 = \frac{LA}{N}\left[\ln\left(\frac{N}{M}\frac{H}{h}\frac{V_3{}^2}{V_s{}^2} + 1.0\right)\right] \qquad (66)$$

Pumping continues until the kinetic energy in the drive pipe is exhausted the velocity becomes zero and pumping stops and delivery valve closes.

Analysis of period 4.

The delivery valve is completely closed.Near the delivery valve the water is under the pressure head (H+h+hr) while at the other end of the drive pipe,the pressure is due to the static supply head,H.The pressure difference and the elasticity of water and pipe material causes the water to recoil towards the supply tank.The recoil causes a reduction of pressure in the hydram to below atmospheric pressure causing a small quantity of air to be sucked into the hydram through the snifter valve. The waste valve opens due to its own weight and reduction of pressure in the hydram.The opening of the waste valve is assumed to be instantaneous.The pressure in the drive pipe returns to the static supply pressure and the next cycle of operation begins.Experiments by Utahara [63] showed that the waste valve opened on the first pressure wave after the delivery valve closed(i.e first recoil).Therefore,the duration of period 4 can be estimated as the time for a complete reflection of an acoustic wave,

$$t_4 = \frac{2L}{c} \tag{67}$$

after which the remaining pressure waves in the drive pipe are considered negligible in comparison to the main flow and average pressure difference.

Krol[32] and Lansford and Dugan [37] argue that the energy converted from kinetic energy to 'strain' energy when the waste valve closes,is again reconverted into kinetic energy when the delivery valve closes and accelerates the flow towards the supply tank.This action causes physical water separation from the waste valve during recoil.It is difficult to see why all the energy should be reconverted to kinetic energy since some of the energy was expended forcing open the delivery valve,compressing air and water in the air chamber,creating pressure waves in the drive pipe,straining the drive pipe pump material and possibly the waste and delivery valve materials (if resilient types are used).The lag in energy conversion between the components listed above plus the overlap of events during this period makes it difficult to assume a consistent 'energy flow' in one direction only and the reconversion of energy without any 'losses'. It is,therefore,difficult to estimate the actual energy available to accelerate the flow towards the supply tank.Secondly,if the supply reservoir is big,the recoil effect will be minimized and pressure waves will dampen out fast.

3.3 OPERATING CHARACTERISTICS -SUMMARY.

Duration of the pumping cycle, $T = t_1 + t_2 + t_3 + t_4$ (68)

$$= \left(\frac{L^2}{2gHM}\right)^{0.5} \ln\left(\frac{1+\frac{V_0}{V_S}}{1-\frac{V_0}{V_S}}\right) + \frac{4L}{c} + \left(\frac{2L^2}{ghN}\right)^{0.5} \tan^{-1}\left[\frac{NV_3^2}{2gh}\right]^{0.5} \qquad (69)$$

(in seconds).

Average pumped flow per second, $Q = \forall_3/T$ (70)

$$= \frac{LA}{NT}\left[\ln\left(\frac{NV_3^2}{2gh} + 1.0\right)\right] \qquad (71)$$

(litres/second).

Flow ratio $FR = Q/Q_w = \forall_3/\forall_1$ (72)

$$= \frac{M}{N}\left[\ln\left(\frac{NV_3^2}{2gh} + 1.0\right)\right] \Big/ \left[\ln\left(\frac{1.0}{1.0 - \frac{V_0^2}{V_S^2}}\right)\right] \qquad (73)$$

Pump efficiency (Rankine), $e = \dfrac{\forall_3\, h}{\forall_1\, H}$ (74)

$$= \frac{h}{H}\frac{M}{N}\left[\ln\left(\frac{NV_3^2}{2gh} + 1.0\right)\right] \Big/ \left[\ln\left(\frac{1.0}{1.0 - \frac{V_0^2}{V_S^2}}\right)\right] \qquad (75)$$

Pump power, $P = Q\gamma h$ (76)

$$= 9.81h\frac{LA}{NT}\left[\ln\left(\frac{NV_1^2}{2gh} + 1.0\right)\right] \text{ (watts)} \quad (77)$$

Valve beat frequency, VB =60.0/T (beats per minute) (78)

Total wasted flow in litres/second

$$= \frac{LA}{MT}\left[\ln\left(\frac{1.0}{1.0 - \frac{V_0^2}{V_S^2}}\right) + \frac{2V_0}{c}\right] \text{ (litres/second)} \quad (79)$$

Total supply flow to the pump (Q_s) in litres/second

$$Q_S = Q_W + Q + \frac{V_2}{T} \quad (80)$$

Head ratio (HR) = $\frac{h}{H}$ (81)

Chapter IV

EXPERIMENTAL PROCEDURE

4.1 APPARATUS AND EXPERIMENTAL SET UP

The apparatus was set up as shown in figure 8,to measure the total flow volume supplied to the pump and that delivered by the pump during its operation for variable supply and delivery heads.

The flow pumped into the supply tank (A),was controlled by a valve (C).The water level in the supply tank was kept at the desired constant level by two 102 mm diameter overflow pipes that drained away the excess water.The supply tank had a capacity of about 2.5 cubic meters and was supported at about 1.0m above the laboratory floor.

A steel drive pipe(schedule 40),with a nominal diameter of 32 mm and a length of 15.50m connected the supply tank and the hydram.The drive pipe was horizontal and supported at regular intervals.The drive pipe was fitted with a gate valve (A),15.3m from the supply tank.

A small copper pipe of nominal diameter 19 mm was connected at the other end of the hydram and acted as the discharge pipe.It was fitted with a Bourdon pressure gauge near the pump and a stop-cock,(valve B).

A large tank,(B),1.97mx0.90mx0.90m was the collecting tank,and collected all the flows that escaped through the waste valve and through the discharge pipe.The tank was fitted with a short 102 mm overflow PVC pipe at a point 0.60m from the bottom,which discharged into the calibrated measuring tank(C).The measuring tank was fitted with a glass gauge tube (D) with a scale from which the water level in the tank was measured.The tank had a surface area of 0.392 square meters and was 0.5 m high.There was also a 32 mm drain pipe fitted with a gate valve to drain the measuring tank(C)after each measurement.

Across the large tank (B),two metal bars were bolted on the top edges at both ends.The hydram and end of drive pipe were supported and anchored on these metal bars for extra support against horizontal and lateral movement.In the experiment,the supply was varied by changing the height of the overflow pipes in tank A.Three values of supply head(H) of 1.0m,1.5m and 2.0m were possible.Higher supply heads of up to 2.5m could only be obtained by slanting the drive pipe and lowering the test hydram into the tank B.The delivery head and delivery flow were varied with valve B.The waste valve stroke length and weight (or spring tension) of the test hydrams could also be varied.

A simple cover of polythene sheet on a steel-wire cage was used to stop water from splashing so that all the water escaping from the waste valve fell into the tank B as required.

Two pressure gauges,were used,one with a range of 60 psi (414 kPa)and the other with the range of 160 psi(1103 kPa).

4.2 MEASUREMENTS

With the experimental set up shown in figure 8,the performace of various hydraulic ram pumps was investigated.

To start the experiment valve A was opened and water flowed from the supply tank to the hydram.The valve A was also used to stop the flow when the hydram pump was being changed.Valve A was also used to regulate the flow in the determination of V_0,the velocity at which the waste valve started to close,and the drag coefficient of the waste valve.

The valve B was adjusted to obtain the desired delivery flow rate for the corresponding steady total delivery head(Hd).The delivery head was then read off the pressure gauge.The discharge or delivery flow was measured with calibrated measuring cylinders of 1.0 litre and 2.0 litre capacity.Time measurements were taken with two stop-watches. A spring balance was used to measure the spring tension for the hydrams whose waste valve was controlled by a spring.The total flow volume supplied to the hydram (wasted and delivered) was measured volumetrically with the measuring tank C.

4.2.1 Basic parameters measured.

In the experiment,the following were measured:-

1. Supply head (H) in meters.
2. Total delivery head (Hd) in meters.
3. Delivery flow (Q) in litres per minute.
4. Total supply flow (Q_s) in litres per minute.
5. Valve beat frequency (VB) in beats per minute.

The total delivery pressure could be measured to the nearest 3.5 kPa on the 60-psi pressure gauge and 7.0 kPa on the 160-psi pressure gauge. The water level in the measuring tank could be read to the nearest 1.0 mm.The discharge flow (Q) could be measured to the nearest 5.0 milliliters with the 1.0-liter cylinder and 10.0 milliliters with the 2.0-liter cylinder.Time could be measured to the nearest 0.10 seconds with the stop watches.

4.3 HYDRAULIC RAM PUMP TESTS.

The tests consisted of two main parts.In the first part,five small hydrams were tested to establish their operating characteristics. Some of the hydrams tested like the Blake represented the smallest models produced by the company.The tests were to compare small hydrams for small scale applications.Some of the hydrams fitted this category better than the others.The Blake for instance was designed for large scale applications compared to the Davey or the Fleming hydrams. In the second part,extensive tests were done on the locally made hydram to improve its design and performance and establish its operating characteristics.

4.3.1 Tests on the five hydraulic ram pumps.

Four 32 mm (1.25 inch) hydrams were tested to compare their operating characteristics.These were the BLAKE (smallest size)(fig. 9),the Davey (model 4) (fig. 4),the RIFE (model 10/15 BU) (fig. 10) and the original ITDG (Intermediate Technology Development Group) hydram from pipe fittings with spring-type impulse waste valve (figure 12) hydrams.The fifth hydram tested was a 25.4 mm (1 inch) Fleming hydram (model FR102)(fig. 11).The tests were to investigate how the various small hydrams of different make and design compare in operation.A comparative study of these hydrams had not been found in the literature.The tests were also to identify what aspects of a design make a particular hydram operate better than the others.The hydrams were tested for supply heads of approximately 1.0m,1.5m and 2.0 m except for the Blake hydram,which due to its design neccesitated an angle in the drive pipe thereby increasing the maximum supply head to 2.5 m.The actual supply head values differed slightly in each hydram depending on the waste valve design.The waste valve opening was taken as the datum in each hydram.The delivery head was varied from the highest which the pump could achieve with zero delivery flow to the minimum head at which the pump could still work.The range,in general,varied from

600 kpa to 20kpa. The waste valve weight or tension and valve stroke were also varied. Thus the perfomance of the pumps under the varying conditions was examined. Operating characteristics were established and general comparisons made. All pumps were compared on the basis of maximum efficiency at the maximum supply head used.

4.3.2 <u>Design modification and tests on the ITDG hydram</u>

From the tests and analysis in part 4.3.1, the locally made ITDG hydram was modified for better operation. The pump adjustment was found to be difficult and hydram operation was not smooth. The pump was weak. The spring-operated waste valve (fig 13) was, therefore replaced with a weighted-impulse valve (fig.15). Steel valve spindles did not work well and were replaced with a brass spindle which worked well and moved freely through the spindle guide without sticking. Several tests on the following aspects of the pump proportions were then investigated for better pump design and performance:-

1. <u>Stroke of the waste valve(S)</u>. The stroke of the waste valve was varied to investigate its effect on the hydram efficiency and pumping capacity and general operating characteristics. The stroke was varied from 1.0 mm to 8.0 mm.
2. <u>Weight of the waste valve</u> (<u>W</u>). The weight was varied between 0.350 kgs to 1.157 kgs to investigate its ef-

fect on the pump operation and capacity. Valve stroke of 1.0mm was used.

3. <u>Waste valve orifice</u> (A_0). The diameter of the waste valve opening was changed and three different diameters of 2.5 cms, 3.0 cms and 3.5 cms used to investigate the 'choking' effect of A_0 on pump characteristics.

4. <u>Waste valve area</u> (A_v). The diameter of the waste valve was varied and three diameters of 3.9 cms, 4.6 cms and 5.0 cms were used. The three sizes were tested in combination with each of the three waste valve orifice sizes and a waste valve stroke of 2.0 mm.

5. <u>Frictional head loss through the delivery valve</u>. The resistance of the delivery valve was changed by varying the thickness of the rubber on the flat perforated plate of the valve. Six different rubber thicknesses of 1.8 mm, 2.6 mm, 3.1 mm, 6.0 mm, 9.1 mm and 16 mm were used in the tests.

6. <u>Air chamber size</u>. The air chamber length was varied for a standard 50.8 mm diameter steel pipe to investigate the effect of volume and height of the air chamber on hydram operation. Four lengths of 0.30 m, 0.61 m, 1.0 m and 1.52 m were used in the experiment.

7. <u>Size of the air(snifter) valve</u>. Tests were done to investigate the effect of the air valve size on the

performance of the hydram.Three diameters of 0.41 mm,0.72 mm and 1.41 mm were used.

4.3.3 Frictional head loss factors

Experiments were done to estimate the friction head loss factor(XM) for the drive pipe alone under steady flow conditions with the hydram removed.Tests were also done to estimate the friction head loss factors for the waste valve(RS) as a function of the valve stroke for the Davey and modified locally-made hydrams.The measurements were done by keeping the stroke length constant and measuring the flow through the waste valve under steady flow conditions.Similarly,the head loss factors (HVD) for the delivery valves were determined with air chambers removed and the waste valve fully closed.All the flow was through the delivery valve under steady flow conditions.Finally,an attempt was made to measure V_0,the velocity at which the waste valve started to close.The flow in the drive pipe was regulated with the valve C until a flow rate was obtained through the waste valve such that the valve began to close.The flow was then reduced slightly in such a way that a steady flow rate through the waste valve, was maintained without the valve closing.These measurements were approximative and could only give the estimates rather than precise values.These values were later used in the theoretical models.These values were determined for the supply head of 2 m only.The average flow velocity in the drive pipe was used in these computations.

Chapter V

EXPERIMENTAL RESULTS AND ANALYSIS.

5.1 RESULTS FOR INDIVIDUAL HYDRAM TESTS.

5.1.1 The Fleming hydram.

The Fleming hydram was tested for supply heads of 1.90 m, 1.41 m and 0.96 m. The original waste valve weight of 0.047 N and stroke of 1.0 cm were not changed. The total friction head loss factor for the waste valve and drive pipe(M) was estimated to be 24.0 and that of the delivery valve(one-way check valve) and drive pipe(N) determined to be 33.0

The effect of supply head on pump performance is demonstrated in figures 16,17 and 18. The complete operating characteristics of the pump for the supply head(H) of 1.9 m are given in figure 19.

5.1.2 The Rife hydram

The hydram was tested at supply heads of 2.0m, 1.51m and 1.06m. The most efficient operating characteristics were obtained with a spring tension of 1.13 N and a stroke length of 4 mm at a supply head of 2.0 m. The same valve settings were used for H=1.51m and H =1.06 m. The total head loss factors for the waste valve and pipe was determined to be 19.0 and 720 for the delivery valve including the drive pipe.

The operation of the pump for various supply and delivery heads is displayed in figures 20 to 22.The complete operating characteristics of the pump for the supply head(H) of 2.0 m are given in figure 23.

5.1.3 The Blake Hydram.

The model used in the tests required little stroke adjustment.Test runs were done for conditions of highest efficiency.The tests were done for H=2.5m and 2.0m.

The results were plotted in figures 24 and 25 for different supply and delivery heads.Variation of valve beat frequency with delivery flow for H =2.0 m is shown in figure 26.The complete operating characteristics of the pump for H =2.0 m are given in figure 27.

5.1.4 The ITDG (spring valve) hydram.

The locally made hydram with the original spring-operated valve was tested for supply heads of 1.90m, 1.41m and 0.95m.The spring tension and stroke were first changed to obtain the most efficient operating conditions for the supply head of 1.90m. The spring tension of 0.6 N and a stroke length of 2 mm were used.

The operation of the pump for the three supply heads and variable delivery heads is shown in figures 28 and 29. The variation of the valve beat frequency with delivery flow is given in figure 30. The complete operating characteristics of the hydram for the supply head(H) of 1.90m are given in figure 31.

5.1.5

The Davey hydram

The pump was tested for the supply heads of 1.96m, 1.41m. and 0.95m. The most efficient operation was obtained with the valve stroke of 8mm with the original waste valve weight of 0.357 N for H=1.96m. The effect of stroke length on pump operation was also investigated for stroke lengths of 2mm, 6mm, 8mm, 10mm and 12mm and a supply head of 1.96 m.

The operation of the pump for variable supply and delivery heads is shown in figures 32 to 34. The total operating characteristics of the pump for H =1.96 m are given in figure 35. The effect of stroke length on pump efficiency and capacity is shown figures 36,37 and 38. The friction head loss factors for the waste and delivery valves including the drive pipe were determined as 40.0 and 21.0 respectively. V_0 was estimated to be 0.30 m/sec and the valve drag coefficient, C_D =52.0 for the waste valve stroke of 8 mm. The variation of the total friction head loss factor(M), V_0 and C_D with valve stroke is shown in figure 39.

5.2 TEST ON THE MODIFIED ITDG(WEIGHTED-IMPULSE) HYDRAM.

5.2.1 Effect of delivery valve on hydram operation,

All the tests were done for a supply head of 1.90m.The effect of delivery valve head loss factor on the pump efficiency and capacity was tested.The results are shown in figures 40 and 41.The variation of valve beat frequency with delivery valve head loss is shown in fig. 42. The variation of friction head loss with rubber thickness is given in figure 43.

5.2.2 The effect of waste valve orifice and valve diameters.

The effect of waste valve orifice was tested in combination with three sizes of the waste valve diameters of 3.9 cms, 4.6 cms and 5.0 cms.The results are shown in figures 44 and 45 for the waste valve diameter of 4.6 cms.Similar results were obtained with the 3.9 cms diameter valve. Tests with the 5.0 cms diameter valve gave poor results and they were discontinued.

For each orifice and waste valve diameter the total friction head loss of the valve and drive pipe(M),V_0,V_s and drag coefficient(C_D) of the valve were determined.The results are summarized in table 1 below.

Figures 46,47 and 48 show the effect of waste valve size on pump efficiency,valve beat frequency and pump capacity in that order for the orifice diameter of 3.0 cms.

TABLE 1

Waste valve characteristics for the ITDG hydram

valve diameter (cms)	parameter	orifice diameter (A_0) in cms. 2.5	3.0	3.5
3.9	Rs	78	49	40
	C_D	5.4	3.8	2.5
	V_0	0.30	0.40	0.50
	V_s	0.65	0.83	1.0
4.6	Rs	80	67	42
	C_D	5.2	4.2	3.0
	V_0	0.30	0.40	0.50
	V_s	0.60	----	0.90
5.0	Rs	158	106	81
	C_D	38.0	9.5	5.2
	V_0	0.10	0.20	0.25
	V_s	0.50	----	0.55

5.2.3 <u>Effect of waste valve stroke length (S)</u>.

The valve stroke length was varied from 1.0 mm to 8.0 mm and the pump tested with an orifice diameter of 3.0 cms and waste valve diameter of 4.6 cms. The effect of stroke length on pump efficiency and capacity is given in figures 49 and 50. Figure 51 shows the variation of valve stroke length and pump power.

For each valve stroke length the total friction head loss factor(M), V_0,V_s and drag coefficient(C_D) were determined.Figure 52 shows the variation of these parameters with the stroke length.

5.2.4 <u>Effect of waste valve weight</u>

The waste valve weight was changed from the original 0.357 N up to 1.157 N for a valve stroke of 1.0 mm.The valve diameter of 4.6 cms and orifice diameter of 3.0 cms were used.

The results were plotted in figures 53 and 54 to demonstrate the effect of valve weight on pump efficiency and pumping capacity.

5.2.5 <u>Effect of air chamber size</u>.

The diameter of the air chamber was kept constant at 50.8 mm diameter and its length varied to increase its volume.Tests for air chamber lengths of 0.3 m, 0.61 m, 1.0 m and 1.3 m were done.The effect of air chamber volume on hydram performance is given in figure 55.For the various sizes tested,there was no significant effect on hydram efficiency, valve beat or pumping cycle duration.

5.2.6 Effect of air valve size.

Three sizes of the air valve were tested, each in combination with the four air chamber sizes used in 5.2.5. The effect of the air valve size on hydram efficiency and pumping capacity is shown in figures 56 and 57.

5.2.7 Operating characteristics of the modified ITDG hydram.

The operating characteristics of the modified locally made hydram were obtained for A_v =4.6 cms, A_0 =3.0 cms and an air chamber length of 1.0 m (fig. 58).

5.3 COMPARISON OF ALL SIX HYDRAMS.

The operating characteristics of the selected modified design of the locally made hydram was compared with the operating characteristics of the other five hydrams that had been tested. Parameters that were compared included efficiency, pumping capacity, wasted flow requirements, valve beat frequency and power delivered. The graphical comparison is done in figures 59 to 63.

Chapter VI

COMPARISON OF THE THEORETICAL MODEL AND THE OBSERVED RESULTS.

6.1 GENERAL.

The Ottawa hydram model together with the models by Krol [32] and Iversen [28] were tested against the observed data from the locally-made and the Davey hydrams.For the locally-made hydram with weighted-impulse valve,results for stroke lengths of 2 mm and 6 mm were used.For the Davey hydram,the results for the valve stroke length of 8 mm were used.Comparison was made for pump efficiency, capacity,cycle duration,valve beat frequency and power.

Lastly,an attempt was made to use the Ottawa model for design and investigation of possible hydram installations.

In order to apply the models,the values of N,M,C_D and V_0 in figures 39 and 52 were used.The actual values used are summarised in table 2

TABLE 2

Parameters used in the theoretical models.

Pump	Stroke (mm)	M	N	V_0 m/sec)
ITDG	2	66	82	0.40
ITDG	6	33	82	0.70
Davey	8	42	21	0.40

6.1.1 Pump efficiency.

Figure 64 gives the test results for the locally-made hydram for the stroke length of 2.0 mm.The maximum estimation errors for the three models are given in table 3 below.

TABLE 3

Efficiency estimation errors.

Pump	stroke (mm)	Ottawa model	Iversen model	Krol model
ITDG	2	<10 %	<25 %	< 30%
ITDG	6	15- 20 %	< 15 %	< 40 %
Davey	8	12- 15 %	< 25 %	20-25 %

6.1.2 Delivery or pumped flow (l/minute).

Figure 65 shows the estimation results for the locally-made hydram for a 2 mm stroke length.The maximum error differences for all three are given in table 4 below.

TABLE 4

Delivery flow estimation errors.

Pump	Stroke (mm)	ottawa model	Iversen model	Krol model
ITDG	2	< 30 %	> 50 %	< 5 %
ITDG	6	< 30 %	30 -35 %	< 16 %
Davey	8	< 50 %	> 50 %	< 30 %

6.1.3 Valve beat frequency (beats/minute).

The valve beat frequency was more difficult to predict by the simple models.Krol's model gave good results with an error less than 20 %. The Ottawa model overestimated by up to 60 % while the Iversen model overestimated the valve beat frequency by about 75 %.All these results were for the locally-made hydram with a stroke of 2 mm.Figure 66 shows how the three models performed.errors of similar order of magnitude were obtained for the Davey hydram.

6.1.4 Cycle duration (T) in seconds

Figure 67 shows the differences between the observed and predicted by the three models for the locally-made hydram. With the ITDG hydram, Krol's model gave an average maximum error less than 16%. The Ottawa model underestimated the cycle duration with an error less than 36% while Iversen's model underestimated by about 50%. The models were also tested with the Davey hydram data and Krol's model gave an average error less than 10% while the Ottawa model underestimated by less than 43% and Iversen's model by less than 46%.

6.1.5 Pump power (watts).

The error differences between the observed and the estimated values are summarized in table 5. The results for the ITDG with stroke length of 2mm are compared in figure 68 for all the three models.

TABLE 5

Errors in the estimation of pump power

pump	stroke (mm)	Krol's model	Ottawa model	Iversen's model
ITDG	2	< 18 %	< 32 %	over 170 %
ITDG	6	< 14 %	< 38 %	< 50 %
Davey	8	< 27 %	< 27 %	over 200 %

6.2 APPLICATION OF THE MODEL TO HYDRAM DESIGN AND INSTALLATIONPROBLEMS.

6.2.1 General approach.

An attempt was made to use the Ottawa model for hydram design and investigation of hydram installations.The effect of increased friction head loss factors for the waste and delivery valves was also investigated.The effect of increased supply head(H) and drive pipe length in hydram installations was examined.V_0 =0.40 m/sec or a velocity ratio of 0.60 were used.Drive pipe friction loss factor(XM) was increased in proportion to the increase in drive pipe length.

6.2.2 Friction head loss factor of the delivery valve(HVD).

This value of HVD was varied from 3 up to 238 and its effect on hydram operation analysed.N,therefore,varied from 15 up to 250.Its effect on pump power and delivery flow was negligible and it was only noticeable at low delivery heads.The effect on pump efficiency was significant as shown in figure 69.There were no noticeable changes in the head - flow ratio relationship,valve beat frequency and cycle time(T).

6.2.3 Friction head loss in the waste valve(RS).

The value of Rs was increased from 1 up to 118. The value of M, therefore, increased from about 13 up to 130. Efficiency increased by about 9% as RS increased from 1 to 50. Increasing RS from 50 to 118 increased the peak efficiency by 2-3 %. Increasing Rs from 1 to 90 reduced peak pump power to about one third. Pumped or delivery flow decreased as Rs increased(figure 70). The effect on valve beat and cycle duration was not significant.

6.2.4 Increased supply head(H).

The supply head was increased from 2m to 5m. Peak power increased from 2 watts to over 9 watts, an increase of over 400 %. Pumped flow increased very much as shown in figure 71. The effect of increased supply head on pump efficiency is shown in figure 72. Cycle duration decreased with the increase of H with the bigger increase for H between 2.0 and 3.0 m. The rate of decrease was smaller for H between 3m and 4m and even smaller for higher H values.

6.2.5 Drive pipe length (L).

The length was increased from 1.0m up to 121m or an L/D ratio of from 28 up to 3460. The effect of increased drive pipe length on hydram efficiency and the dimensionless head ratio-flow ratio curve was negligible. Peak pump power reduced by about 40 % as length increased from 1m to 121 m.Pumped flow also decreased by 30-40 % as L increased from 1-91m as shown in figure 73.

Cycle duration increased by almost 250% as L increased from 1-91m. Maximum valve beat frequency decreased from over 1900 to 28 beats/minute as L increased from 1-91m.Increasing the length from 1-31m decreased the maximum valve beat from over 1900 to 66 beats/minute.

6.2.6 Increase of V_0 (m/sec).

A test was made to see the direct effect of increased V_0 on pump operation.V_0 was increased from 0.30 m/sec to 0.60 m/sec.Peak power doubled and peak pump efficiency was almost halved particularly for high delivery flows as indicated on figure 74 .Figure 75 shows the effect of increased V_0 on pumped flow.Cycle duration(T) was almost tripled as V_0 doubled.

Chapter VII

DISCUSSION AND CONCLUSIONS

7.1 DISCUSSION

7.1.1 Test results for individual hydrams for variable H

7.1.1.1 Pump efficiency.

Generally speaking increasing the supply head increased the pump efficiency,particularly at low delivery heads (i.e high delivery flows). Peak efficiency values increased by about 10-20% when supply head was doubled.The increase was due to increased delivery flows (Q) and delivery heads which accompany the increase in supply heads.The wasted flows (Q_w) did not increase much due to the reduced acceleration period since velocities in the drive pipe were also increased.

The efficiency curves started from zero,went through a peak and then fell continuously.This was more evident in the hydrams whose waste valve weights (or tension) are high like the Blake and Rife hydrams.For hydrams with light waste valves like the Fleming and Davey, the efficiency curves did not fall gradually but remained high and then fell abruptly.The falling limb was,however,difficult to observe in these measurements.

Efficiency was higher in the hydrams with light waste valves like the Fleming compared to those with heavy ones like the Rife and the Blake.This was probably due to the fast waste valve closure so that Q_w was small and Q was of the same order of magnitude.This resulted in high efficiencies.The head at which the highest efficiency was obtained was variable for each hydram.This value appeared to be related to the weight or spring tension of the waste valve.The efficiency of the hydram is a function of waste valve design,valve weight,valve stroke and the supply and delivery heads.Therefore,efficient operating conditions can only be obtained experimentally by adjusting valve weight and stroke for any given hydram design and installation.

For a hydram already operating with maximum efficiency,increasing valve stroke or weight will reduce its efficiency.This is mainly due to increased Q_w.In well-designed hydrams like the Davey,the difference is minimal (Fig.36).

7.1.1.2 Pumping capacity of the hydrams.

Delivery flow generally decreased with the increase in delivery head.The amount pumped by the various hydrams operating with highest efficiency was directly proportional to the supply head.

Delivery heads to which the pump could pump increased with supply head.For higher supply heads,the pump will deliver the same quantity of water to a higher delivery head than for a lower supply head.The increase is not proportional to the supply head increase.

The maximum delivery head the pump reached (h_{max}) increased with the supply head, valve weight and /or valve stroke. The minimum head (h_{min}) at which the pump could still operate appeared to be of the same order of magnitude irrespective of valve weight (or tension) and valve size. The Fleming hydram, however, had very high h_{min} values; and these increased with the increase in supply head (Fig.17). This may be due to the check valve between the waste valve and the air chamber which required a minimum back-pressure to keep it closed during recoil, rather than the weight or size and stroke of the waste valve.

7.1.1.3 Head ratio-flow ratio curves.

The best pumping rates were obtained with head ratios below 6.0 for most of the hydrams (table 6).

The Fleming and the Blake hydrams did not operate for head ratios below 2. The Blake could still pump about 1.0 l/minute at a head ratio of 10. In general the flow ratio increased but did not double when the supply head was doubled.

It was noticed that at high head ratios, the flow ratios for H = 1.0 m and H = 2.0 were about the same for most hydrams, except for the Blake and Rife hydrams. This was probably due to the fact that at high delivery heads the delivery head had more effect on the operation of the hydram. The discharge was greatly reduced due to the high delivery pressures which had a retarding effect. The recoil effects were

TABLE 6

Head ratio and flow ratio variation for various hydrams.

HEAD RATIO	FLOW RATIO								
	Fleming		Rife		Davey		ITDG(S)		Blake
	H =2m	H =1m	H =2m	H =1m	H=2m	H =1m	H =2m	H =1m	H =2m
1	--	--	0.016	0.32	0.56	0.80	0.31	0.57	--
2	--	--	0.125	0.23	0.30	0.40	0.18	0.23	--
6	0.11	0.14	0.06	0.10	0.10	0.12	0.06	0.08	0.050
8	0.075	0.11	0.04	0.08	0.08	0.09	0.04	0.05	0.041
10	0.06	0.07	0.03	0.06	0.06	0.06	0.025	0.03	0.032
12	0.045	0.055	0.03	0.04	0.05	0.05	0.02	0.025	0.028

also more pronounced.The effect of the supply head was thus reduced.

For low delivery heads and head ratios below 6.0,the flow ratios for supply heads of 1 m and 2 m were significantly different.This was true in the same hydram and for different hydrams.Therefore any empirical formulations based on head ratio and flow ratio may only be applicable for head ratios over 6 or preferably 8 and only for hydrams of similar or comparable waste valve designs. Waste valve weight or spring tension should,preferably,be about the same.There is a minimum supply head needed for the hydram to operate. It was found that the Blake hydram could not operate at supply heads below 1.5 m.

7.1.1.4 Valve beat frequency and cycle duration.

For all hydrams doubling the supply head(H) doubled the valve beat frequency for medium high and low delivery heads. This was due to the high velocities in the drive pipe that resulted when H increased. Since V_0 was still the same and was obtained in the hydram more often with increased supply head, the waste valve was now closing more rapidly. Cycle duration decreased as the supply head increased for the same reason.

7.1.1.5 The effect of waste valve design and adjustment on hydram operation.

The effect of valve stroke length on hydram performance was difficult to detect and interpret for well-designed commercial hydrams. This was particularly so for efficiency as seen in figure 36, for the Davey hydram. Similar results were obtained with the Rife hydram.

Normally short valve strokes give higher pump efficiency due to fast waste valve closure and less wasted flow (Q_w). In the Davey hydram this rule was not true, probably due to the shape of the waste valve and its orifice (Fig. 4). Figure 39 shows that the valve characteristics were almost constant for stroke lengths greater than 8 mm. The Fleming hydram operated with a stroke length of 10 mm which was much longer than for the other hydrams and yet had the maximum efficiency.The Rife hydram gave best efficiencies with a stroke length of 4 mm.

Increased stroke length increased delivery flow as seen in figure 37 for the Davey hydram.This was likely due to the reduced friction loss in the waste valve and higher velocities in the drive pipe.All these resulted in pumping more flow and to higher delivery heads by the hydram.The waste valve took longer to close as V_0,the velocity required to start waste valve closure,increased with valve stroke length.Therefore,Q_w increased as waste valve stroke increased.As a result,the flow ratio remained almost the same and the head ratio-flow ratio curve remained the same for all valve stroke settings (Fig.38).Since it was noticed that increasing valve stroke or valve weight had almost a similar effect on the hydram operation,the head ratio-flow ratio curve is expected to remain the same for increased valve weight.This was observed in the locally made hydram.

Valve beat frequency was low at very high delivery heads and increased to a peak and then fell continuously.The number of valve beats per minute was higher in the hydrams with light waste valves like the fleming.Valve beat frequency was therefore a function of the waste valve weight,stroke,size(diameter) and its design shape.The design of the waste valve has a controlling effect on the operation of the hydram.

7.1.2 Design modifications of the locally-made hydram

7.1.2.1 - Delivery valve design.

The delivery valve design affected the pump operation, particularly the pump efficiency, valve beat frequency and capacity (Fig. 40-42). When the rubber on the perforated delivery valve plate was thin, it was greatly affected by the back pressure from the delivery pipe and air chamber during recoil. The rubber tended to get sucked into the perforations thus increasing the friction head loss through the valve. The effect was most obvious at high delivery heads. As the delivery head reduced, the recoil effects and back pressure were minimal and the pump operated normally. When the rubber thickness was increased greatly, its resistance to the delivery flow became great and the pump could not work.

Except for very thick rubbers greater than 10 mm, which gave very high friction head losses, the pumping capacity of the hydram was not much affected. Thin rubbers weaken very fast being cut by the perforated plate. Therefore, a reasonable thickness is needed that will not affect pump operation and last longer. Alternatively, the perforations should be enlarged if a thicker rubber valve is to be used. It is, therefore, considered that it is the resistance to the flow and friction loss through the valve that affect the hydram operation when thicker rubbers are used.

7.1.2.2 : Waste valve diameter and opening.

The tests on the effect of waste valve orifice indicated that it had very little effect on pump efficiency and the general operating characteristics of the hydram (figures 44 and 45). Increasing the orifice by about 40% did not affect pump characteristics.

Table 1 shows that there was a change in the valve characteristics including RS and V .However, their overall effect on the flow pumped and wasted was not significant. Probably the measuring techniques used were not precise enough to detect the changes.

The results indicated that the area through the waste valve being smaller than the drive pipe area (D = 0.351 m) had little effect on the hydram. The "choking" effect on the flow due to this area being smaller than the area of the drive pipe, pointed out by Bergeron [7], was not observed in the experiment.

In another test, the orifice diameter of 3.0 cms was used and the waste valve diameter changed. The pump worked properly for waste valve diameters of 3.9 cms and 4.6 cms and 5.0 cms. There was very little change in hydram efficiency and pumping capacity except for the waste valve diameter of 5.0 cms. There were noticeable differences in the valve beat frequency particularly for the 5.0 cms diameter valve. The 5.0 diameter valve did not work properly because it was almost the same size as the waste valve 'pot' or 'housing' which had a diameter of 50.8 cms.Therefore,the increased friction losses and the reduced V_0 value limited the flow velocities in the drive pipe and made the valve close faster.Table 1 shows how V_0 was greatly reduced and figure 47 shows the increase in valve beat frequency.Therefore,for good hydram operation,the valve diameter should not exceed 70-75% of the valve 'pot' or 'housing' for the locally-made hydrams.

7.1.2.3 Waste valve stroke length.

In a non-commercial hydram,like the ITDG,changing the valve stroke affected pump operation significantly,particularly pump efficiency.Figure 49 shows that pump efficiency decreased,and pumped flow increased with increased stroke length.The test results indicated that for stroke lengths longer than 4 mm,efficiency of the pump was more stable.Increasing the valve stroke length had less effect on pump efficiency for stroke lengths greater than 4 mm.For stroke lengths less than 4 mm,increasing valve stroke had a bigger influence on hydram efficiency.With a stroke length of 1.0 mm,the pump had very high efficiency but operated within small limits and its pumping capacity and power were very small.

Increasing valve stroke increased pumped flow.However,as with the Davey hydram,the increased pumped flow and delivery head were counterbalanced by increased Q_w so that the head ratio-flow ratio characteristic was almost unchanged (fig. 50).

Figure 51 shows that the pump power increased with the increase of waste valve stroke length.The figure also shows that the effect of increasing stroke length on pump power was greater for stroke lengths below 4 mm. For stroke lengths greater than 4 mm,the effect was less particularly on the peak power.With a stroke length of 1 mm,the pump power was very small.Therefore,as pointed out by the earlier researchers like Krol [32],efficiency should not be the only operating criterion on which the pump design and installation is based.The pumping capacity of the pump is also important.

7.1.2.4 Waste valve weight or spring tension.

Increasing the valve weight reduced pump efficiency and increased the pumping capacity of the pump as given in figures 53 and 54.This was due to the increased V_0 value so that higher velocities were obtained in the drive pipe and the resultant higher waterhammer pressures that occurred in the hydram.The effect was similar to increasing valve stroke length.However,the results indicated that increasing valve stroke gives better pump operation and higher pumped flows than increasing valve weight.

7.1.2.5 Air chamber size.

Tests with the air chamber size showed that the air chamber volume had no significnat effect on the hydram operation.With air chamber lengths of 0.3 m to 1.3 m and 50.8mm in diameter,the pump efficiency and delivery flow were the same.Figure 55 indicates that the hydram operation was not altered by increasing the air chamber volume.Probably at higher supply heads,it may make a difference,but at the test supply head of 2.0 m,the effect was not noticeable.Krol[32] reported a 10% increase in efficiency when the air chamber size was doubled.If this were true, then hydrams with big air chambers like the Blake (about 7.6 litres air chamber volume) would give higher efficiencies. It is probable that if supply heads are large,waterhammer pressures are great and large air volumes in the air chamber may be necessary to absorb the surges.This would necessitate larger air chambers.At low supply heads,smaller air chambers can be used with no or little effect on the hydram operation.

7.1.2.6 Air (snifter) valve size.

Another aspect of the hydram design that was investigated was the size of the air valve.It was found from the experiments that the size of the smifter valve was not critical.Only when the hole became so large that significant amounts of water pumped by the hydram were escaping through the air valve did the effect become noticeable.Figure 56 shows that the change in hydram efficiency was less than 5%.The effect on pumped flow was not significant (figure 57).The noticeable effect on efficiency was probably due to increased Q_w while Q remained almost the same. Therefore, for practical purposes a small hole not exceeding 1.0 mm and with no nails or cotter pins is sufficient. The tests also indicated that the effect of the air valve size did not depend on air chamber size for the sizes tested.

7.1.3 Operating characteristics of different hydrams.

The operating characteristics of six hydrams including the two versions of the locally-made hydram were compared at a supply head of about 2 m. A supply head of 2 m was considered a medium supply head which can easily be obtained in the field with a simple small dam.

7.1.3.1 Pump efficiency.

The maximum efficiencies with which the hydrams operated varied greatly as seen in figure 59. Hydrams with light waste valves gave higher efficiencies. The Fleming hydram gave very high efficiencies but its pumping capacity was much smaller than the others. Hydram efficiency, therefore, depends on weight of the waste valve, friction head loss through the valves and the general design of the waste and delivery valves. The effect of valve stroke length on efficiency is secondary in a well-designed hydram.

In this experiment, the Blake gave the lowest efficiencies. This was possibly due to the fact that the supply head was not sufficient and it would operate better under higher supply heads. This is also possibly true for the Rife hydram.

7.1.3.2 Pumping capacity.

Comparing the pumping capacity of hydrams operating at their maximum efficiency,the hydrams with large heavy waste valves, large air chambers like the Blake and Rife hydrams pumped larger quantities particularly at higher delivery heads.The simpler locally-made hydrams did not perform as well.

The pumping capacity of a hydram,however,does not depend on waste valve weight alone.The Davey and modified ITDG and Fleming hydrams have ligher waste valves than the original ITDG hydram and yet gave better results.Figure 60 shows that with heavy rugged commercial hydrams like the Blake,the flow pumped exceeded 3 times that of the locally-made hydram with spring-type waste valve. At high delivery heads the small hydrams have comparable pumping capacity.The Blake hydram could not operate at head ratios below 4.The Fleming, also, had a limited operating range between the head ratios of 2.5 and 10.0.The pumping capacities could be increased if the pumps operated with lower efficiences.This would be achieved by changing the waste valve weight(or spring tension) and/or stroke length.

For efficient hydram operation,the pumping capacity of the hydram does not depend on valve weight or stroke alone but on many factors related to the design of waste valve and its 'housing'.These factors are very difficult to single out.Since hydrams with large air chambers gave high flows at higher delivery heads,it is possible the air chamber size was partly responsible.In the experiment with the air chamber of the locally-made hydram,no effect of air chamber size was detected.This was possibly due to the fact that the volumes of 0.3-0.45 litres investigated were too close to give different results.These air chamber volumes were very small compared to 7.6 litres for the Blake,about 6 litres for the Rife and about 3 litres for the Davey hydrams.

7.1.3.3 Head ratio-flow ratio curves.

While the flow ratio-head ratio relationship is suitable for describing the operating characteristics of a hydram,applying it for comparison of several hydrams can lead to wrong conclusions. From figure 61,it appears that the Fleming is a better pump than the Blake or the Rife.This is because these hydrams require different amounts of supply and wasted flow (Q_w).The Fleming uses about one third of what the Blake uses.For pumps which use drive flow of the same order of magnitude like the Davey and the improved ITDG hydrams, the curves were almost similar.

The curves,however,are useful for the comparison of the limiting conditions under which the various hydrams will operate.Secondly,where the supply flow is limited,the curves indicate which hydram(s) would be suitable for the planned installation.Figure 62 shows the waste flow requirements for the various hydrams tested.

Pumping power is related to pumping capacity.The hydrams which pump more flow to higher heads, like the Blake will give higher power as shown in figure 63.

7.1.3.4 Waste valve design and pump adjustment (tuning).

The comparison of operating curves showed that the original ITDG hydram was operating poorly and the improved version had operating characteristics similar and comparable to the small commercial hydrams.The improvement was due to improved general design of the waste valve,the delivery valve and smooth waste valve movement.Adjustments were easier to do than with the spring impulse valve.Therefore,unless heavy-duty springs of good quality are used,weighted impulse valves operate better and smoothly. Weighted impulse valves are easier to adjust and can last longer while cheap spring valves fail soon.This was experienced in the course of the experiments.

7.1.4 <u>Testing the theoretical models against the observed data</u>..

The results indicated that the model predictions were good taking into account any hydram design differences and hydram adjustments like changing stroke length.Therefore,the models can be considered to be of general applicability.

7.1.4.1 Estimation of pump efficiency.

In general, the model by Krol is more powerful and accurate than the other two simpler models. This is because Krol's model includes the details of the retardation and discharge periods including the effects of 'recoil'. The other models are simpler approximations and not as detailed. In all cases tested the Ottawa model gave better prediction results for efficiency than the other two. Iversen's model is totally unrelated to the observed efficiency curve and did not even indicate the trend. This is because the model does not consider the waterhammer effects so that the discharge velocities were overestimated. As a result the discharge flow was overestimated particularly for the high delivery heads. Krol's model reaches peak efficiency fast and goes down fast. As far as efficiency prediction was concerned, the Ottawa model gave the best results.

7.1.4.2 Estimation of pumped or delivery flow.

Krol's model gave very good estimates of the pumped flow and Iversen's model highly overestimated the flow. The Ottawa model was close but understimated the flow for medium to high delivery heads. Again Iversen's model neglects waterhammer effects and assumes acceleration velocities are equal to the discharge velocities which is not correct. It is possible that when the details of retardation and recoil are not included in the analysis, the drive or wasted flow is overestimated.The energy available for pumping during subsequent periods is also underestimated and therefore the Ottawa model underestimated the pumped flow.The Ottawa model with a maximum error of less than 40% is accurate enough to predict general trends.Iversen's model also gave a good estimate of the trend but had a much bigger error in the estimation of the actual values.It is,however,not clear why Krol's model which predicted pumped flow very accurately,failed to give good efficiency estimates.

7.1.4.3 Pump power.

The third most important characteristic was the pump power.Krol's model gave good results with less than 30% error for all tests.The Ottawa model was second best with a maximum error less than 40% for the the hydrams used in the test.Iversen's model grossly overestimated the pump power for delivery heads greater than 5m.This is again related to the errors in overestimating the delivery flow and cycle time (T) by the simpler models like Iversen's and the Ottawa models.The Ottawa model did,however,predict the trend fairly well.The error increased with increase in delivery head for all models but at different rates.

7.1.4.4 Cycle duration and valve beat frequency.

Cycle time (T) and valve beat frequency (VB) are interrelated.Krol's model performed very well with a maximum error less than 16%.Iversen's model and the Ottawa model had much larger errors.This was due to neglecting the details of retardation and recoil,particularly at high delivery heads.It was observed that the cycle duration decreased very fast as the delivery head increased.In the experimental data,the decrease in T was gradual and small.At high delivery heads, the pumping period is shorter,acceleration time may change and the time for the pressure waves to die out may be longer.Also,due to large pressure head differences at both ends of the drive pipe immediately after the discharge valve closes,there may be more energy available for pumping during the subsequent periods that is ignored in the simpler models.Krol's model considers the negative velocities in the drive pipe during recoil and time for velocities to become positive again,so that it indirectly takes into account the energy changes in the hydram installation.As a result,the simpler models greatly underestimate the duration of the cycle and consequently overestimate the valve beat frequency.

7.1.4.5 Summary.

Since,for a given installation,the major parameters required are pumped flow (Q) efficiency (e) and possibly pump power (P),the simpler Ottawa model should suffice.The tests also confirmed what earlier researchers had observed; with the parameters like V_0, M, N and C_D determined experimentally,the operation of the hydram could be estimated with reasonable accuracy.

It was also seen that when V_0 was increased, the efficiency decreased and the estimation error increased. However, better estimates of Q,T and VB were obtained. The velocity ratio (V_0/V_s) was usually about 0.6. If friction losses in the drive pipe are neglected and the maximum theoretical velocity given by equation(4) is used, the velocity ratio will be much less than 0.1. The velocity ratios of 0.2-0.4 given by Bergeron[7] cannot be obtained, except possibly for larger hydrams with heavy waste valves and long valve strokes.

7.1.5 Using the Ottawa model for hydram and installation design.

7.1.5.1 The effect of V_0 on hydram performance.

The tests indicated that V_0 had a big effect on pump operation, because it affected the quantities of waste flow (Q_w) pumped flow (Q) and delivery head (h) that the pump could develop. Therefore, V_0 is the main factor controlling hydram operation. V_0 is a function of waste valve design, stroke and weight for a given hydram installation.

7.1.5.2 Delivery valve design.

The tests indicated that with small head loss through the delivery valve, the delivery valve had little effect on hydram operation. With increased friction losses through the valve, the pumped flow was reduced and pump efficiency decreased, particularly, for low delivery heads. At low delivery heads the effect of back pressure from the delivery pipe and air chamber is low and the effect of the delivery valve dominates the pump operation. Therefore high head losses in the delivery valve should be avoided.

7.1.5.3 Effect of waste valve design.

The friction head loss factor(RS) of the waste valve had a big effect on hydram operation. The head loss affected V_0, which reduced as RS increased. Thus efficiency tended to increase while power and pumped flow decreased. Therefore, high friction head losses in the waste valve design should be avoided. This is possible by good proportioning of the waste valve orifice and valve diameter in relation to the size of the 'housing'. The hydram should also be designed with smooth passages for the flowing water. High friction losses in the drive pipe should be avoided. Tests with the locally-made hydram showed there was a wide range of waste valve orifice and diameter sizes that can be used in combination without affecting the operation of the pump.

7.1.5.4 The effect of increased supply head.

Increasing supply head increased pumped flow and power. Valve beat frequency increased and cycle duration decreased. Generally, the effect of increasing H, was to increase velocities in the drive pipe. Thus, V_0 was attained much faster and more frequently at the increased H values. This resulted in fast closing of the waste valve, unless its weight or stroke were increased accordingly. The fast closure of the waste valve reduced Q_w and increased Q and h so that peak efficiency also increased. The delivery flow at which peak efficiency was reached increased with the increase of supply head(H). The increased velocities in the drive pipe increased the pumping capacity of the hydram and the pumped flow(Q) increased.

7.1.5.5 The effects of increased drive pipe length.

Increased length of the drive pipe reduced peak power and pumped flow by a maximum of 40% in each case for the tested range. The main factor was the increase of friction losses with the increaesd drive pipe length(L). There was very little effect on pump efficiency mainly because a fairly smooth drive pipe was assumed in the test and the length(L) of 91 m did not affect the flow velocity significantly. This length, however, with an L/D ratio of over 2800 was outside the limits given by Calvert [12] and significantly different results were expected. A velocity ratio of 0.60 was assumed based on model test results. The length of the drive pipe (L) is not included in the expressions for flow ratio (Q/Q_w) and, therefore, has no direct effect on the pump efficiency. With a drive pipe length of 1.0m, the valve beat frequency was very high and the cycle duration very short averaging 0.034 seconds. Pumped flow was, however, little affected.However,this is not a physically realistic situation.No valve could vibrate at this high frequency due to its inertia.Therefore the pump wouldn't work at this condition even though the theoretical model says it could.

In the equations for acceleration and discharge times (t_1)and(t_3),drive pipe length(L)is directly included.The length (L)is,also,directly used in the expressions for t and t .As a result L has a significant influence on the duration of the cycle (T) and therefore on valve beat frequency.However,since the model was already known to underestimate cycle time (T),and all the expressions for time durations (t) are direct multiples of L,the model can only give trends rather than exact values of the effect of the drive pipe length on hydram operation.This is mainly for the effect on cycle duration and valve beat frequency.

7.2 CONCLUSIONS.

The conclusions from this study can be summarized as follows:-

1. A good design of the waste valve is essential for efficient hydram operation. High friction losses through the waste valve are undesirable.
2. Depending on the design of the waste valve assembly, a hydram will operate with maximum efficiency even with long stroke lengths. Increasing waste valve stroke length has the same effect as increasing its weight. Increasing stroke length, however, gives better and smoother hydram operation and higher delivery flows compared to increasing the valve weight.
3. The flow area through the waste valve can be much smaller than the area of the drive pipe with no significant effect on the operating characteristics of the hydram. There is a wide range of waste valve orifice and valve diameters and their combinations with which the hydram will operate normally.
4. V_0, the velocity to initiate waste valve closure, is the most important parameter in hydram operation. This value is generally a function of waste valve design and adjustment. Other researchers [37] reached the same conclusion. Increasing V_0 reduces hydram efficiency but increases its pumping capacity and power delivered.

5. The delivery valve should be well designed with minimum friction head losses and resistance to the flow. The grade and hardness of the rubber used should be able to withstand the back pressure from the delivery pipe and air chamber.
6. Increase in hydram capacity, and valve beat frequency is proportional to increase in the supply head.
7. For a given hydram design, the head ratio-flow ratio characteristic curve describes the hydram operation for the given installation and is not affected by changes in the valve stroke length or weight. It is not suitable for comparison of different hydrams except if their drive flow requirements are comparable or about the same. Better hydram operation is obtained with head ratios between 4 and 8. The curves can be used to select suitable hydrams for installations in which the water source is limited.
8. Different hydrams of the same size, operating with maximum efficiency, have different pumping capacities which are proportional to the total supply flow requirements of the hydrams. A well designed locally-made hydram has comparable operating characteristics as the commercial hydrams of the same general size.
9. The size of the air chamber has no significant effect on hydram operation particularly at small supply heads.

10. The size of the air valve is not critical in hydrams. A simple small-sized (< 1 mm) hole is sufficient.
11. With parameters estimated under steady flow conditions, theoretical models can be used to predict hydram operation with reasonable accuracy. The models can also be used in the investigation of hydram design and installations.
12. Neglecting waterhammer effects in the theoretical models grossly reduces their accuracy and capabilities. Neglecting the details of recoil and retardation in the theoretical analyses has the greatest effect on the underestimation of the pumping cycle duration and the overestimation of the valve beat frequency.
13. The Ottawa model gives a better estimation of results than the Iversen model. It is, therefore, an improvement on Iversen's model and is simpler to understand and operate than Krol's model.

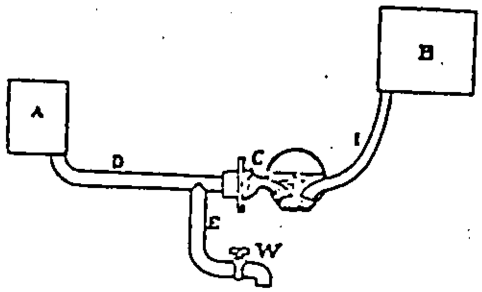
7.3 RECOMMENDATIONS FOR FURTHER WORK.

Further studies should investigate the hydram's operating limitations,test the durability of the locally made hydrams and popularize their use,particularly,in developing countries.This would include:-

1. Durability studies in the field for the locally-made hydrams.
2. Testing hydrams with larger waste valves to establish the upper practical size limits for efficient hydram operation.
3. Testing the hydrams under different operating conditions,particularly, higher supply heads and different waste valve sizes to establish limiting conditions.
4. Test the Ottawa and Krol models for the operating conditions in (2) and (3) to establish the limitations of the models.Also,verify the results obtained by the models on the effects of increased supply head and drive pipe length.
5. Further tests on the effect of the air chamber size with significantly larger pipes,in relation to drive pipe length and supply head.

Appendix A

FIGURES.



LEGEND.

A- Supply Source.
D- Drive pipe.
W- Waste Valve.
C- Delivery or Check valve.
I- Delivery pipe.
B- Header tank.

FIG. 1 ORIGINAL WHITEHURST'S HYDRAM (MEAD 1933)

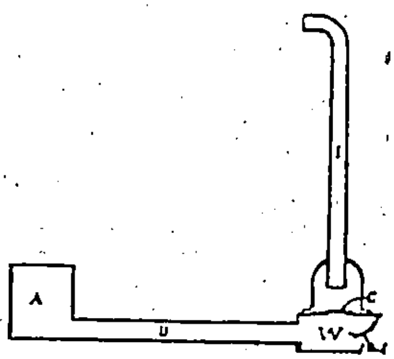


FIG. 2 ORIGINAL MONTGOLFIER'S HYDRAM (MEAD 1933)

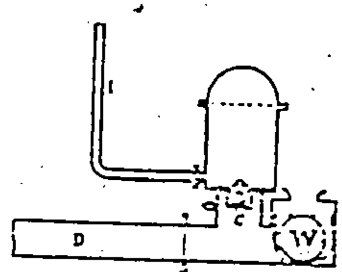


FIG. 3 MONTGOLFIER'S IMPROVED HYDRAM (MEAD 1933)

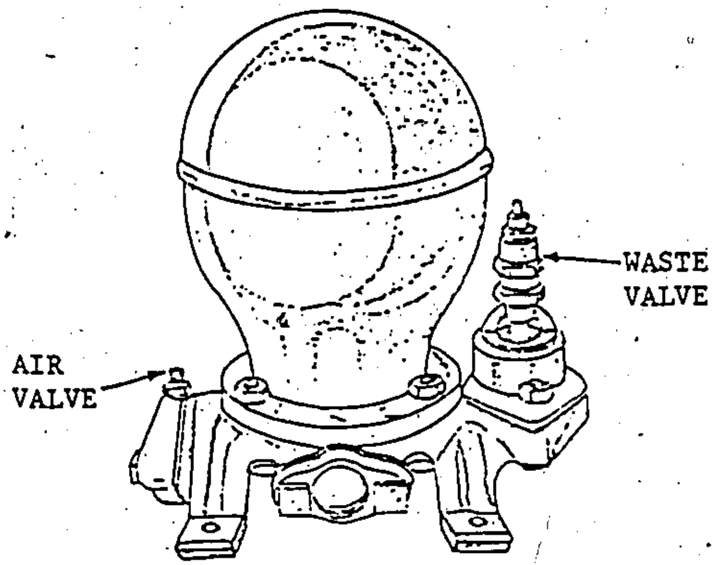


FIG. 4(a) SIDE VIEW

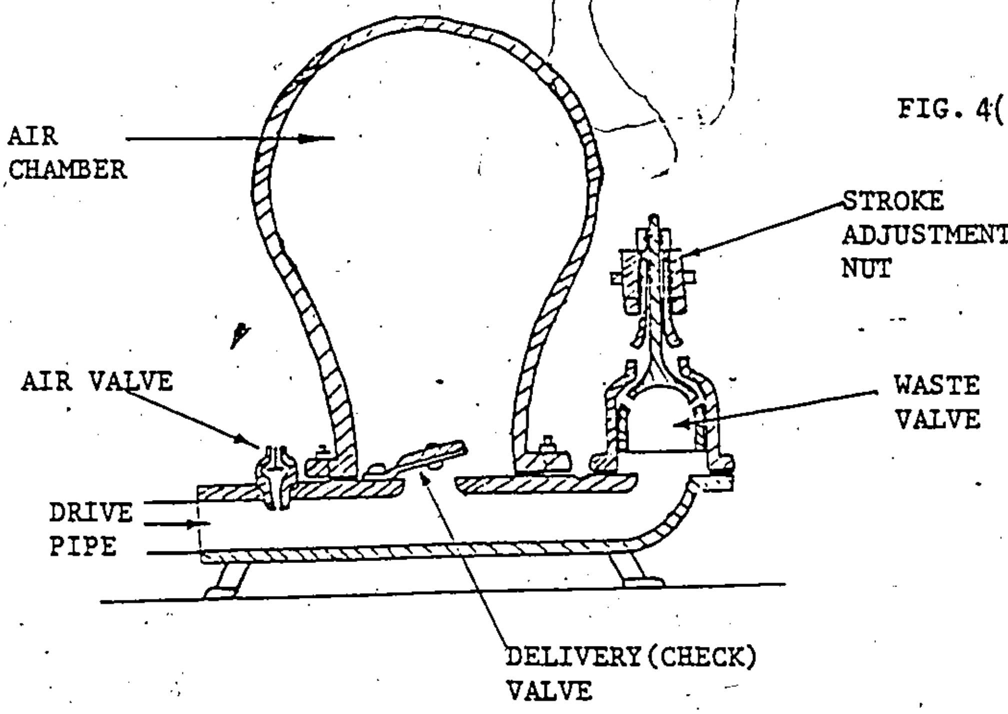


FIG. 4(b) DETAILS OF THE WASTE AND CHECK VALVES

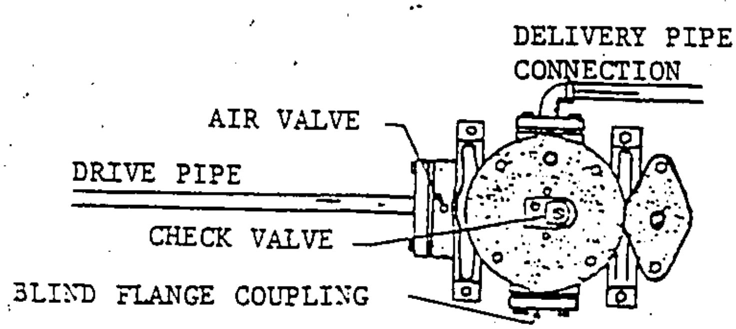


FIG. 4(c) TOP VIEW WITH AIR CHAMBER& WASTE VALVE REMOVED

FIG. 4 DAVEY HYDRAM

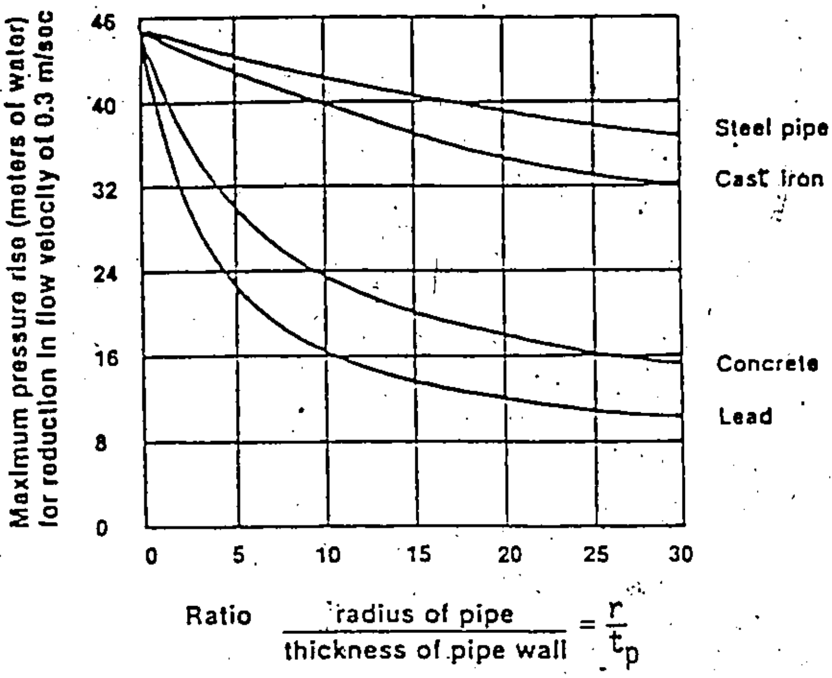


FIG. 5 DRIVE PIPE SPECIFICATIONS (Watt 1975)

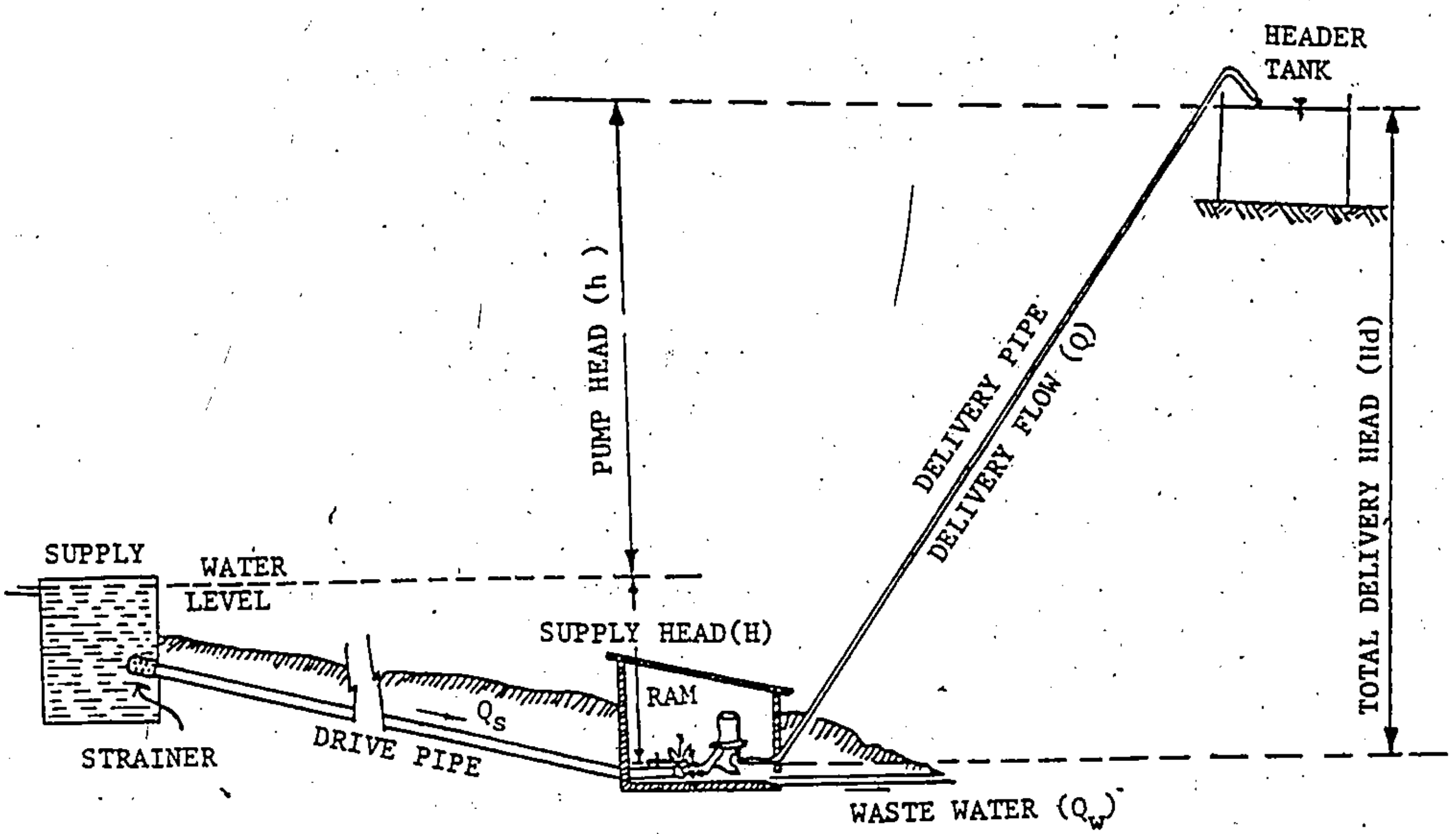


FIG. 6 THE ARRANGEMENT OF A TYPICAL HYDRAM INSTALLATION

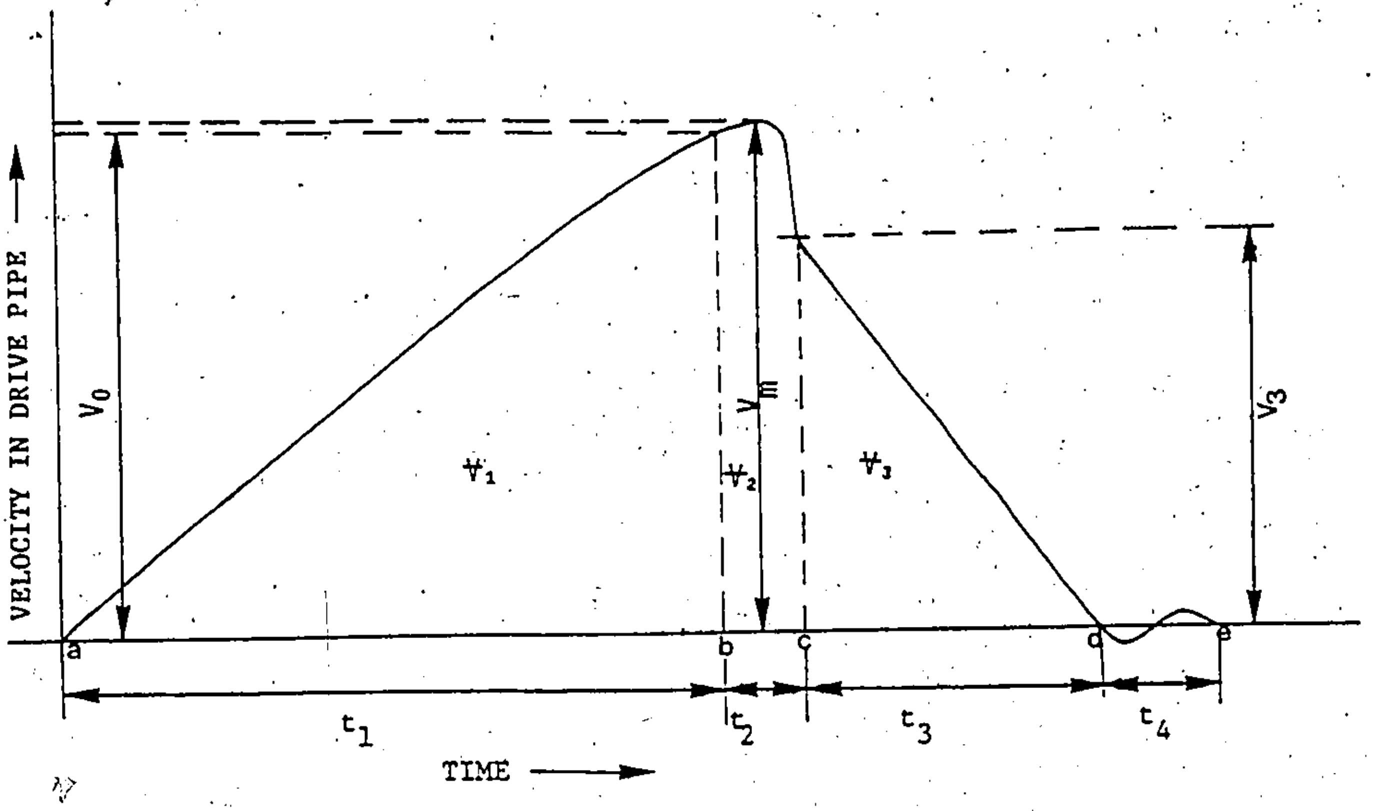


FIG. 7 SCHEMATIC REPRESENTATION OF THE TIME -VELOCITY VARIATION IN THE DRIVE PIPE DURING ONE PUMPING CYCLE

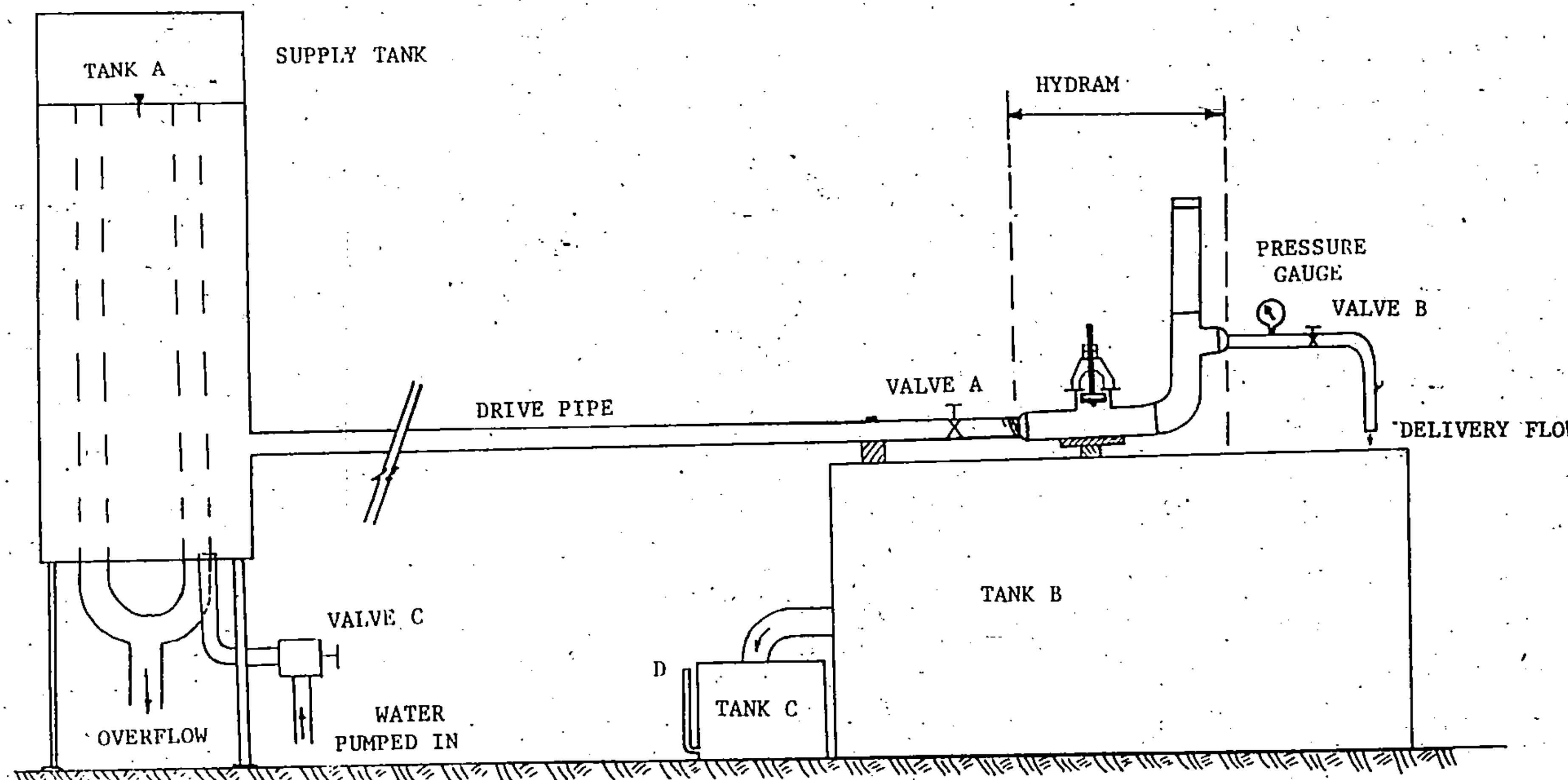


FIG. 8 APPARATUS AND EXPERIMENTAL SET UP

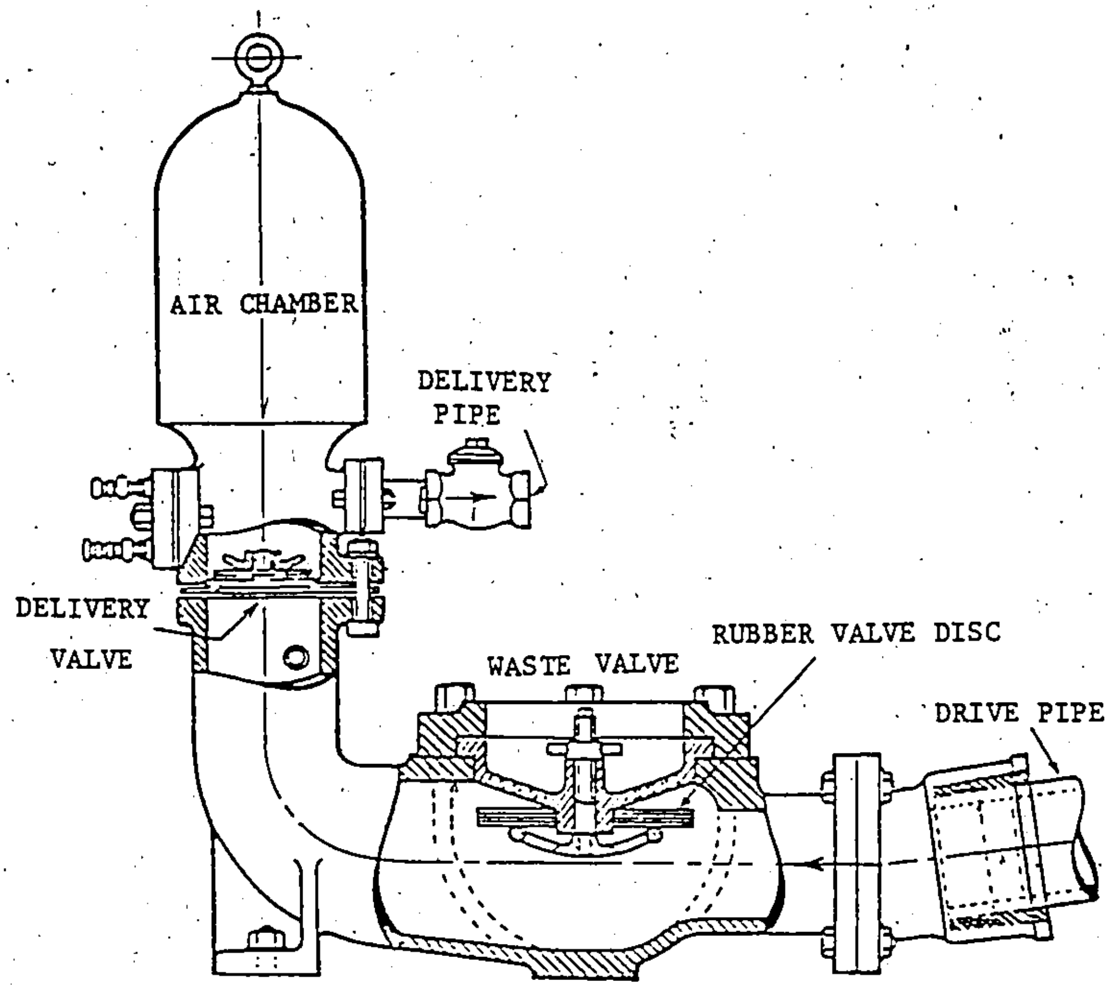


FIG. 9 BLAKE HYDRAM

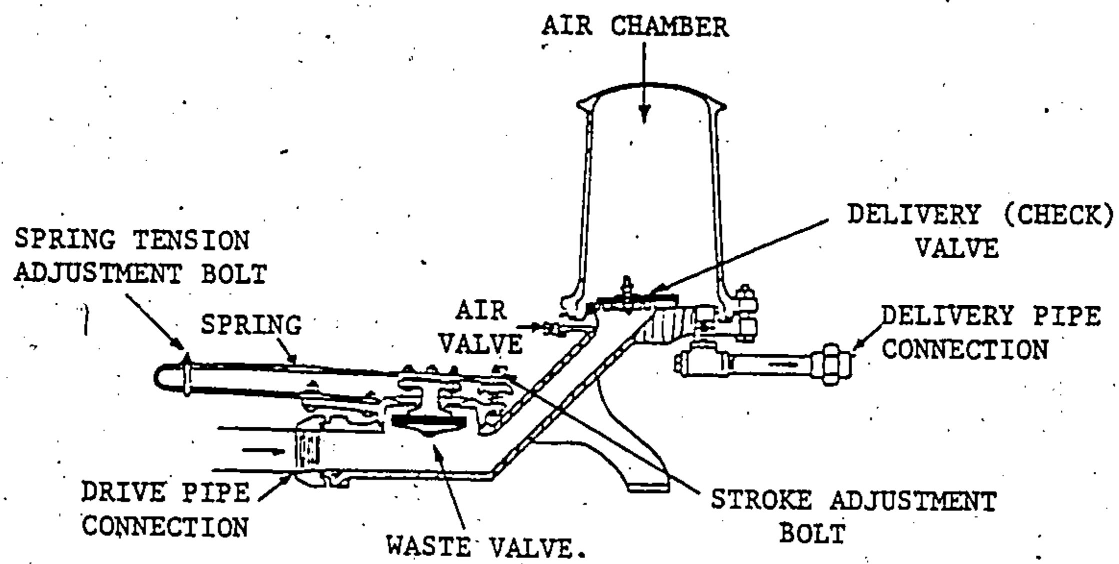


FIG. 10 RIFE HYDRAM

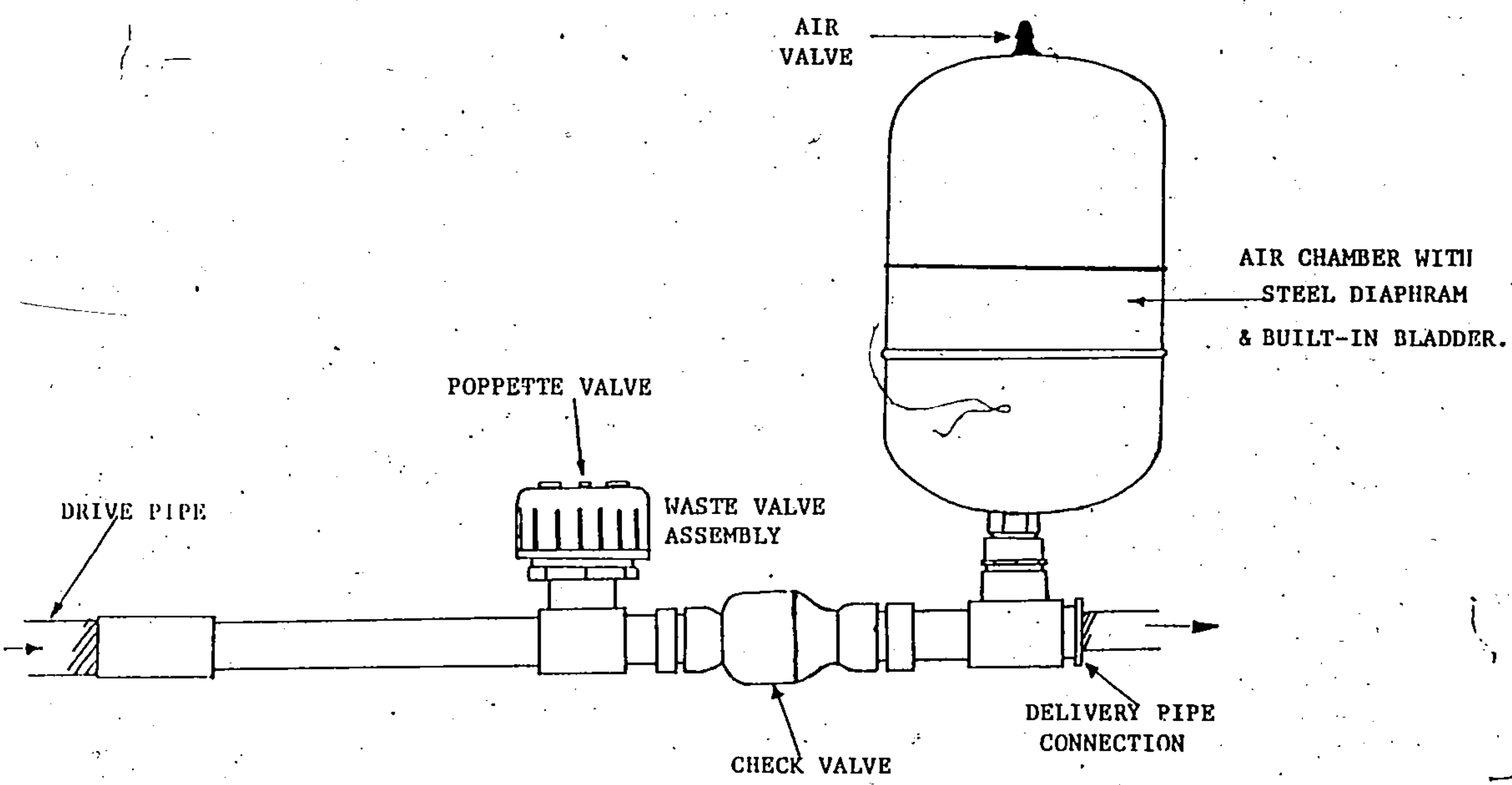


FIG. 11 FLEMING HYDRAM

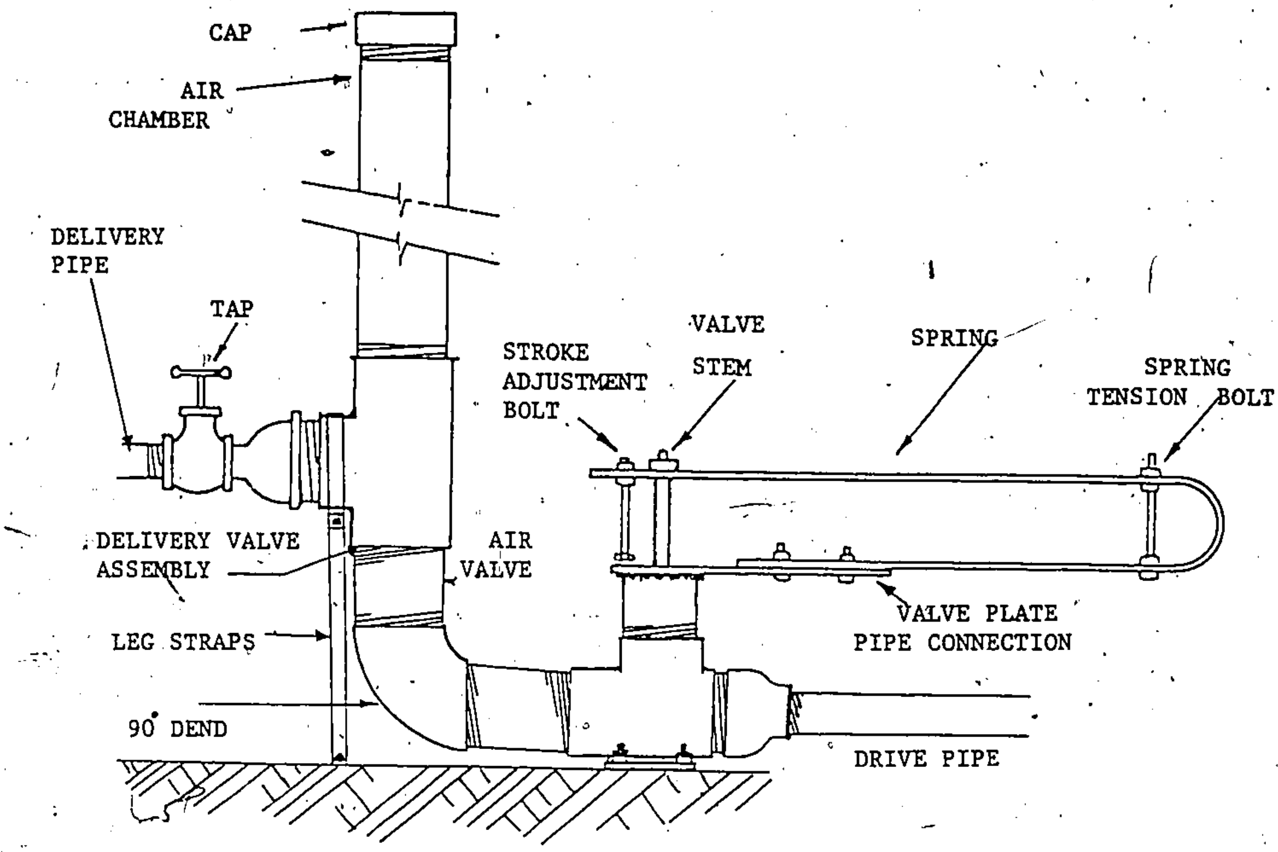


FIG.12 THE LOCALLY MADE HYDRAM WITH SPRING-CONTROLLED WASTE VALVE(Watt 75)

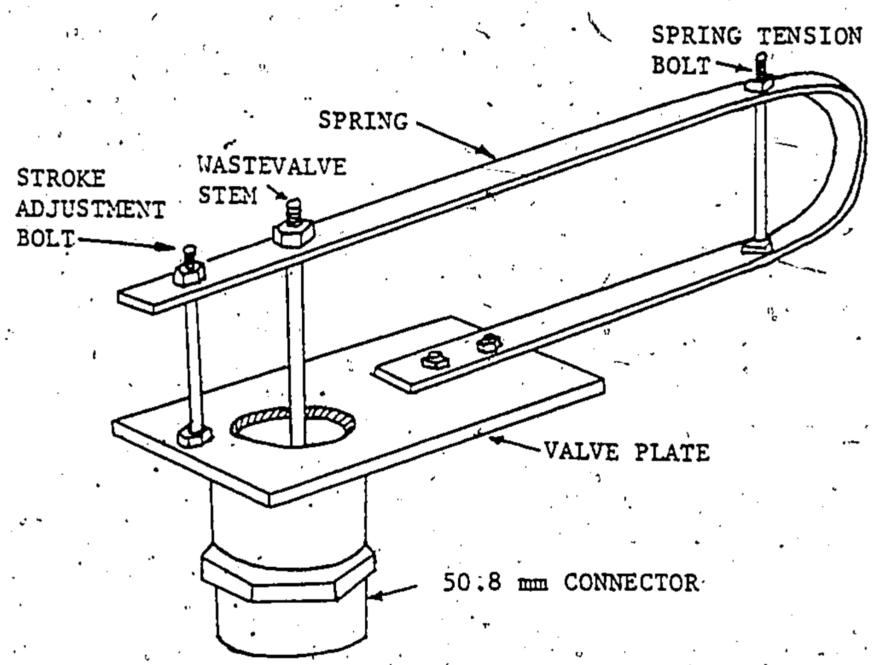


FIG. 13 ORIGINAL SPRING-TYPE IMPULSE VALVE FOR THE LOCALLY MADE ITDG HYDRAM

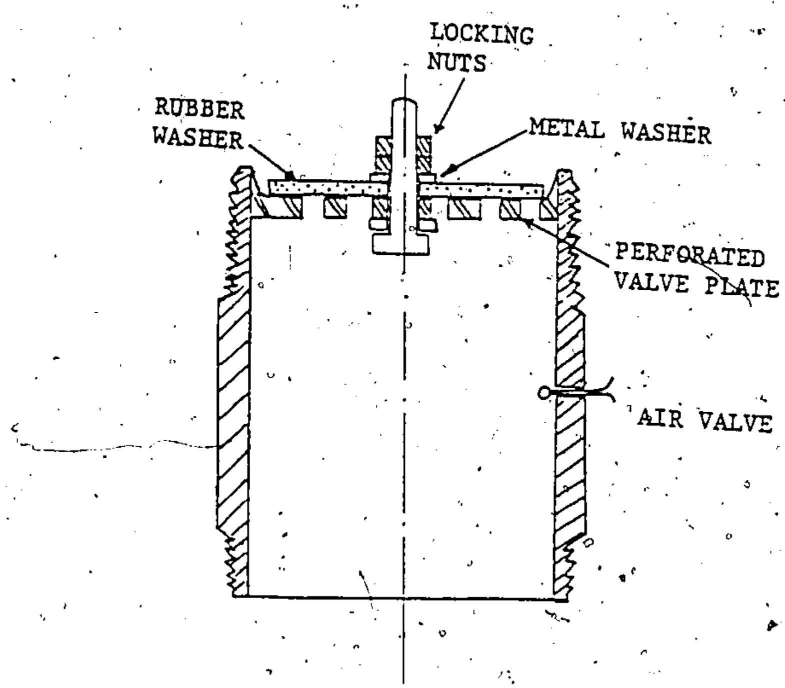


FIG. 14 DELIVERY (CHECK) VALVE ASSEMBLY FOR THE LOCALLY MADE HYDRAM

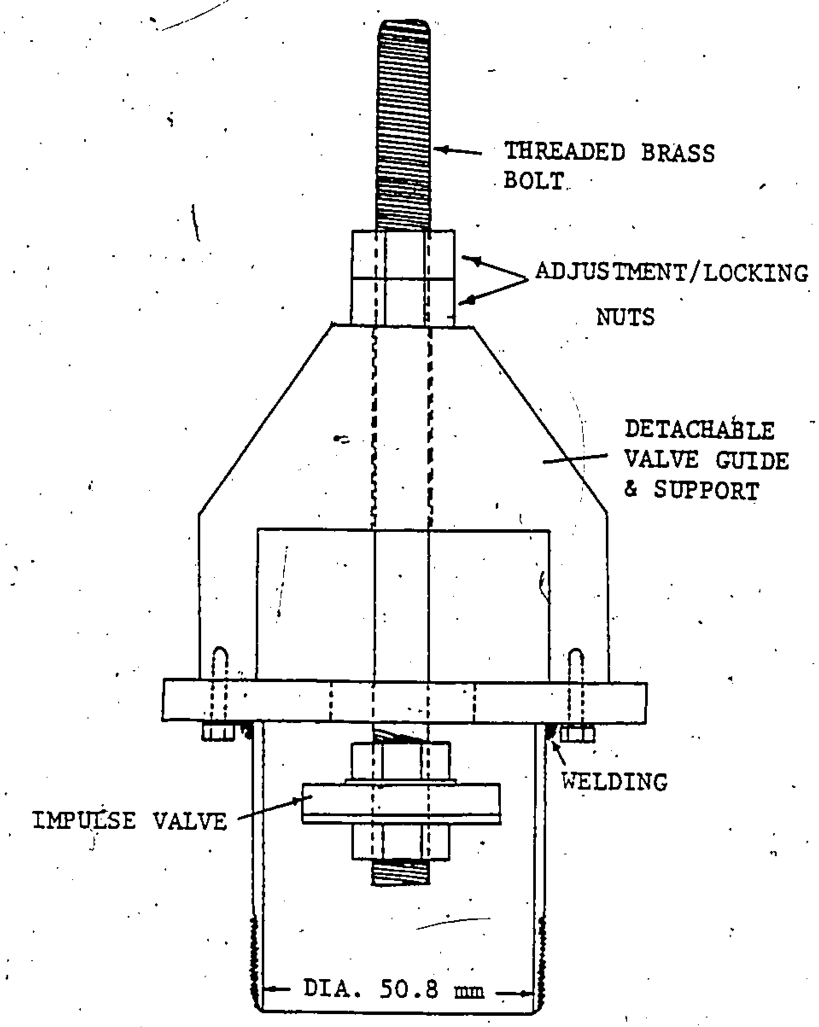


FIG. 15 WEIGHTED IMPULSE VALVE FOR THE LOCALLY MADE HYDRAM

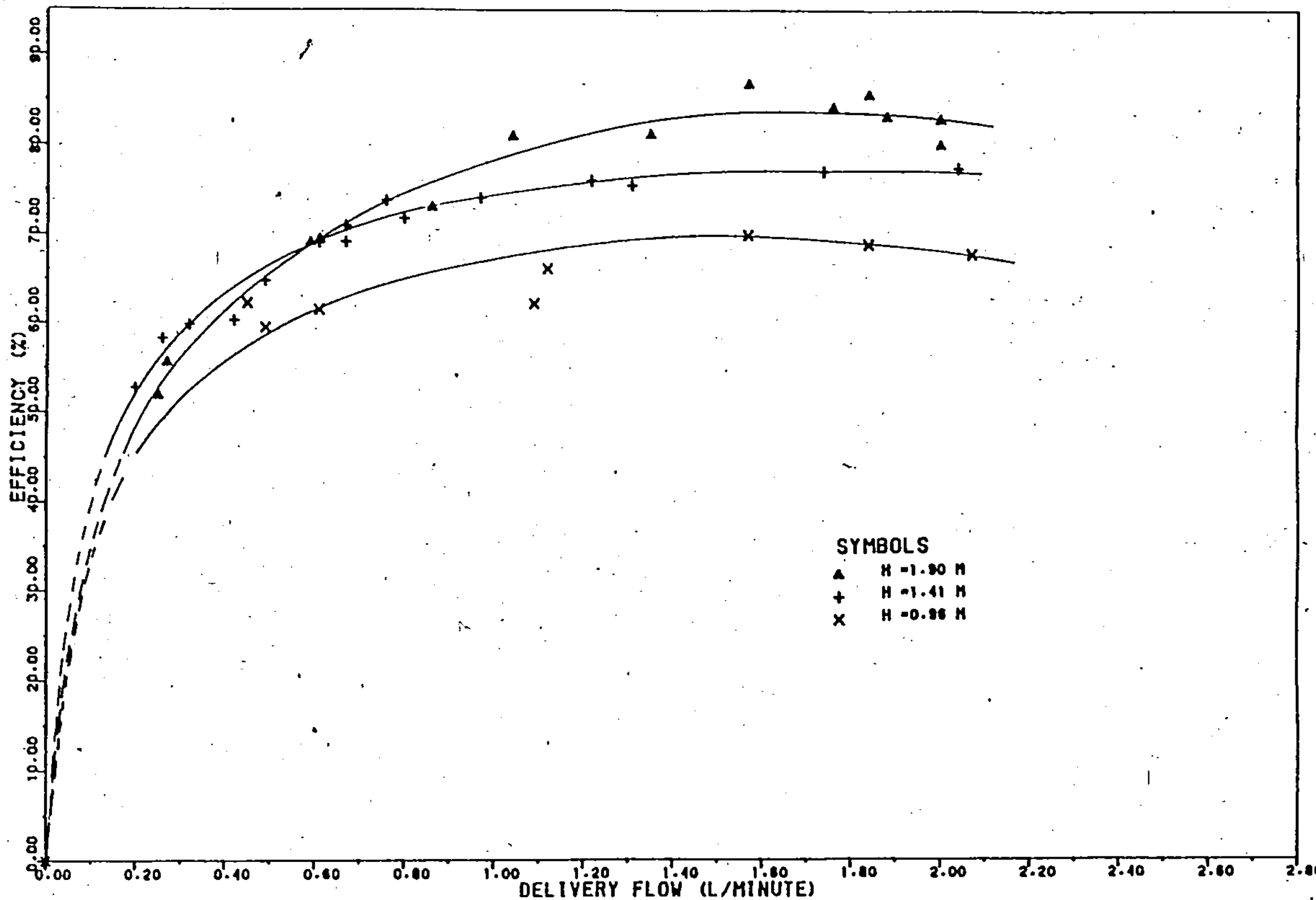


FIG. 16 FLEMING HYDRAM: EFFICIENCY VS DELIVERY FLOW

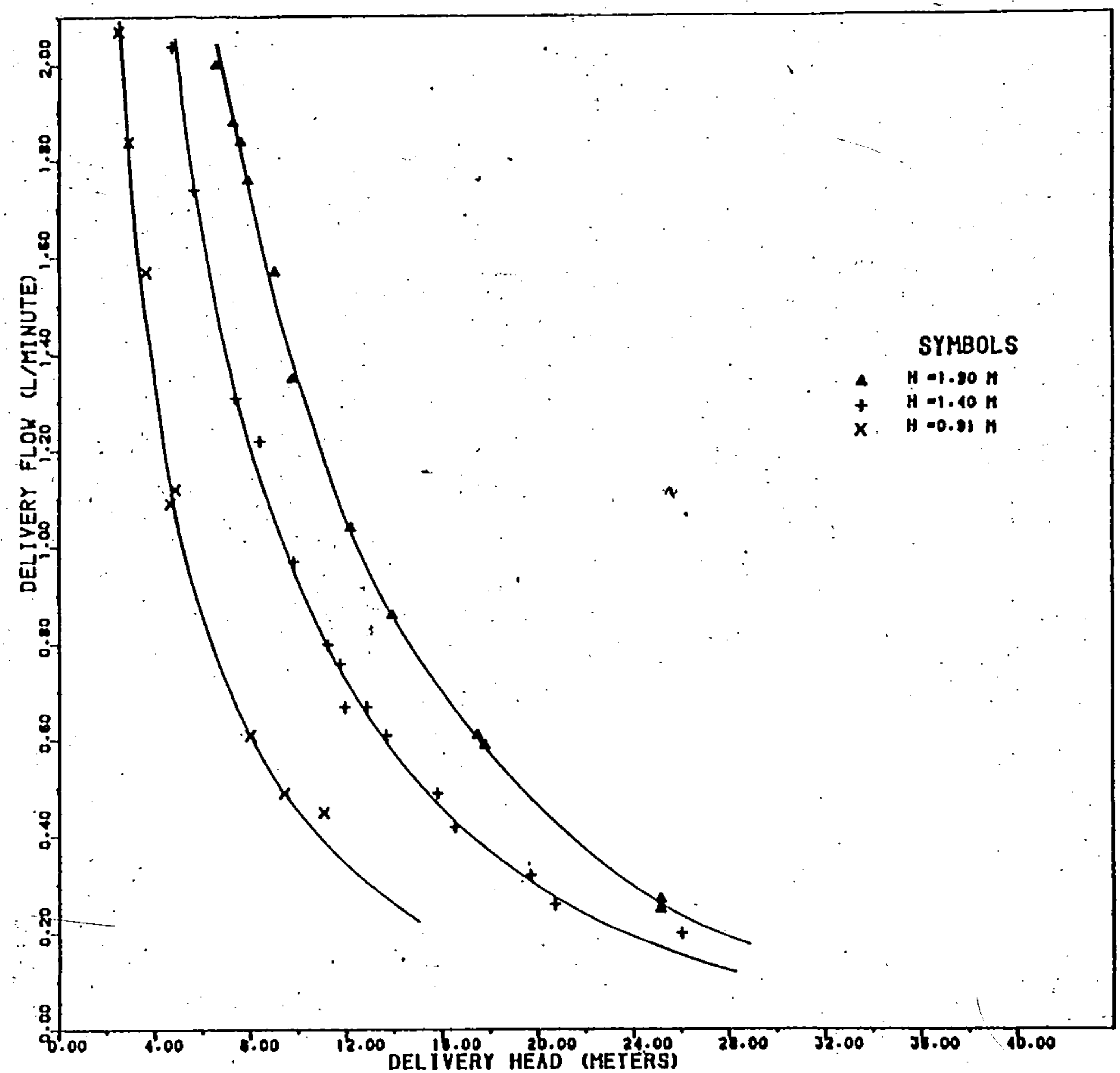


FIG. 17 FLEMING HYDRAM: DELIVERY FLOW VS DELIVERY HEAD

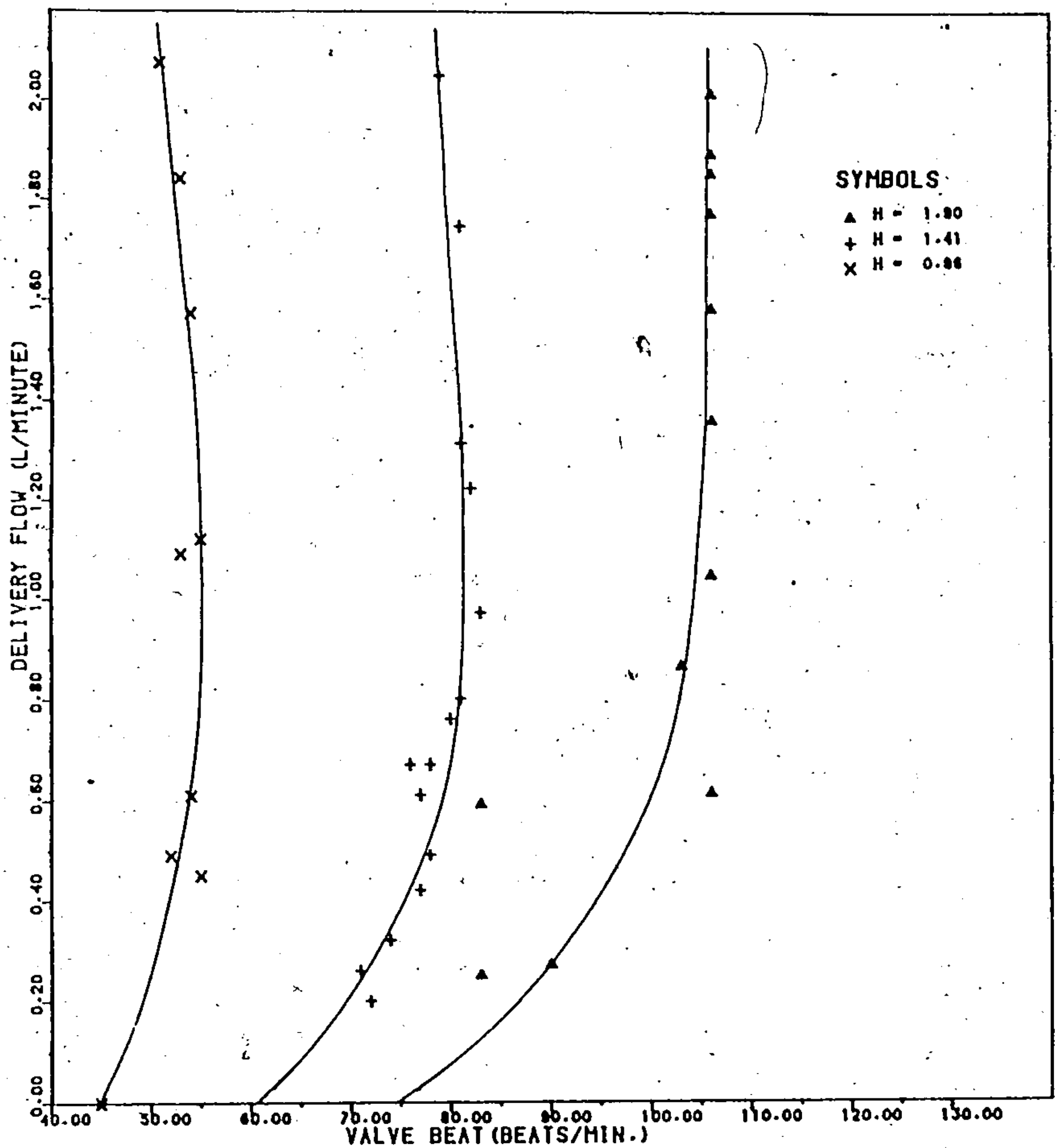


FIG.18 FLEMING HYDRAM-DELIVERY FLOW VS VALVE BEAT

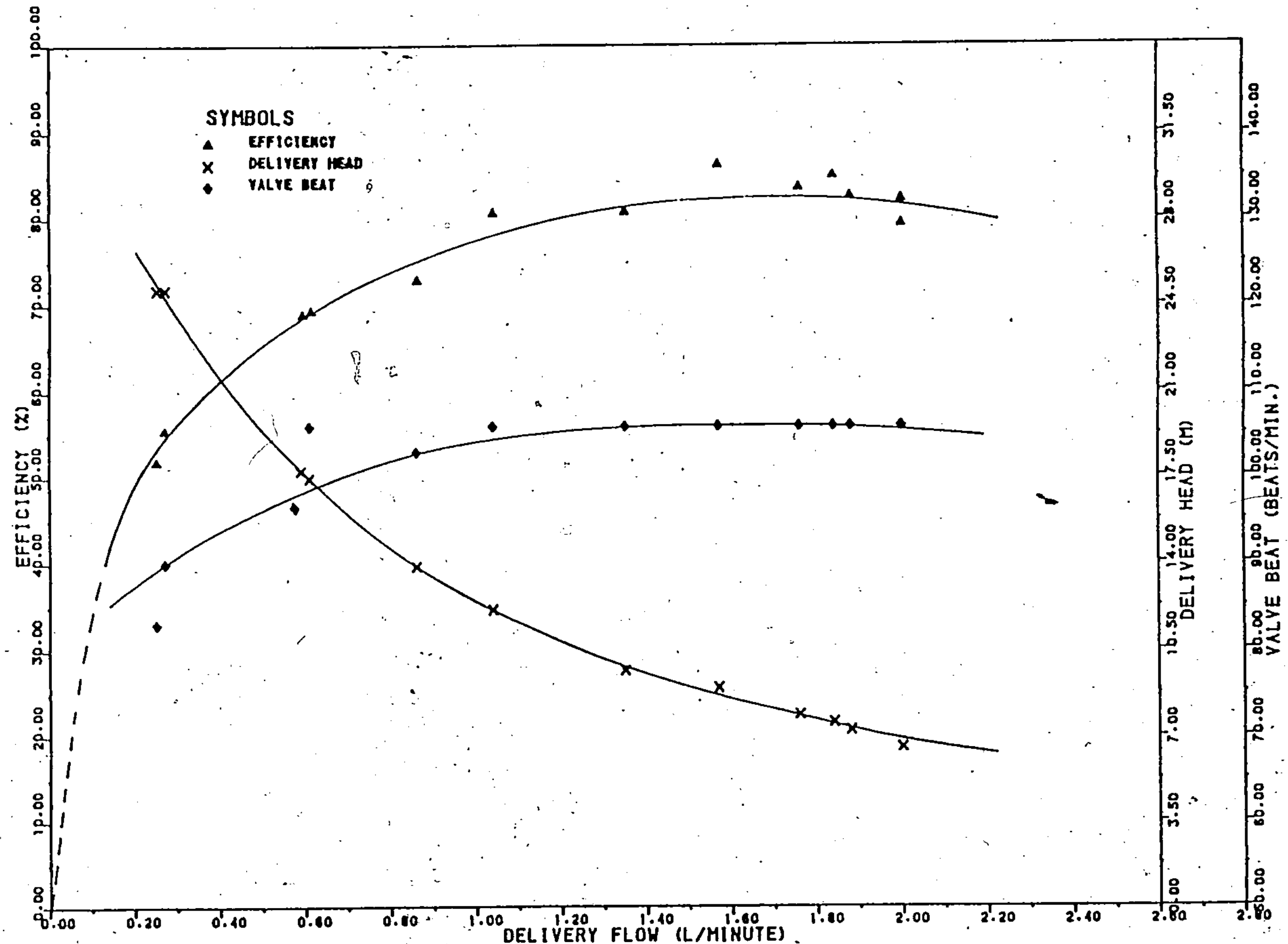


FIG. 19 FLEMING HYDRAM OPERATING CHARACTERISTICS H = 2M

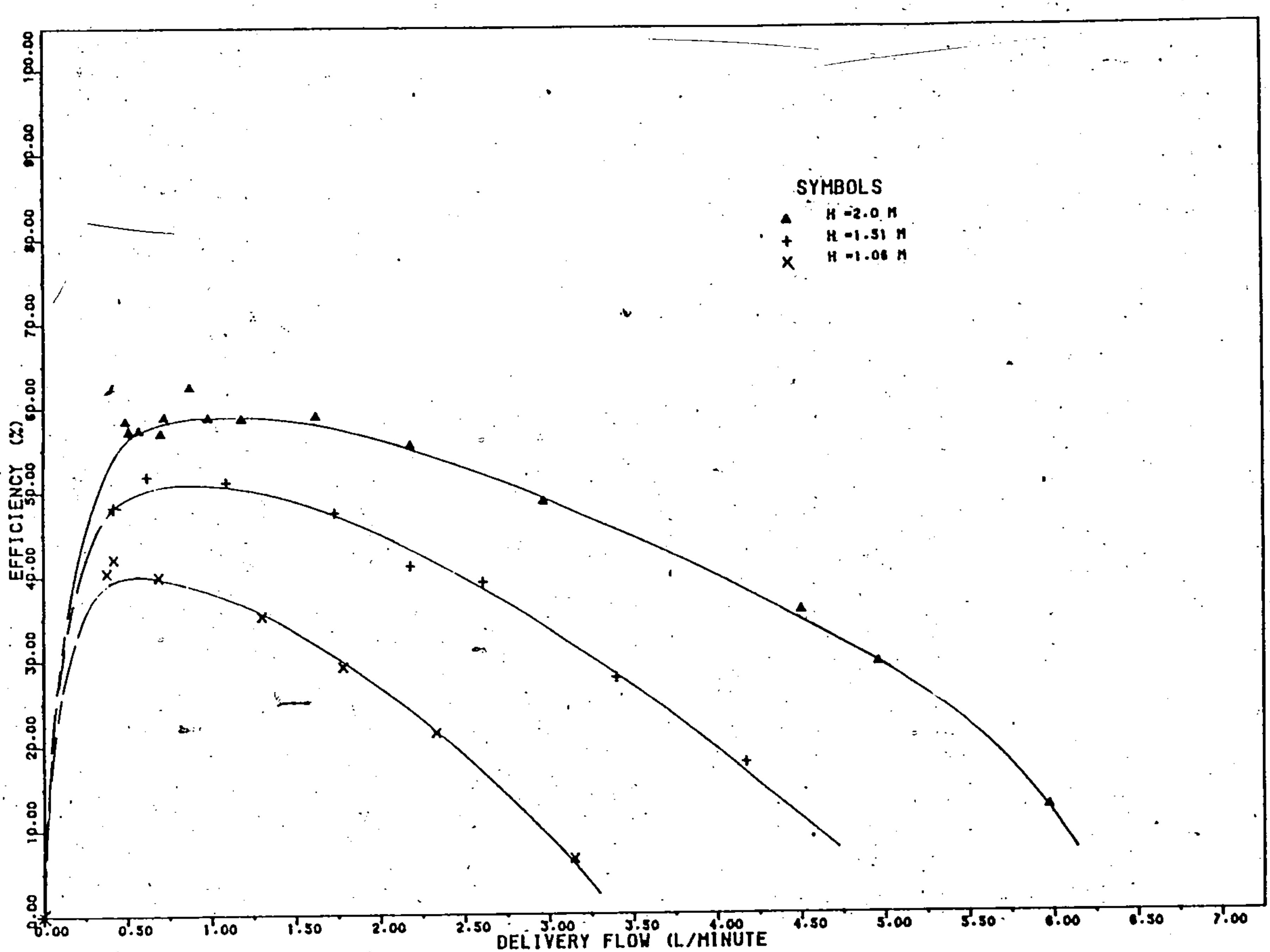


FIG20 RIFE HYDRAM: EFFICIENCY vs DELIVERY FLOW

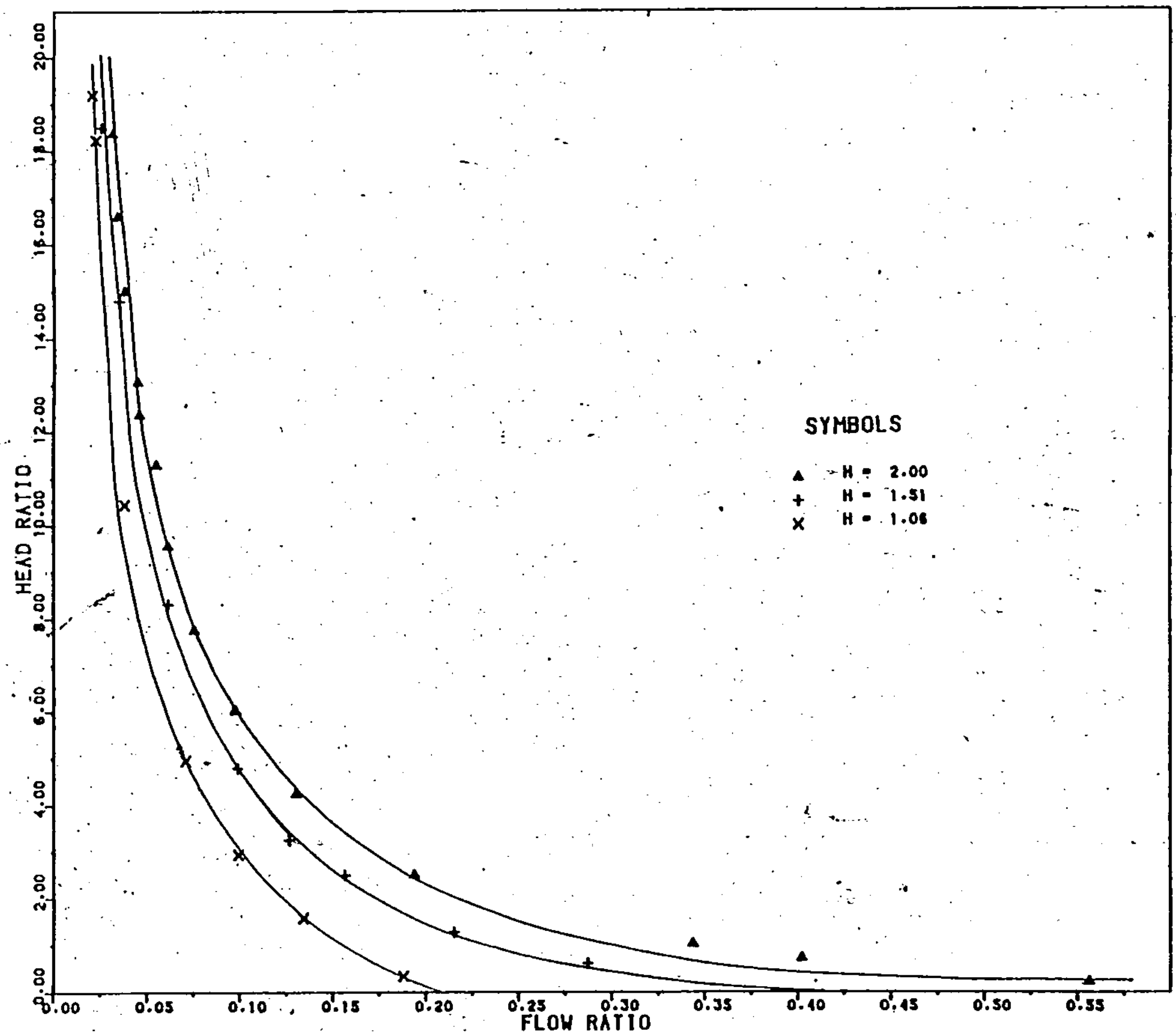


FIG. 21 RIFE HYDRAM: HEAD RATIO VS FLOW RATIO

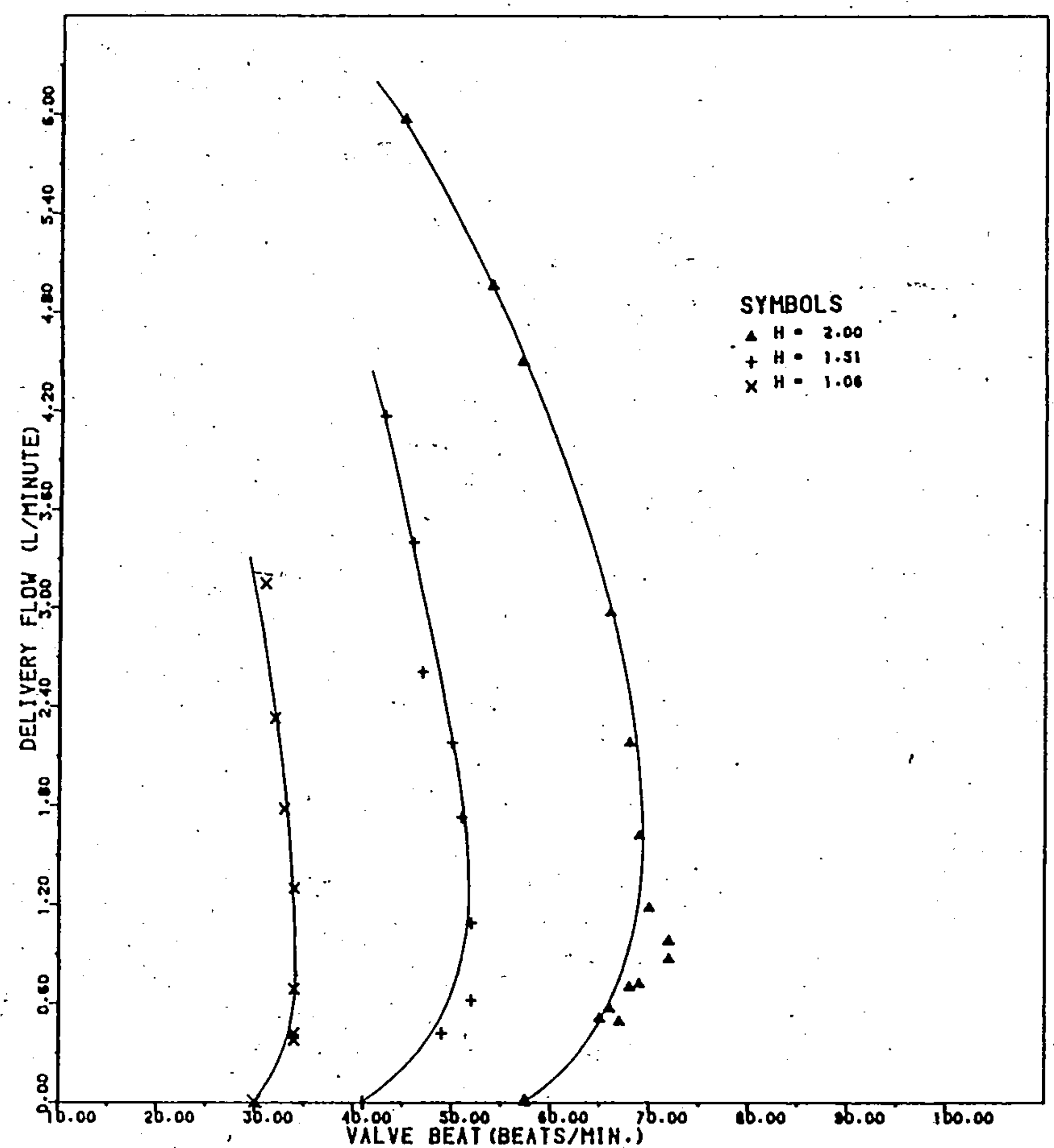


FIG. 22 RIFE HYDRAM: DELIVERY FLOW VS VALVE BEAT

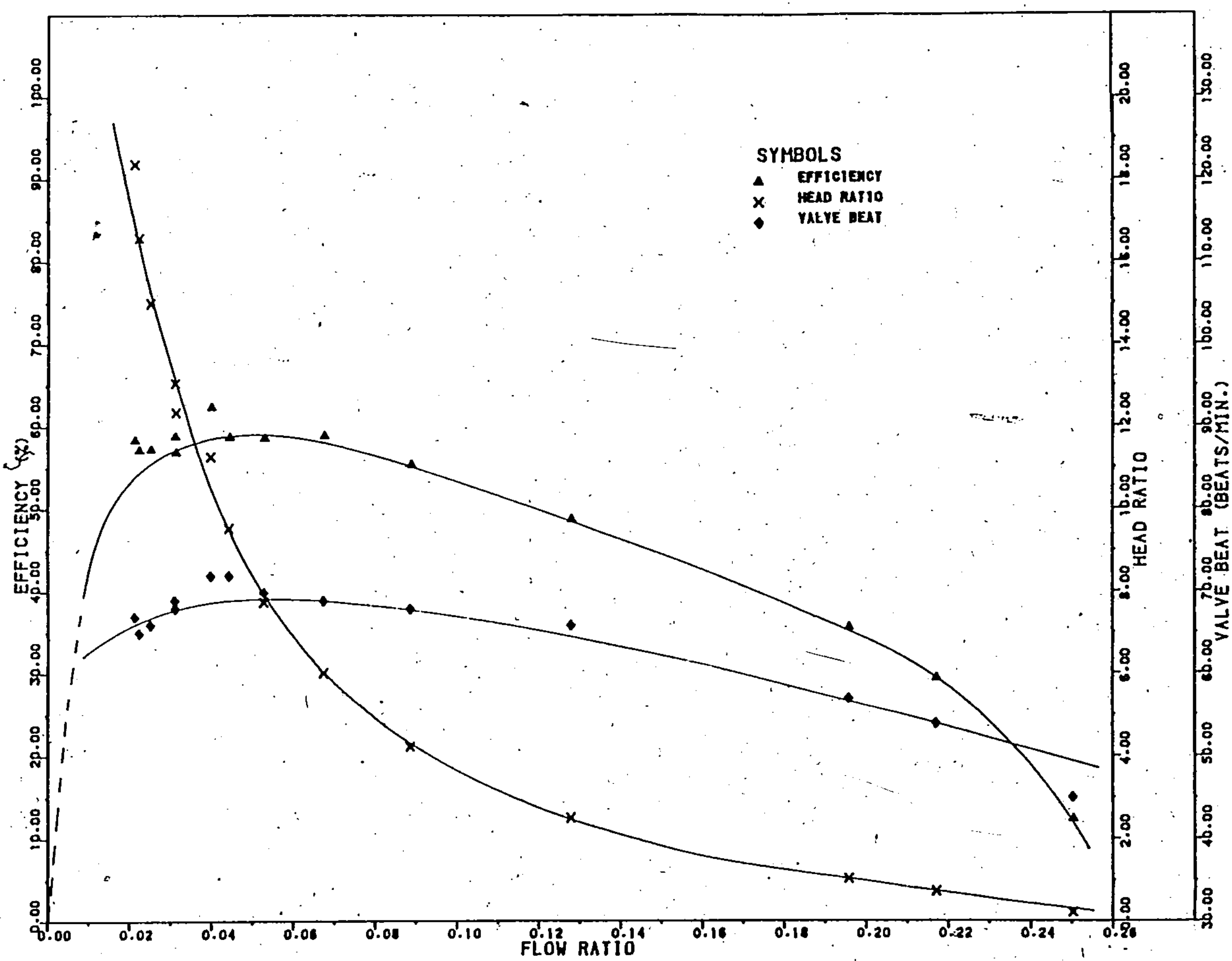


FIG. 23 RIFE HYDRAM: OPERATING CHARACTERISTICS FOR H =2 M

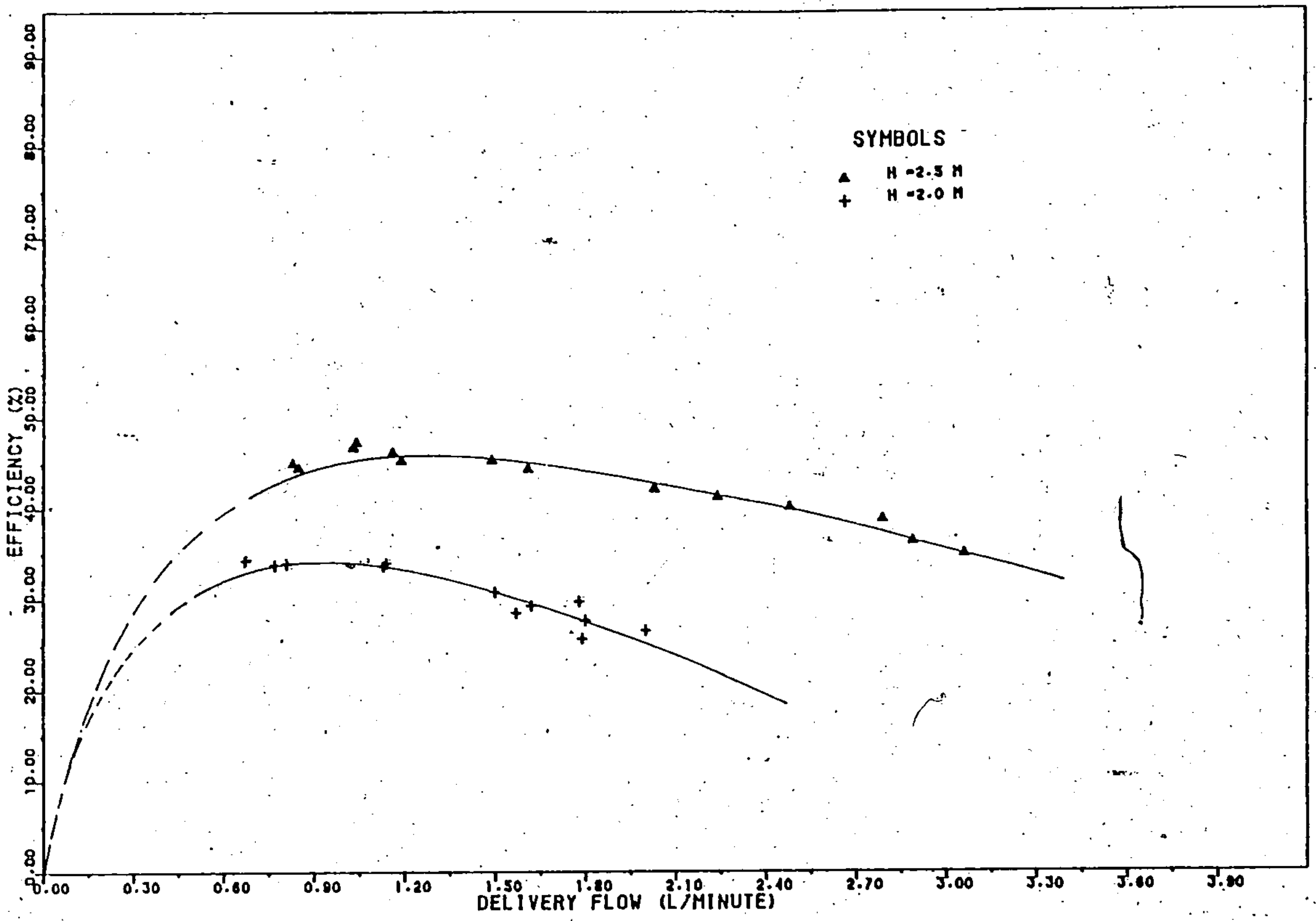


FIG. 24 BLAKE HYDRAM: EFFICIENCY VS DELIVERY FLOW

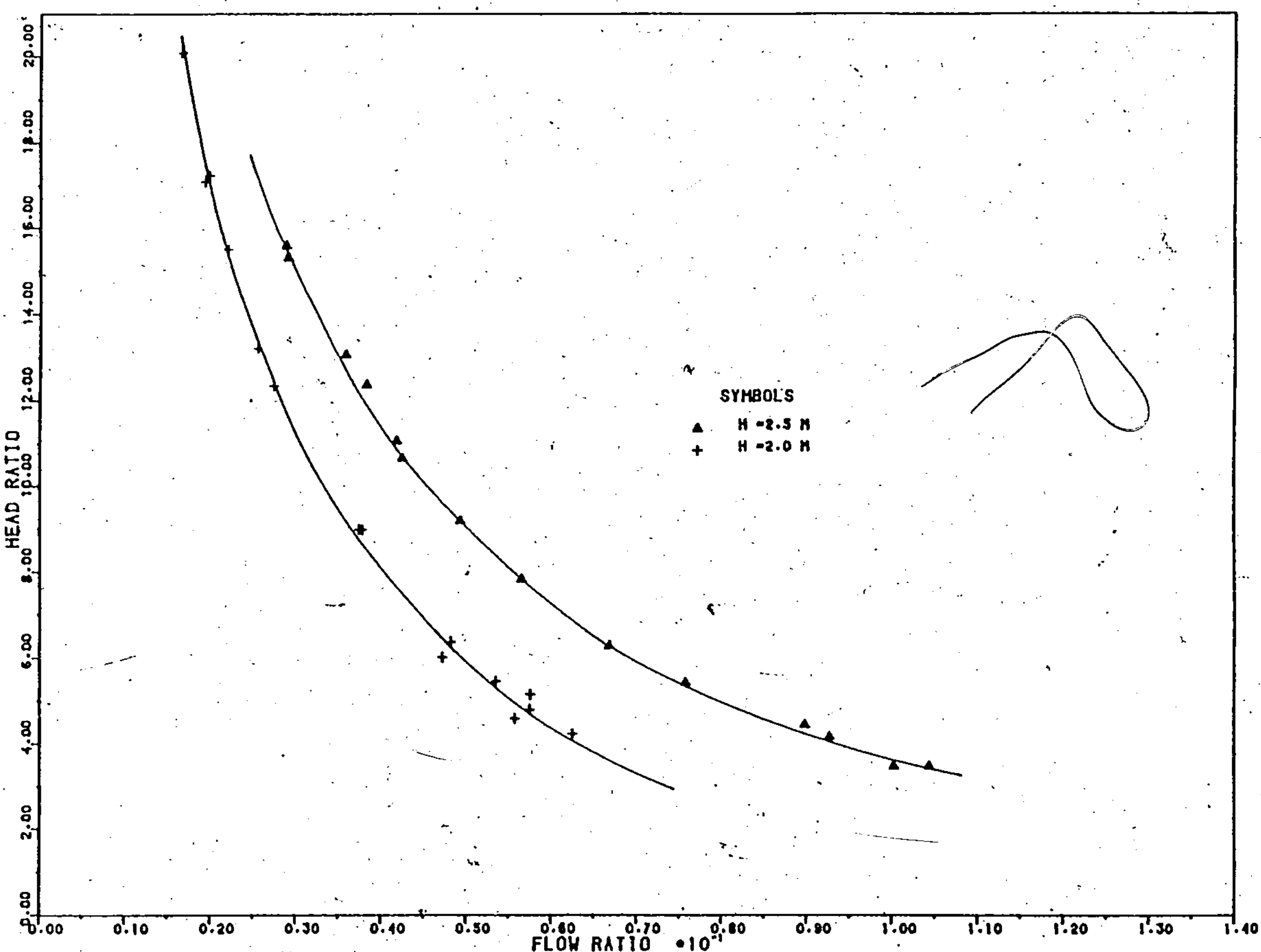


FIG. 25 BLAKE HYDRAM: HEAD RATIO VS FLOW RATIO

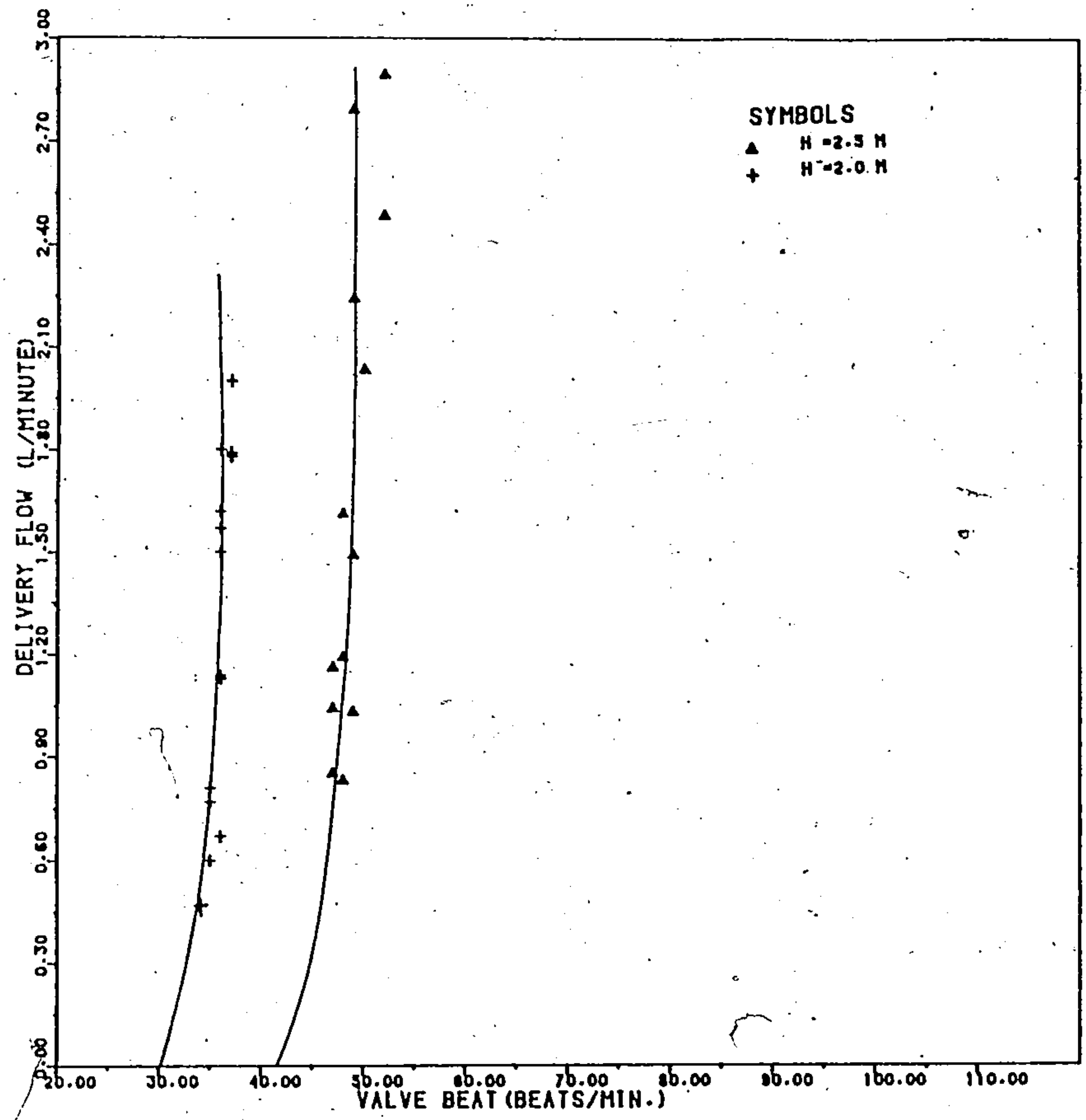


FIG.26 BLAKE HYDRAM: DELIVERY FLOW VS VALVE BEAT

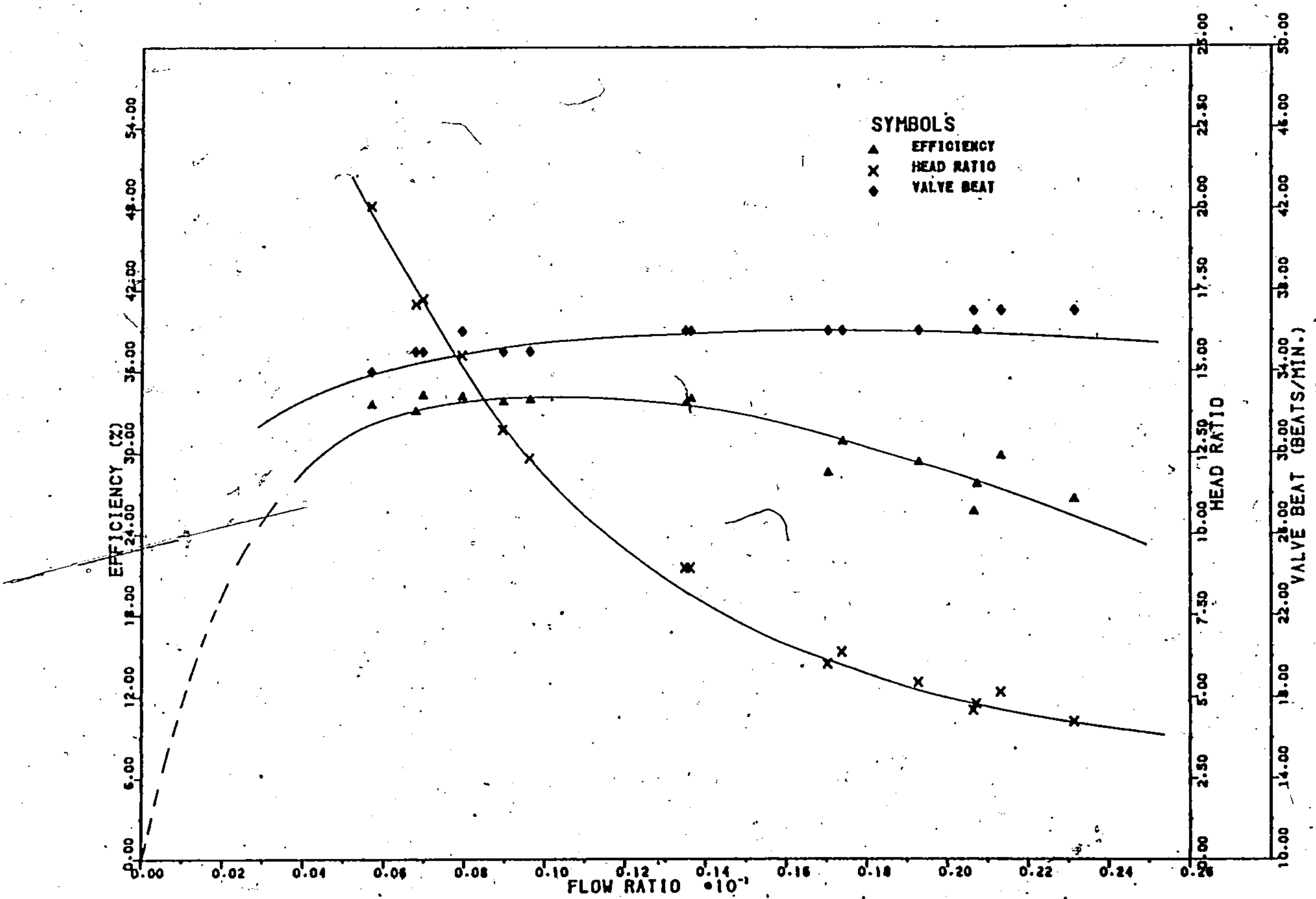


FIG.27 BLAKE HYDRAM: OPERATING CHARACTERISTICS FOR H = 2 M

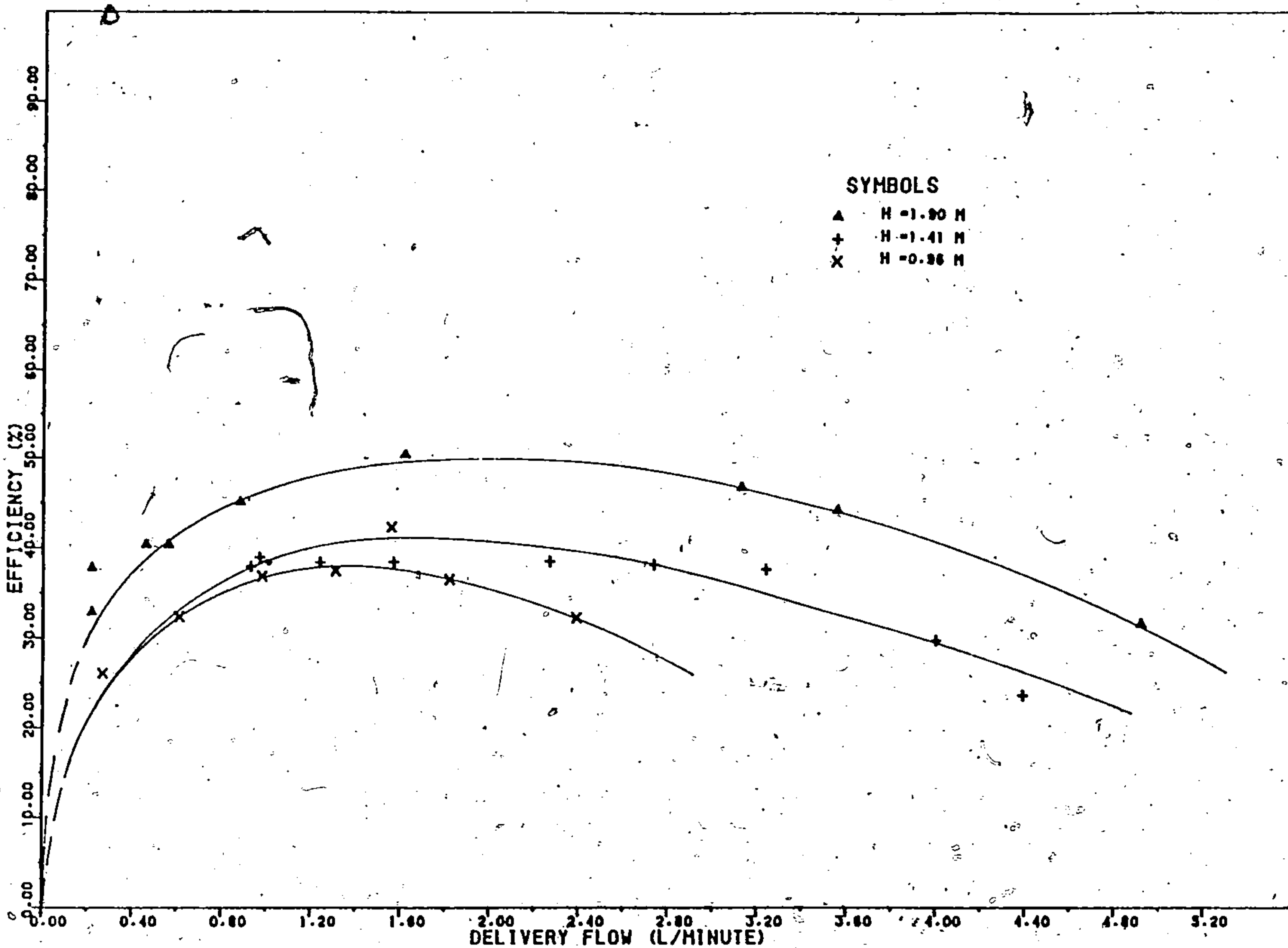


FIG.28 ITDG HYDRAM: EFFICIENCY VS DELIVERY FLOW

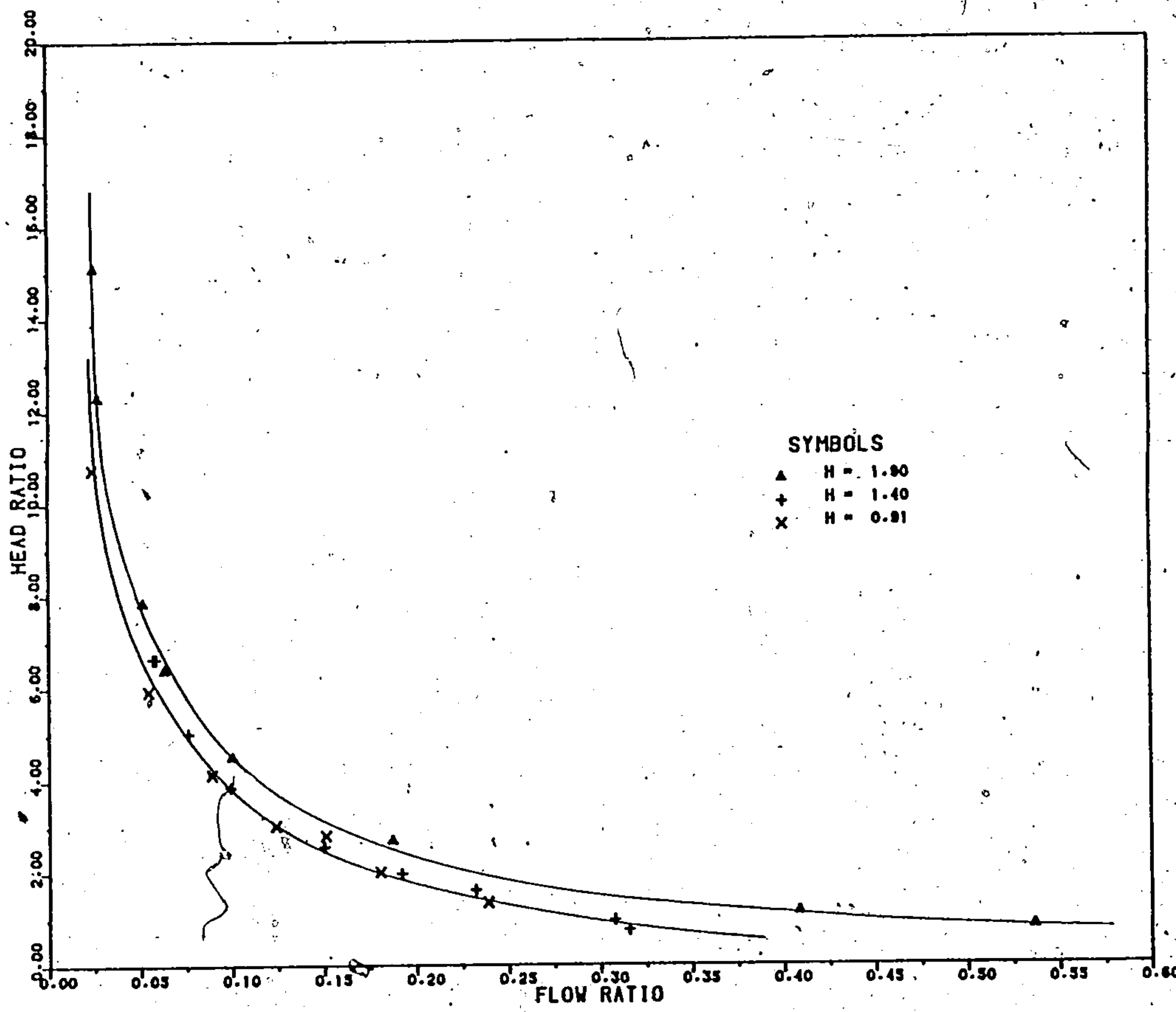


FIG. 29 ITDG HYDRAM: HEAD RATIO VS FLOW RATIO

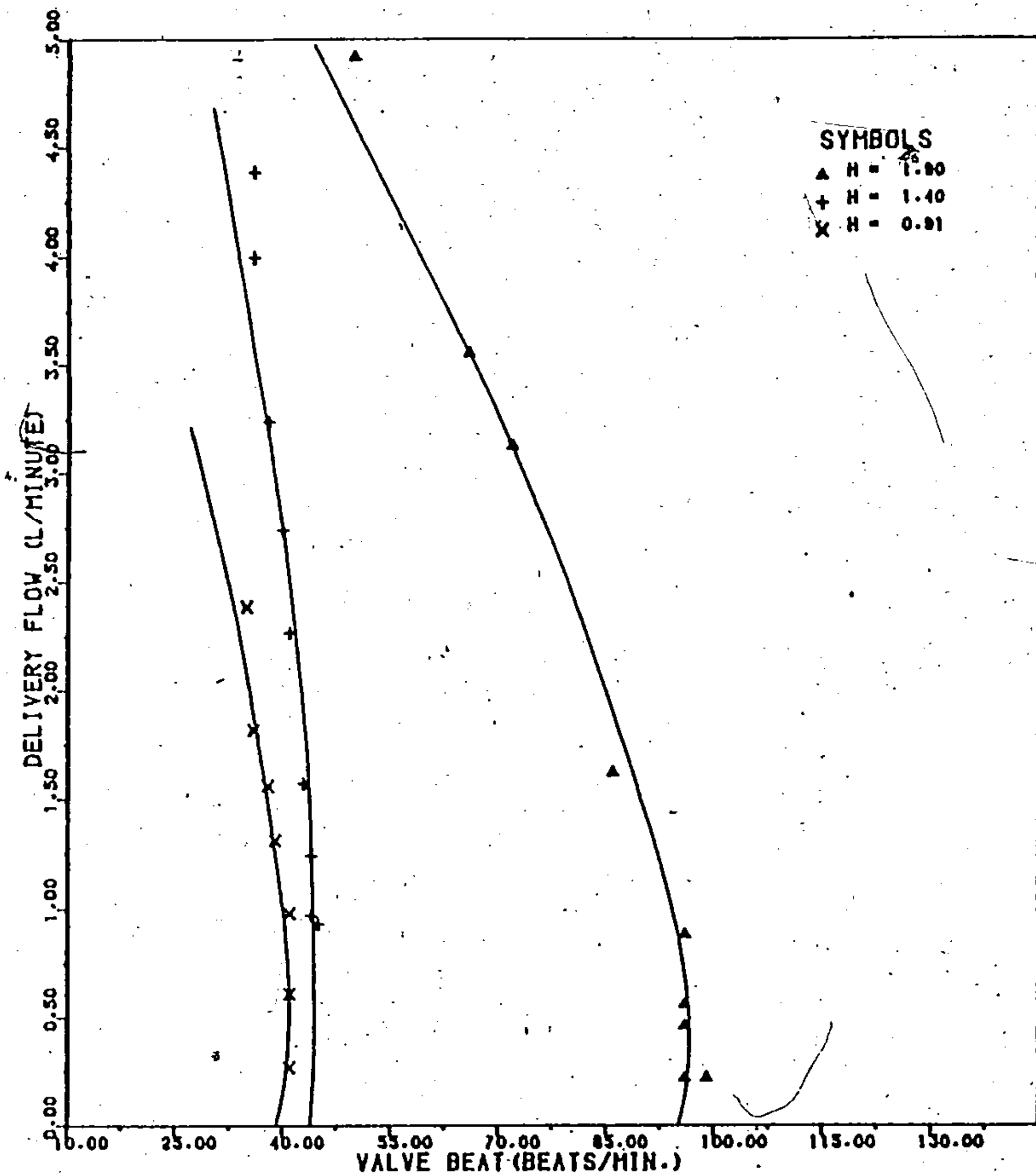


FIG. 30 ITDG HYDRAM: DELIVERY FLOW VS VALVE BEAT

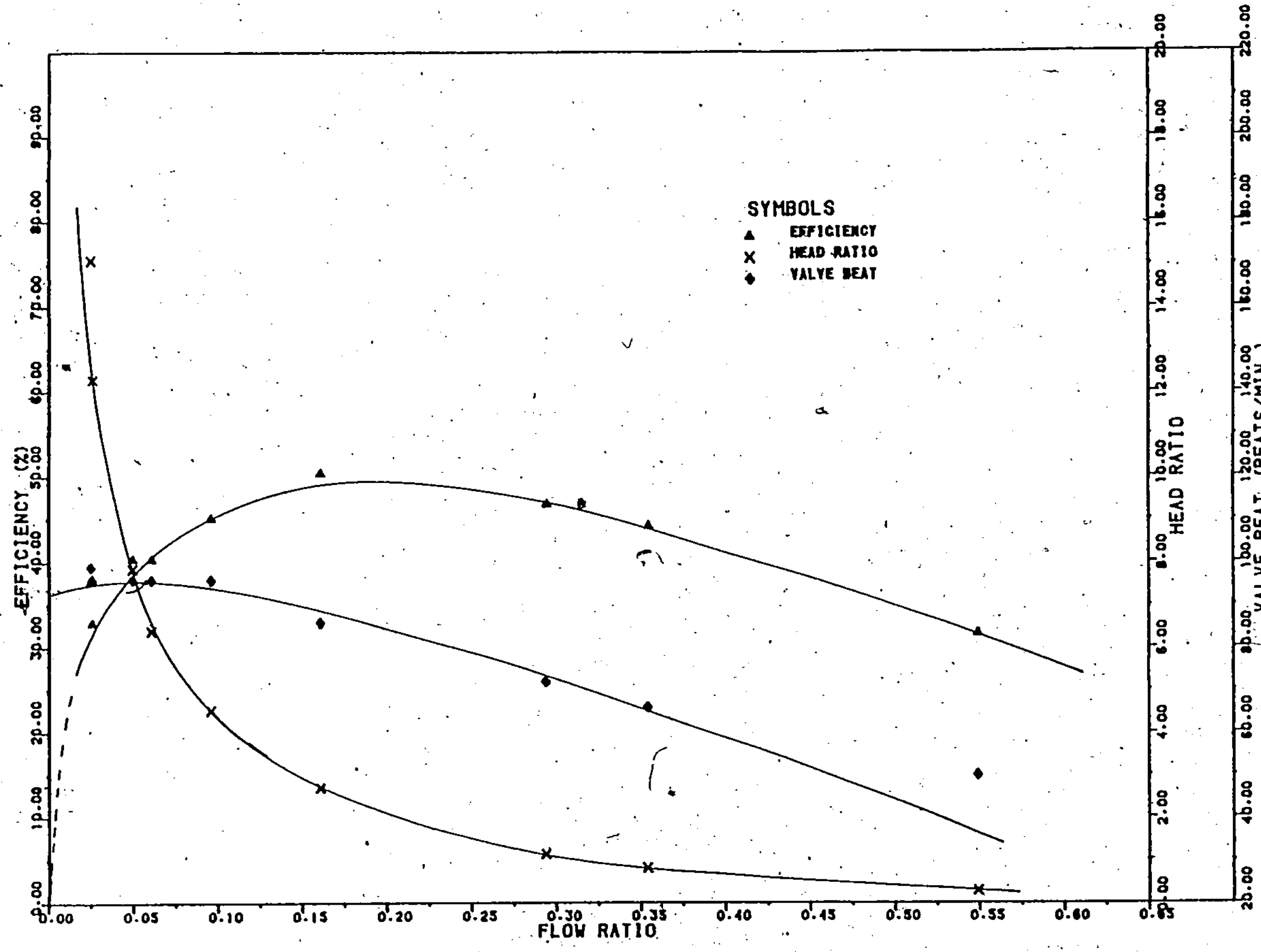


FIG. 31 ITDG HYDRAM: OPERATING CHARACTERISTICS FOR H =2 M

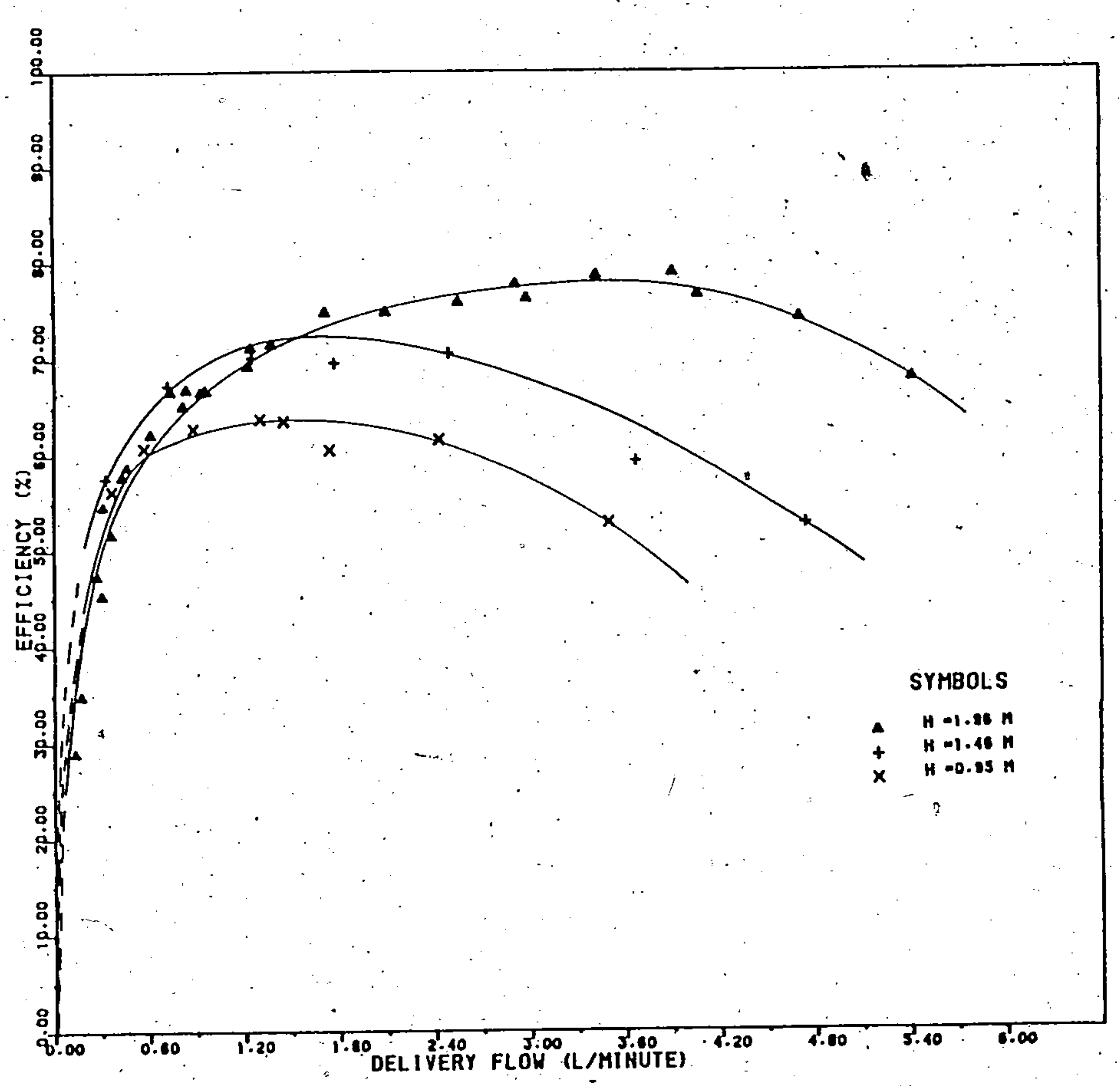


FIG. 32 DAVEY HYDRAM: EFFICIENCY VS DELIVERY FLOW

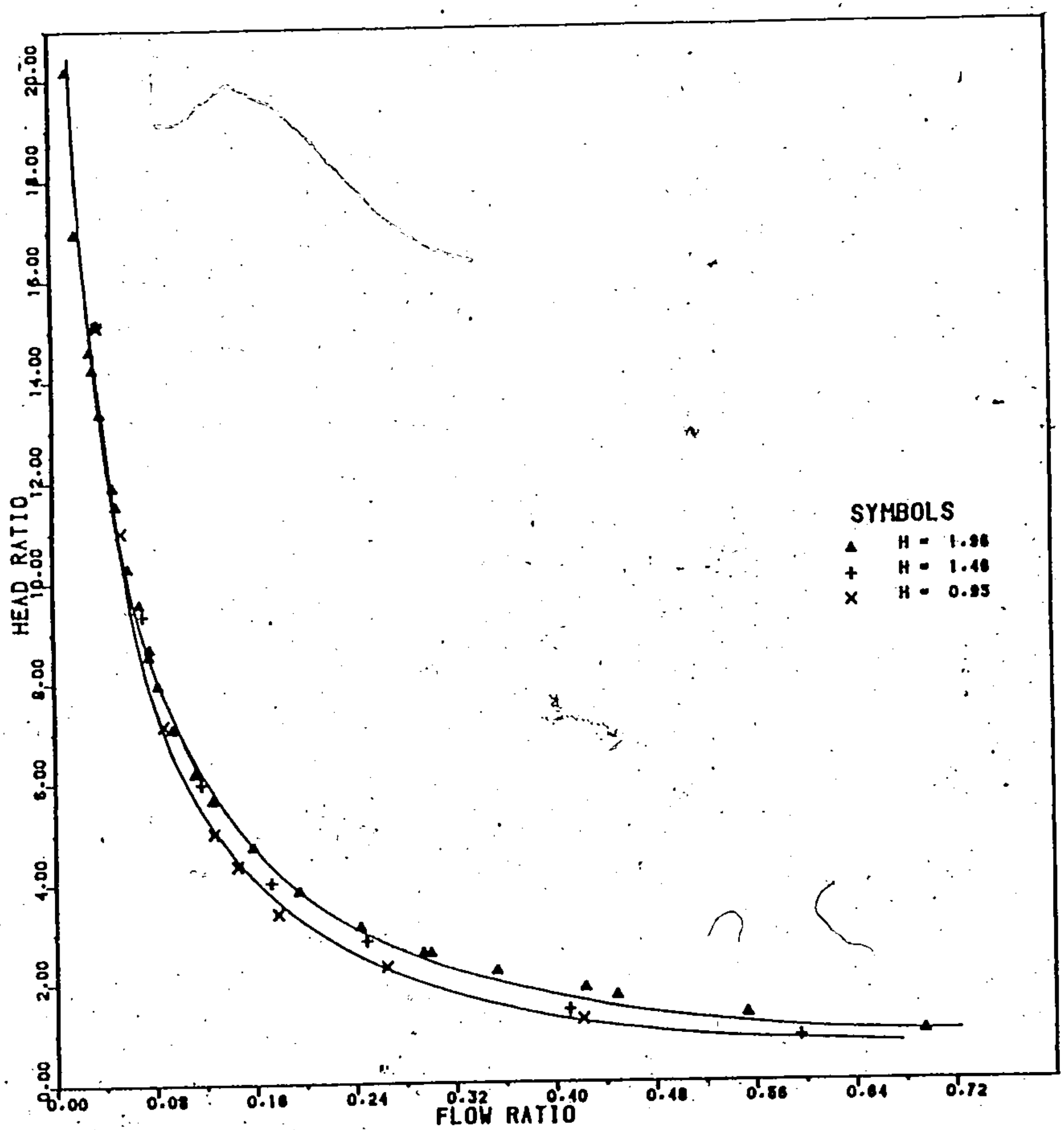


FIG. 33 DAVEY HYDRAM: HEAD RATIO VS FLOW RATIO

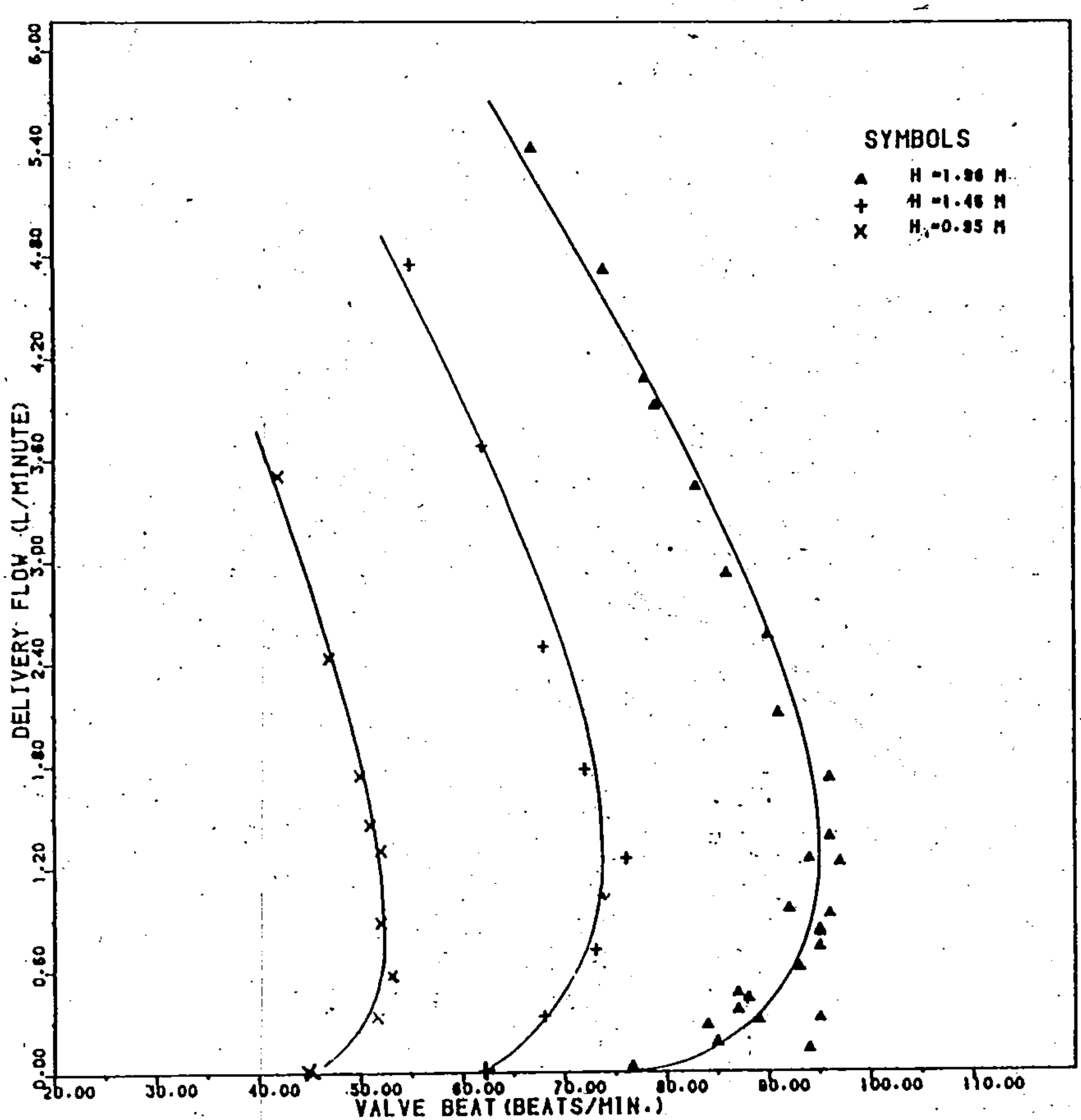


FIG.34 DAVEY HYDRAM:DELIVERY FLOW VS VALVE BEAT

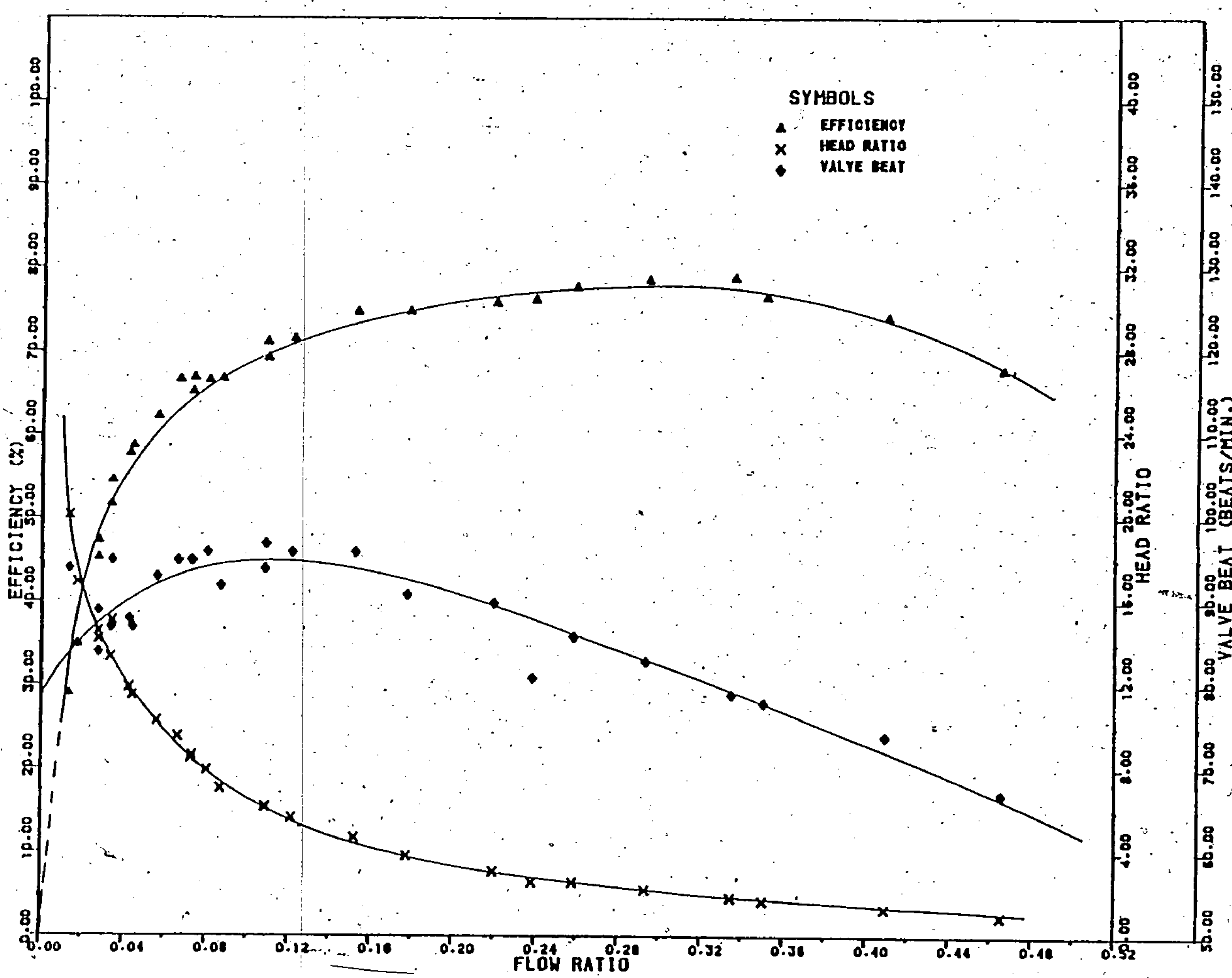


FIG-35 DAVEY HYDRAM: OPERATING CHARACTERISTICS FOR H =2 M

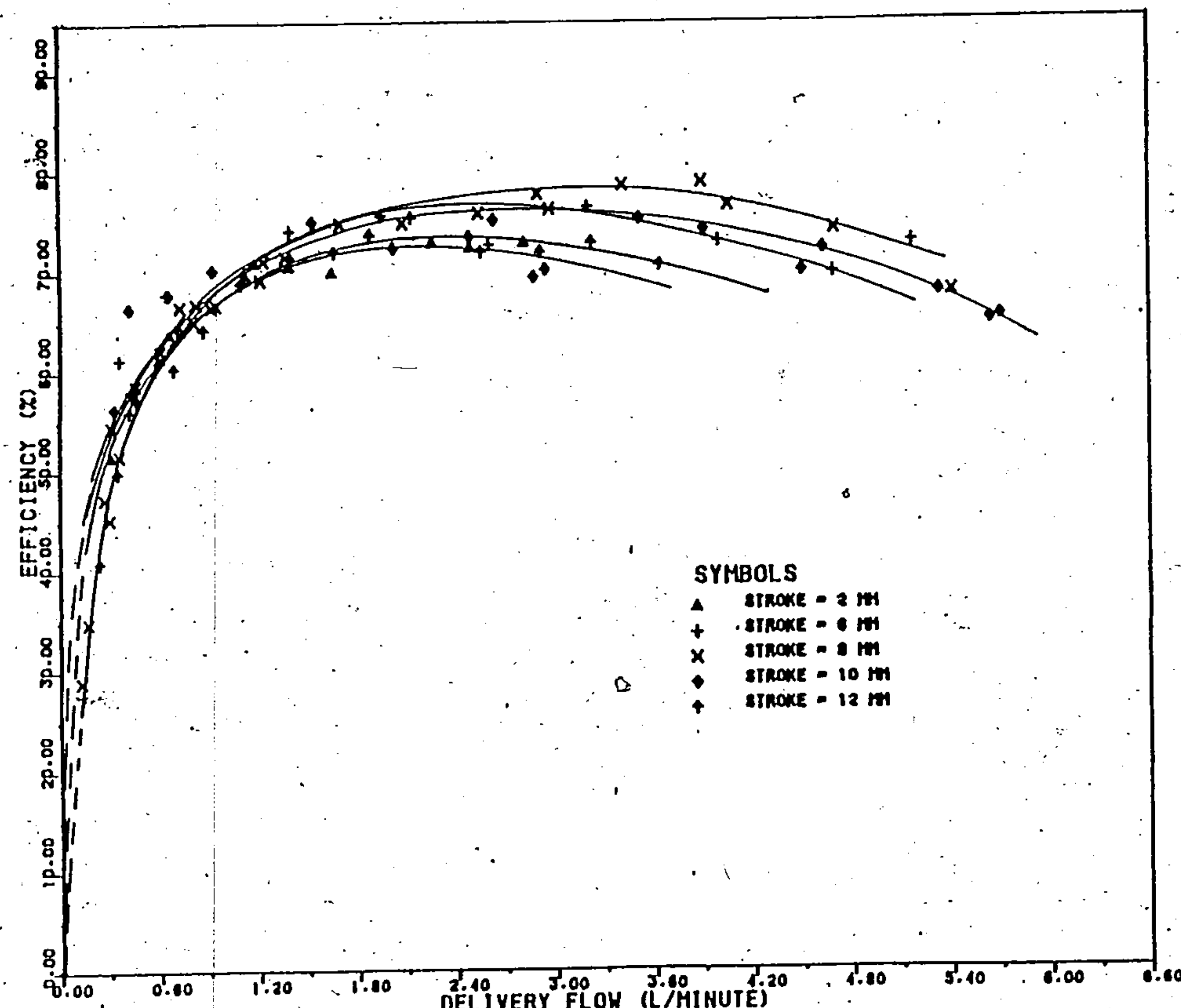


FIG. 36 DAVEY HYDRAM: VARIATION OF PUMP EFFICIENCY WITH VALVE STROKE

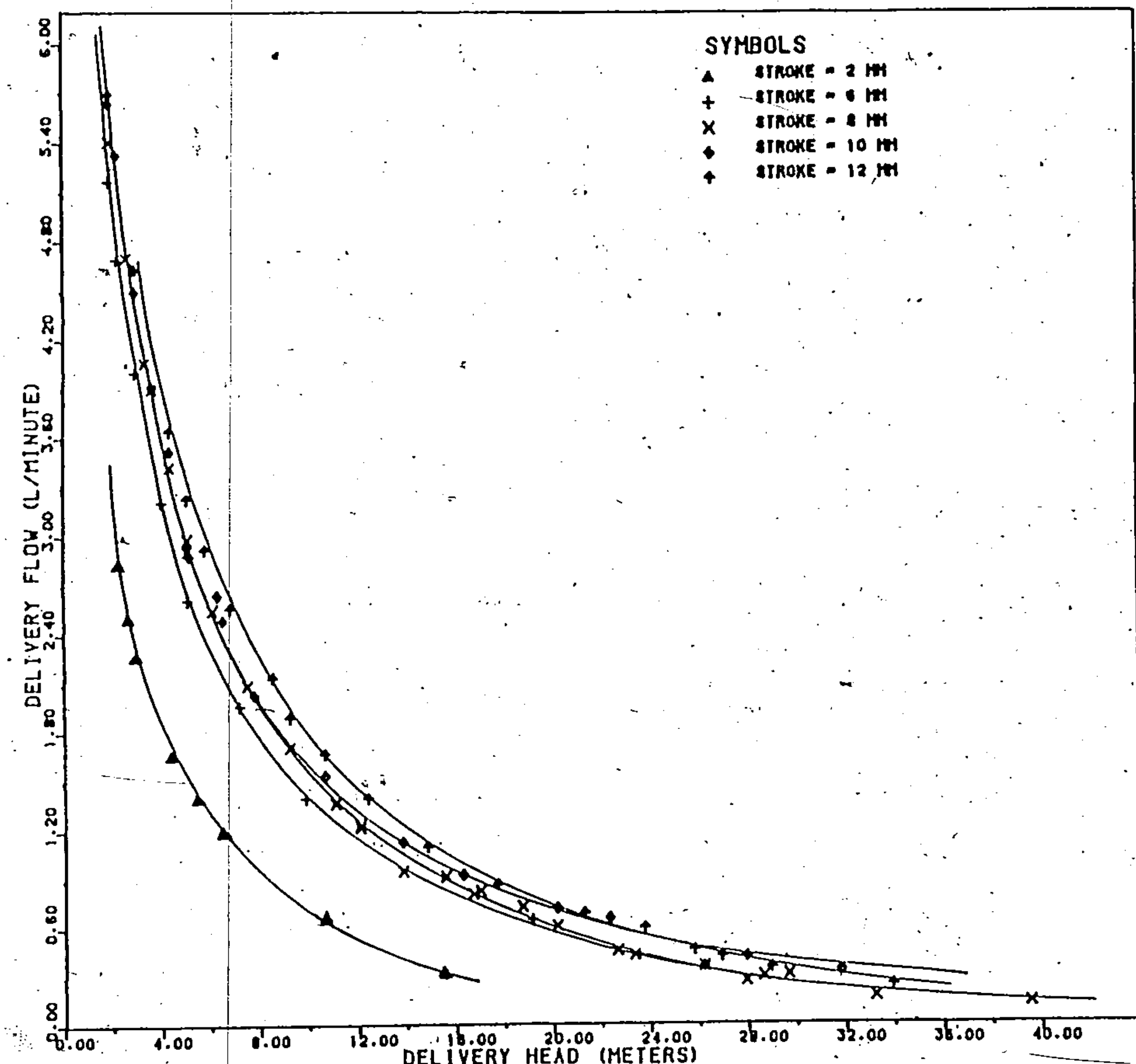


FIG. 37 DAVEY HYDRAM EFFECT OF VALVE STROKE ON PUMP CAPACITY

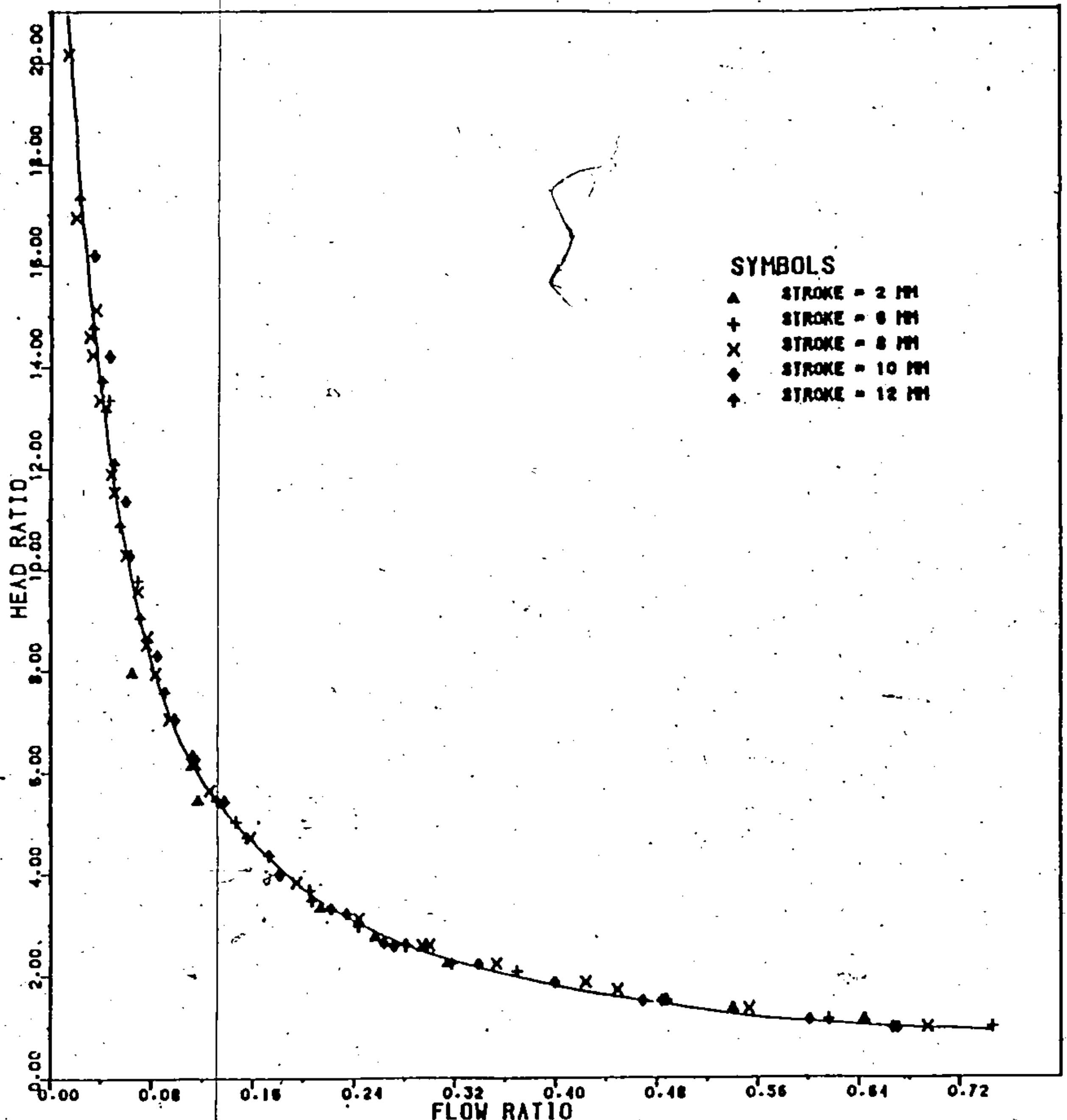


FIG. 38 DAVEY HYDRAM: HEAD RATIO VS FLOW RATIO FOR DIFFERENT STROKE LENGTHS

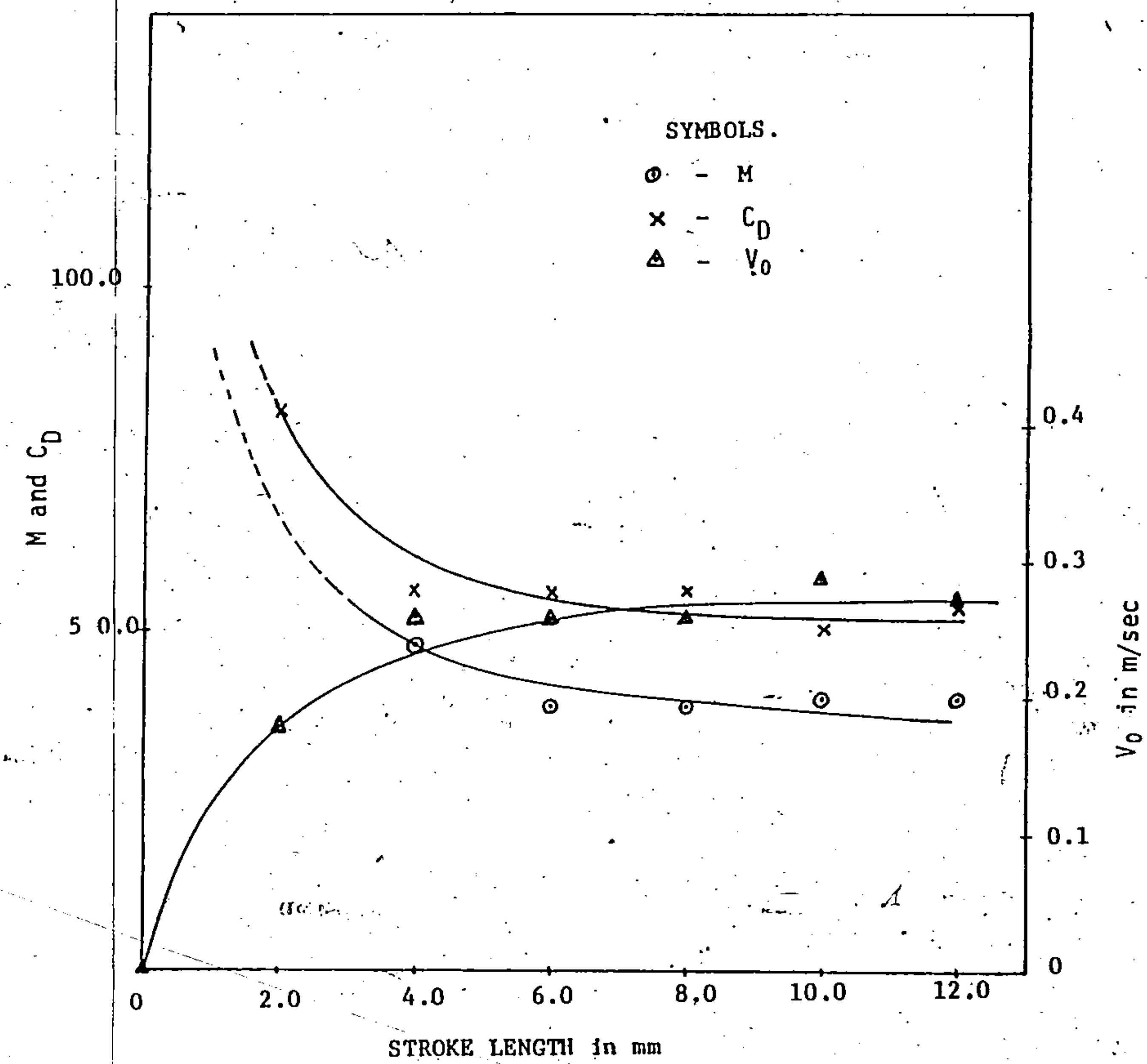


FIG. 39 VARIATION OF M, C_D AND V_0 WITH STROKE LENGTH FOR THE DAVEY HYDRAM

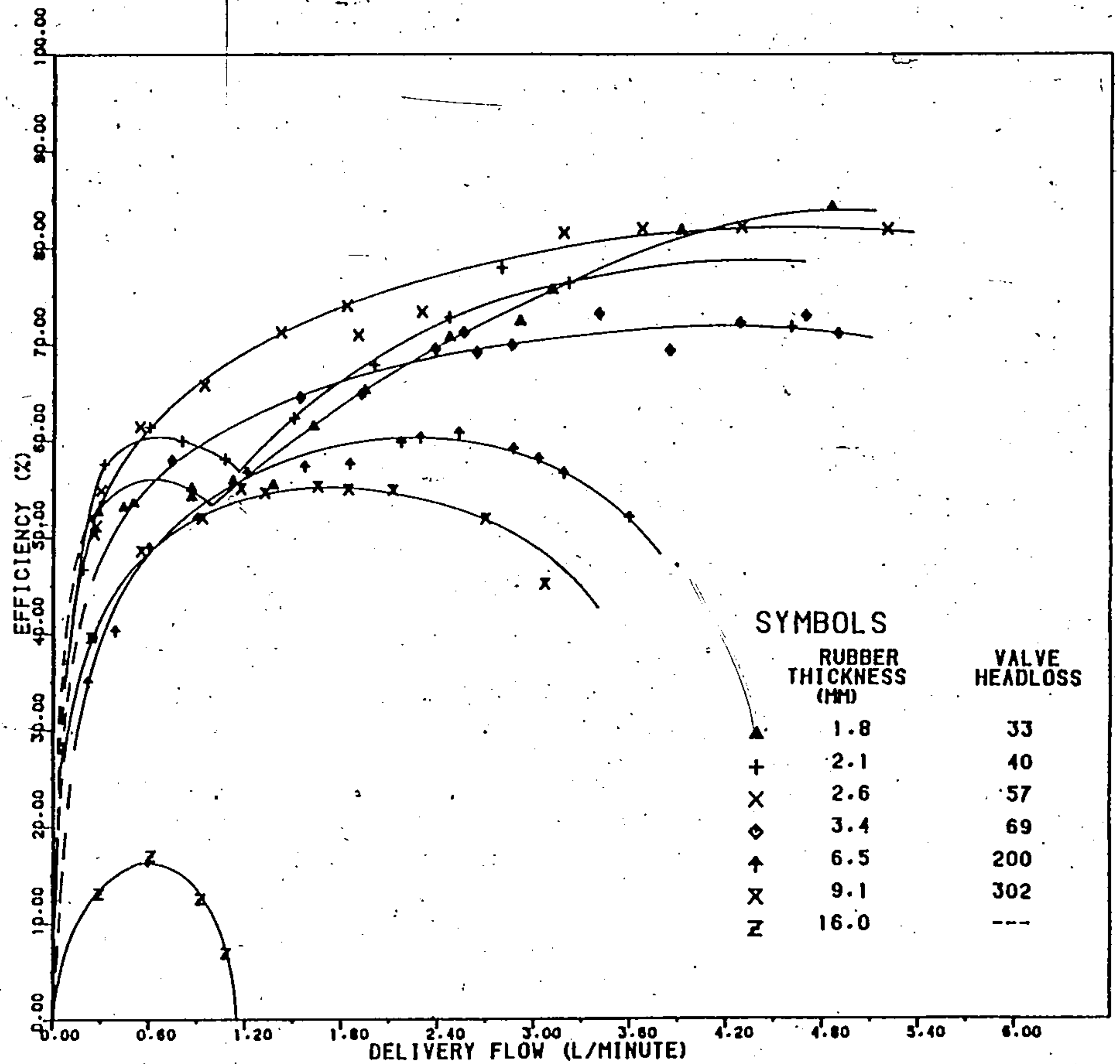


FIG.40 ITDG(IMPULSE) HYDRAM:EFFECT OF DELIVERY VALVE ON PUMP EFFICIENCY

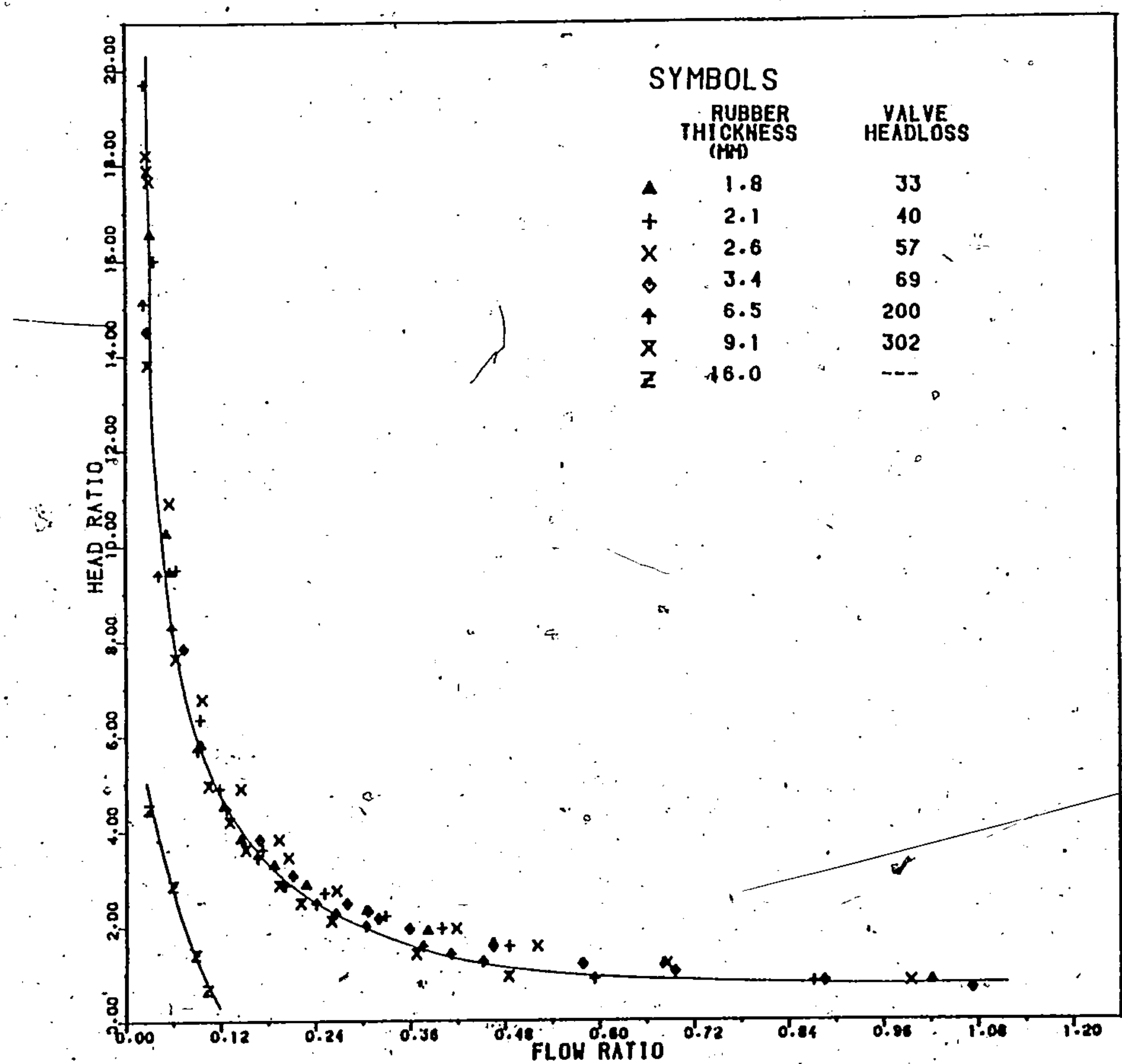


FIG. 41 ITDG (IMPULSE) HYDRAM: EFFECT OF DELIVERY VALVE ON PUMP

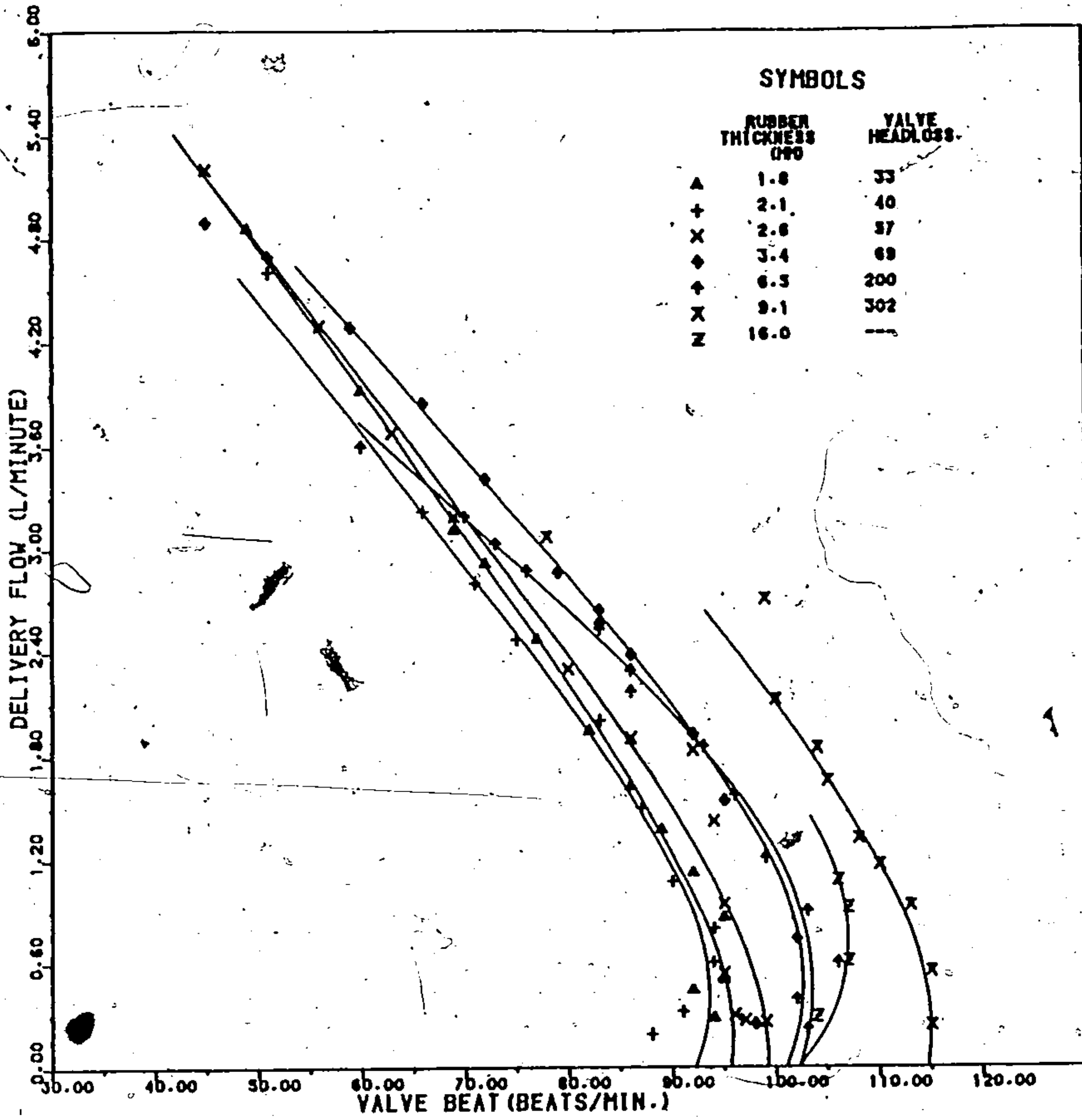


FIG. 42 ITDG (IMPULSE) HYDRAM EFFECT OF DELIVERY VALVE ON VALVE BEAT FREQUENCY

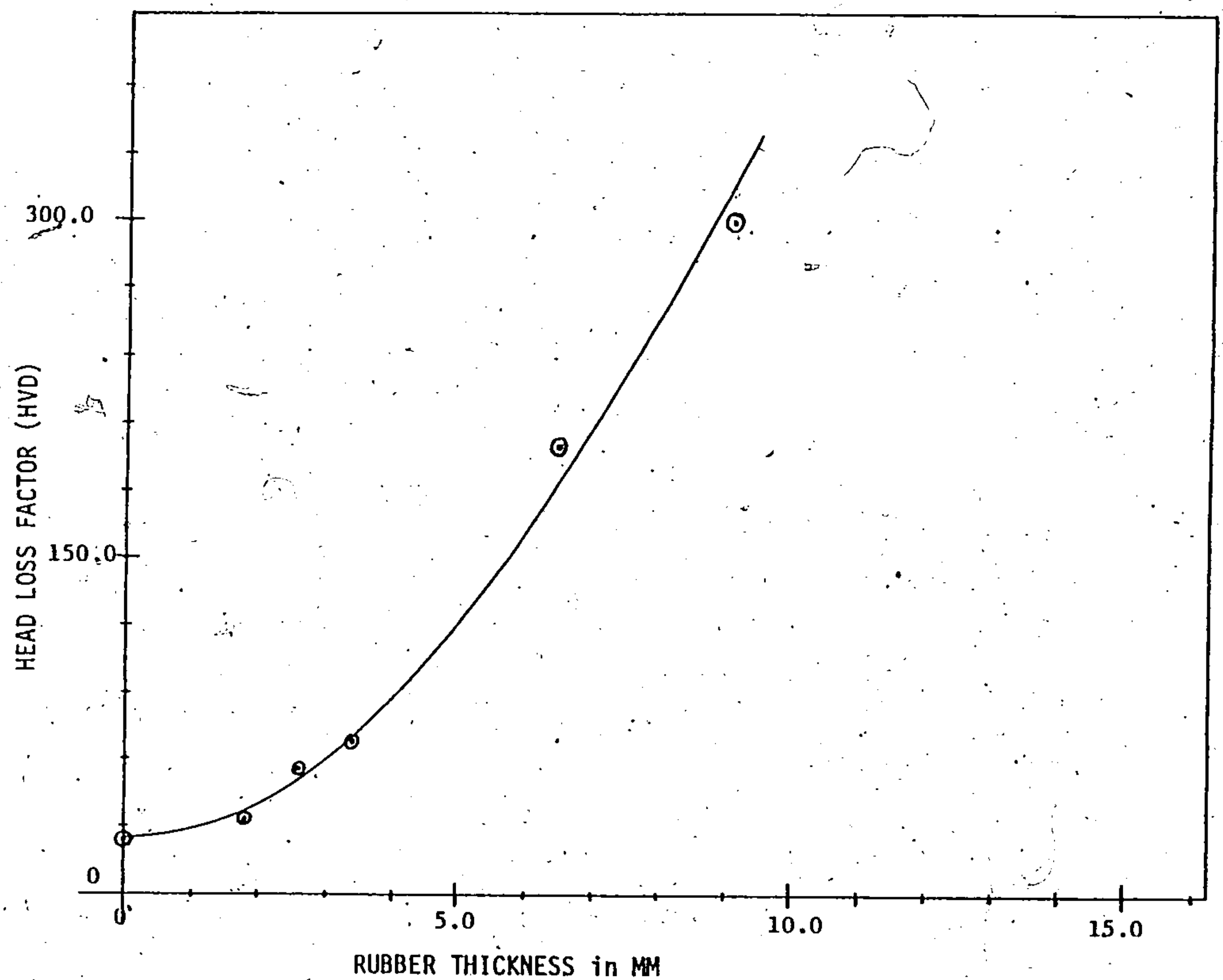


FIG. 43 VARIATION OF DELIVERY VALVE HEAD LOSS FACTOR(HVD) WITH RUBBER THICKNESS

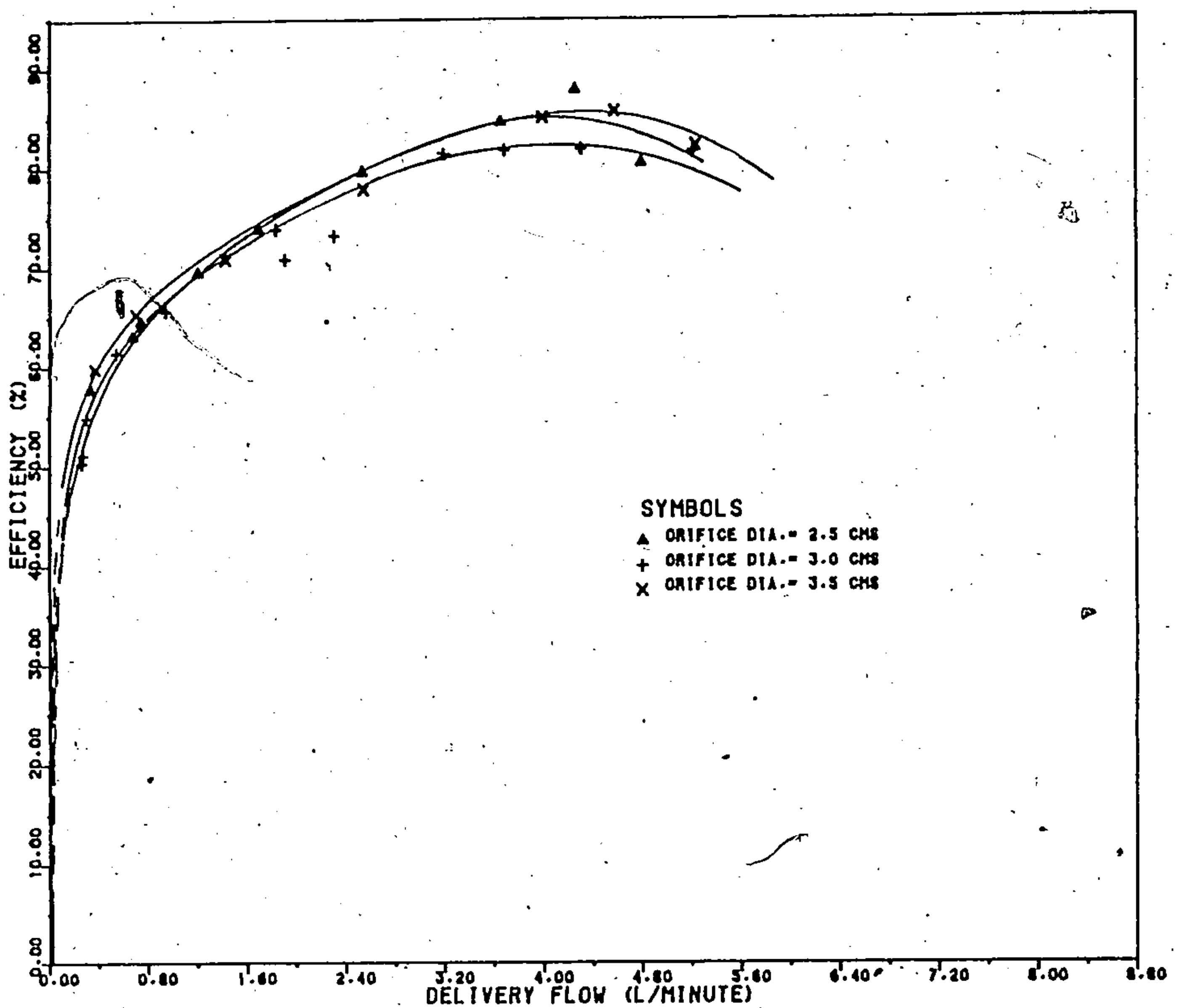


FIG. 4.4 ITDG (IMPULSE) HYDRAM: EFFECT OF WASTE VALVE ORIFICE ON PUMP EFFICIENCY. VALVE DIAMETER = 4.6 CMS

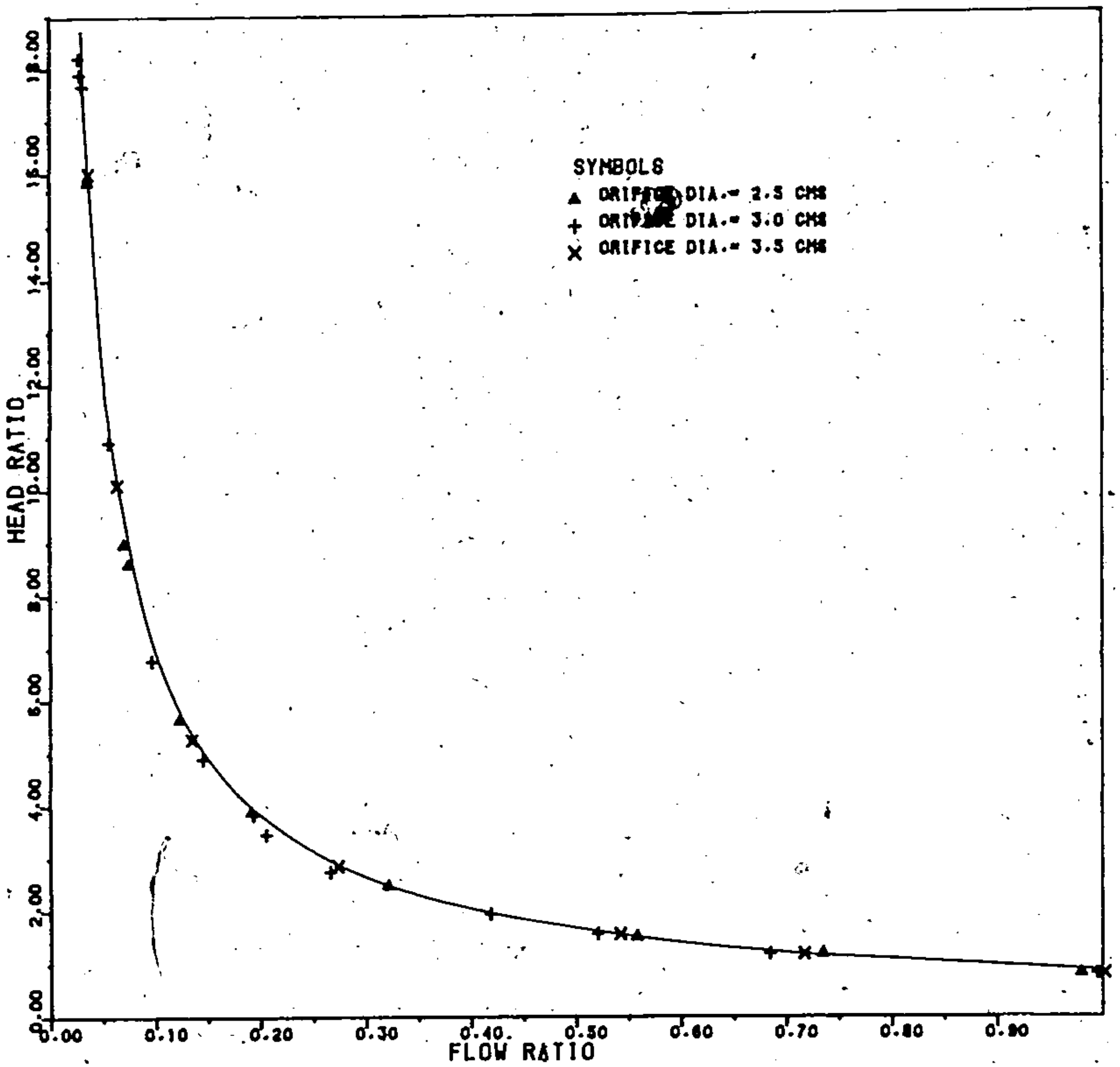


FIG.4 5 ITDG (IMPULSE) HYDRAM EFFECT OF WASTE VALVE ORIFICE ON PUMP PERFORMANCE. VALVE DIAMETER =4.6 CMS

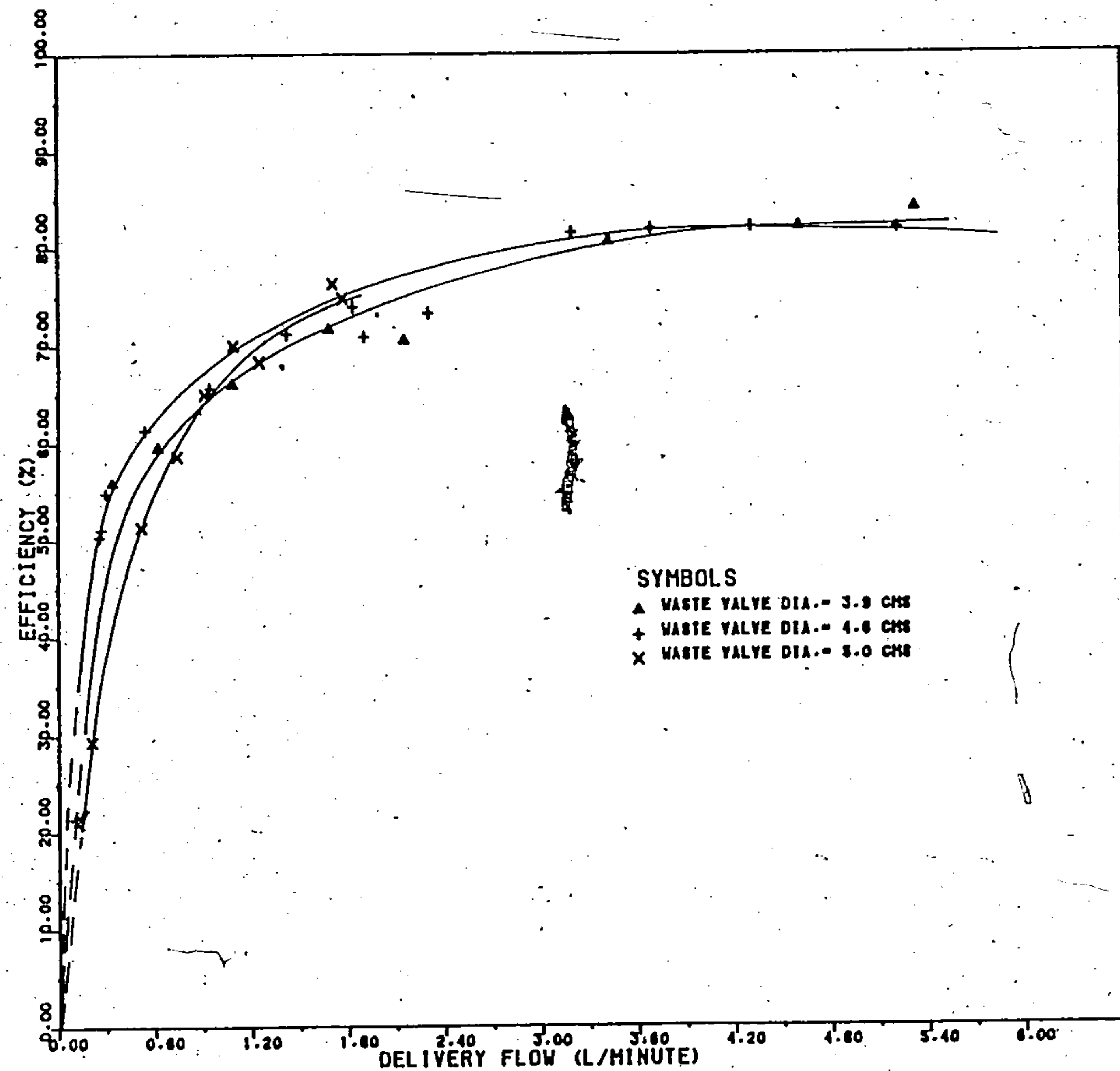


FIG. 46 ITDG (IMPULSE) HYDRAM: EFFECT OF WASTE VALVE DIAMETER ON PUMP EFFICIENCY. ORIFICE DIAMETER =3.0 CMS

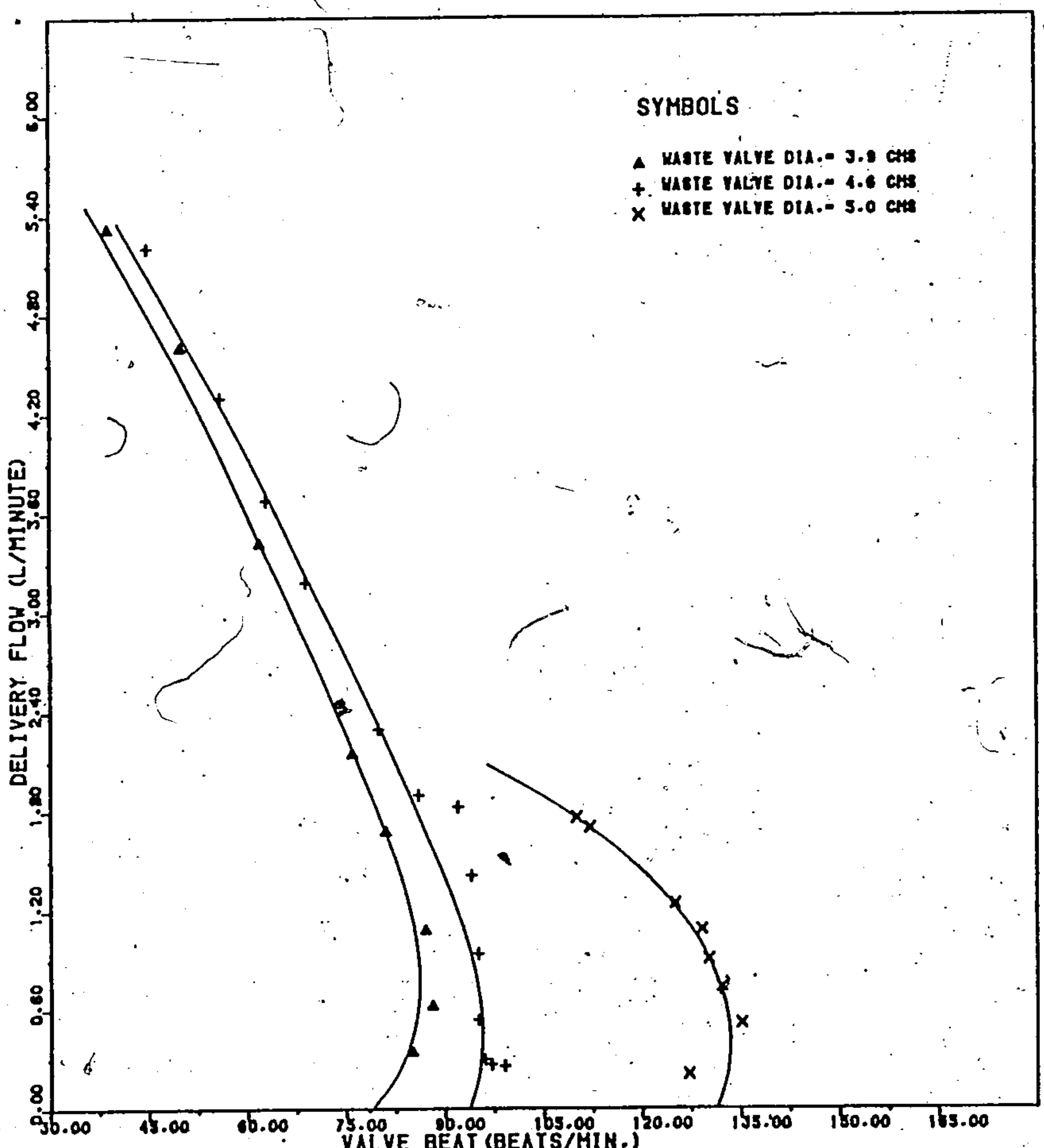


FIG. 47 ITDG (IMPULSE) HYDRAM: EFFECT OF WASTE VALVE DIAMETER

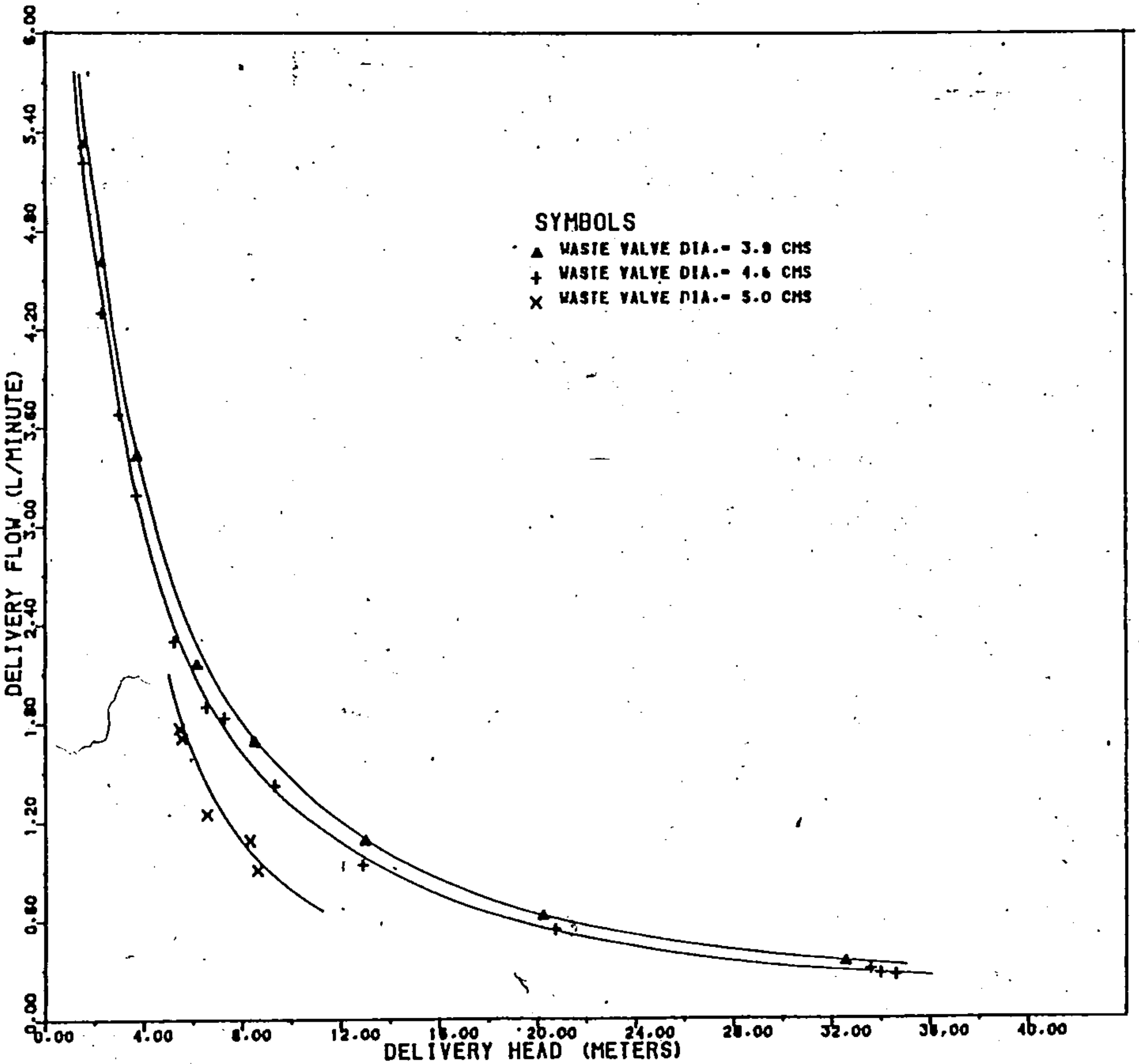


FIG. 48 ITDG (IMPULSE) HYDRAM: EFFECT OF WASTE VALVE DIAMETER

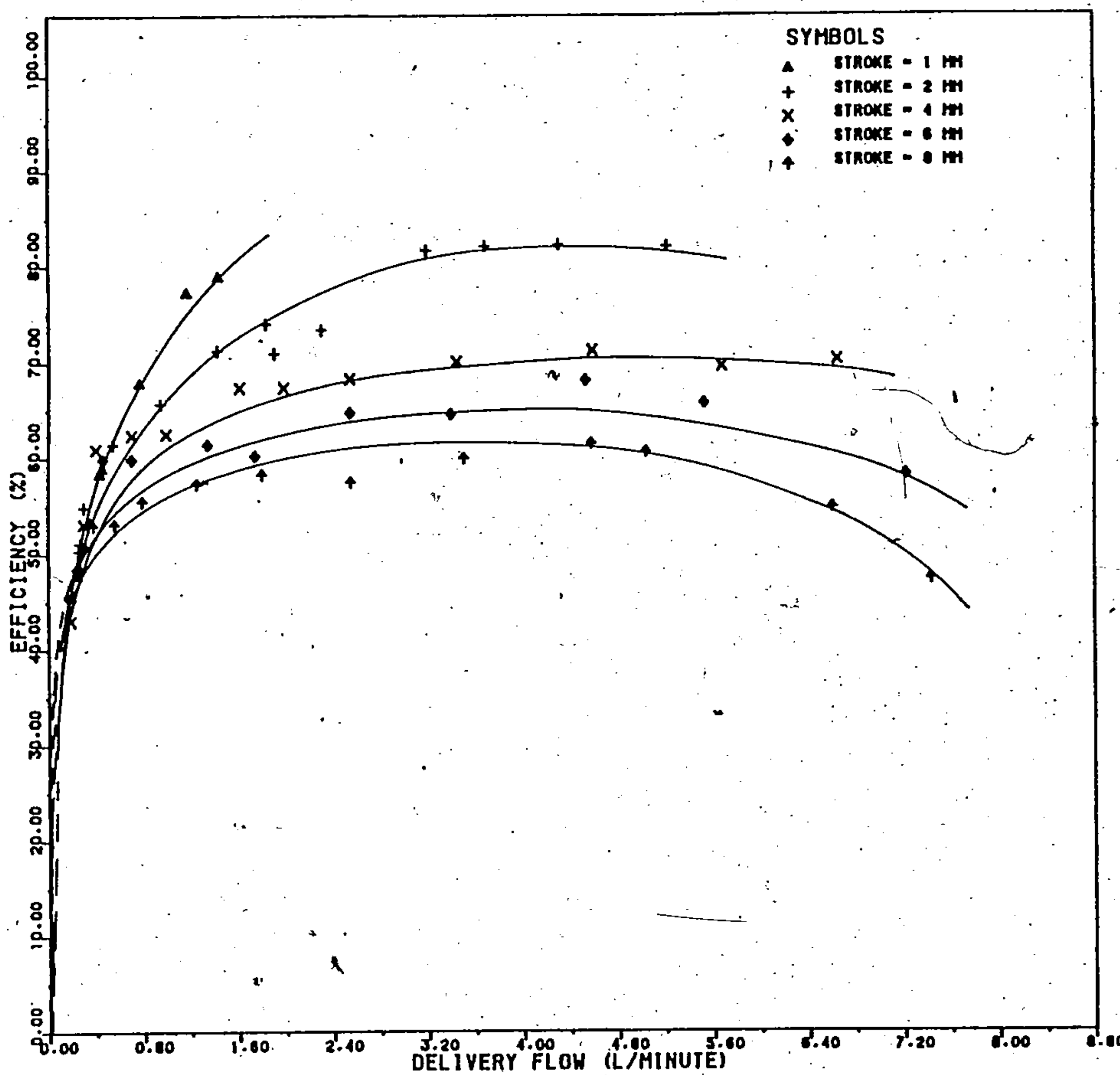


FIG. 49 ITDG (IMPULSE) HYDRAM: EFFECT OF VALVE STROKE ON PUMP EFFICIENCY

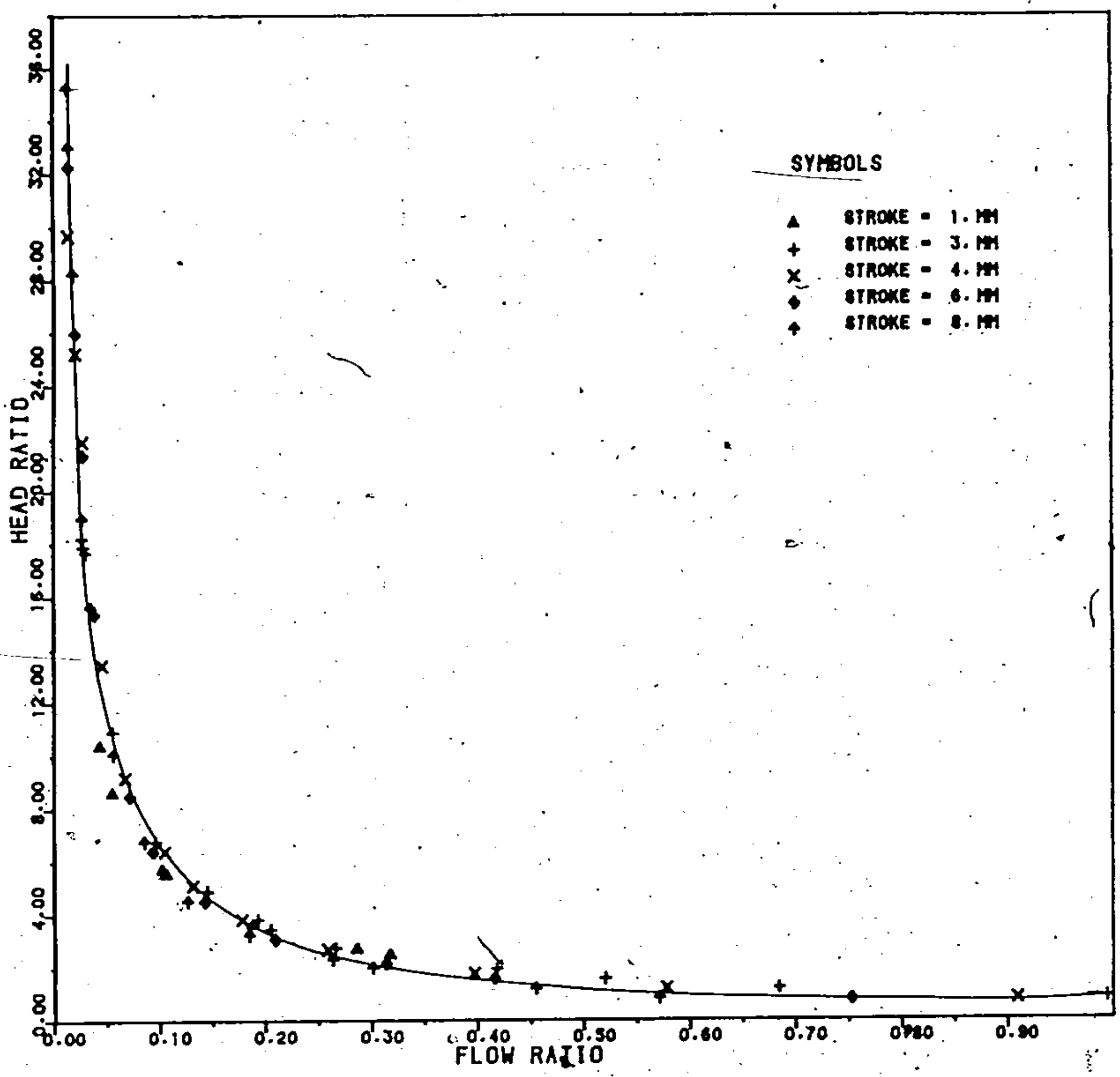


FIG. 50 ITDG (IMPULSE) HYDRAM: HEAD RATIO VS FLOW RATIO

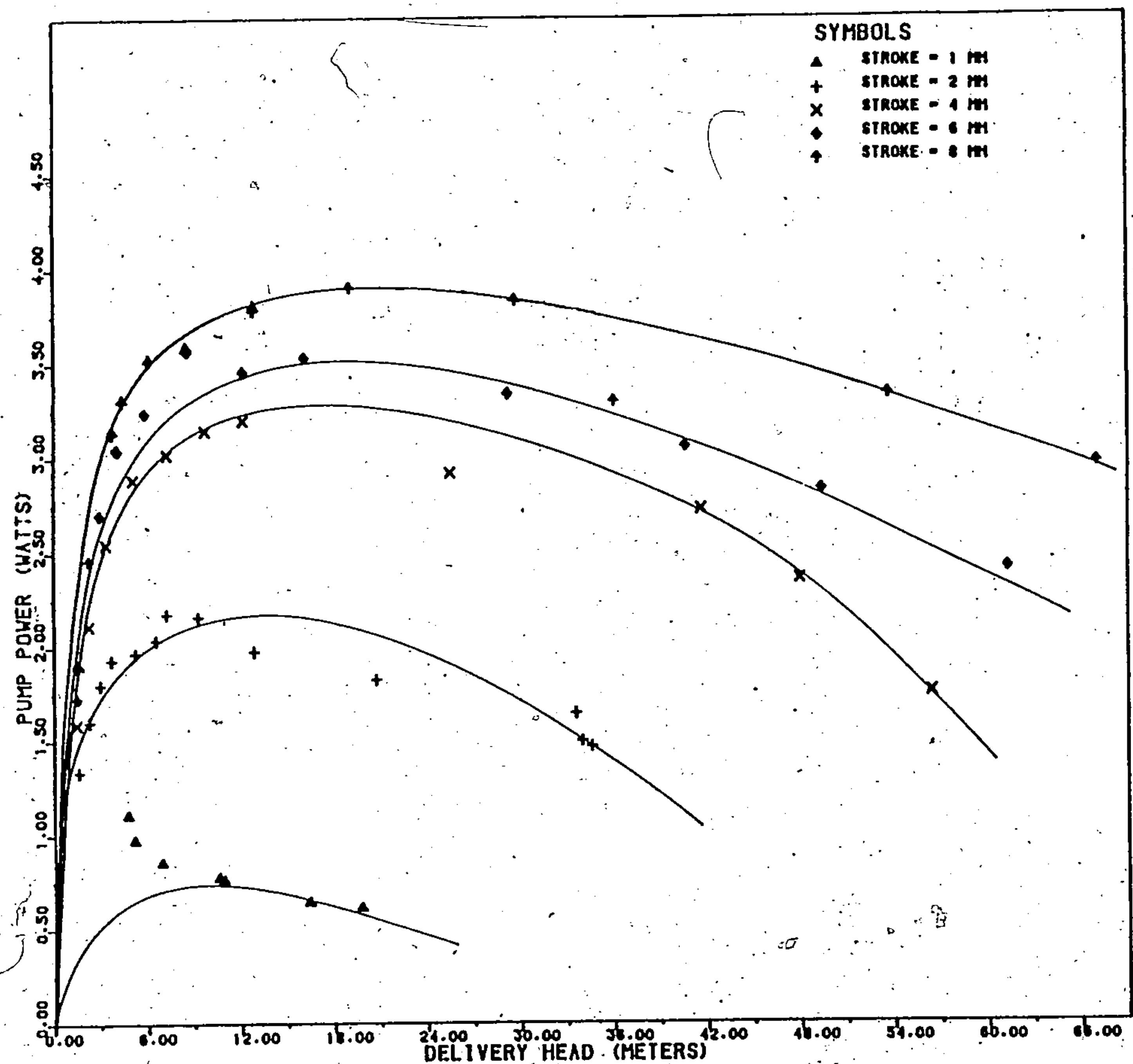


FIG.51 ITDG(IMPULSE) HYDRAM:EFFECT OF VALVE STROKE ON PUMP POWER

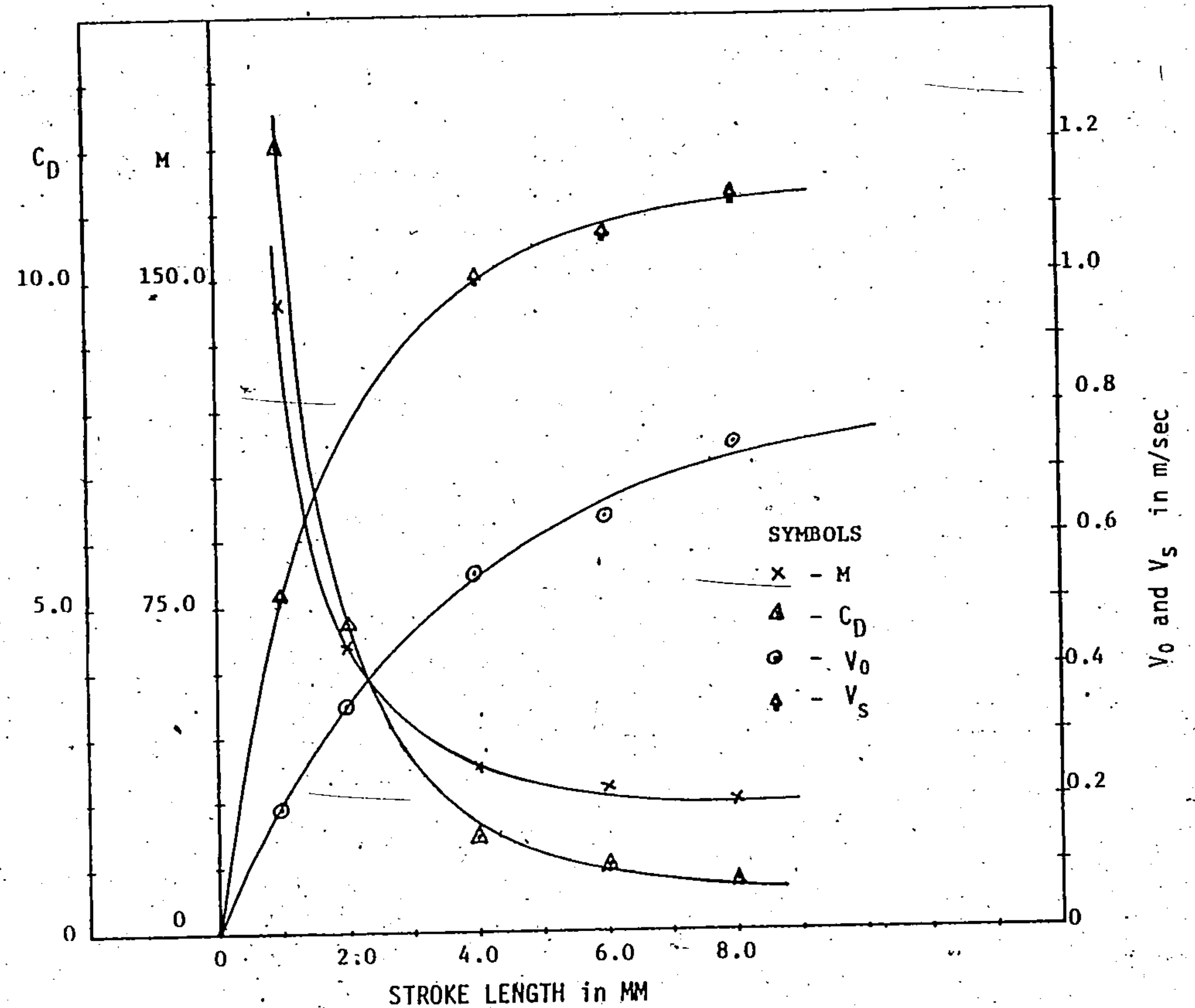


FIG. 52 VARIATION OF M, C_D, V_0 AND V_s WITH STROKE LENGTH FOR THE ITDG (WEIGHTED IMPULSE VALVE) HYDRAM

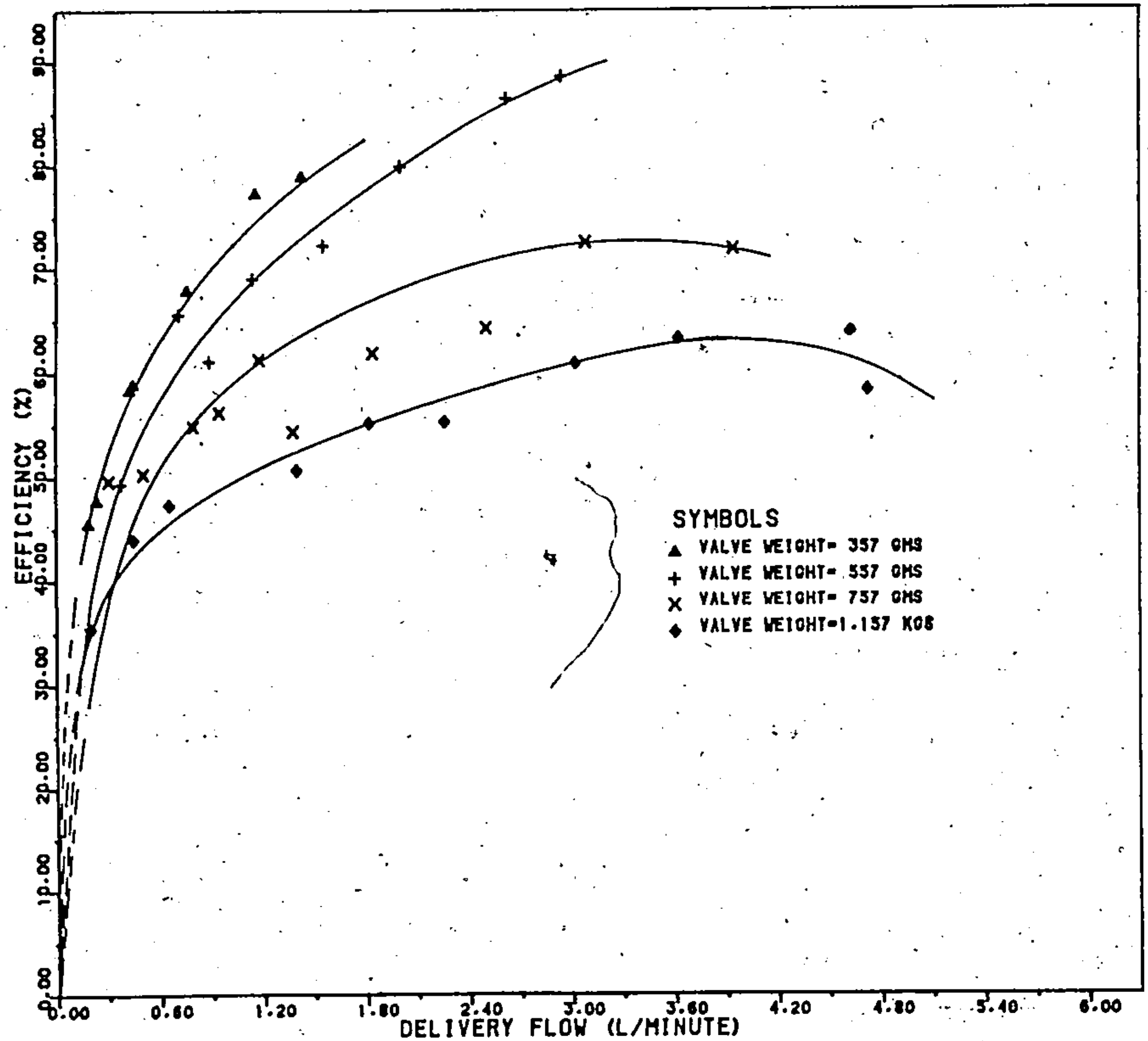


FIG. 53 ITDG (IMPULSE) HYDRAM: EFFECT OF WASTE VALVE WEIGHT ON PUMP EFFICIENCY. VALVE DIAMETER =4.6 CMS

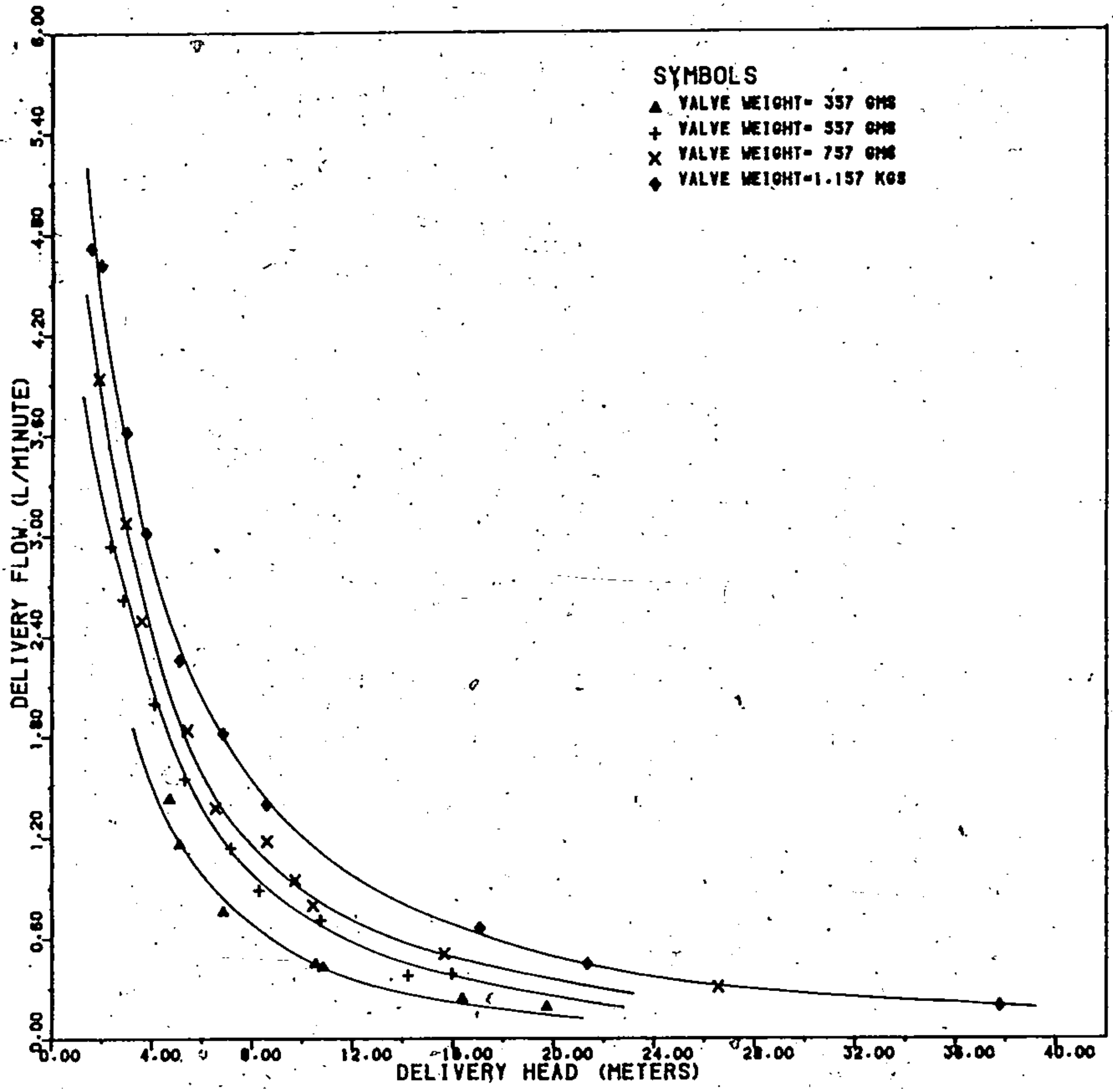


FIG. 54 ITDG (IMPULSE) HYDRAM: EFFECT OF WASTE VALVE WEIGHT ON PUMP CAPACITY. VALVE DIAMETER =4.6 CMS

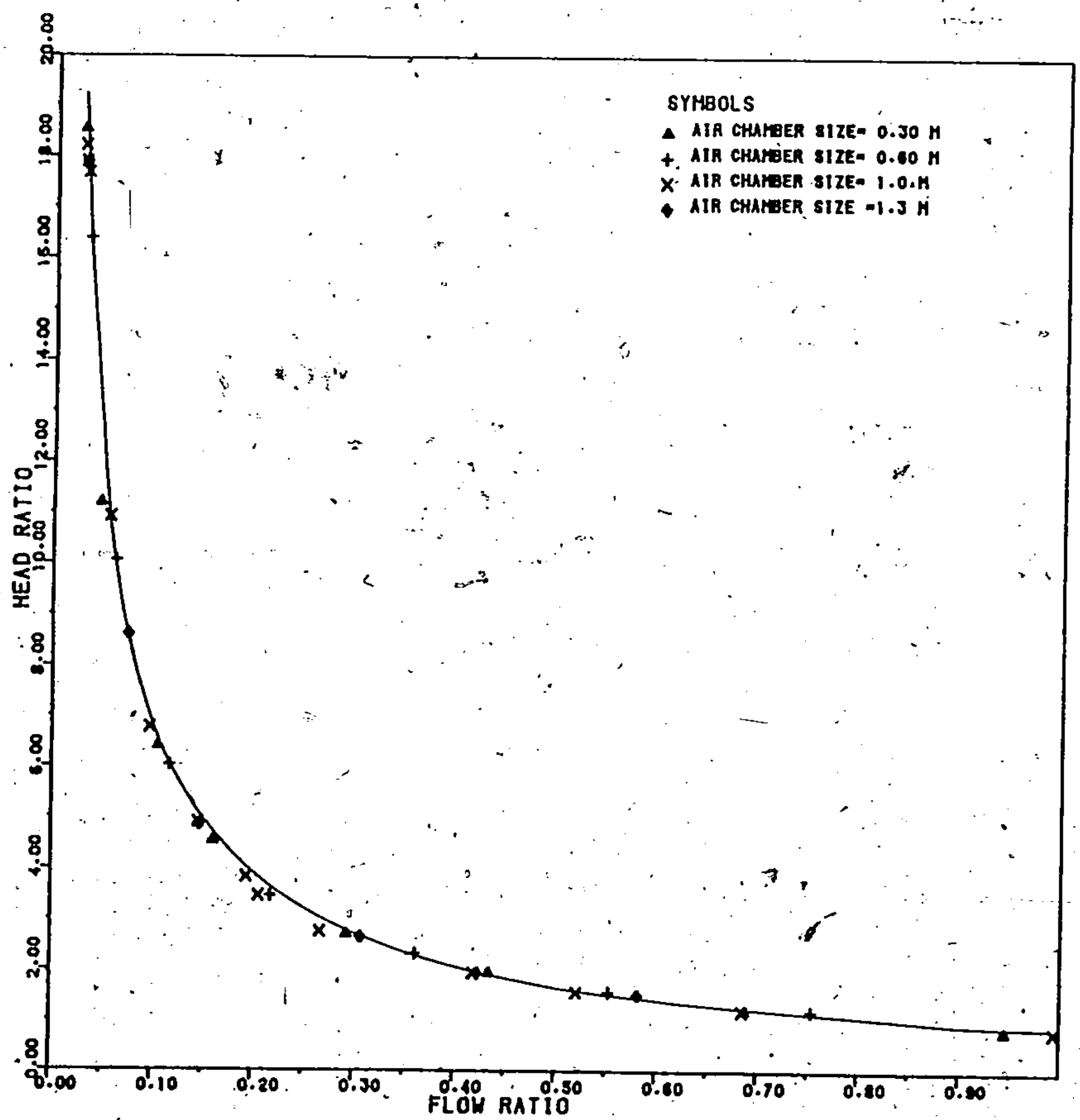


FIG.55 ITDG(IMPULSE) HYDRAM:EFFECT OF AIR CHAMBER SIZE ON PUMP PERFORMANCE.

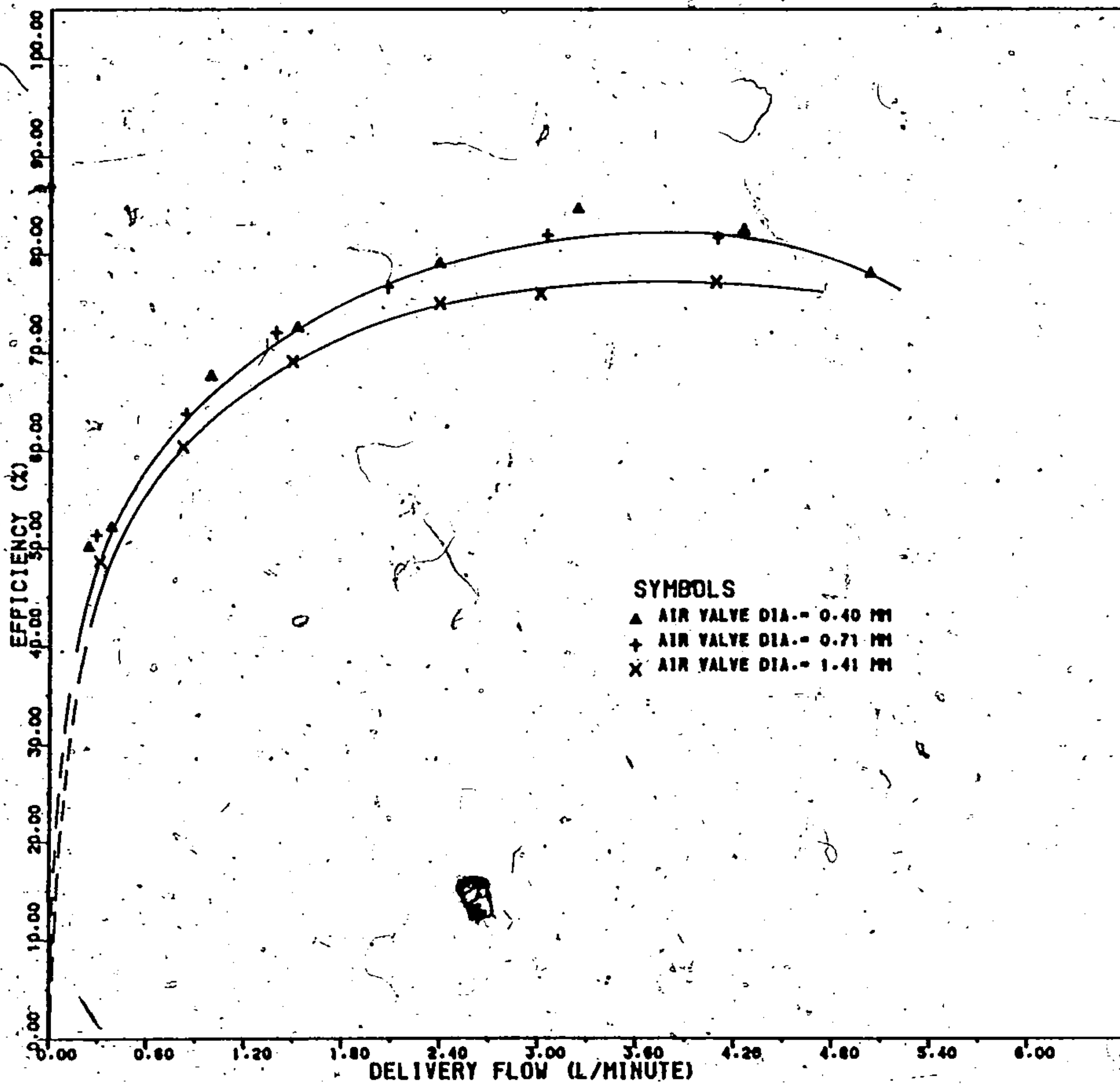


FIG. 56 ITDG (IMPULSE) HYDRAM-EFFECT OF AIR VALVE ON PUMP EFFICIENCY.

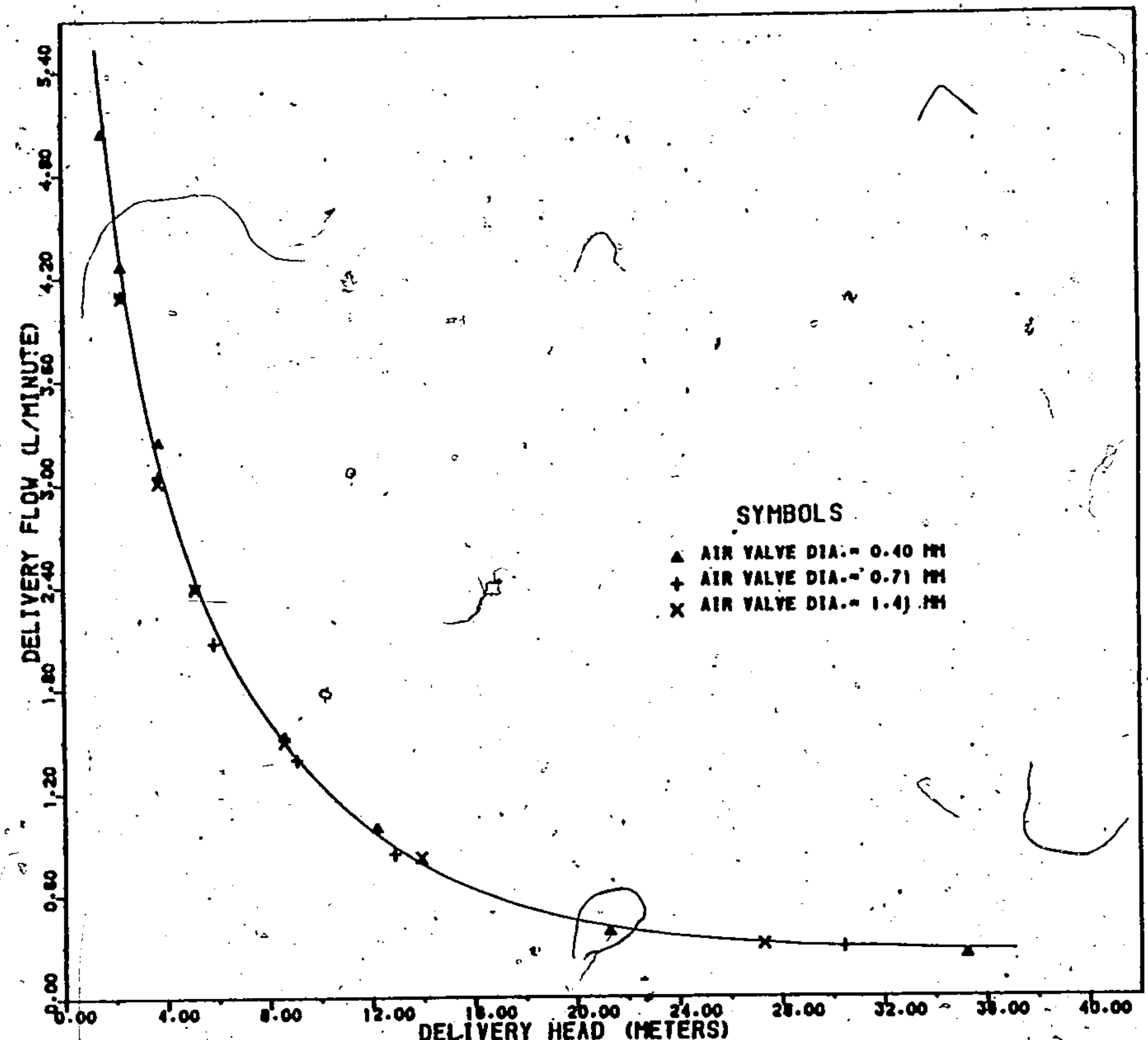


FIG.57 ITDG(IMPULSE) HYDRAM:EFFECT OF AIR VALVE ON PUMP CAPACITY.

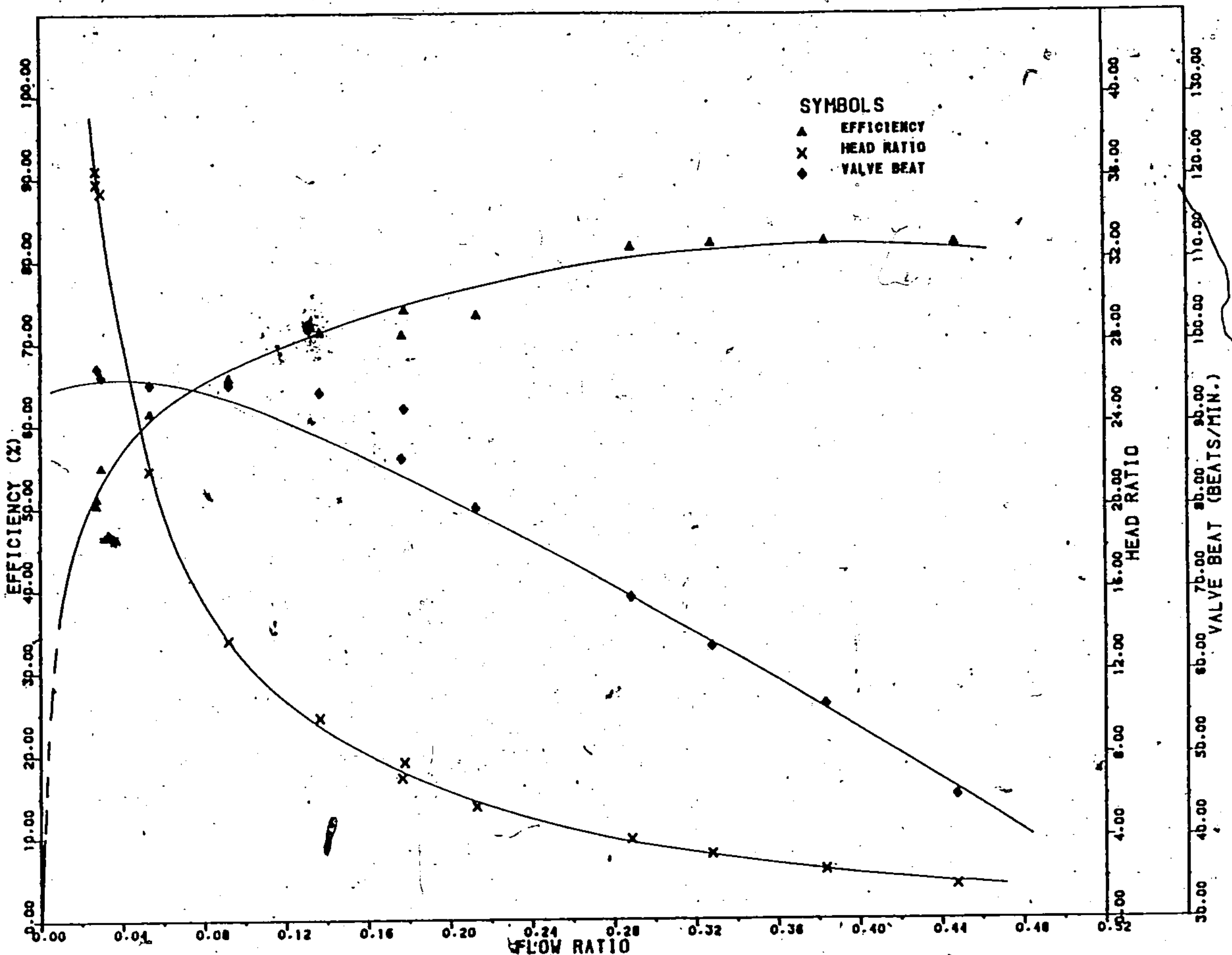


FIG. 58 ITDG (IMPULSE) HYDRAM: OPERATING CHARACTERISTICS. H = 2M

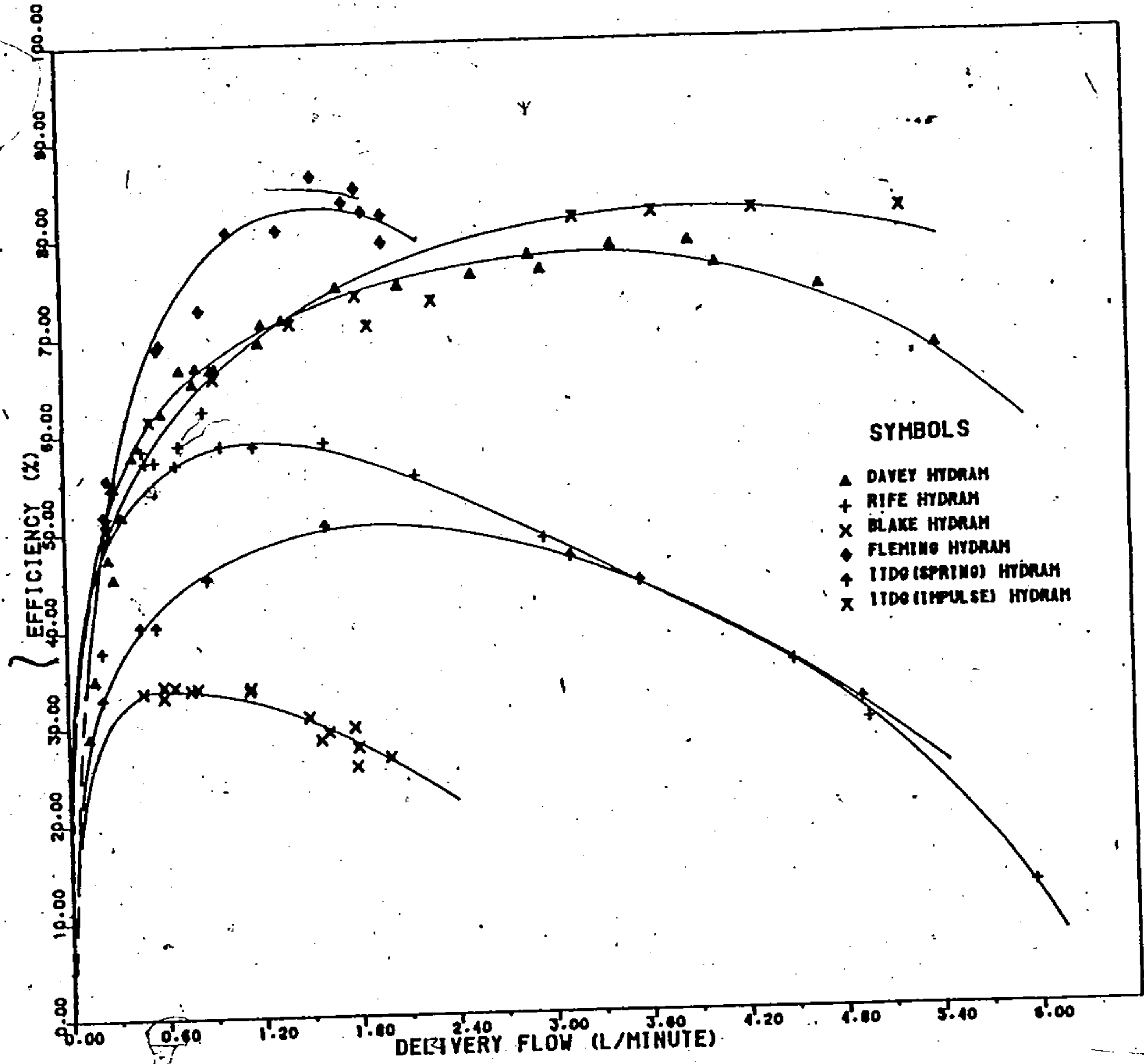


FIG.59 EFFICIENCY VS DELIVERY FLOW FOR VARIOUS HYDRAMS. H=2m

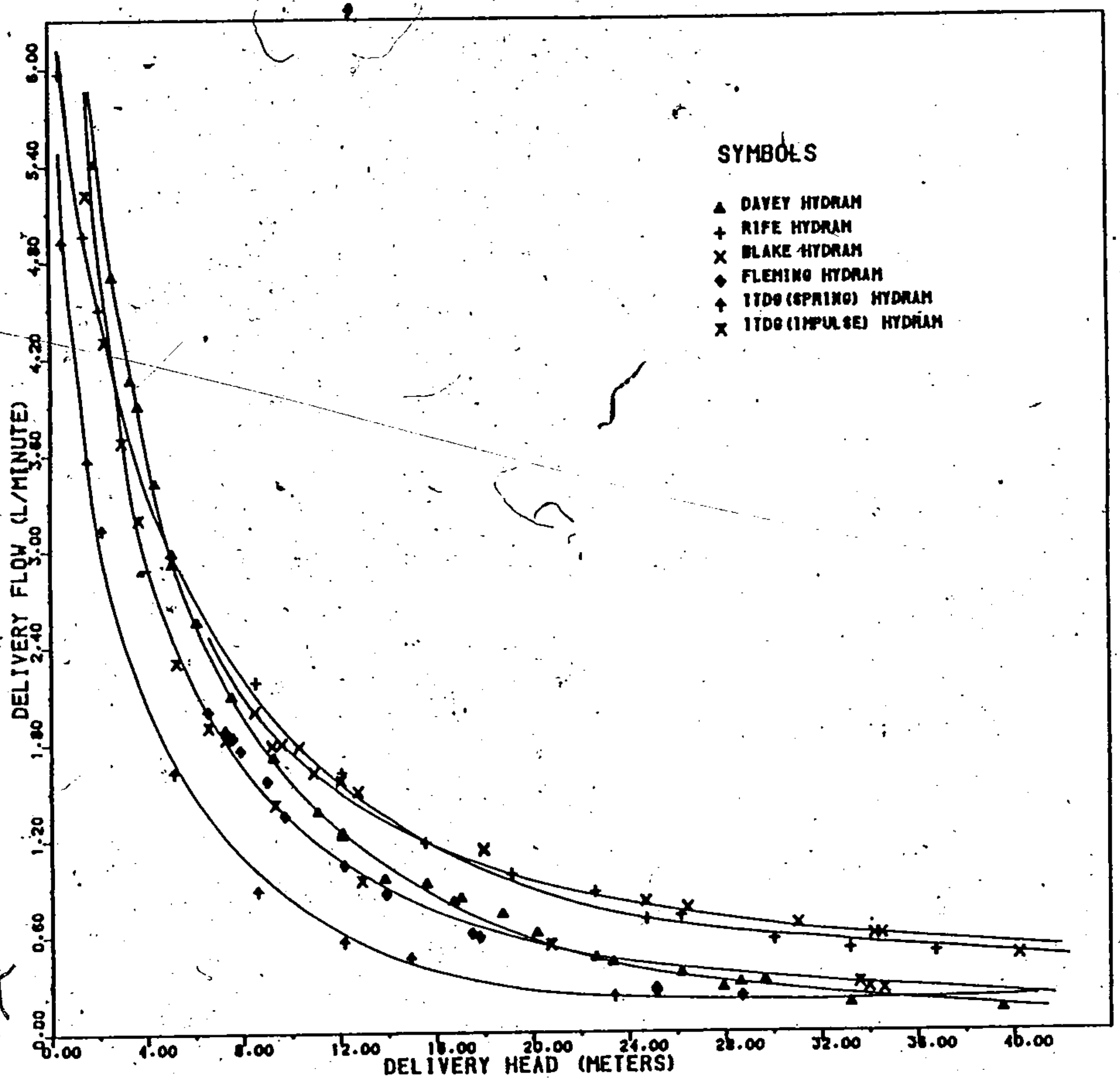


FIG. 60 DELIVERY FLOW VS DELIVERY HEAD FOR VARIOUS HYDRAMS. H=2 m

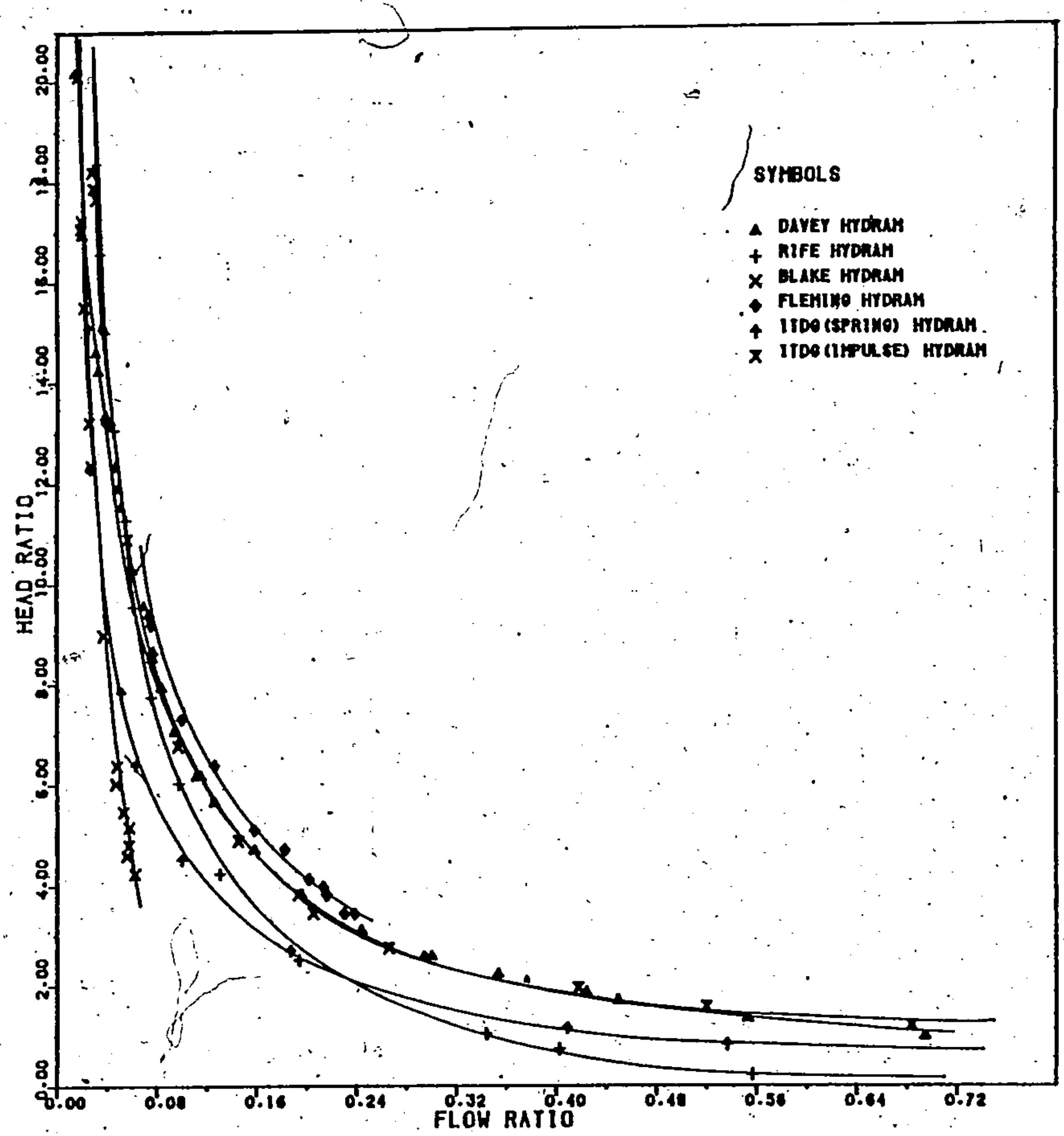


FIG. 61 HEAD RATIO VS FLOW RATIO FOR VARIOUS HYDRAMS, H = 2 m

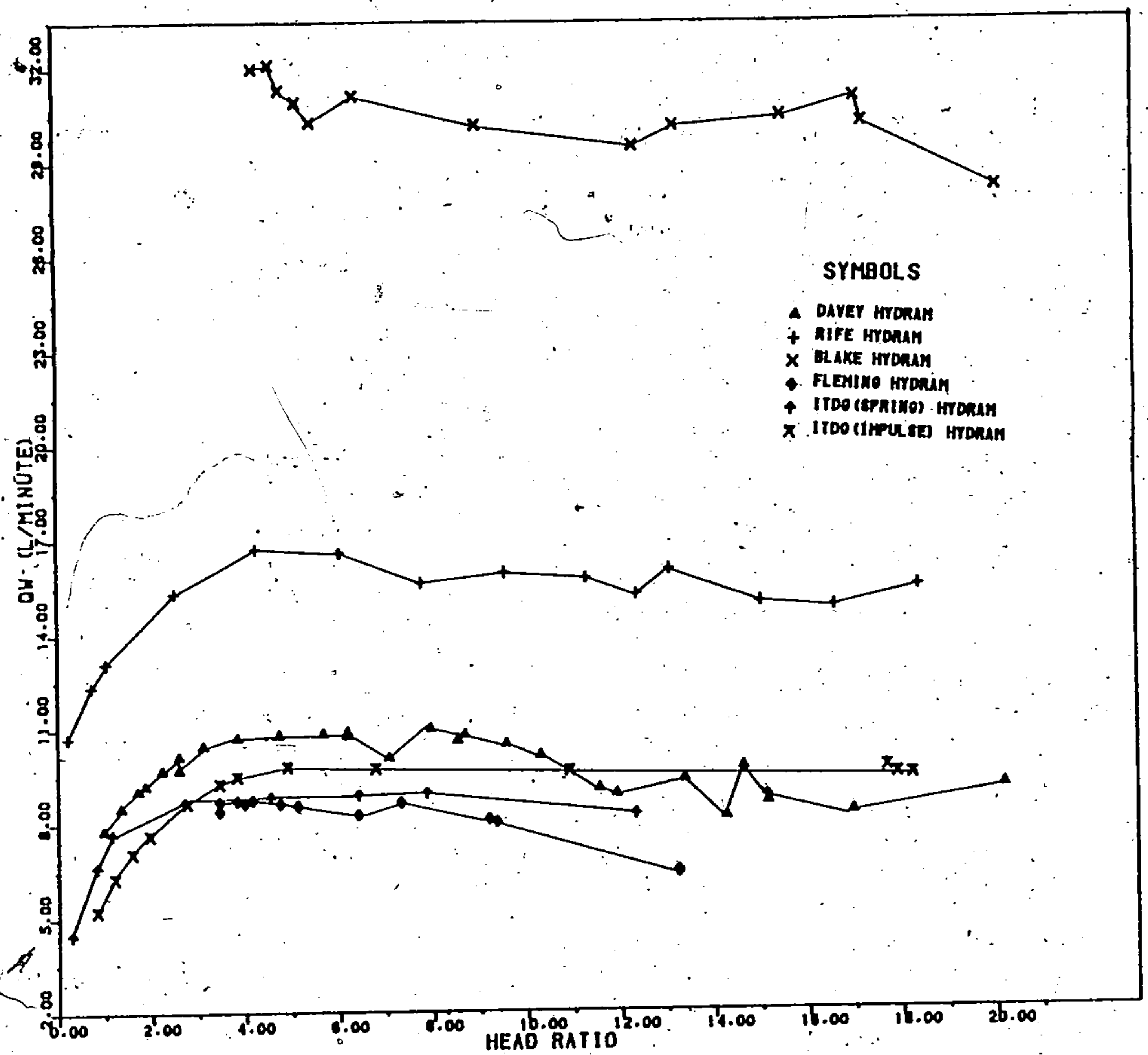


FIG. 62 VARIATION OF QW WITH HEAD RATIO FOR VARIOUS HYDRAMS. H=2m

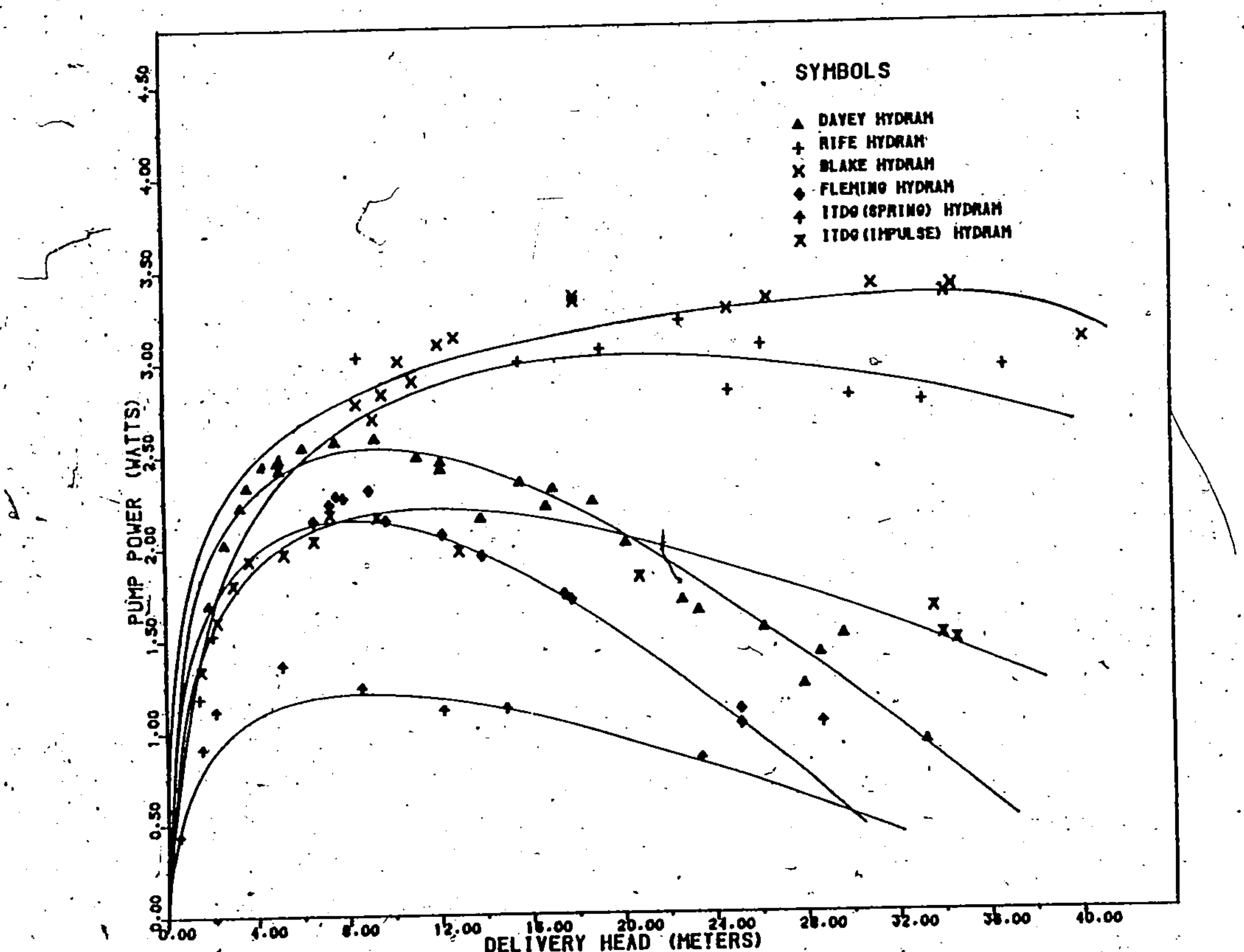


FIG. 63 PUMP POWER VS DELIVERY HEAD FOR VARIOUS HYDRAMS. H=2m

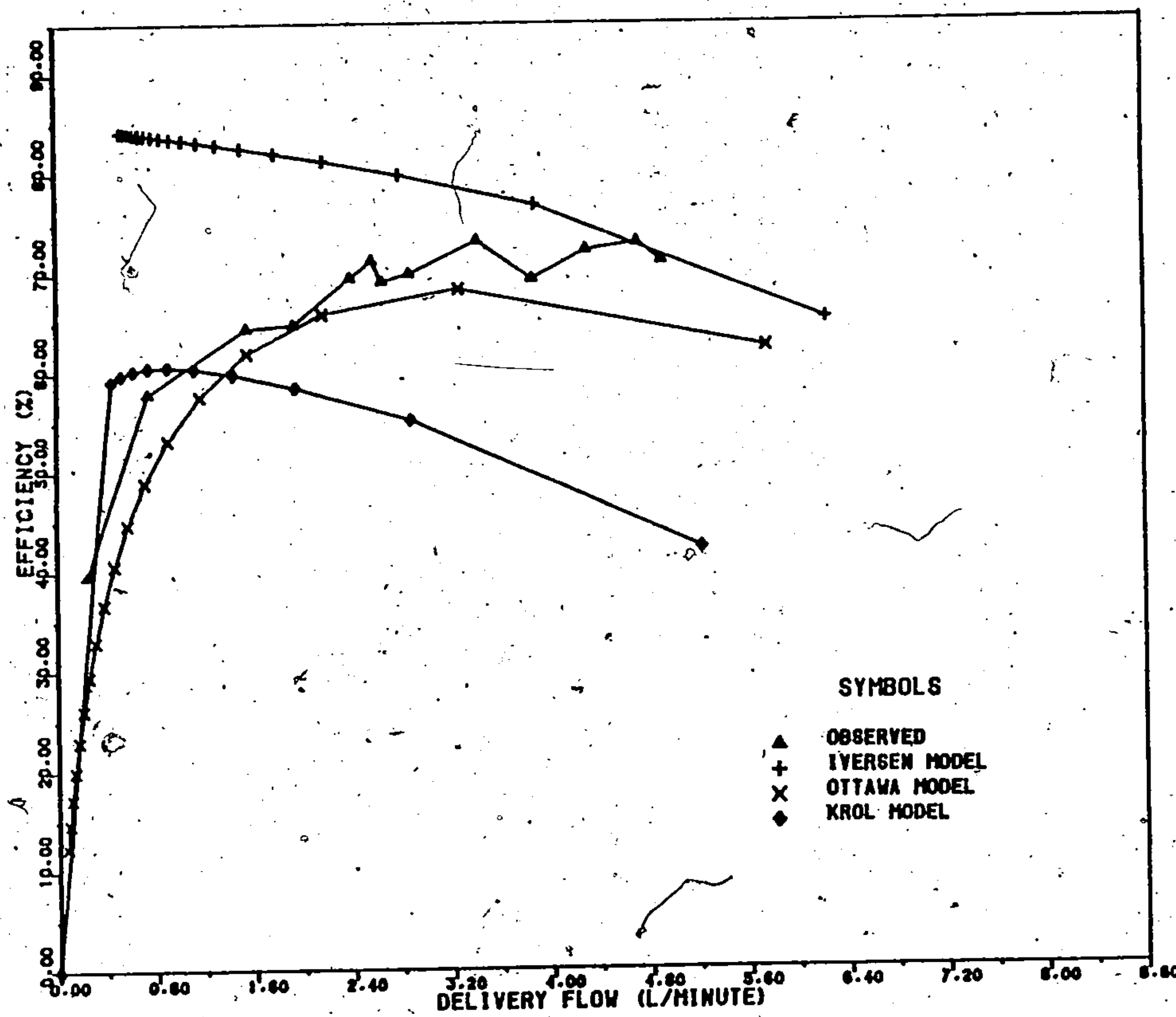


FIG. 64 COMPARISON OF OBSERVED AND COMPUTED RESULTS FOR PUMP EFFICIENCY

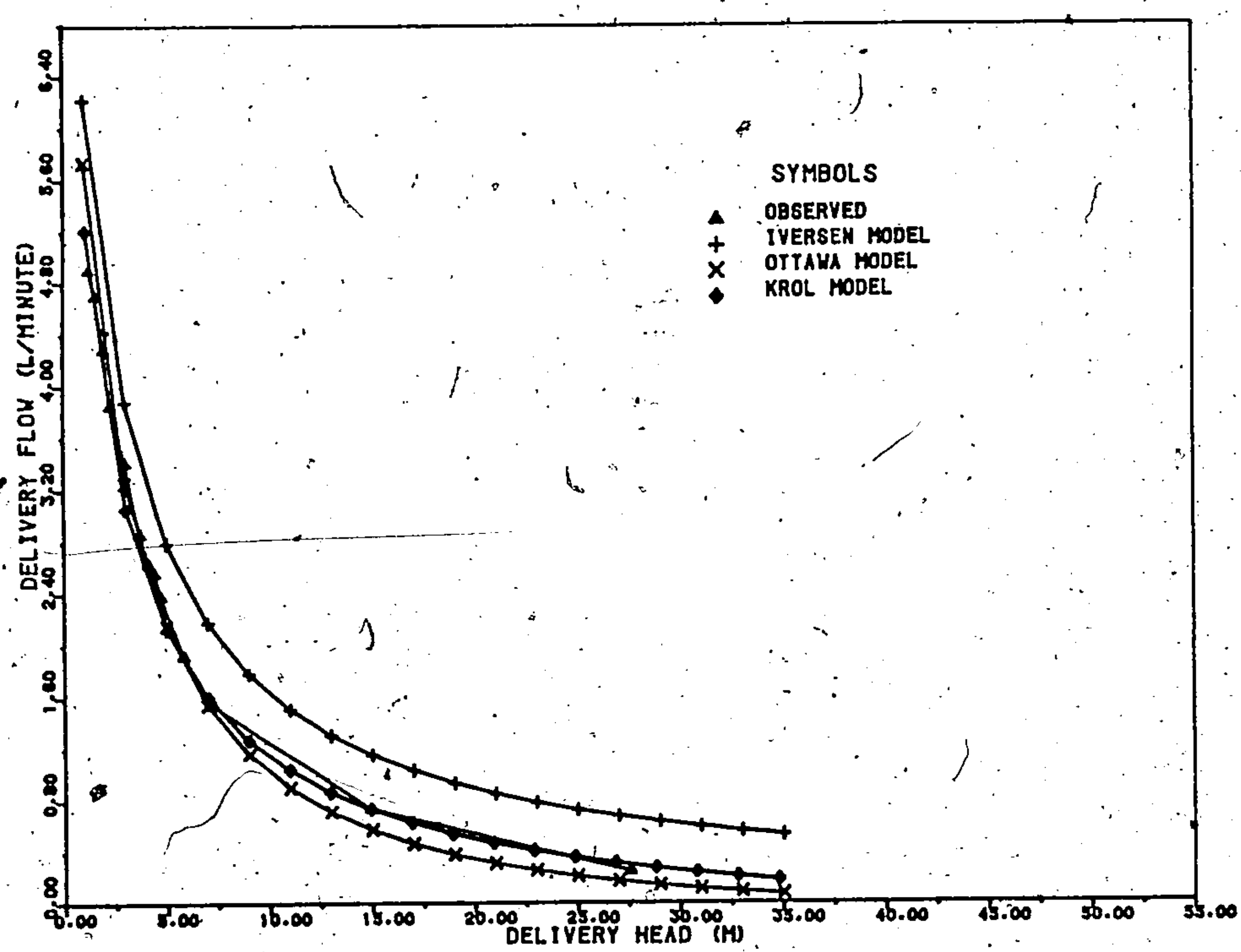


FIG. 65 COMPARISON OF OBSERVED AND COMPUTED RESULTS FOR DELIVERY FLOW

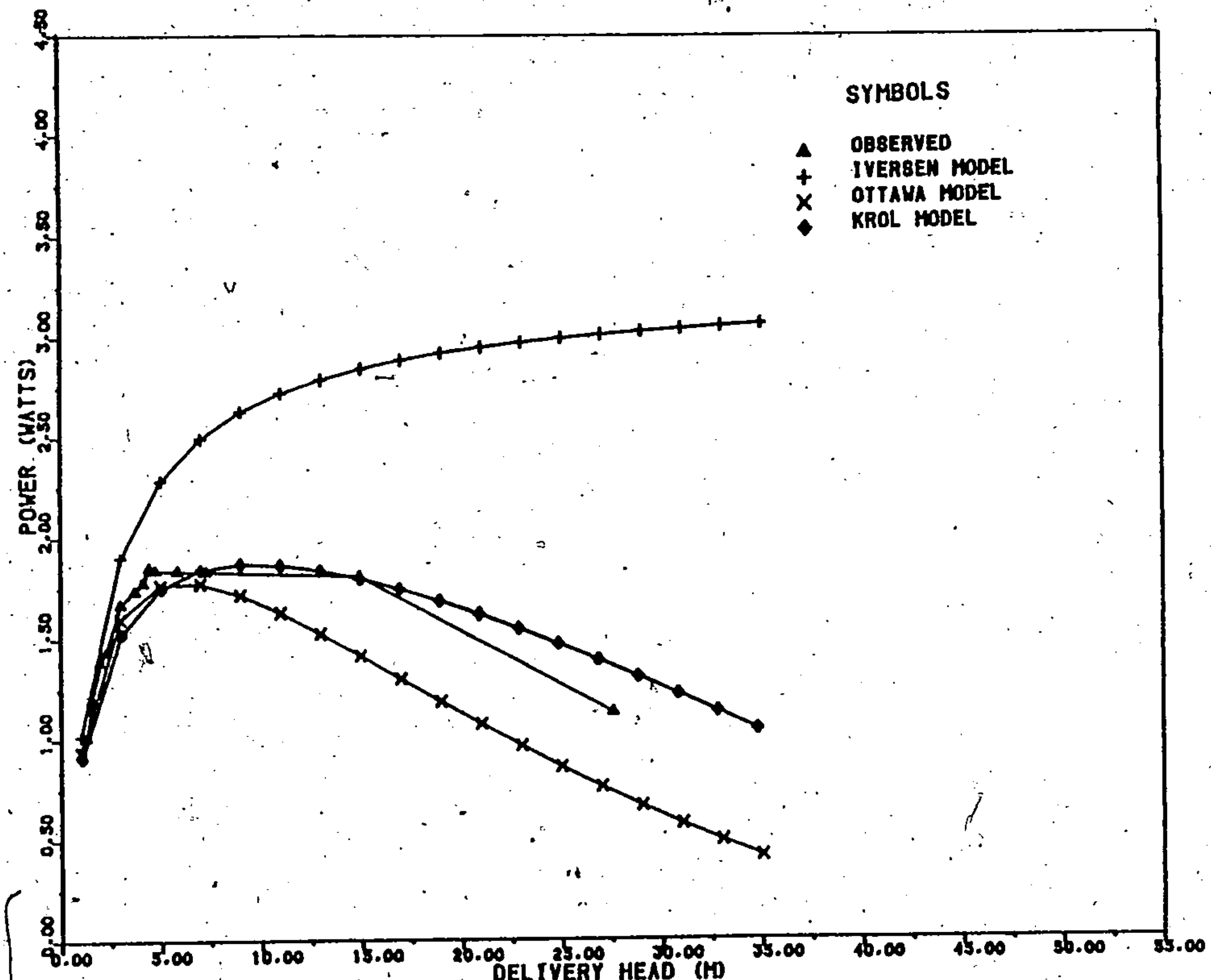


FIG. 66 COMPARISON OF OBSERVED AND COMPUTED RESULTS FOR PUMP POWER

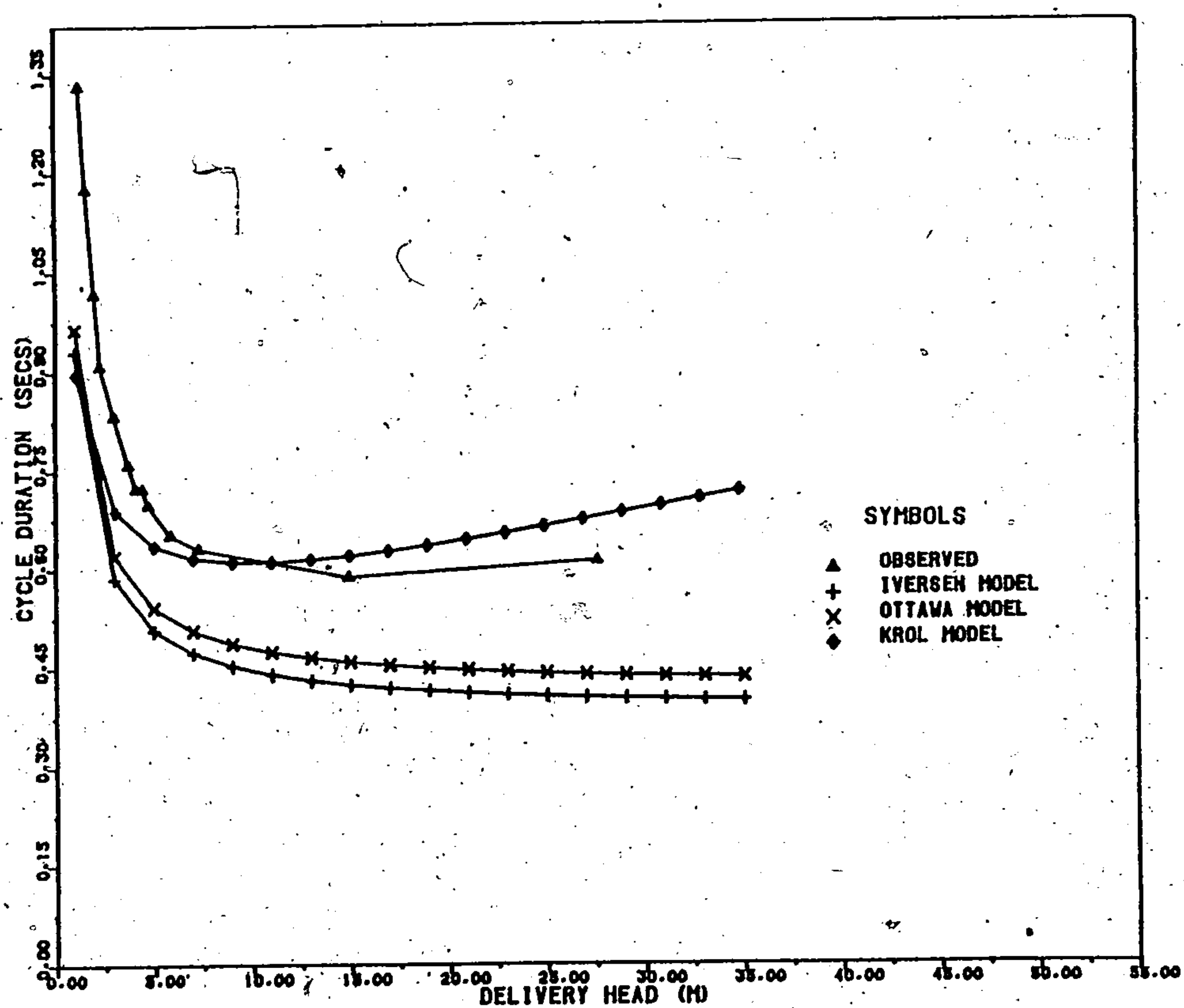


FIG. 67 COMPARISON OF OBSERVED AND COMPUTED RESULTS FOR CYCLE DURATION

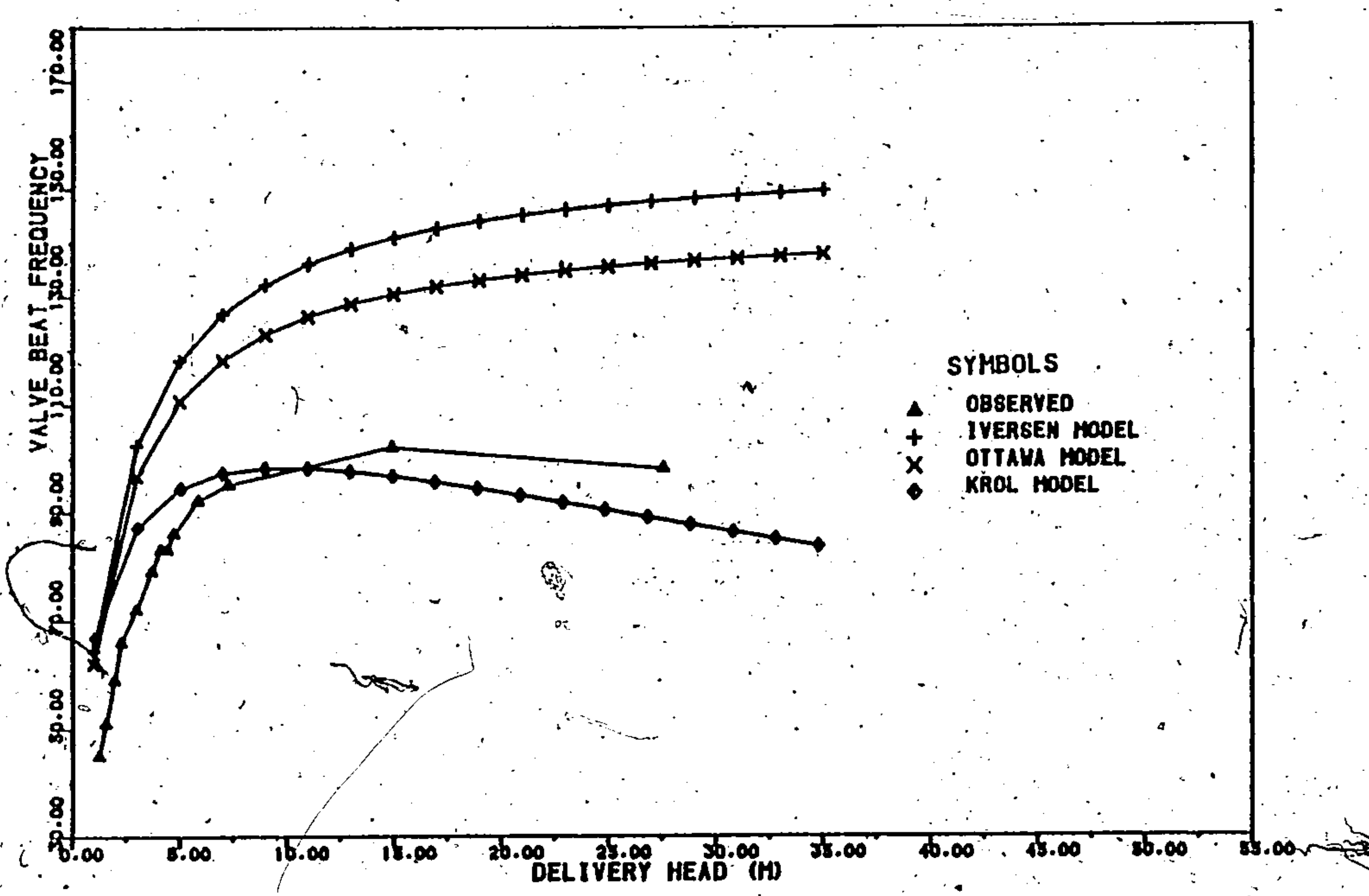


FIG. 68 COMPARISON OF OBSERVED AND COMPUTED RESULTS FOR VALVE BEAT FREQUENCY

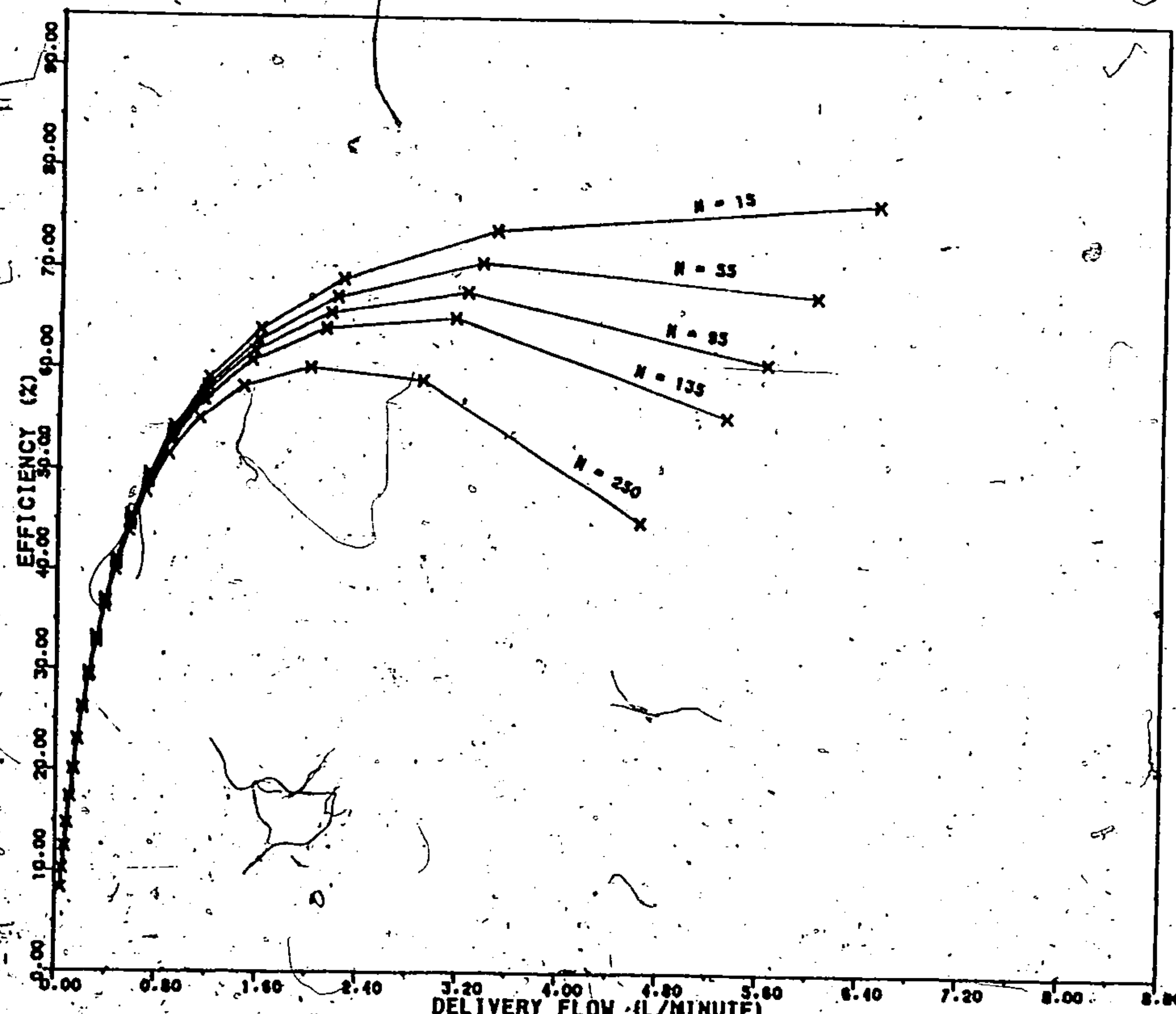


FIG. 69 EFFECT OF FRICTION HEAD LOSS OF THE DELIVERY

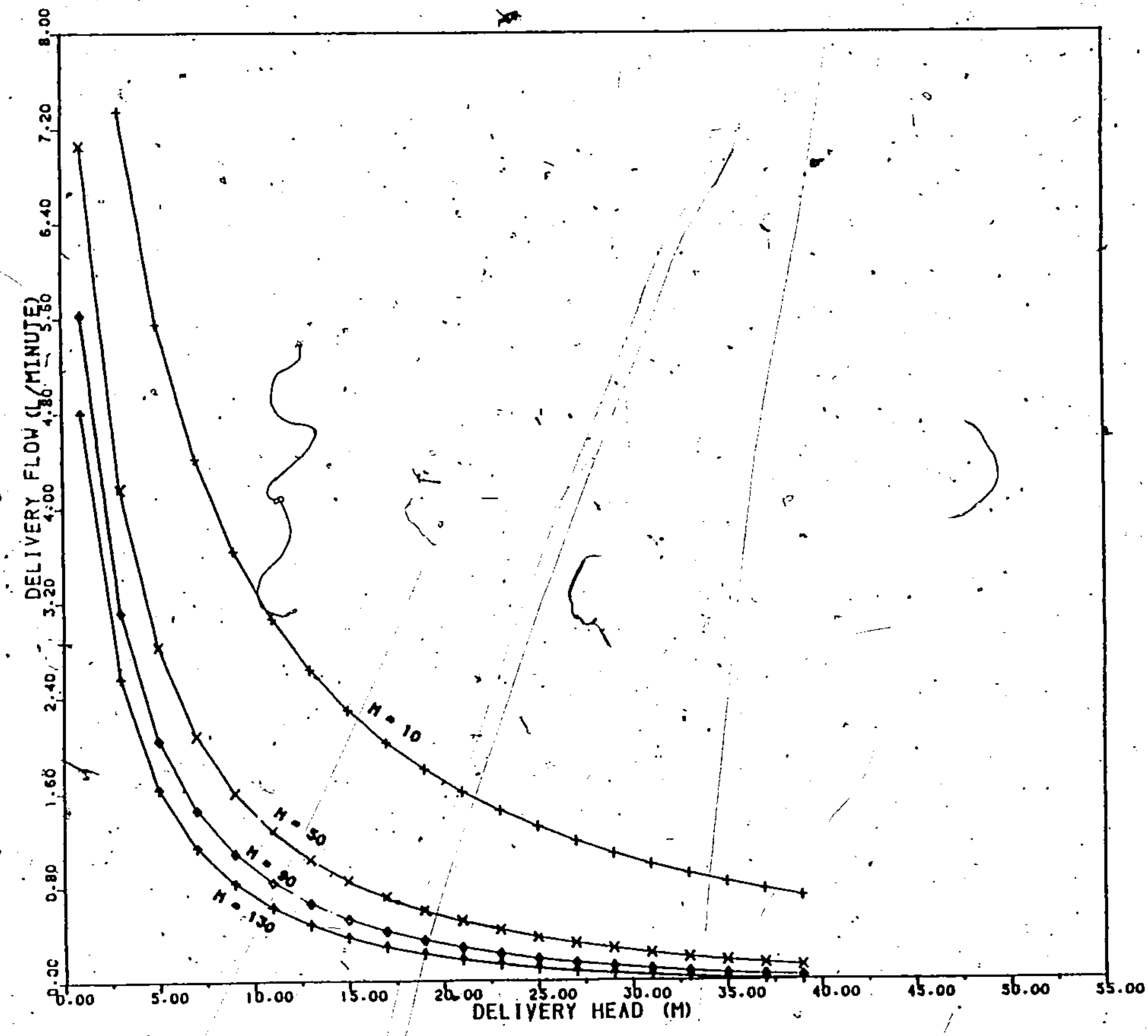


FIG. 70 EFFECT OF THE FRICTION HEAD LOSS OF THE WASTE VALVE ON PUMP CAPACITY

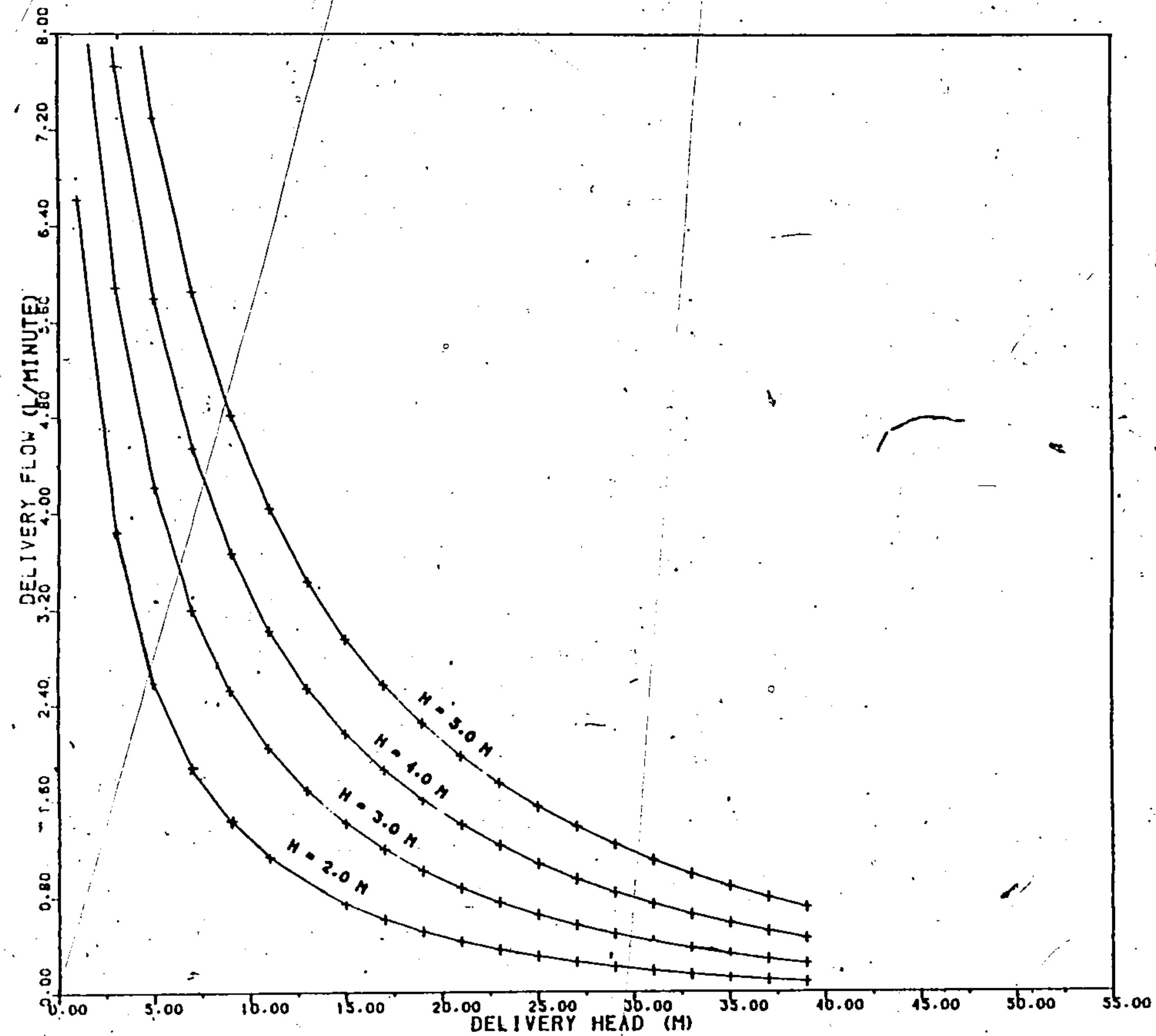


FIG.71 EFFECT OF SUPPLY HEAD ON HYDRAM CAPACITY

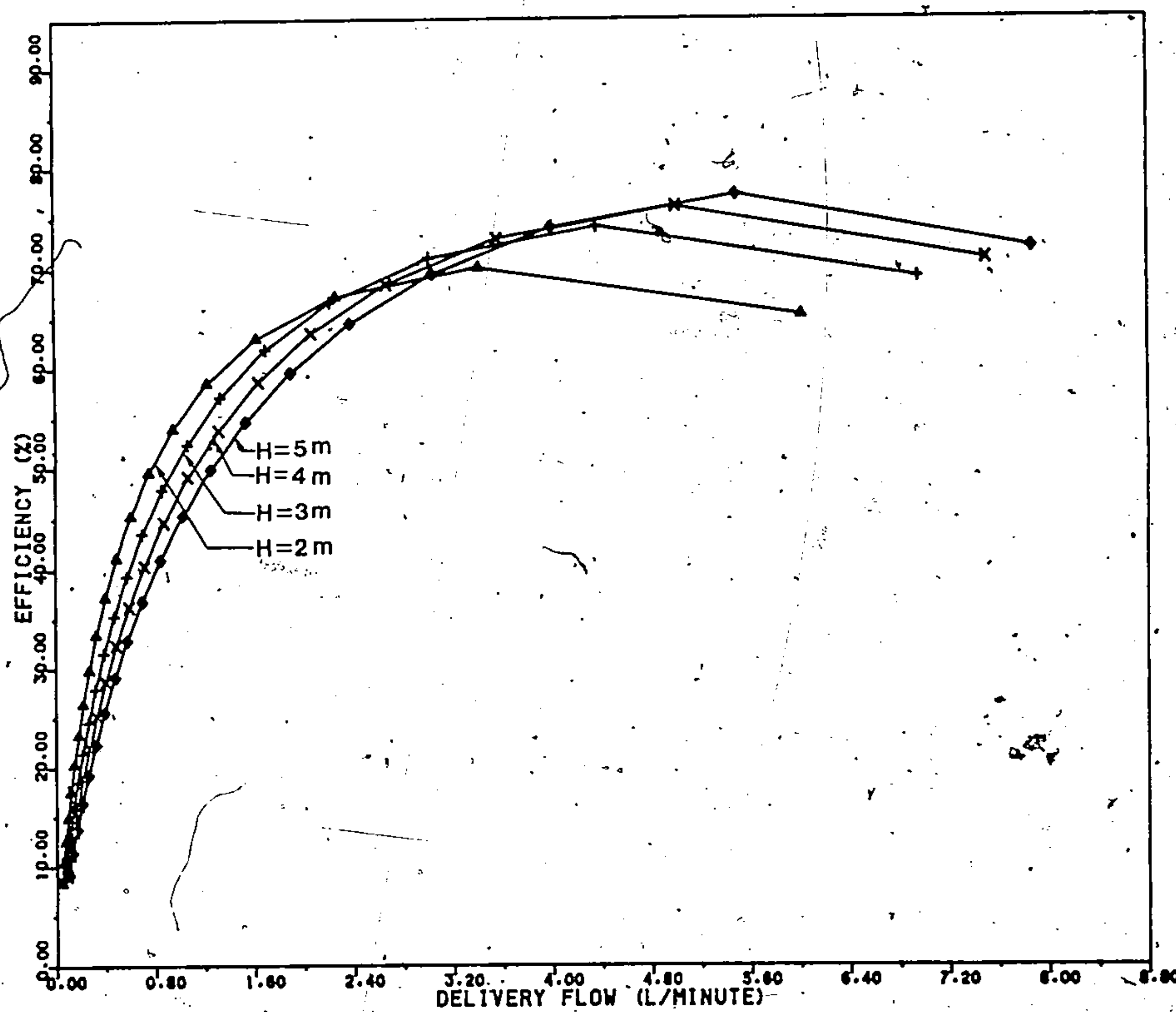


FIG. 72 EFFECT OF SUPPLY HEAD ON HYDRAM EFFICIENCY

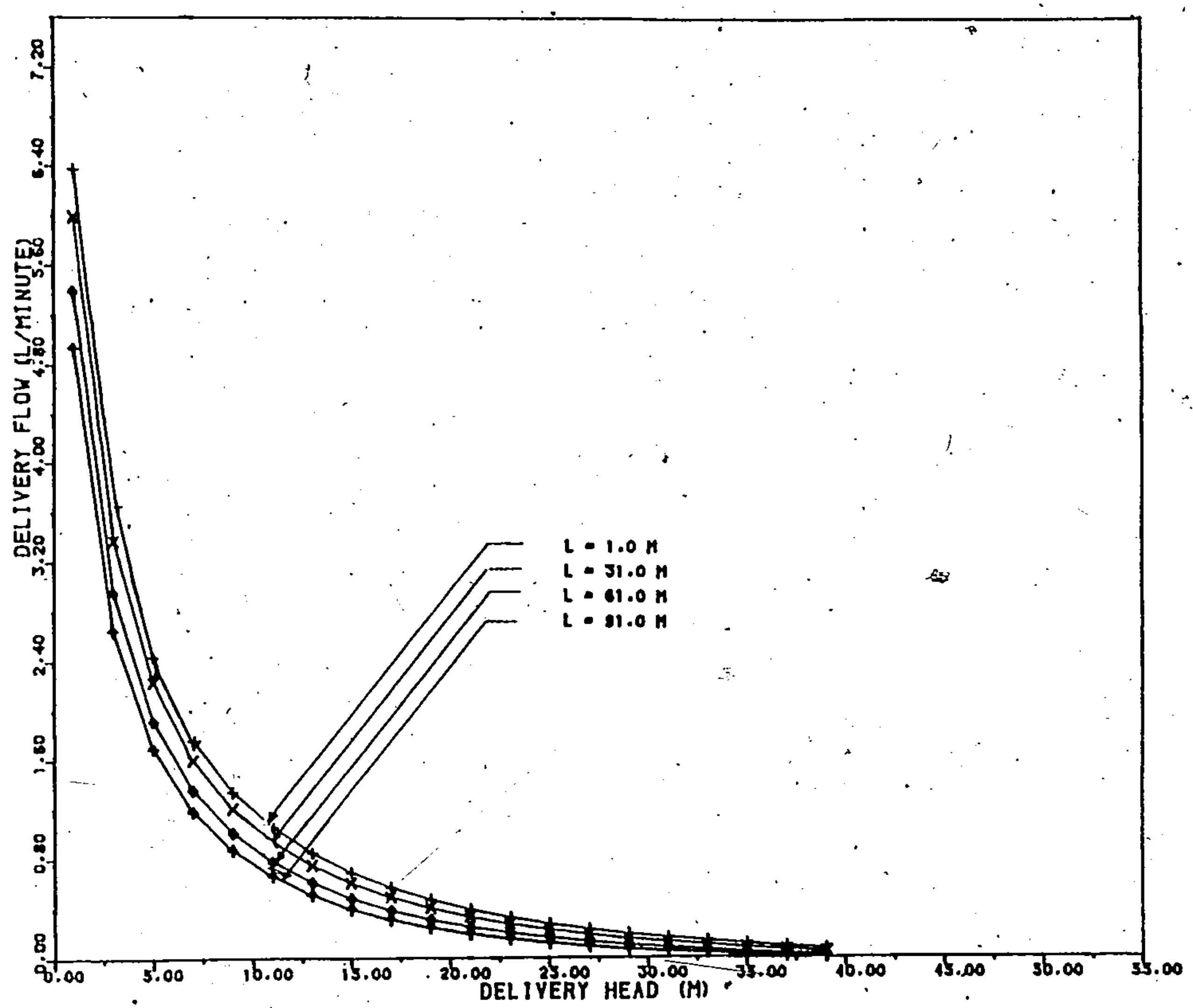


FIG. 73 EFFECT OF DRIVE PIPE LENGTH ON PUMP CAPACITY

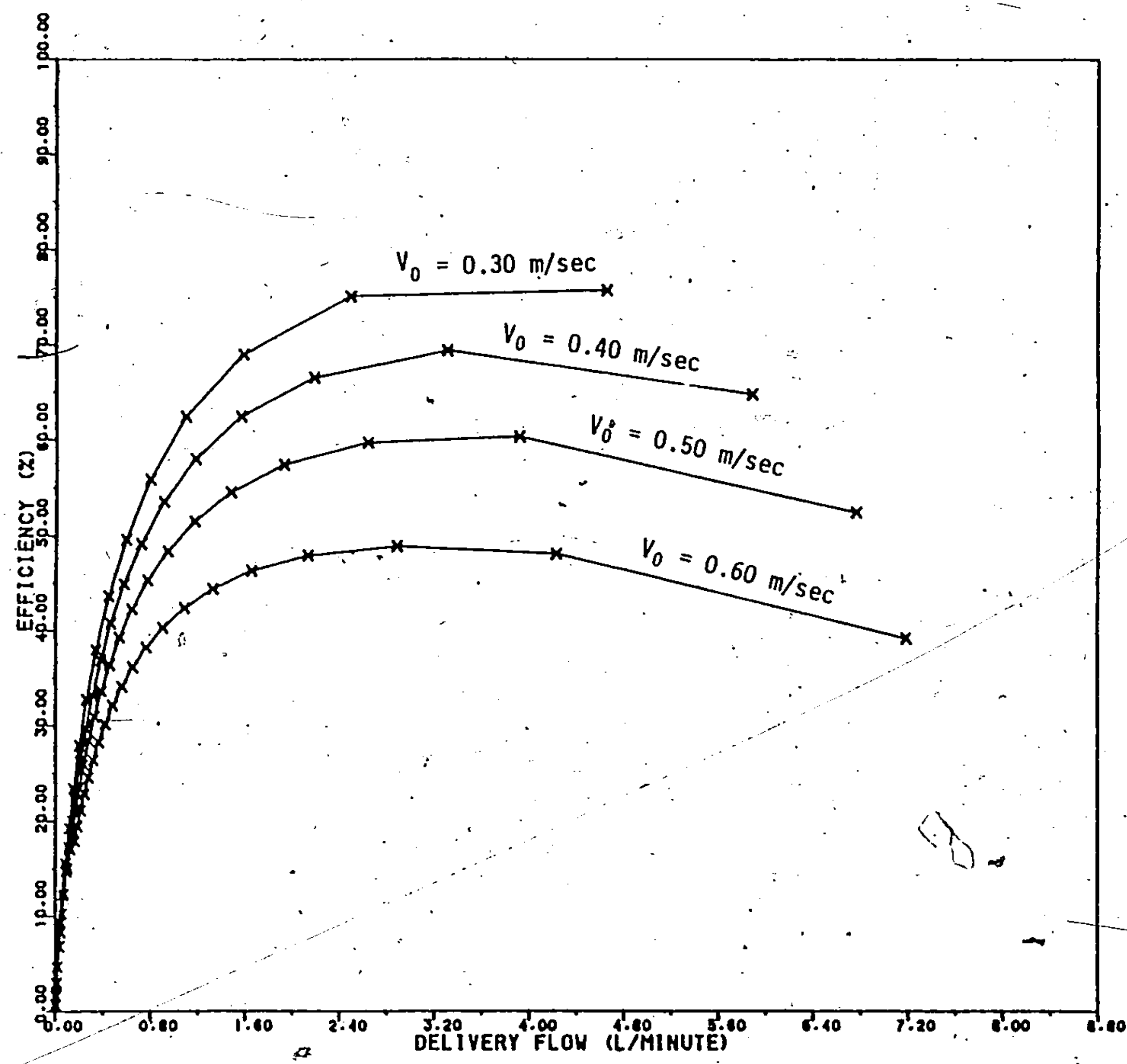


FIG. 74 EFFECT OF V_0 ON PUMP EFFICIENCY

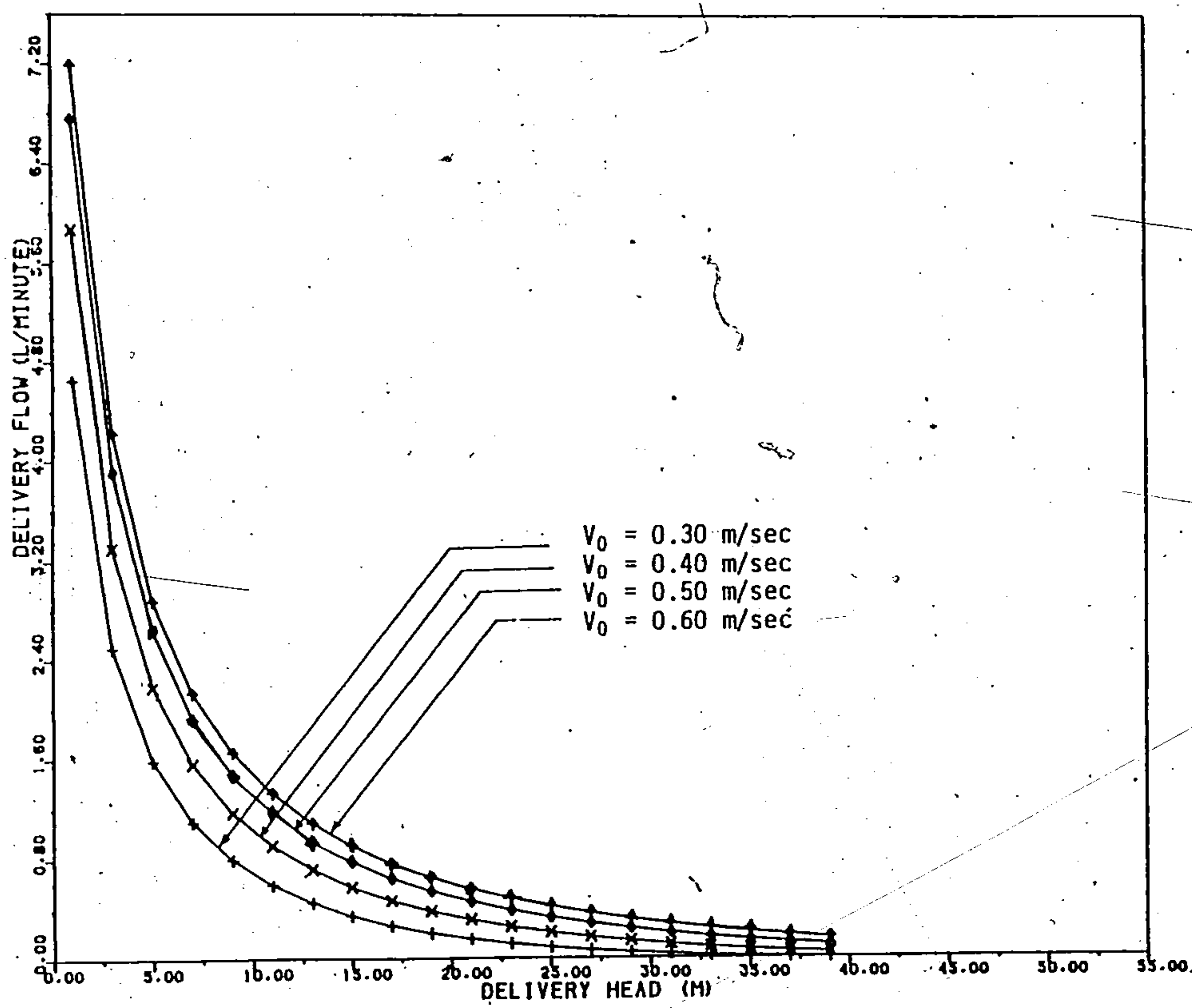


FIG.75 EFFECT OF V_0 ON DELIVERY FLOW

Appendix B

COMPUTER PROGRAM LISTING.

```
C       ***********************************************************
C       *  THEORETICAL MODELS FOR HYDRAM OPERATION                 *
C       *  USES MODELS BY KROL,IVERSEN AND KAHANGIRE (OTTAWA)      *
C       *  COMPARES AND PLOTS THE OBSERVED AND COMPUTED RESULTS *
C       *                  OTTAWA 1983                             *
C       ***********************************************************
C          H - SUPPLY HEAD (M)
C          L - DRIVE PIPE LENGTH (M).
C          M - FRICTIONAL HEAD LOSS FACTOR DURING ACCELERATION
C          N - TOTAL HEAD LOSS FACTOR DURING DISCHARGE.
C         DL - DIAMETER OF THE DRIVE PIPE (M).
C         PT - THICKNESS OF THE DRIVE PIPE (M).
C          S - STROKE LENGTH OF THE WASTE VALVE (M).
C         HD - DELIVERY HEAD (M).
C         WT - WEIGHT OF WASTE VALVE (N).
C        HVD - FRICTION FACTOR OF THE DELIVERY VALVE ALONE.
C         AV - CROSS-SECTIONAL AREA OF THE WASTE VALVE (M ² ).
C       DRAG - DRAG COEFFICIENT OF THE WASTE VALVE.
C         VO - VELOCITY TO START VALVE CLOSURE (M/SEC.).
C       AREA - CROSS-SECTIONAL AREA OF THE DRIVE PIPE (M ²).
C          G - ACCELERATION DUE TO GRAVITY (M/SEC/SEC).
C       GAMA - SPECIFIC WEIGHT OF WATER =9810 (N/M ³ ).
```

```
C        E - BULK MODULUS OF ELASTICITY OF PIPE MATERIAL
C        K - BULK MODULUS OF ELASTICITY OF WATER
C       KC - COMPOSITE MODULUS OF ELASTICITY OF WATER AND PIPE
C            MATERIAL
C        C - WAVE CELERITY (M/SEC.).
C      RHO - DENSITY OF WATER (KGS/M ³ ).
C
C*******************************************************************
C   READ IN THE PARAMETERS
C===================================================================
                REAL TITLE(80)
                REAL M,N,L,K,KC
            READ(5,1) H,L,DL,PT,AREA
            READ(5,2) AV,S
            READ(5,3) VO,DRAG,M,N,HVD
               RHO =1000.0
                G =9.81
                E =190000000000.0
                K =2170000000.0
                GAMA =9810.0
                  ICODE =1
C    ---- IF ICODE =1   RESULTS ARE PRINTED AND PLOTTED.
C    ---- IF ICODE >1   RESULTS ARE NOT PLOTTED.
C-------------------------------------------------------------
C       CALL THE SUBROUTINE TO READ THE OBSERVED DATA
C=============================================================
        CALL OBSVD(H,ICODE)
```

```
C
C
C============================================================
C   CONVERT DATA TO IMPERIAL UNITS FOR KROL MODEL
C*************************************************************
                 HK =H*3.28
                 SK =S *3.28*12.0
                 DLK =DL *3.28
                 WTK =WT *2.12
                 AREAK =AREA *10.76
                 XLK =L *3.28
                 AVK =AV *10.76
                 PTK =PT *3.28
C------------------------------------------------------------
C  START COMPUTATIONS FOR THE THEORETICAL MODELS
C------------------------------------------------------------
                 VS =SQRT(2.0*G*H/M)
                 VR =VO/VS
                 VOK =VO *3.28
                 VSK =VS *3.28
C    COMPUTE HMAX,THE MAXIMUM HEAD THE PUMP DEVELOPS  AND  C
                 X25 =(1.0/K) +(DL/(E*PT))
                 XX25 =(X25*GAMA/(144.0*G))
                 KC =1.0/X25
                 X76 =KC/RHO
                  C =SQRT(X76)
                 HMAX =C*VO/G
```

```
                WRITE(6,4) C
C=====================================================================
                WRITE(6,5) VS,VO,VR,HMAX,M,N
            CALL IVA(VS,VO,H,AREA,M,N,L,HMAX,ICODE)
            CALL KAHA(VS,VO,H,M,N,L,S,C,HMAX,HVD,AREA,ICODE)
            CALL KROL(HK,M,N,XLK,SK,DLK,WTK,VOK,VSK,PTK,HVD,AVK,ICODE)
1             FORMAT(2F4.1,3F10.7)
2             FORMAT(F10.5,F5.3)
3             FORMAT(F4.2,4F6.1)
4             FORMAT(20X, 'C =',F10.0,/15X,'===================',//)
5         FORMAT(1H1,//,10X,'VS =',F5.2,5X,'VO =',F5.3,5X,'VR =',F5.3,5X
     1'HMAX =',F5.1,/15X,'M =',F10.1,5X,'N =',F10.1,//)
                STOP
                END
C**************************************************************
C        *  SUBROUTINE TO READ THE OBSERVED DATA                *
C        *  AND COMPUTE THE PUMP CHARACTERISTICS                *
C        *                                                      *
C**************************************************************
         SUBROUTINE OBSVD(HS,ICODE)
C**************************************************************
         DIMENSION QS(50),QD(50),EFD(50),HD(50),VB(50),EFR(50)
         DIMENSION HR(50),FR(50),T(50),POWER(50),QW(50)
           REAL TITLE(80)
22      FORMAT(//,20X,'TEST DATE',15X,'N=',I5,5X,'HS =',F8.5,5X,'STROKE=
     1,F6.0,/)
C
```

```
      READ(5,11) TITLE
11    FORMAT(80A1)
      WRITE(6,12) TITLE
12    FORMAT(///,20X,40A1,//)
      READ(5,16) N,S
16    FORMAT(I5,F4.1)
      WRITE(6,22) N,HS,S
      WRITE(6,77)
77    FORMAT(/,10X,'HD',5X,'VB',5X,'QS',5X,'QD',5X,'EFD',5X,'EFR',
     15X,'HR',5X,'FR',5X,'T',5X,'QW',5X,'POWER',/)
      DO 10 I =1,N
      READ(5,2) (HD(I),VB(I),QS(I),QD(I))
      HD(I) =HD(I)/9.81
      EFD(I) =(HD(I)*QD(I)*100)/(QS(I)*HS)
      EFR(I) =QD(I)*(HD(I) -HS)*100.0/((QS(I) -QD(I))*HS)
      HD(I) =HD(I)-HS
      HR(I) =HD(I)/HS
      QW(I) =QS(I) -QD(I)
      FR(I) =QD(I)/QW(I)
      T(I) =1.0/VB(I)*60.0
      POWER(I) =QD(I)*HD(I)*9.81/60.0
C     POWER IN WATTS
      WRITE(6,23)(HD(I),VB(I),QS(I),QD(I),EFD(I),EFR(I),HR(I),FR(I),T(
     1,QW(I),POWER(I))
23    FORMAT(6X,2F7.1,2F7.2,2F8.1,2F7.2,F7.3,2(2X,F7.2),/)
10    CONTINUE
2     FORMAT(F5.0,F5.0,F6.2,F6.2)
```

```
C    CALL THE PLOTTING SUBROUTINE >>>>>>>PLOTN.
C-----------------------------------------------------------------
          IF(ICODE.GT.1) GO TO 38
         KK =2
       CALL PLOTN(N,KK,HD,QD)
38       RETURN
        END
C=========================================================
C    =  SUBROUTINE IVA FOR THE IVERSEN MODEL              =
C    =  USES FULL EQUATIONS WITH DIFFERENT N &M VALUES    =
C    =                                                    =
C=========================================================
        SUBROUTINE IVA1(VS,VO,H,A,M,N,L,HMAX,ICODE)
C**********************************************************
          DIMENSION HD(50),HR(50),FRATIO(50),QP(50),VB(50)
          DIMENSION VP(50),VT(50),T(50),POWER(50),EFF(50)
        REAL L,M,N
           WRITE(6,1)
1          FORMAT(15X,'IVERSEN MODEL RESULTS,FULL EQUATIONS',//)
          G =9.81
          NN= 20
          PI =3.142
           HD0 =1.00
             ADD =2.0
              WRITE(6,17)
17           FORMAT(11X,'I',5X,'HD',5X,'QP',4X,'HR',4X,'FRATIO',2X,'E
     1,4X,'T',6X,'VB',/)
```

```
C    START THE LOOP FOR DISCHARGES AT VARIOUS DEL. HEADS.
          DO 10 I =1,NN
           IF(I.LE.1) HD(I) =HD0
           IF(I.GT.1) HD(I) =HD(I-1) +ADD
C    PERIOD OF ACCELERATION
C==============================================================
C    COMPUTE DURATION            ----TA
C    COMPUTE FLOW VOLUME         ----VT
           X1 =(L**2)/(2.0*H*G*M)
           VR =VO/VS
           X2 =(1.0 +VR)/(1.0-VR)
           TA =SQRT(X1)*ALOG(X2)
C
C    VOLUME OF FLOW WASTED DURING PERIOD 1  -------> VT
           VM =VO
            X4 =1.0/(1.0-(VR**2))
            VT(I) =((L*A)/M)*ALOG(X4)*1000.0
C
C    PERIOD OF RETARDATION       ----> TD
             X5 =(2.0*(L**2))/(G*HD(I)*N)
             X6 =N*(VM**2)/(2.0*G*HD(I))
             TD =SQRT(X5)*ATAN(SQRT(X6))
C
C    PUMPED FLOW VOLUME      ------> VP
              X7 =((N*(VM**2))/(2.0*G*HD(I))) +1.0
              VP(I) =((L*A)/N)*ALOG(X7)*1000.0
C    FLOW RATIO  =  VOLUME RATIO
```

```
                FRATIO(I) =VP(I)/VT(I)
C     HEAD RATIO
                HR(I) =HD(I)/H
C   MECHANICAL EFFICIENCY OF THE PUMP      ----> EFF
                EFF(I) =FRATIO(I)*HR(I)*100.0
C    DURATION OF THE CYCLE
                 T(I) =TA +TD
C    PUMPING RATE -- LITRES PER MINUTE
C ===============================================================
                 QP(I) = (VP(I)/T(I))*60.0
C       COMPUTE PUMP POWER
                  POWER(I) =QP(I)*HD(I)*9.81/60.0
C    VALVE BEAT FREQUENCY         -----> VB
                  VB(I) =60.0/T(I)
C    WRITE THE RESULTS
            WRITE(6,20)I,HD(I),QP(I),HR(I),FRATIO(I),EFF(I),T(I),VB(I),
     1POWER(I)
20          FORMAT(10X,I2,2X,F5.1,2X,F5.3,2X,F4.1,5X,F5.3,2X,F4.1,2X,
     1F5.3,2X,F5.0,2X,F4.2,/)
10               CONTINUE
C    CALL THE PLOTTING SUBROUTINE >>>>>PLOTN.
C-------------------------------------------------------------------
                 IF(ICODE.GT.1) GO TO 38
                  KK =3
                CALL PLOTN(NN,KK,HD,QP)
38               RETURN
                 END
```

```
C==========================================================================
C      * THE OTTAWA MODEL BY KAHANGIRE    1983                          *
C      *                                                                *
C*********************************************************************
C
      SUBROUTINE KAHA(VS,VO,H,M,N,L,S,C,HMAX,HVD,A,ICODE)
C--------------------------------------------------------------------------
       DIMENSION HD(50),HR(50),QP(50),FRATIO(50),EFF(50)
        DIMENSION VB(50),T(50)
      REAL M,N,L,TITLE(80),POWER(50)
        GAMA =9810.0
      G =9.81
      WRITE(6,5)
5      FORMAT(//,20X,'KAHANGIRE MODEL RESULTS ',//)
         WRITE(6,19)
19      FORMAT(6X,'I',5X,'HD',4X,'HR',4X,'QP',4X,'FRATIO',4X,
     1'EFF',4X,'VB',4X,'T',/)
      HD0 =1.00
       ADD =2.0
      NN =20
      VR =VO/VS
      DO 30 I =1,NN
       IF(I.LE.1) HD(I) =HD0
       IF(I.GT.1) HD(I) =HD(I-1) +ADD
      HR(I) =HD(I)/H
C  PERIOD 1
C    COMPUTE T1
```

```
      X1 =SQRT((L**2)/(2.0*G*H*M))
      X2 =(1.0 +VR)/(1.0-VR)
      T1 =X1*ALOG(X2)
C     COMPUTE FLOW VOLUME FLOWING TO WASTE    --------->Q1
      Q1 =(L*A/M)*ALOG(1.0/(1.0-(VR**2)))
      Q1 =Q1*1000.0
C     PERIOD 2
C     COMPUTE T2
      V2 =VO
      T2 =2.0 *L/C
C     COMPUTE FLOW VOLUME,Q2
      Q2 =A*V2*T2
      Q2 =Q2*1000.0
C     TOTAL VOLUME OF WATER FLOWING TO WASTE,VT =Q1 +Q2
      VT =Q1 +Q2
C     COMPUTE HEADLOSSES DURING RETARDATION,HRD
      HRD =(HVD*(V3**2)/(2.0*G))*(1.0-(HD(I)/HMAX))
C     COMPUTE VELOCITY ,V3
      V3 =V2-((HD(I)+HRD)*G/C)
C     COMPUTE T3
      X6 =SQRT(2.0*(L**2)/(N*G*HD(I)))
      X7 =SQRT(N*(V3**2)/(2.0*G*HD(I)))
      T3 =X6*ATAN(X7)
C     COMPUTE DISCHARGE VOLUME Q3
      X8 =(N*(V3**2)/(2.0*G*HD(I))) +1.0
      X9 =L *A/N
      Q3 =X9*ALOG(X8)
```

```
            Q3 =Q3*1000.0
             VP(I) =Q3
C  COMPUTE FLOW RATIO
            FRATIO(I) =Q3/Q1
C  COMPUTE EFFICIENCY        ---->EFF
            EFF(I) =FRATIO(I)*HR(I)*100.0
C  ALTERNATIVE COMPUTATION FOR QP
         QPD =(A*L*GAMA*(VO**2))/(2.0*G*(HD(I)+HRD))
         QPP =QPD*1000.0
C----------------------------------------------------------------------
C   PERIOD 4
C  COMPUTE T4
            T4 =2.0*L/C
C   COMPUTE TOTAL CYCLE TIME     ------> T
             T(I) =T1 +T2 +T3 +T4
C   COMPUTE THE PUMPING RATE QP IN LITRES PER MINUTE.
             QP(I) = VP(I) *60.0/T(I)
             POWER(I) =QP(I)*HD(I)*9.81/60.0
C   COMPUTE CYCLE FREQUENCY IN BEATS/MINUTE
             VB(I) =60.0/T(I)
C    WRITE THE RESULTS.
          WRITE(6,40) I,HD(I),HR(I),QP(I),FRATIO(I),EFF(I),VB(I),
    1T(I),POWER(I)
40         FORMAT(5X,I2,2X,F5.1,2X,F4.1,2(2X,F6.3),2X,F5.1,2X,
    1F5.0,2X,F5.3,2X,F4.2,/)
30           CONTINUE
C   CALL THE PLOTTING SUBROUTINE >>>>>> PLOTN
```

```
      IF(ICODE.GT.1) GO TO 38
       KK =4
      CALL PLOTN(NN,KK,HD,QP)
38    RETURN
      END
C===================================================================
C  ******************************************************************
C *   KROL SUBROUTINE USES THE MODEL BY KROL IN IMPERIAL UNITS      *
C *   THE RESULTS ARE CONVERTED INTO METRIC UNITS                   *
C  ******************************************************************
C-------------------------------------------------------------------
      SUBROUTINE KROL(HS,HM,N,XL,S,DL,WT,VO,VS,PT,HVD,AREA,ICODE)
C===================================================================
      DIMENSION HD(50),QD(50),QW(50),EFF(50),POWER(50),HR(50)
      DIMENSION T(50),VB(50),VR(50),FRATIO(50),HRL(50)
      DIMENSION QDM(50),QWM(50),HDM(50)
       REAL M,N,L,HD0
      PI =3.142
      E =28000000.0
      G =32.2
      XKK =300000.0
      GAMA =62.4
       FN =0.0169
       SUMJ =0.5
        XM =12.0
C*******************************************************************
C              XL =L = LENGTH OF DRIVE PIPE
```

```
C              HS ---- SUPPLY HEAD
C              HD -----DELIVERY HEAD
C                FN ---- FRICTIONAL COEFFICIENT OF DRIVE PIPE
C*******************************************************************
                 DRAG =WT*G/(AREA*GAMA*VO**2)
                 WRITE(6,324)
324              FORMAT(1H1,///,20X,'KROL MODEL RESULTS.',//)
              RS =HM-XM
           WRITE(6,441) S,RS,DRAG
441        FORMAT(20X,'STROKE =',F10.5,10X,'RS =',F10.5,10X,'DRAG= ',F1
     15,//)
C  **     COMPUT MAX HEAD THE PUMP CAN DEVELOP
            X25 =(1.0/XKK) +(DL/(E*PT))
             XX25 =(X25*GAMA/(144.0*G))
             XK1 =1.0/X25
            X35 =X25*AREA*(GAMA**2)*DRAG
            X36 =144.0*WT/X35
            HDMAX3 =SQRT(X36)
               C =SQRT(144.0*XK1*G/GAMA)
               HMAX =C*VO/G
C*****************************************************************
             WRITE(6,439) HDMAX3,HMAX
439          FORMAT(20X,'HDMAX3 =',F10.1,5X,'HMAX =',F10.1,//)
              WRITE(6,800)
800           FORMAT(15X,'OPERATING CHARACTERISTICS OF HYDRAM',/)
            WRITE(6,18)
18          FORMAT(11X,'I',5X,'HD',5X,'HR',4X,'QP',2X,'FRATIO',
```

```
     14X,'EFF',4X,'VB',4X,'T',/)
C *************************************************************
C   START COMPUTATIONS FOR DELIVERY FLOW AT DIFFERENT DEL. HEADS.
C-------------------------------------------------------------
           HD0 =3.5
             ADD =6.5
           DO 20 I =1,20
             VR(I) =VO/VS
           IF(I.LE.1) HD(I) =HD0
           IF(I.GT.1) HD(I) =HD(I-1) +ADD
          HRL(I) =(HVD*(VO**2)/(2.0*G))*(1.0-(HD(I)/HDMAX3))
C ****************************************************************
C  **     COMPUTE DELIVERY FLOW  LBS PER CYCLE
C ***************************************************************
                 HR(I) =HD(I)/HS
            X41 =PI*(DL**2)*XL/(8.0*(HD(I) +HRL(I)))
            X42 =WT/(AREA*DRAG)
             X43 =((GAMA**2)/144.0)*X25*((HD(I)+HRL(I))**2)
            QD(I) =X41*(X42-X43)
C **     COMPUTE THE AMOUNT WASTED >>>>>QW
             X44 =GAMA*PI*(DL**2)/4.0
             X45 =XL/HM
             X46 =2.0*G*HS/HM
             X47 =(G*GAMA/144.0)*X25*((HD(I)+HRL(I))**2)
             X48 =WT*G/(AREA*GAMA*DRAG)
             X49A =ABS((X46-X47)/(X46-X48))
             X49 =X45*ALOG(X49A)
```

```
      X50 =S*XL*(X48**2)
      X51 =2.0*AREA*HS*GAMA*DRAG
      X52 =4.0*(G**2)*HS*(1.0-(WT*HM/X51))
       X53A =X50/X52
      X53 =(X50/X52)**0.3333333333333
     QW(I) =X44*(X49+X53)
     FRATIO(I) =QD(I)/QW(I)
C **      COMPUTE DURATION OF TOTAL CYCLE  >> T << SECS/CYCLE
      X54 =XL/SQRT(2.0*G*HS*HM)
      X55 =1.0 +SQRT(WT*HM/X51)
      X56 =1.0-SQRT(WT*HM/X51)
      X57A =ABS(X55/X56)
      X57 =X54*ALOG(X57A)
      X58 =S*XL*SQRT(X48)
      X59 =4.0*(G**2)*HS*(1.0-(WT*HM/X51))
      X60A =X58/X59
      X60 =(X58/X59)**0.3333333333333
      X61 =XL*SQRT(X48)/(G*(HD(I) +HRL(I)))
      X62 =SQRT(X25*GAMA/(144.0*G))
      X63 =X62*XL*(((HD(I)+HRL(I))/HS)+3.0)
      T(I) =X57 +X60 +X61 +X63
      VB(I) =(1.0/T(I))*60.0
C  **     COMPUTE EFFICIENCY
      EFF(I) =(QD(I)*HD(I)/(QW(I)*HS))*100.0
C=====================================================================
C   CHANGE RESULTS TO METRIC UNITS   >>>>>>>>>>>>>>>>>>>>>>>>>>>>>>>>
C--------------------------------------------------------------------
```

```
      QDM(I) =QD(I)*1000.0*60.0/(GAMA*35.3*T(I))
      QWM(I) =QW(I)*1000.0*60.0/(GAMA*35.3*T(I))
      HDM(I) =HD(I)/3.28
      POWER(I) =QDM(I)*HDM(I)*9.81/60.0
C **   WRITE OUT THE RESULTS
      WRITE(6,25) I,HDM(I),HR(I),QDM(I),FRATIO(I),EFF(I),VB(I),
     1T(I),POWER(I)
25    FORMAT(10X,I2,2X,F5.1,2X,F5.1,2X,F5.3,2X,F5.3,2X,F5.1,
     12X,F5.0,2X,F5.3,2X,F4.2,/)
20    CONTINUE
C   CALL THE PLOTTING SUBROUTINE >>>>>>>> PLOTN
      IF (ICODE.GT.1) GO TO 38
      NN =10
      KK =5
      CALL PLOTN(NN,KK,HDM,QDM)
38    RETURN
      END
C==============================================================
C    *                                                         *
C    *    SUBROUTINE FOR PLOTTING OBSERVED AND COMPUTED RESULTS *
C    *                                                         *
C    *    USES THE CALCOMP PLOTTER                             *
C**************************************************************
      SUBROUTINE PLOTN(N,KK,X,Y)
C**************************************************************
      DIMENSION X(50),Y(50)
      K1 =N +1
```

```
      K2 =N +2
        X(K1) =0.0
        X(K2) =2.5
        Y(K1) =0.0
        Y(K2) =0.40
       IF(KK.GT.2) GO TO 66
      CALL PLOTS(35.0,27.5)
      CALL PLOT(2.0,4.0,-3)
       CALL FACTOR(0.75)
       CALL RECT(0.0,0.0,23.0,22.0,0.0,3)
     CALL AXIS(0.0,0.0,17HDELIVERY HEAD (M),-17,22.0,0.0,X(K1)
    1,X(K2))
     CALL AXIS(0.0,0.0,13HDELIVERY FLOW,13,20.0,90.0,Y(K1),Y(K2))
      CALL SYMBOL(2.0,-2.0,0.30,'FIG.    DELIVERY HEAD VS DELIVERY F
    1W',0.0,39)
66    CALL FLINE(X,Y,N,1,1,KK)
      IF (KK.LT.5) GO TO 39
       CALL FLINE(X,Y,N,1,1,KK)
        CALL SYMBOL(25.0,15.5,0.25,2,0.0,-1)
        CALL SYMBOL(26.0,15.5,0.25,8HOBSERVED,0.0,8)
        CALL SYMBOL(25.0,15.0,0.25,3,0.0,-1)
        CALL SYMBOL(26.0,15.0,0.25,13HIVERSIN MODEL,0.0,13)
        CALL SYMBOL(25.0,14.5,0.25,4,0.0,-1)
        CALL SYMBOL(26.0,14.5,0.25,12HOTTAWA MODEL,0.0,12)
        CALL SYMBOL(25.0,14.0,0.25,5,0.0,-1)
        CALL SYMBOL(26.0,14.0,0.25,10HKROL MODEL,0.0,10)
      CALL PLOT(0.0,0.0,999)
```

```
39      RETURN
        END
//GO.SYSIN  DD  *
>>>> INSERT DATA HERE >>>>>>>
/*
=====================================================================
```

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