UNIVERSITY OF MINNESOTA

ST. ANTHONY FALLS HYDRAULIC LABORATORY

Project Report No. 201

EFFECT OF AIR INGESTION ON PERFORMANCE OF A CENTRIFUGAL PUMP

by

John M. Killen and Joseph M. Wetzel

Final Report for
Contract N00024-80-C-5944

Prepared for

NAVAL SEA SYSTEMS COMMAND
Engineering Test Services
Naval Ship Engineering Center
Department of the Navy
Washington, D. C. 20362

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TABLE OF CONTENTS

LIST OF FIGURES

EFFECT OF AIR INGESTION ON PERFORMANCE OF A CENTRIFUGAL PUMP

INTRODUCTION

The full scale model test described here was initiated to examine the effect of suspended air bubbles on the performance of a CGN 38 sea-water circulating pump. A Carver pump*Type 13N, Serial No. 110709, was chosen as the test model. This is a single suction, vertical discharge, horizontal suction pump. It has a single stage impeller and is capable of delivering 3000 gpm at a total dynamic head of 10 psi at 1150 rpm. The impeller was trimmed by the pump manufacturer to provide the desired head-discharge curve near the rated flow condition.

The pump suction line was connected to the overhead St. Anthony Falls Hydraulic Laboratory supply channel, which conveys Mississippi River water into the building. After passing through the pump, the water was discharged into the laboratory weighing tank system,as shown on Fig. 1. Flow rates can be measured with an accuracy of better than .1 per cent depending on the duration of the sample. A gate valve at the downstream end of the discharge pipe was used to control flow through the system. Air was injected into a vertical leg of the pump section line through an injector nozzle supplied by the sponsor.

The pressure at the pump inlet and the pressure differential across the pump was measured with mercury manometers. Precision Bourdon tube gages with a .1 psi resolution were used to measure pressure associated with air flow rate measurement. Air flow rate was measured with venturi meters, supplied by the sponsor, and the manufacturer's calibration curves. The air volume flow rate referenced to the pump inlet was determined from the measured mass flow rate of the injected air and the air density as calculated from the measured inlet pressure.

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*Carver Pump Company, 2415 Park Ave., Muscatine, Iowa. Ref. Carver Drawing No. D-13N-098-004.

The overhead supply channel, which was 16 feet above the pump inlet, provided sufficient head to assure a positive gage pressure in the suction line under the rated flow conditions. The following comparisons can be made between model and prototype:

	Model	Prototype
Suction Conduit	8 in. Dia.	12 in. Dia.
Discharge Conduit	10 in. Dia.	10 in. Dia.
Suction Pressure	1 psig	4 to 4½ psig
Discharge Pressure	10 psig	10½ to 12 psig
Fluid	River Water	Sea Water
Discharge	3000 gpm	Unknown
Impeller	Cast Steel	Alloy

The most significant difference between the model test conditions and the prototype conditions was the possibility of maintaining flow through the model pump even though the pump's ability to develop a head was lost due to air ingestion. This capability in the model test made it possible to extend values of the air void fraction beyond the unstable region which will be noted on the accompanying figures.

In order to duplicate the flow condition at values other than the rated condition, it would have been necessary to duplicate the flow resistance of the prototype system. Since this was not available, the downstream valve in the test loop was set to give rated conditions and was assumed to approximate the system flow resistance at other flow rates.

RESULTS

The results of initial tests with clear water at and above the rated discharge are shown on Fig. 2, and good agreement was obtained with the manufacturer's curve. The head, H, is the total dynamic head calculated in accordance with the *Hydraulic Institute Standard*, 1975.

$$H = \frac{\left[P_d - P_s\right]\left[\frac{1}{W_d} + \frac{1}{W_s}\right]}{2} + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} + Z_d - Z_s$$

where P = unit pressure,

W = specific weight,

V = velocity,

Z = elevation,

and the subscripts d and s refer to discharge and suction tap locations, respectively.

Valid comparison of total dynamic head can be made between model and prototype if the changes of suction pipe size and suction tap location are taken into account in the calculation of total dynamic head.

The use of a pressure control on the air supply gave a nearly constant volume flow rate, Q_a, over a range of water flow rates, Q_w. As the pump began to lose head, the water flow rate reduced and the void fraction, as a result, increased which caused further loss of head developed by the pump. This cumulative effect resulted in an unstable region, shown by hash lines in Fig. 3, in which equilibrium could not be maintained for a sufficient period of time for valid measurements to be made. If a large mixing tank for combining air and water had been employed so that pump performance would not influence the void fraction, instability would be reduced, although possibly not entirely eliminated as further discussion will indicate.

Figure 4 is a replotting of the data of Fig. 3 with the ordinate now expressed as a ratio of total dynamic head with air to total dynamic head without air, H_o, each at the same flow rate. The latter was taken from Fig. 2. A test condition more similar to that in field operation is shown in Fig. 5. The numbers next to each data point are the actual water flow rates in gpm. The downstream control valve was set to give rated flow and pressure with clear water and was left unchanged while air was added to the flow. It must be realized that the lowest flow conditions are maintained by gravity flow rather than by any contribution by the pump.

The ability of the pump to recover from a condition of failure due to the presence of air was examined by starting the test at a void fraction greater than 0.1 and gradually reducing the air content of the flow. The data are plotted in Fig. 6. Essentially the same performance curve is produced for increasing as for decreasing the void fraction.

Other experimental results (Ref. 1) have indicated that the loss of efficiency by a centrifugal pump was not as great, with the addition of air, if the pump was operated at a higher flow rate. To investigate this possibility some tests were made at a discharge exceeding the design value, and the results for an initial no air flow rate of 3815 gpm are shown in Figs. 7 and 8. The marked difference in the shape of Figs. 7 and 8 is due to the very rapidly falling head with discharge above design flow. Only a slight improvement in developed head can be noted for this particular pump. The unstable region was not present since the flow change with air addition was not as great.

DISCUSSION

The tests have established that a sudden loss of head developed across the pump which occurs between 4 and 5 per cent volume concentration of air; however, the pump performance will reliably recover if the void fraction is reduced below .04. It will now be proposed that the observed reduction of pump head with the addition of air is a reliable measure of the behavior of the prototype system where the flow is entirely dependent on the head developed by the pump. Recall that the flow in the laboratory test setup was only partly dependent on the head developed by the pump and partly by the difference in elevation between the head tank and discharge tank.

An exploratory effort was made to determine the physical mechanism for reduction in pump head due to air ingestion. A small laboratory centrifugal pump in a transparent case with impeller vane contours similar to those in the Type 13N impeller was available. A flow was passed through this small pump in a manner similar to that in the model test, and the flow in the impeller was observed with a strobe light.

The following is a tentative explanation of the pump performance shown in Figs. 3 through 8 in terms of the usual observations on the small pump. Sketches of the observed flow in the impeller are shown in Fig. 9. As air was added to the flow, a separation zone at the leading edge of the vane began to grow. This phenomena is believed to correspond to the small reduction in developed head between zero and .04 void fraction. At about

.04 void fraction, the separation zone filled the flow passage and an interface formed along a circumference which would expand to enclose all or part of the impeller. This interface would move in or out radially with a very small change in air flow rate and is believed to correspond to the unstable region shown in the figures. The pump acted as a very efficient air separator in this mode. The separated air is fed into the central part of the impeller at a rate equal to its removal by the imposed flow. Eventually the efficiency of the pump as an air separator was impaired, and the leveling out of the curve above .06 void fraction occurred. When the interface is located near the outer perimeter of the impeller, flow runs into the eye of the impeller as a jet and spreads uniformly over the back of the pump impeller giving a layered flow in the passages, except at the very tip of the vanes.

An effort was made without success to remove air in the immediate vicinity of the eye of the impeller in the Type 13N model test. In view of the visual observation with the transparent model, it appears that an optimum location of a vent would be difficult to specify from the geometry of the system and that a rather large volume of air must be removed.

Based on the above discussion, it is concluded that either an air removal system should be incorporated in the suction line to assure an air content of less than 4 per cent by volume, or an alternative type of pump should be sought which would be more tolerant of high void fractions, or both. A passive type air removal system, such as a gravity separator, rather than an active type such as a hydrocyclone, would be preferred. A degradation of pump performance would reduce the effectiveness of the hydrocyclone separator, which would result in the same type of instability observed in the figures of the test reported here. The acceptable void fraction of .04 would considerably reduce the size of a gravity separator from the usual water tunnel application in which air removal down to a fraction of a per cent is desired.

No experimental data are known to be available as to the effect of bubble size at a given concentration on pump performance. Experience has shown that bubbles shear down to a maximum diameter of approximately 1.5 mm (Ref. 2) in a turbulent boundary layer. These bubbles have a rise velocity of about 10 cm per sec. A gravity air separator is proposed to replace the suction line that is presently 12 in. in diameter with a pipe of 24 in. diameter to give an average flow velocity of approximately 2 ft per second at a discharge of 3000 gpm. The cross section of the suction line would be filled with a bundle of diamond-shaped tubes with each tube approximately 3/4 in. on the side. A total tube length of 24 in. would be sufficient, particularly if the 24 in. diameter suction line was offset with its center line above the center line of the pump impeller.

An effort was made to utilize and calibrate two K-Ray radioactive type concentration meters supplied by the sponsor. Initially, only one of the units functioned properly. Calibration against measured flow rates showed a dependency on liquid flow rate which does not seem reasonable. Available time and other priorities of the contract limited investigation of the performance of these meters.

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2. Cain, P., Personal communication relating to his thesis, Department of Civil Engineering, University of Canterbury, New Zealand, 1975.

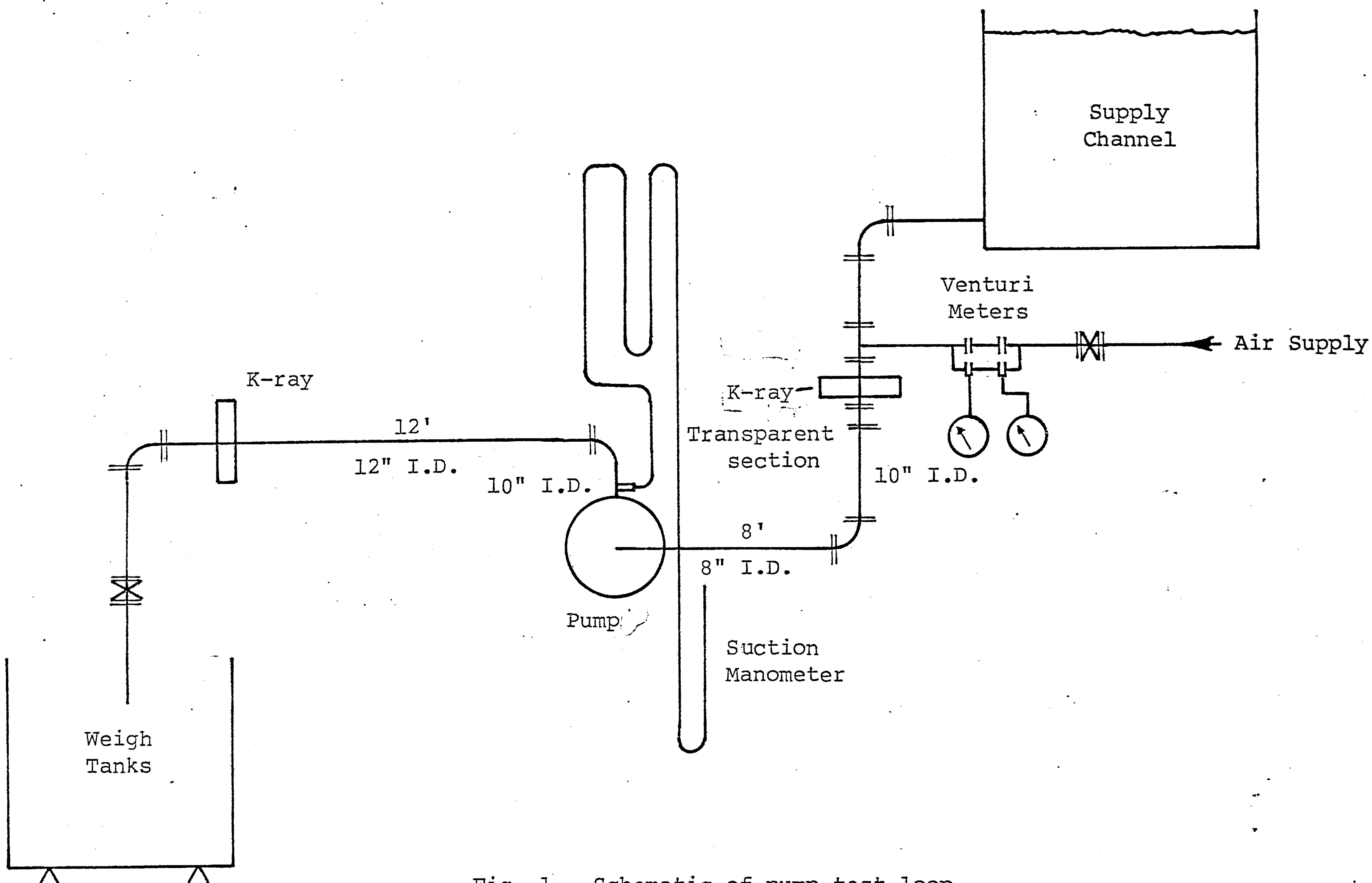

Fig. 1. Schematic of pump test loop.

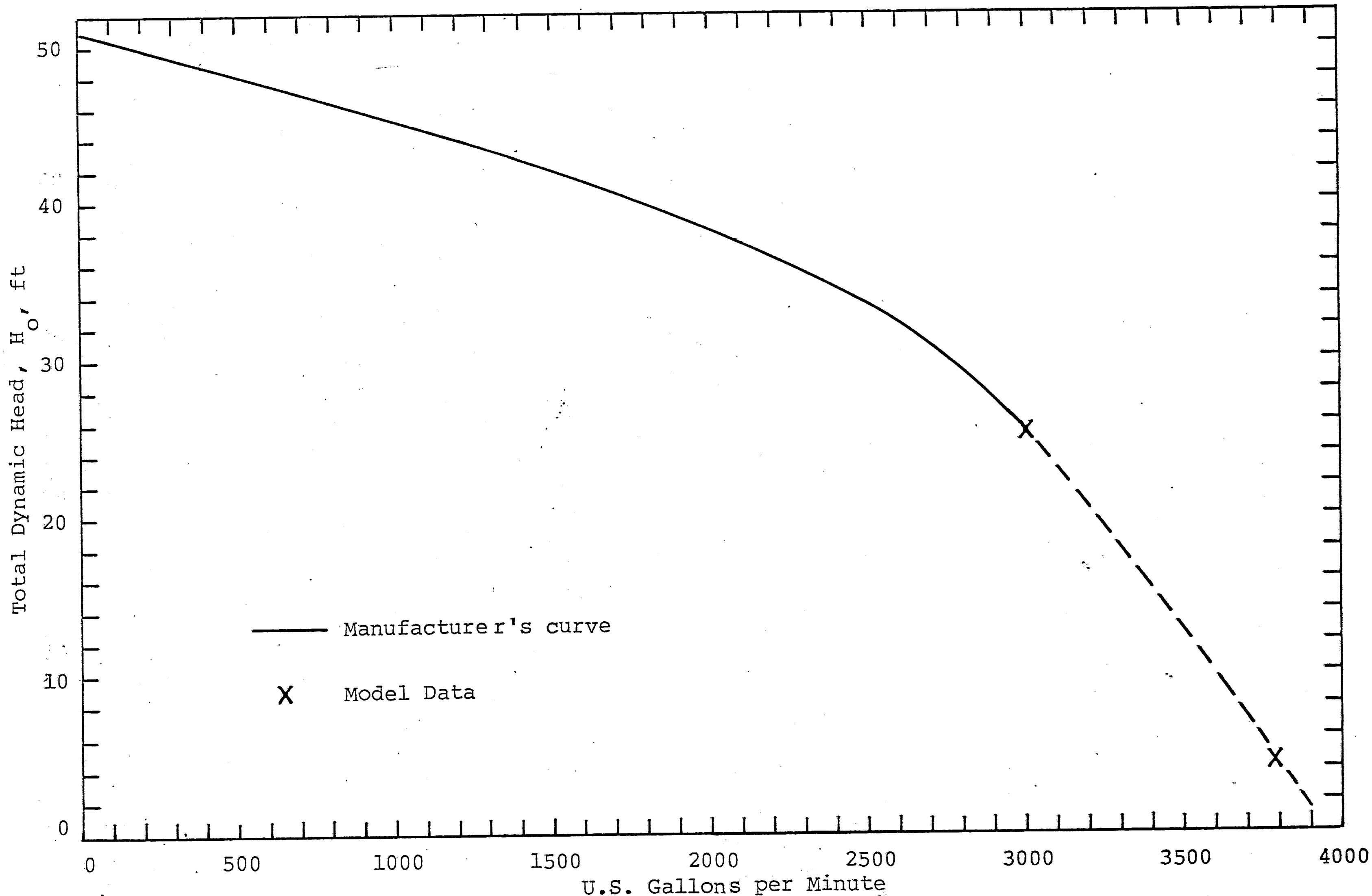

Fig. 2. Pump performance with no air added.

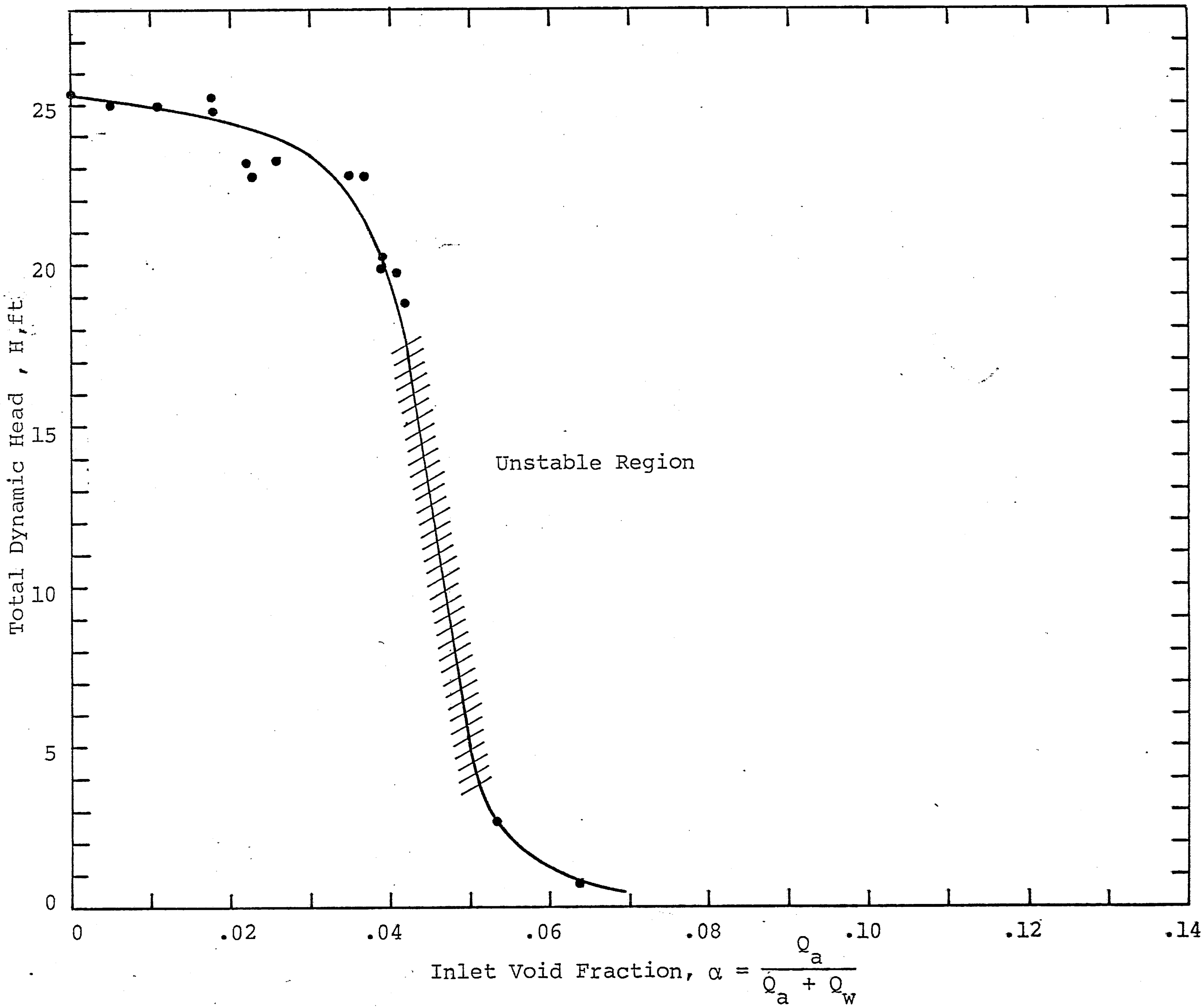

Fig. 3. Variation of pump head with inlet void fraction, Q_w = 3000 gpm.

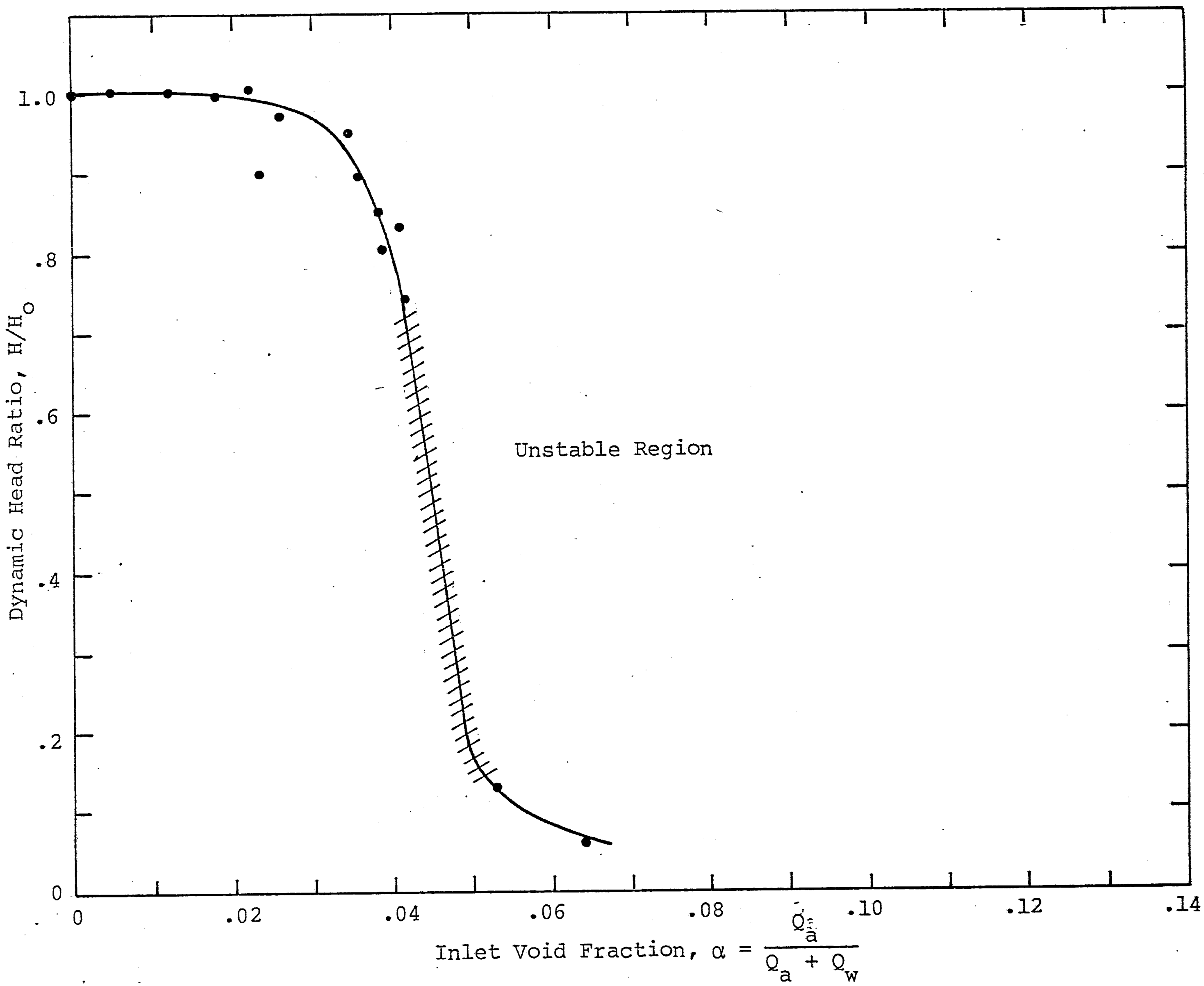

Fig. 4. Variation of pump head ratio with inlet void fraction, Q_w = 3000 gpm.

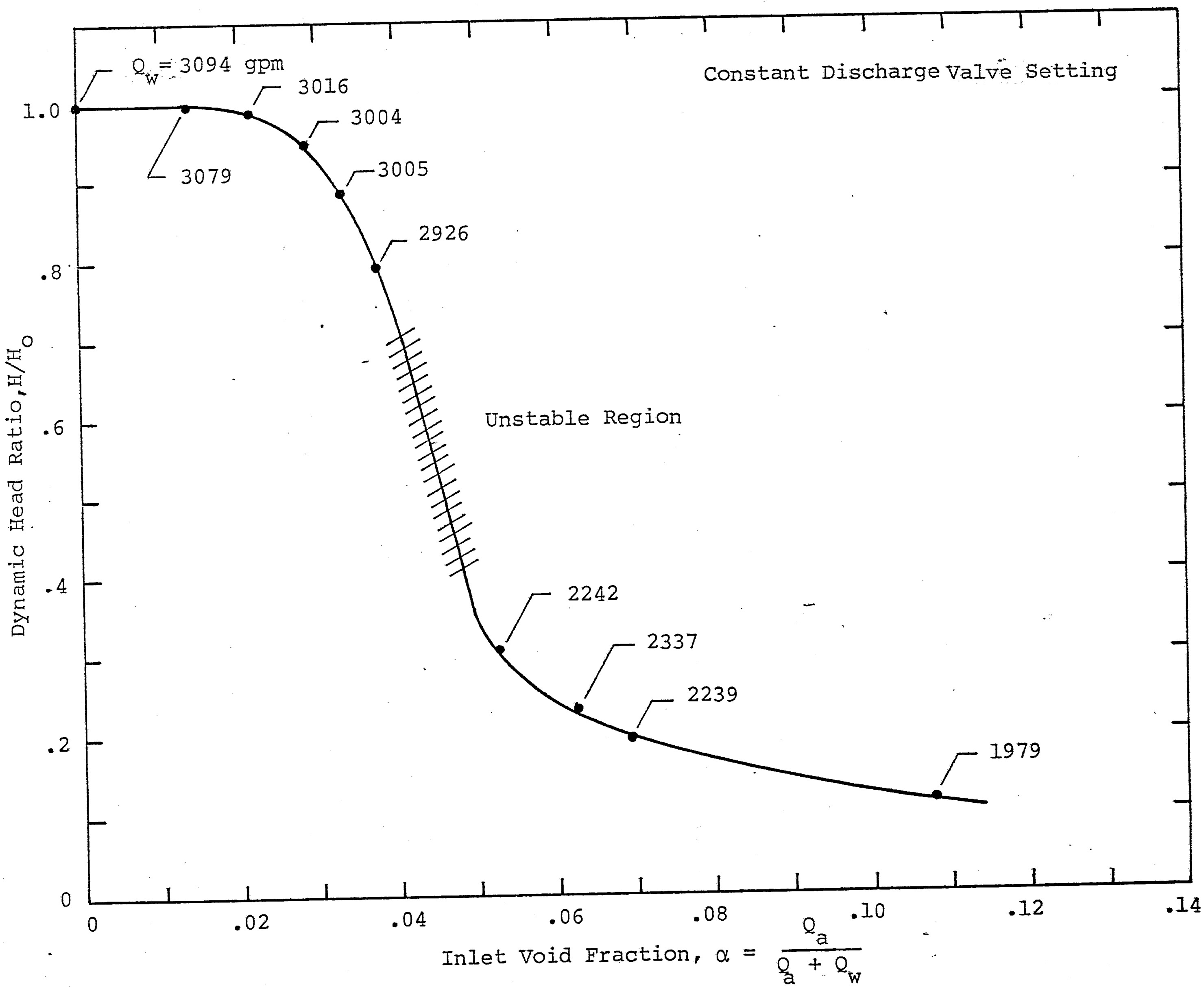

Fig. 5. Variation of pump head ratio with inlet void fraction.

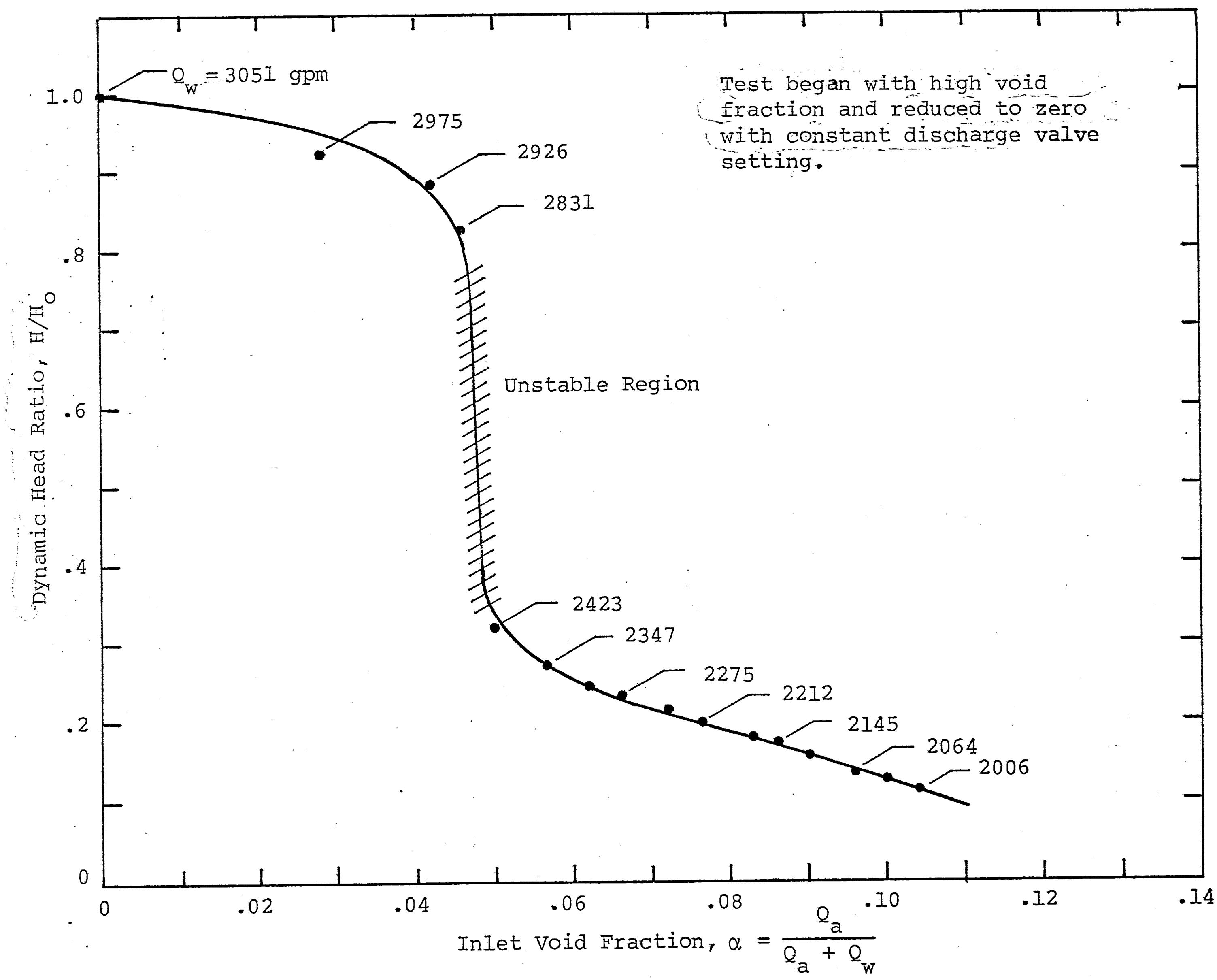

Fig. 6. Variation of pump head ratio with void fraction, hysteresis effect.

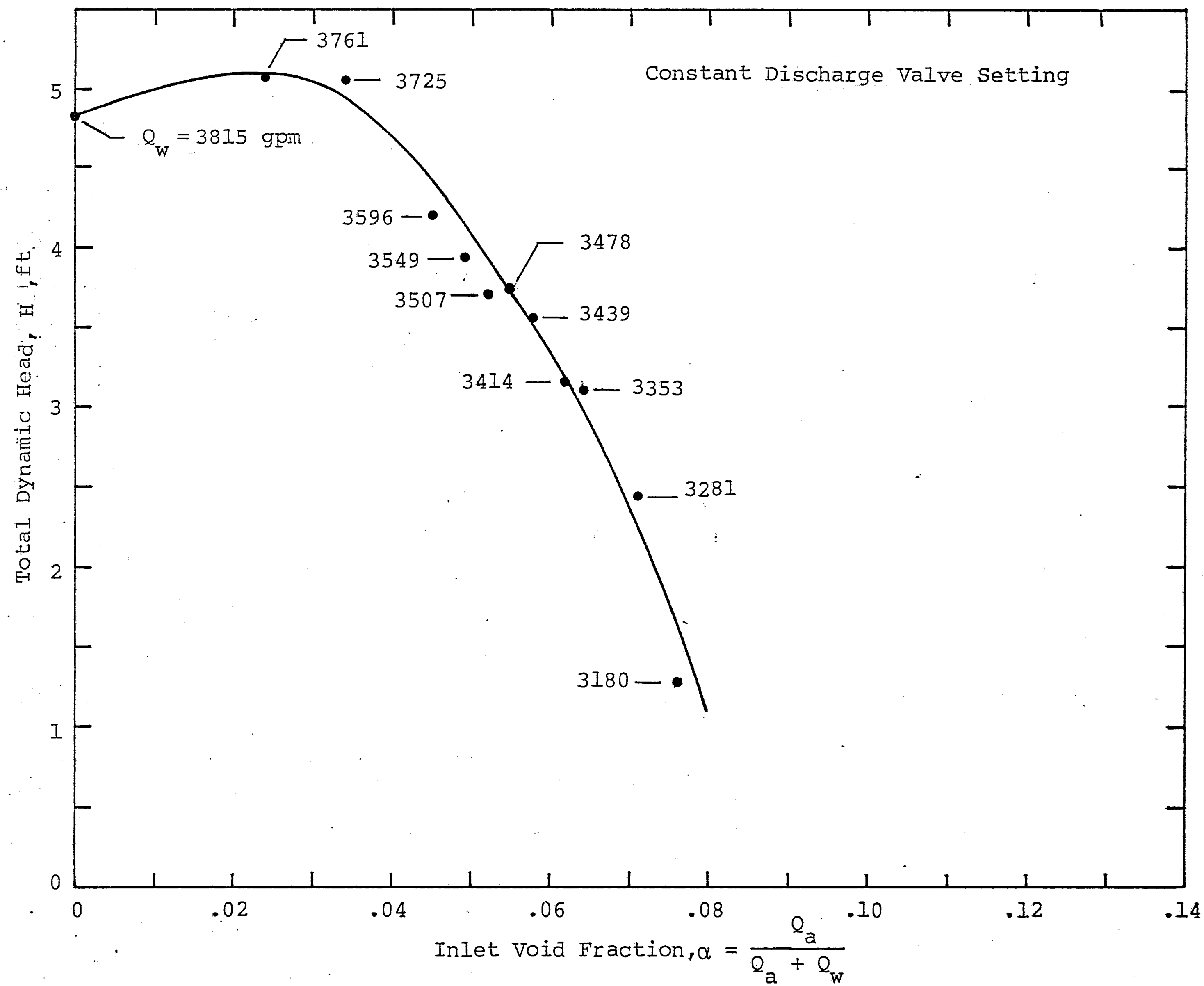

Fig. 7. Variation of pump head with void fraction, Q_w > 3000 gpm.

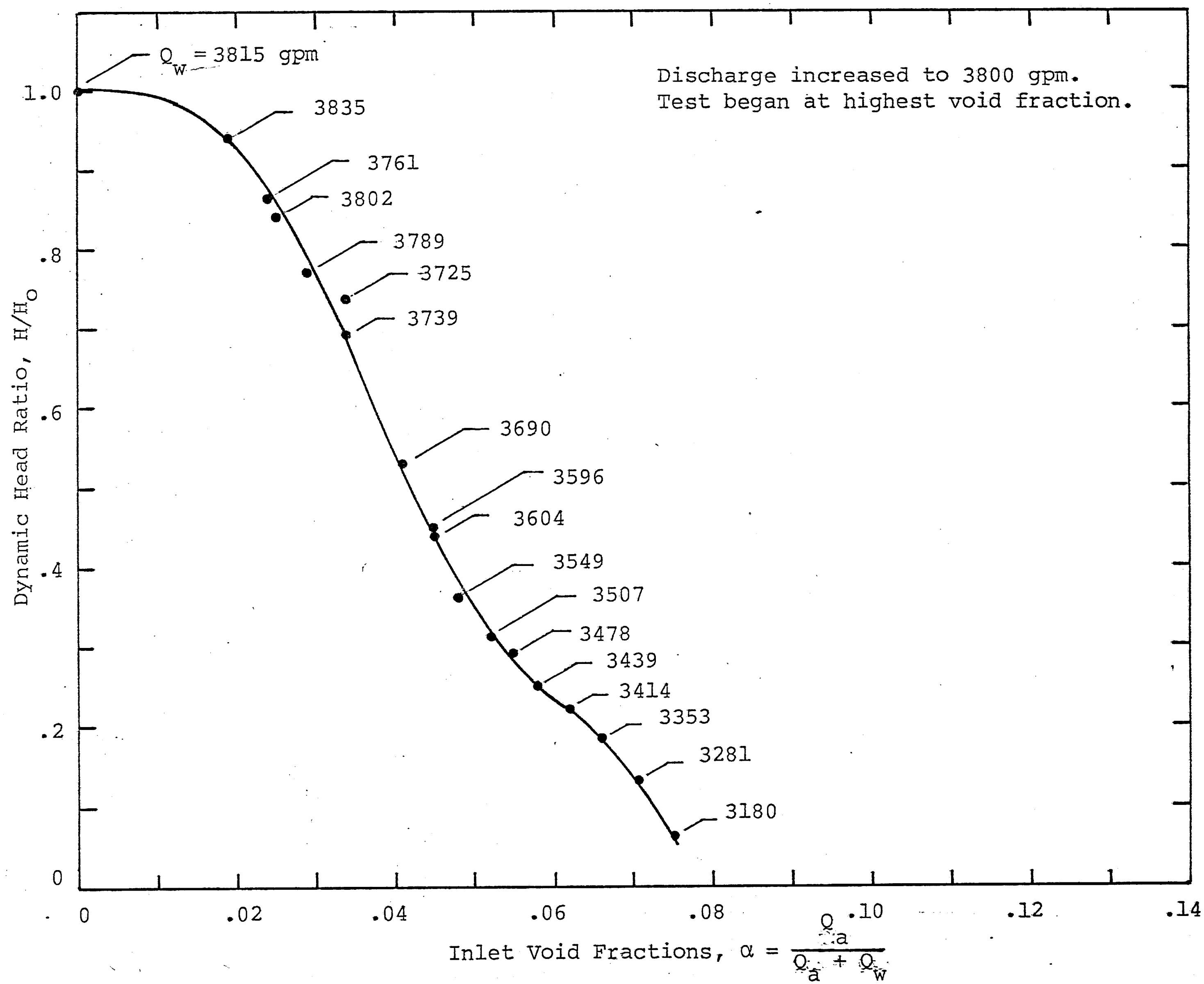

Fig. 8. Variation of pump head ratio with void fraction, hysteresis effect, $Q_w > 3000$ gpm.

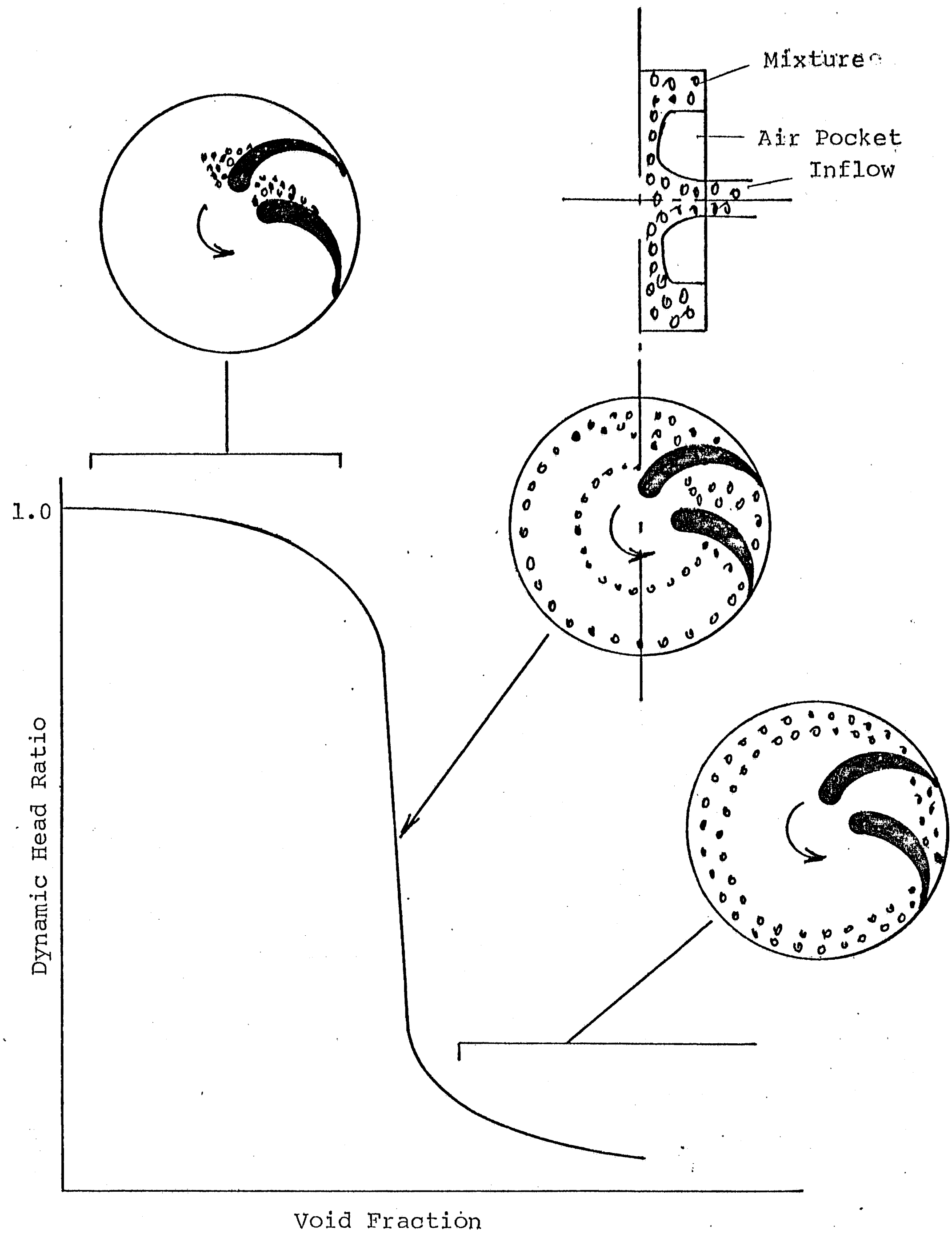

Fig. 9. Air distribution in small pump impeller.