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Refrigerant-Pressure-Actuated Water Regulating Valves
for Refrigerant Condensers

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A STUDY OF THE CHARACTERISTICS
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FOR REFRIGERANT CONDENSERS

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1.0 Introduction

At the request of the Quartermaster Research and Engineering Command, the National Bureau of Standards investigated the performance characteristics of refrigerant-pressure-actuated valves designed for regulating the flow of water through refrigerant condensers. This type of valve, widely used in water-cooled refrigerating systems, has two principal functions, namely, (a) to maintain a refrigerant condensing pressure in the compression system that will produce an adequate flow of liquid refrigerant through the expansion device, and (b) to conserve water. These same purposes can be accomplished by temperature-actuated valves which control the condenser water flow in response to the temperature of the refrigerant in the condenser or to the difference in water temperature between condenser inlet and outlet. However, each type of valve has somewhat different operating characteristics.

A proposed Military Specification, "Valves, Water Regulating, Refrigerant R-12", prepared by the Quartermaster Corps and dated August 31, 1960 classifies these water-regulating valves as follows:

Size 1 - 3/8 inch National Pipe Thread (NPT), 6 gallons per minute (GPM) capacity
Size 2 - 1/2 inch NPT, 10 GPM capacity
Size 3 - 3/4 inch NPT, 25 GPM capacity

In addition, the specification incorporates material requirements designed to provide strength, durability, corrosion resistance, stability of performance, and ease of maintenance and repair.

A proposed Military Standard, "Valves, Water Regulating, Refrigerant R-12", also prepared by the Quartermaster Corps and dated August 31, 1960 cites the above classification and includes the following rating conditions:

Condenser Hot Gas Pressure to be Maintained	126.55 psig
Pressure Drop across Valve for Rated Capacity	25 psig
Regulator Opening psig below Condenser Pressure	25 psig
Range of Operating Pressures	65 to 200 psig

Allowable Water Pressures 125 psig (Max), 35 psig (Min)
Maximum Water Temperature in Condenser Based on Temperature Rise in Condenser of 10F - 85F.

The principal purpose of this investigation was to study the performance characteristics of refrigerant-pressure-actuated water-valves that are of significance with respect to the two broad functions of refrigerant condensing pressure control and water conservation, and included the following characteristics:

(a) The range of condensing pressure adjustment provided by the valve.
(b) The change in condensing pressure required to move the valve from fully closed to fully open position for a range of inlet water pressures.
(c) The water flow rate through the valve in its fully open position for a range of pressure drops across the valve.
(d) The hysteresis in the control mechanism on rising and falling condensing pressures.
(e) The difference between the condensing pressure required to commence opening the valve and that required to fully stop the water flow.
(f) The effect of inlet water pressure on the condensing pressure required to open and close the valve for a given setting of the adjustment spring.

Pressure-actuated valves of 3/8-inch, 1/2-inch, and 3/4-inch pipe size from each of three valve manufacturers were used for the study. The relation of condensing pressure to water flow rate was observed for each valve for three settings of the adjustment spring and for three values of water pressure drop through the valve.

In addition, the results obtained from the test specimens were compared with the requirements of the proposed Military Specification and Military Standard, as a basis for possible revision of these documents and as a basis for suggesting new design criteria to the manufacturers of this type of valves.

2.0 Description of Test Specimens

Nine refrigerant pressure-actuated water flow regulating valves submitted by Quartermaster Research and Engineering Command were investigated. These valves were manufactured by the following companies:

1. Controls Company of America, Milwaukee, Wisconsin
2. Marsh Instrument Company of America, Skokie, Illinois
3. Penn Controls, Inc., Goshen, Indiana

The valve sizes (nominal N.P.T.), and manufacturer's identification were as follows:

Mfr.	Nominal Pipe Size, In.	Manufacturer's Identification
CC of A	3/8	Device 70365-108, Mod. 65A
	1/2	Device 70265-115, Mod. 65A
	3/4	Device 70265-116, Mod. 65A
Marsh	3/8	Type 56-30
	1/2	Type 56-30
	3/4	Type 56-30
Penn	3/8	Type 246P93AR, Mod. 2300
	1/2	Type 246P04AR, Mod. 2300
	3/4	Type 246P06AR, Mod. 2300

All of the valve specimens used a bellows-type power element responsive to condensing pressure of the refrigerant to modulate the degree of valve opening by opposing the closing action of an adjustable spring mechanism. The spring tension could be adjusted by turning an adjusting stem projecting from the spring assembly. The balance of forces between the spring, the refrigerant power element and inlet and outlet water pressures determined the position of the regulating valve disk.

The valves were intended for use with refrigerant R-12 and were 3/8 inch, 1/2 inch, and 3/4 inch in size, based on the female pipe threads at the inlet and outlet water openings. The screw-type spring adjustment and the bellows-type power element were at opposite ends of the valve body.

Each power element was provided with a small-bore soft-copper tube, commonly called a capillary tube, about 30 inches long with a 1/4-inch SAE flare nut for connection to the source of refrigerant condensing pressure. Each spring adjustment mechanism was provided with a suitable end to receive a simple adjusting tool, or could be positioned by hand. The valves were designed to be installed in any position. The rated refrigerant pressure ranges for the three makes were:

Marsh	60 to 270 psi
Penn	70 to 260 psi
CC of A	65 to 300 psi

The maximum exterior dimensions of the nine valves are listed in table 1. In this table the length is the distance from the face of the water inlet to water outlet boss, the height is the distance from the base of the power element capillary tube (not including the strain-relief device) to the end of the spring adjustment means, and the width is the exterior dimension at right angles to the length and height.

Table 1

Maximum Exterior Dimensions of Nine Water Valves, Inches

Valve Make	Nominal Size	Length	Height	Width	Capillary Tube Length
CC of A	3/8	2 9/16	5	1 5/8	32
	1/2	3 1/4	6 3/8	2 1/4	32
	3/4	3 1/4	6 1/4	2 1/4	32
Marsh	3/8	2 13/16	5 1/8	2	32
	1/2	3 3/16	5 5/8	2	31 1/2
	3/4	3 3/8	6 1/4	2 1/4	31
Penn	3/8	2 13/16	6	1 5/8	30
	1/2	3 3/16	6 5/16	2	30
	3/4	3 9/16	7 1/8	2 1/8	30 1/2

Figures 1, 2, and 3 show features of the three makes tested.

2.1 Construction of Controls Company of America Valves

Figure 1 is a photograph of the 3/8-, 1/2-, and 3/4-inch valves manufactured by the Controls Company of America. The 1/2-inch size has been disassembled to show the relation of the parts. The valve body is shown near the left center of Figure 1 and just below it is the main valve disk or sliding block. This block fits over two projections on the cross arm fastened to the push rod between the two rubber fabric diaphragms which are also fastened to the pushrod. This assembly is shown just below the sliding valve disk in Figure 1. To assemble, the push rod and one diaphragm are forced through the valve body so that there is a diaphragm above and below the valve body. The bellows-type power element is held tight against the diaphragm with four screws. On the top side, the retainer plate covers the diaphragm and in turn is covered with the cup, spring and cover forming the spring adjusting assembly. This assembly is attached to the valve body by four screws. Spring compression adjustment is accomplished by turning the square slotted stem projecting from the end of the spring retainer.

The sliding-block valve disk is held against the valve orifice or seat by the inlet water pressure and is held in alignment by the push rod guided in turn by the diaphragms and retaining plates.

The 3/8-inch valve has one valve orifice slot and the 1/2- and 3/4-inch sizes have two valve orifice slots, each sealed by one of two bars of the sliding-block valve disk when the valve is closed.

According to literature furnished by the manufacturer, the valve seat was stainless steel, the valve disk was called a "sliding graphitar block", and the valve body was cast brass.

2.2 Construction Details of Marsh Valves

Figure 2 is a photograph of the three Marsh valves with the 1/2-inch size disassembled to show the approximate relative position of the parts. Shown near the left center of the figure is the globe-type valve body. The seating surface of the replaceable valve seat is not visible because the body has been inverted to show the two slots in the body cavity that guide the two projections of the push-rod assembly pictured just above the valve body. The rubber fabric diaphragm with the hole in the center is held to the pushrod assembly by a machine screw and washer. The diaphragm is covered by the metal upper retainer covered in turn by the bellows-type power element containing the bellows pushrod. Four screws hold the power element to the valve body. Pictured below the valve body, the valve disk and disk retainer unit is screwed to the push rod assembly and is in contact with the solid rubber fabric diaphragm. The diaphragm is covered with the retainer and guide plate, and the spring push-plate assembly is inserted into the guide plate. The concentric inner and outer springs are placed over the rod of the push-plate and inserted in the adjusting-cap assembly attached to the valve body by four screws.

Spring compression adjustment is accomplished by rotating the knurled cap by hand or by wrench or screwdriver.

The valve seat appeared to be stainless steel and the valve disk was a rubber-type material inserted in a metal retainer.

The direction of water flow through the valve was such that it opposed the closing motion of the valve disk.

2.3 Construction Details of Penn Valves

Figure 3 shows the three Penn valves with the 1/2-inch size disassembled to illustrate the relation between parts. Just below the left center of the figure is the valve body and above it is the valve disk and retainer assembly. Two each of the four rubber

fabric type diaphragms cover the top and bottom openings, respectively, of the body and are held in place to make a watertight fit at each end of the valve-disk assembly by means of the long machine screw which extends through the entire pushrod assembly and screws into the hexagonal power-element pushrod shown below the valve body. The power-element assembly is held in place with four screws, and seals one pair of diaphragms against the valve body. The two diaphragms on the spring end are held in place by the pushrod guide plate which forms the base for the spring adjustment assembly, consisting of the spring push-plate, spring, and spring retainer. The spring retainer was held to the body by the four long screws. Spring adjustment is accomplished by turning the square stem projecting from the end of the spring retainer.

The replaceable valve seat appeared to be of brass and the valve disk was a rubber-type material in a brass retainer.

The indicated direction of water flow through the valve was such that the flow of water opposed the closing motion of the valve disk.

3.0 Description of Test Apparatus

The valve specimens were connected in turn to a water supply of suitable pressure and capacity with pressure-regulating valves, pressure gages, and valves to control and measure the performance characteristics as indicated diagrammatically in Figure 4. The discharge water from the test specimen was collected in a tank mounted on a platform scale. Stop watch readings were taken to determine average rates of water flow. Compressed dry nitrogen was used to provide the desired power element pressures.

Referring to Figure 4, the nipples, tees, and unions between the test specimen and the reducing couplings on the inlet and outlet sides of the specimen were of the same nominal pipe size as the threaded connections of the specimen itself. The distance between the two unions was kept constant at about 18 inches by selecting pipe nipples of different lengths for the various specimens. The pipe nipples between the unions and reducing couplings on either side of the test specimen were about one foot long. The pipe and fittings between the reducing couplings and the supply and discharge ends of the test apparatus were of nominal 2-inch size. The pressure gage on the inlet side was a calibrated Bourdon-type test gage with a range of 0 to 300 psi, a 6-inch dial, and 2 psi graduations, whereas that on the outlet side was a calibrated Bourdon-type gage with a range of 0 to 100 psi, a 4-inch dial, and one psi graduations.

The capillary tube to the power element of the test specimen was connected to a cylinder of dry nitrogen gas through a pressure-reducing valve. Pressure at the inlet to the capillary tube was measured with a gage of the same range and size as that at the water inlet to the test specimen. The pressure-reducing valve on the nitrogen cylinder was adjusted to maintain the desired pressure at the inlet to the capillary tube. A vent valve on the nitrogen line was used in some tests to waste a little nitrogen gas in order to maintain a floating condition in the pressure-reducing valve.

4.0 Test Procedure

Each specimen valve was similarly installed in the test apparatus as shown in Figure 4. Each valve was operated at three settings of the pressure-adjusting spring and three inlet water pressures for a total of nine tests for each valve. The three spring settings were:

1. Full retraction of spring adjustment mechanism.
2. A setting which produced an initial valve opening at about 80 psig nitrogen pressure on the power element, and 30 psig inlet water pressure.
3. A setting which produced an initial valve opening at 150 to 160 psig nitrogen pressure on the power element, in most cases, and 30 psig inlet water pressure.

The three inlet water pressures were:

1. 30 psig
2. 60 to 70 psig
3. 90 to 100 psig

In addition, the water flow rate was measured for a range of water pressure drops from 0 to above 80 psi with the valve set for an open position. For these tests, the open position was obtained by maintaining a nitrogen gas pressure of 300 psig in the power element and the spring adjustment was in the fully-retracted position.

The following procedure was used for obtaining the three spring adjustments described above. The spring adjustment device was first set in the fully-retracted position. The minimum gas pressure required to open the valve was determined by slowly increasing the nitrogen gas pressure, starting at 0 psig, and observing the pressure at the instant the inlet water pressure started to drop from an initial value of 30 psig. This pressure drop indication was made very sensitive by closing the shutoff valve on the inlet water line (see Fig. 4) with water at 30 psig pressure trapped between the inlet water shutoff valve and the test specimen. A similar procedure was used for the two higher settings of the spring adjustment. Several

adjustments of spring compression were usually required to obtain the desired nominal opening gas pressures of 80 and 160 psig. Once the desired opening pressure was obtained, the same spring compression setting was used for all tests at that nominal opening pressure.

At each of the three settings of the spring adjustment, water flow rate measurements were made at three inlet water pressures-- 30, 60 to 70, and as near to 100 psig as the laboratory supply could provide. While each of these inlet water pressures was maintained on the upstream side of the valve specimen, the outlet valve on the downstream side was manually adjusted to maintain a pressure of 10 psig at the outlet of the test specimen, so the pressure drop across the valve was 10 psi less than the entering water pressure at all times. This was done to test all the valves under similar conditions and to simulate, for each data point, the resistance to the flow of water of a condenser downstream of the test valve.

Each series of measurements at a particular inlet water pressure was conducted generally in the following manner. After determining the minimum nitrogen gas pressure required to open the valve under test, as described above, the gas pressure on the power element was raised an increment of 5 psi, the inlet water shutoff valve was opened, and the outlet water shutoff valve partially closed until 10 psig was indicated by the gauge on the outlet side of the test specimen. When the conditions had stabilized, a moving tare weight was obtained for the water receiving tank and the time required, not less than 2 minutes, to collect an even multiple of 100 pounds of water was recorded, using a stop watch. The stop watch was started as the tare weight tipped the scale arm and stopped when the collected water again tipped the scale arm, with the necessary added weights. The receiving tank was then drained while the gas pressure was increased an increment of 5 to 10 psi. The water pressures were adjusted again to the desired inlet and 10 psig outlet, and when stabilized, the weighing procedure was repeated. This procedure was repeated at selected increments of higher nitrogen pressure up to a maximum of 300 psig.

Water flow rates were then similarly determined for descending values of nitrogen pressure from 300 psig to a pressure which allowed the valve to close. The nitrogen pressure in the power element of the valve was reduced by manual operation of the vent valve shown in Figure 4. The water valve under test was considered closed at the final 5 psi incremental setting of nitrogen pressure at which no water flowed from the outlet pipe into the receiving tank after the previous higher increment at which a very slow flow was obtained. In most cases, this procedure resulted in a definite closing pressure determination within a 5 psi range.

5.0 Test Results

5.1 Water Flow Rate at Full-Open Valve Position

Water flow rates were determined for the nine valve specimens for a range of water pressure drops with the valve in the open position produced by a high power element pressure. The water flow rate is plotted in Figures 5 and 6 as the dependent variable against the pressure drop across the valve in the water circuit. In Figure 5, each group of three curves shows the performance of the three valve sizes from one manufacturer. In Figure 6, each group of three curves shows the performance of three similar-sized valves, one from each of the three manufacturers.

The water flow rate data shown in Figures 5 and 6 were obtained with the spring adjustment device fully-retracted and a nitrogen gas pressure on the power element of 300 psig. For most of the valves, this produced a maximum valve opening. For the 1/2-inch and 3/4-inch valves manufactured by Controls Company of America, the maximum water flow rate was obtained with a valve disk travel slightly less than the maximum possible travel of the disk pushrod assembly. The values plotted in Figures 5 and 6 for these two valves are the water flow rates obtained with the maximum travel of the valve disk pushrod.

Each value of water flow rate was determined by collecting, for a measured duration of about two minutes, the water discharged through the valve at the various inlet water pressures against a constant outlet water pressure of 10 psig. The water pressure drop across the valve was increased by increments of 10 psi to a maximum of about 100 psig water pressure at the inlet side of the valve.

Figure 5 indicates that the water flow rate curves for the different-sized valves of each manufacturer, were generally parabolic in shape and that water flow rate increased with valve size, other conditions being equal. It was determined that the water flow rate for each valve in the full-open position was essentially proportional to the square root of the water pressure drop across the valve.

Figure 6 provides a comparison of the water flow rates of valves of the same nominal pipe size from different manufacturers. It will be noted that the 1/2-inch size valves from the three manufacturers had almost the same capacity throughout the range of pressure drop employed in the tests. For example, at a water pressure drop of 75 psi the water flow capacity of the three 1/2-inch valves differed by only 10 percent, with the Penn valve having the highest capacity and the Controls Company of America valve, the lowest capacity.

In the 3/8-inch size, the water flow capacity of the three makes of valves differed widely. At a water pressure drop of 25 psi the highest capacity was 1.8 times the lowest, whereas at a water pressure drop of 75 psi this ratio was 2.1. In this size valve, the Penn valve also had the highest capacity and the Controls Company of America valve had the lowest capacity.

For the three 3/4-inch size valves, the ratio of the highest water flow capacity to the lowest capacity ranged from 1.24 at a water pressure drop of 25 psi to 1.41 at a pressure drop of 75 psi. In this size valve, the Marsh valve had the highest capacity and the Controls Company of America valve had the lowest capacity.

It appears that the single orifice opening in the 3/8-inch valve of the Controls Company of America did not permit water flow rates comparable to the 3/8-inch globe-type valves of the other two makes to the same degree that the double orifice opening permitted in the larger size valves.

5.2 Valve Performance for Modulated Water and Refrigerant Pressures

The water flow rates of each of the nine valve specimens for a range of refrigerant condensing pressure and at three settings of the spring adjustment mechanism are shown graphically in Figs. 7 to 15, inclusive. In these figures the water flow rate is plotted as a dependent variable against the refrigerant condensing pressure, produced by the pressure of dry nitrogen gas in the power element of the valve during these tests. In most cases the performance of the valves was determined for both increasing and decreasing values of refrigerant condensing pressure. The test procedures used in this investigation were developed during the course of the tests of the 3/8-inch valves, so the results on this size valve are not quite as comprehensive as those for the other sizes. Only the test results of the 3/8-inch valves that were obtained by procedures essentially similar to those finally used for the larger valves are shown in this group of figures.

In each of Figures 7 through 15, the observed data are presented in three groups of curves corresponding to the three settings of the spring adjustment mechanism, except in a few cases where the performance of the valve prevented completing the three series of tests. Different symbols were used in the figures to show the results for increasing and decreasing refrigerant condensing pressures, and also for different pressure drops across the valves.

The several performance features of the refrigerant-pressure-actuated water valves that were of interest in this investigation included the following:

(a) The ratio of the change in water flow rate to the change in refrigerant-condensing pressure in the central modulating range of the valve, from about 10% to 90% of the maximum water flow rate.

(b) The total change in refrigerant condensing pressure required from 90% maximum water flow rate to valve shutoff.

(c) The opening and closing characteristics of the valves, from shut-off to about 10% of the maximum water flow rate.

(d) The water flow versus refrigerant condensing pressure characteristics from 90% to 100% of the maximum water flow rate.

(e) The frictional characteristics of the movable elements of the valves, as indicated by the difference in water flow rate for identical increasing and decreasing values of refrigerant condensing pressure.

(f) The effect of supply water pressure on the performance of the valves.

An inspection of the curves in Figs. 7 to 15, inclusive, shows that the water flow rate was approximately proportional to the refrigerant condensing pressure throughout a broad central range of water flow rate, since the central portions of the curves approximate straight lines. This approximation appeared to be less exact for the 3/8-inch Marsh valve than for the other specimens, and appeared to be less exact for the lowest inlet water pressure than for the higher water pressures for several of the specimens.

For purposes of analysis and discussion, each of the performance curves in Figs. 7 to 15 can be divided into three sections, namely: the section from valve shutoff to 10 percent maximum flow rate, the section from 10 percent to 90 percent flow rate, and the section from 90 percent to 100 percent maximum flow rate. The first section indicates the opening and closing characteristics of the valve. The middle section comprises the operating range of the valve and is the section in which the water flow rate is most nearly proportional to the refrigerant condensing pressure. The third section reveals the operating characteristics of the valve near its wide open position. For most of the specimens tested, the change in refrigerant condensing pressure required to obtain the last 10 percent of the maximum water flow rate was much larger than that required for a similar increment of water flow rate in the working range. For this reason it is considered undesirable to rate a water regulating valve on the basis of its maximum water flow rate since it would require an abnormally high refrigerant condensing pressure to produce the maximum water flow rate.

5.2.1 Sensitivity Factor

As used in this report, the ratio of the change in water flow rate to the change in refrigerant condensing pressure in the operating range of a water regulating valve from 10 percent to 90 percent of the maximum water flow rate is defined as the 'sensitivity factor' for the valve. It represents the change in water flow rate in gallons per minute per unit change in refrigerant condensing pressure expressed in pounds per square inch. The sensitivity factor is directly related to the slope of the curves in Figs. 7 to 15 in the working range of the valves, that is, a steeper slope represents greater sensitivity and vice versa.

The water flow rates at 10%, 90% and 100% maximum flow rate, the corresponding refrigerant condensing pressures at these flow rates and at valve opening and closing, and the sensitivity factor for each valve specimen are summarized in Tables 2-4, inclusive. Study of these tables indicates that the sensitivity factor differed only slightly with the direction of change of the refrigerant condensing pressure, and only slightly for the range of refrigerant condensing pressure from the minimum setting to the high setting of the pressure adjusting spring.

Table 5 is a summary of the sensitivity factors for the nine valve specimens, in which all the values obtained for a given supply water pressure range were averaged for each valve. This table shows a variation of about 2 to 1 in the sensitivity factors of the different models of the 3/8-inch valves at a supply water pressure of 30 psig and a variation of nearly 3 to 1 for the same size valves at the highest supply water pressure. The sensitivity factors for the two larger size valves varied from 15 to 40% among the different models over the range of supply water pressures used. For any given valve specimen, the sensitivity factor tended to increase as the water supply pressure increased, but the rate of increase in sensitivity was not consistent among different specimens. The sensitivity factor was as low as 0.1 gpm per unit change in refrigerant condensing pressure for one make of 3/8-inch valve and ranged upward to about 1 gpm per unit change in refrigerant condensing pressure for all three 3/4-inch valves at the highest supply water pressure.

5.2.2 Refrigerant Condensing Pressure in Operating Range of Valves

The sensitivity factor for any given make of valve increased rather sharply with the size of the valve. This characteristic might be anticipated since the larger valves had larger water passages. However, the higher sensitivity factors for the larger valves does not necessarily indicate that the operating levels of refrigerant condensing pressure would be lower for larger capacity installations since these require a

TABLE 3

Water Flow and Condensing Pressure Characteristics of Three 1/2-inch Water Regulating Valves

Controls Company of America, 1/2-inch Valve

		Supply Water Pressure, psig		
		30	60 to 70	90 to 100
Maximum Water Flow Rate, Q,	gpm	18.7	31.3	36.0
90% Maximum	"	16.8	28.2	32.4
10% "	"	1.9	3.1	3.6
Δ Q (10 - 90% Flow)	"	14.9	25.1	28.8

		Setting of Pressure Adjusting Spring																	
		Min.		Med.		High		Min.		Med.		High		Min.		Med.		High	
		I*	D**	I	D	I	D	I	D	I	D	I	D	I	D	I	D	I	D
Refrig. Condensing Press at Max. Flow, P, psig		88	83	119	113	204	191	91	85	125	118	210	196	96	88	131	123	210	197
At 90% Maximum Flow	"	84	78	115	108	200	187	87	81	121	113	206	191	91	83	126	117	204	191
At 10% " "	"	49	44	85	78	166	152	49	41	83	74	166	151	51	38	84	73	162	143
At Closing of Valve	"	-	37	-	71	-	145	-	31	-	66	-	139	-	26	-	63	-	134
At Opening of Valve	"	42	-	78	-	158	-	39	-	77	-	160	-	38	-	76	-	159	-
Δ P (10 - 90% Flow)	"	35	34	30	30	34	35	38	40	38	39	40	40	40	45	42	44	42	48
Sensitivity Factor, Δ Q/ Δ P		.43	.44	.50	.50	.44	.43	.66	.63	.66	.64	.63	.63	.72	.64	.69	.65	.69	.60

Marsh Instrument Company, 1/2-inch Valve

		Supply Water Pressure, psig		
		30	60 to 70	90 to 100
Maximum Water Flow Rate, Q,	gpm	18.0	32.0	38.0
90% Maximum	"	16.2	28.8	34.2
10% "	"	1.8	3.2	3.8
Δ Q (10 - 90% Flow)	"	14.4	25.6	30.4

		Setting of Pressure Adjusting Spring																	
		Min.		Med.		High		Min.		Med.		High		Min.		Med.		High	
		I*	D**	I	D	I	D	I	D	I	D	I	D	I	D	I	D	I	D
Refrig. Condensing Press at Max. Flow, P, psig		71	71	190	168			92	92	200	200					202	202		
At 90% Maximum Flow	"	55	55	148	144			66	66	155	155					163	163		
At 10% " "	"	20	20	99	100			22	22	101	101					102	102		
At Closing of Valve	"	-	-	-	80			-	-	-	80					-	76		
At Opening of Valve	"	10	-	80	-			10	-	80	-					78	-		
Δ P (10 - 90% Flow)	"	35	35	49	44			44	44	54	54					61	61		
Sensitivity Factor, Δ Q/ Δ P		.41	.41	.29	.33			.58	.58	.47	.47					.50	.50		

Penn Controls Inc., 1/2-inch Valve

		Supply Water Pressure, psig		
		30	60 to 70	90 to 100
Maximum Water Flow Rate, Q,	gpm	19.4	33.5	39.0
90% Maximum	"	17.5	30.1	35.1
10% "	"	1.9	3.4	3.9
Δ Q (10 - 90% Flow)	"	15.6	26.7	31.2

		Setting of Pressure Adjusting Spring																	
		Min.		Med.		High		Min.		Med.		High		Min.		Med.		High	
		I*	D**	I	D	I	D	I	D	I	D	I	D	I	D	I	D	I	D
Refrig. Condensing Press at Max. Flow, P, psig		130	120	172	162	250	250	123	150	190	170	260	250	180	150	200	200	260	255
At 90% Maximum Flow	"	90	84	138	134	214	209	92	84	137	135	216	209	92	87	138	131	219	214
At 10% " "	"	40	38	89	89	164	163	37	35	86	86	162	161	37	35	82	81	164	160
At Closing of Valve	"	-	34	-	-	-	153	-	27	-	75	-	149	-	23	-	70	-	147
At Opening of Valve	"	34	-	83	-	158	-	28	-	75	-	150	-	24	-	74	-	159	-
Δ P (10 - 90% Flow)	"	50	46	49	45	50	46	55	49	51	49	54	48	55	52	56	50	55	54
Sensitivity Factor, Δ Q/ Δ P		.31	.34	[illegible]	.35	.31	.34	.49	.54	[illegible]	[illegible]	.49	[illegible]	.57	.60	.56	[illegible]	.57	.58

* Increasing Refrigerant Condensing Pressure
**Decreasing Refrigerant Condensing Pressure

TABLE 4

Water Flow and Condensing Pressure Characteristics of Three 3/4-inch Water Regulating Valves

Controls Company of America, 3/4-inch Valve

		Supply Water Pressure, psig: 30	60 to 70	90 to 100
Maximum Water Flow Rate, Q,	gpm	26.7	43.7	50.8
90% Maximum	"	24.0	39.3	45.7
10% "	"	2.7	4.4	5.1
Δ Q (10 - 90% Flow)	"	21.3	34.9	40.6

Setting of Pressure Adjusting Spring

	30: Min.		Med.		High		60 to 70: Min.		Med.		High		90 to 100: Min.		Med.		High	
	I*	D**	I	D	I	D	I	D	I	D	I	D	I	D	I	D	I	D
Refrig. Condensing Press at Max. Flow, P, psig.	58	56	127	120	202	189	99	92	132	123	208	194	102	94	135	126	212	196
At 90% Maximum Flow "	55	52	124	117	198	186	95	87	128	119	203	190	97	89	130	121	207	191
At 10% " " "	25	22	91	85	164	152	57	45	91	80	162	146	56	44	93	76	164	142
At Closing of Valve "	-	14	-	77	-	140	-	36	-	74	-	136	-	36	-	66	-	131
At Opening of Valve "	22	-	80	-	156	-	50	-	84	-	150	-	49	-	80	-	155	-
Δ P (10 - 90% Flow) "	30	30	33	32	34	34	38	42	37	39	41	44	41	45	37	45	43	49
Sensitivity Factor, Δ Q/ Δ P	.71	.71	.65	.67	.63	.63	.92	.83	.94	.89	.85	.79	.99	.90	1.10	.90	.94	.83

Marsh Instrument Company, 3/4-inch Valve

		Supply Water Pressure, psig: 30	60 to 70	90 to 100
Maximum Water Flow Rate, Q,	gpm	32.0	56.8	66.0
90% Maximum	"	28.8	51.1	59.4
10% "	"	3.2	5.7	6.6
Δ Q (10 - 90% Flow)	"	25.6	45.4	52.8

Setting of Pressure Adjusting Spring

	30: Min.		Med.		High		60 to 70: Min.		Med.		High		90 to 100: Min.		Med.		High	
	I*	D**	I	D	I	D	I	D	I	D	I	D	I	D	I	D	I	D
Refrig. Condensing Press at Max. Flow, P, psig	260	230					220	220					215	200				
At 90% Maximum Flow "	106	101					111	109					114	113				
At 10% " " "	61	59					61	59					60	-				
At Closing of Valve "	-	54					-	50					-	-				
At Opening of Valve "	54	-					50	-					50	-				
Δ P (10 - 90% Flow) "	45	42					50	50					54	-				
Sensitivity Factor, Δ Q/Δ P	.57	.61					.91	.91					.98					

Penn Controls Inc., 3/4-inch Valve

		Supply Water Pressure, psig: 30	60 to 70	90 to 100
Maximum Water Flow Rate, Q,	gpm	29.1	50.0	60.0
90% Maximum	"	26.2	45.0	54.0
10% "	"	2.9	5.0	6.0
ΔQ (10 - 90% Flow)	"	23.3	40.0	48.0

Setting of Pressure Adjusting Spring

	30: Min.		Med.		High		60 to 70: Min.		Med.		High		90 to 100: Min.		Med.		High	
	I*	D**	I	D	I	D	I	D	I	D	I	D	I	D	I	D	I	D
Refrig. Condensing Press at Max Flow, P, psig	120	120	165	165	255	205	58	85	140	140	220	201	63	61	137	137	231	227
At 90% Maximum Flow "	45	45	127	121	192	184	46	44	126	124	191	187	53	50	128	126	217	213
At 10% " " "	12	12	89	88	155	151	8	7	86	84	148	146	11	11	84	84	173	167
At Closing of Valve "	-	5	-	83	-	145	-	2	-	77	-	140	-	4	-	75	-	155
At Opening of Valve "	10	-	85	-	150	-	3	-	79	-	142	-	4	-	76	-	160	-
Δ P (10 - 90% Flow) "	33	33	38	33	37	33	38	37	40	40	43	41	42	39	44	42	44	46
Sensitivity Factor, Δ Q/Δ P	.71	.71	.61	.71	.63	.71	1.05	1.08	1.00	1.00	.93	.97	1.14	1.23	1.09	1.14	1.09	1.04

* Increasing Refrigerant Condensing Pressure
**Decreasing Refrigerant Condensing Pressure

Table 5

Sensitivity Factor of Water Regulating Valves

Valve Manufacturer	Supply Water Pressure, psig								
	30			60 to 70			90 to 100		
	Nominal Valve Size, in.								
	3/8	1/2	3/4	3/8	1/2	3/4	3/8	1/2	3/4
Controls Company of America	.11	.46	.67	.14	.64	.87	.14	.67	.96
Marsh Instrument Co.	.16	.36	.59	.19	.53	.91	.23	.50	.98
Penn Controls Inc.	.24	.33	.68	.35	.52	1.01	.40	.58	1.12

Table 6

Decrease in Refrigerant Pressure* Required from 90% Maximum Flow Rate to Valve Shutoff, psi

Valve Manufacturer	Supply Water Pressure, psig								
	30			60 to 70			90 to 100		
	Nominal Valve Size, in.								
	3/8	1/2	3/4	3/8	1/2	3/4	3/8	1/2	3/4
Controls Company of America	60	37	40	61	47	45	87	54	55
Marsh Instrument Co.	60	64	-	74	75	-	92	87	-
Penn Controls Inc.	40	55	38	47	60	47	60	61	51

*Based on data taken at a valve opening pressure of 80 psig.

higher water flow rate. Table 6 summarizes the decrease in refrigerant condensing pressure to change the water flow rate from 90 percent of the maximum rate to valve shutoff. The values in Table 6 were taken from Tables 2-4, inclusive, from the data observed at the medium setting of the pressure adjusting spring, that is, a valve opening pressure of about 80 psig; and for decreasing values of refrigerant condensing pressure in most cases. Table 6 shows a considerable variation in the refrigerant pressure change required to move the valve from 90 percent maximum flow rate to the closed position. This pressure increment was higher for the Marsh valves than for the other two makes in 5 out of the 6 cases for which comparisons could be made. This pressure increment was significantly higher for the 3/8-inch size valve of the Controls Company of America than for the other two sizes of the same make; whereas this pressure increment was highest in the 1/2-inch size of the Penn valves. Reference to Tables 2-4 inclusive show that the refrigerant condensing pressure at 90 percent maximum flow rate would range from 108 psig to 177 psig for eight of the nine valve specimens on which data were observed when they were set for an opening refrigerant condensing pressure of 80 psig.

5.2.3 Opening and Closing Characteristics

It was observed during the tests that the refrigerant condensing pressure had to be reduced to a somewhat lower value to completely stop the water flow in some instances than that required to initiate water flow as the refrigerant condensing pressure was rising. This characteristic is illustrated for the 1/2-inch and 3/4-inch valves of each of the three manufacturers by the plotted data at zero water flow rate in Figs. 8 and 9, 11 and 12, and 14 and 15. A summary of the difference between the opening and closing pressures for the 1/2-inch and 3/4-inch valves is reported in Table 7.

Table 7 indicates that the difference between the opening and closing pressures for the two valves manufactured by Controls Company of America was significantly greater than for the other two valves. The number of possible comparisons with the Marsh valves was limited because malfunctioning of these valves caused early termination of the tests. The valve action in the Controls Company of America specimens was obtained by a slotted movable block sliding over a slotted stainless steel seat with the supply water pressure holding the block against the seat. In the Marsh and Penn valves the supply water exerted forces opposing both the spring pressure and the power element pressure but did not involve sliding further between the moving valve element and the seat. The difference in construction of the valves may account for the variations in opening and closing performance shown in table 7.

Table 7

Difference between Opening and Closing Pressures of 1/2-inch and 3/4-inch Valves, psi

Valve Manufacturer	Supply Water Pressure, psig								
	30			60 to 70			90 to 100		
	Setting of Pressure Adjusting Spring								
	Min.	Med.	High	Min.	Med.	High	Min.	Med.	High
	1/2-inch Valves								
Controls Company of America	5	7	13	8	11	21	12	13	25
Marsh Instrument Co.	-	0	-	-	0	-	-	2	-
Penn Controls Inc.	0	-	5	1	0	1	1	4	12
	3/4-inch Valves								
Controls Company of America	8	3	16	14	10	14	13	14	24
Marsh Instrument Co.	0	-	-	0	-	-	-	-	-
Penn Controls Inc.	5	2	5	1	2	2	0	1	5

The characteristic curves showing the relation between water flow rate and refrigerant condensing pressure near shutoff exhibited three distinct shapes as illustrated most clearly in Figs. 7, 10 and 13, although some other figures have similarities to these. In three of the curves in Fig. 7 a definite discontinuity in the characteristic curves occurred at water flow rates of about 1, 2, and 2 1/2 gpm, respectively. The valve was less sensitive to change in refrigerant condensing pressure below these flow rates in these three instances, although a similar phenomenon was not evident in every test of this valve.

In Figure 10 the characteristic curve flattened out significantly near shutoff for several of the tests. In these instances this 3/8-inch Marsh valve permitted a trickle of water to pass until the refrigerant condensing pressure was lowered well below the value that would be obtained by extrapolating the main portion of the characteristic curve. The reason for this performance was not determined, but it could be caused by foreign matter on the valve seat or a slight misalignment of the moving elements.

In Fig. 13 the slope of the characteristic curve was maintained essentially constant down to shutoff. Similar results were observed in some tests of the 3/4-inch Penn valve, as illustrated in Fig. 15.

The shape of the characteristic curve near shutoff is important with respect to water leakage through the valve when the refrigerant unit is not running. If a water regulating valve has a flattened characteristic curve near shutoff, the pressure adjustment spring must be set at a higher value for any given water temperature to prevent water leakage when the unit is not running. This causes the refrigerating unit to operate at a higher compression ratio and less efficiently than would be necessary if the water regulating valve had a steep pressure curve at shutoff.

5.2.4 Maximum Flow Characteristics

The characteristic curves in Figs. 7-15 inclusive, showing the relation of water flow rate to refrigerant condensing pressure, exhibited three different shapes in the range of flow rate from 90 to 100 percent of the maximum rate. The most frequent shape was a gradual decrease of slope of the curve toward a horizontal line representing the maximum water flow rate, as illustrated by Figs. 7, 10 and 13. Illustrating a second characteristic shape, the water flow rate for the 1/2-inch and 3/4-inch valves submitted by the Controls Company of America reached a maximum value for an intermediate value of refrigerant condensing pressure and then decreased 10 percent or a little more at higher condensing pressures as shown in Figs. 8 and 9. This effect was apparently caused by an overtravel of the sliding valve element past the position of best alignment of the openings in the valve seat and the moving element at high values of refrigerant condensing pressure. Figure 15 illustrates the third shape, a fairly abrupt change from the proportional part of the performance curve to the horizontal line representing maximum water flow rate.

The type of valve performance illustrated in Fig. 15 is probably the most desirable performance near the maximum flow rate, because the valve could be operated up to 100 percent of the maximum flow rate with very little decrease in sensitivity. The 1/2-inch and 3/4-inch valves of the Controls Company of America were also quite sensitive up to the maximum flow rate, as shown in Figs. 8 and 9. However, if the refrigerant condensing pressure continued to rise above the level corresponding to the maximum water flow rate in a system equipped with this type valve, the water flow rate would be reduced by 10 percent or more at the condition of greatest cooling requirement, and the system would be unnecessarily penalized in operating efficiency.

It appears undesirable to operate valves with characteristic curves similar to those shown in Figs. 7, 10 and 13 at the maximum water flow rate because of the large increase in refrigerant condensing pressure required to attain the last 10 percent of the maximum flow rate. Table 8 summarizes the increase in refrigerant condensing pressure required to raise the water flow rate from 90 to 100 percent of the maximum value. These data were taken from Tables 2-4 and Figs. 7-15, inclusive, and show an average value for each water supply pressure and each setting of the pressure adjusting spring.

Table 8 shows that the 1/2-inch and 3/4-inch valves submitted by Controls Company of America and the 3/4-inch Penn valve for water supply pressures above 60 psig required the smallest increase in refrigerant condensing pressure to increase the water flow rate from 90 to 100 percent of the maximum value. The values were in the range from 8 to 20 psig in most instances for these three cases. Table 8

Table 8

Increase in Refrigerant Condensing Pressure Required to Increase Water Flow Rate from 90 to 100 Percent of Maximum Rate, psi

	Supply Water Pressure, psig								
	30			60 to 70			90 to 100		
	Setting of Pressure Adjusting Spring								
	Min.	Med.	High	Min.	Med.	High	Min.	Med.	High
	3/8-inch Valves								
Controls Company of America	24	20	-	22	20	37	22	58	-
Marsh Instrument Co.	-	20	28	53	52	-	109	-	-
Penn Controls Inc.	53	36	-	53	27	-	26	32	42
	1/2-inch Valves								
Controls Company of America*	9	10	13	11	10	9	13	11	13
Marsh Instrument Co.	16	33	-	26	45	-	-	39	-
Penn Controls Inc.	38	31	39	49	44	43	76	66	41
	3/4-inch Valves								
Controls Company of America*	10	8	10	8	9	9	9	9	13
Marsh Instrument Co.	42	-	-	110	-	-	95	-	-
Penn Controls Inc.	75	41	42	27	15	22	11	10	14

*These data based on water flow rate at optimum alignment of valve parts.

shows several instances where the required increase in condensing pressure was in the range from 60 to 110 psig. An examination of Figs. 7-15 shows that these unusually high values corresponded to situations where substantially 100 percent maximum flow rate was obtained with a more moderate rise in refrigerant condensing pressure, but there continued to be a slight increase in water flow rate for greater increases in condensing pressure, perhaps caused by an increased elastic deformation of some of the moving parts other than the springs. Most of the values of pressure increase in Table 8 for all the 3/8-inch valves and for the 1/2-inch Marsh and Penn valves were in the range from 20 to 55 psig.

5.2.5 Hysteresis on Reversal of Refrigerant Condensing Pressure

Data were obtained on each of the 1/2-inch and 3/4-inch valves for both increasing and decreasing values of the refrigerant condensing pressure in the range from shutoff to maximum water flow rate. The results for increasing and decreasing refrigerant condensing pressure are differentiated in Figs. 8, 9, 11, 12, 14 and 15 by using open and closed symbols, and are also tabulated separately in Tables 2-4, inclusive.

An examination of the data in Tables 3 and 4 shows that the sensitivity factor and the change in refrigerant condensing pressure required to change the water flow rate from 10 to 90 percent of the maximum flow rate was comparable for increasing and decreasing values of refrigerant condensing pressure. The variation in sensitivity factor and ΔP (10 - 90% flow) for increasing and decreasing refrigerant condensing pressures was 10 percent or greater in 10 cases out of 43 cases where comparisons were possible, and was less than 10 percent for the other 33 test conditions.

An examination of the difference in refrigerant condensing pressure at 90 percent maximum water flow rate for increasing and decreasing values of refrigerant condensing pressure in Tables 3 and 4 shows that this difference was negligible for the 1/2-inch and 3/4-inch Marsh valves, it averaged from 4 to 6 psig for the two Penn valves of the same nominal size, and it averaged 9 to 10 psig for the same size valves submitted by Controls Company of America. These pressure differences represent the smallest change in refrigerant condensing pressure, produced by a change in refrigerating load or a change in the condensing water temperature, that would cause a readjustment of the valve position and a change in water flow rate.

The greater hysteresis in the valves of the Controls Company of America may have been due to the design of the valves in which sliding friction between the valve seat and moving element was involved in

every change in valve position, whereas in the other two makes the valve was of a globe type in which the turbulence of the water flow would help to overcome static friction in repositioning the moving valve element. The importance of the hysteresis may be judged by the fact that the product of the pressure change required to change the valve position and the sensitivity factor for the valve indicates how much the water flow rate may depart from a design value. For example, a valve with a pressure hysteresis of 10 psig and a sensitivity factor of 0.5 could be using 5 gpm more water than required for limited periods of time when the refrigerant condensing pressure was fluctuating.

5.2.6 Effect of Supply Water Pressure on Valve Performance

A review of Table 5 shows that the sensitivity factor of all the valve specimens increased for increased supply water pressure, but not always throughout the whole range of water pressure. The decrease in refrigerant condensing pressure required to change the water flow rate from 90 percent maximum to shutoff also increased somewhat with increasing water supply pressure as shown in Table 6. The difference between opening and closing pressures for the 1/2-inch and 3/4-inch valves of the Controls Company of America increased with supply water pressure in most instances, and for the Penn valves at some test conditions.

The sensitivity factor increases with water supply pressure because the outlet water pressure was maintained constant for the tests and, as the data in tables 2-4 show, the refrigerant condensing pressure required to obtain 90 percent maximum flow rate was only slightly affected by supply water pressure. Thus the numerator of the ratio $\Delta Q/\Delta P$, defining the sensitivity factor, increased with increasing supply water pressure while the denominator remained nearly constant.

The greater changes in refrigerant condensing pressure required to move the valve between two fixed positions as the supply water pressure was increased are probably related to the greater water pressure difference across the moving element of the valves, which caused greater sliding friction in the case of the specimens submitted by the Controls Company of America and greater direct resistance to closure of the other two makes of valves.

6.0 Comparison of Valve Performance with Military Standard Requirements

The requirements in the proposed Military Standard, summarized in the Introduction, are interpreted to mean that water-regulating valves for refrigerant condensers of 3/8-inch, 1/2-inch, and 3/4-inch pipe size should provide water flow rates of 6, 10, and 25 gpm, respectively, with an increase in refrigerant condensing pressure not to exceed 25 psi above a valve-opening condensing pressure of 101.55 psig for a water pressure drop of 25 psi between valve inlet and outlet.

The maximum water flow rate corresponding to a water pressure drop of 25 psi between inlet and outlet of the test specimens can be determined from Figs. 5 or 6. These values are summarized in Table 9 for comparison with the requirements stated in the proposed Military Standard. It will be seen in Table 9 that the maximum water flow rates ranged from 125% to more than 200% of the requirement for the 3/8-inch valves, averaged about 200 percent of the requirement for the 1/2-inch valves, and ranged from about 115 percent to 150 percent for the 3/4-inch valves.

If the valves were rated at 90 percent of the maximum flow rate to avoid low sensitivity near the wide open position, as discussed earlier in this report, three of the specimens would have less than 17 percent excess capacity and the other six specimens would have from 30 to 100 percent excess capacity. The water flow rates corresponding to 90% maximum are also shown in Table 9 for comparison with the proposed Military Standard requirements.

The rise in refrigerant condensing pressure required to obtain the water flow rates specified in the proposed Military Standard for the various size valves with a water pressure drop of 25 psi is shown in Table 10. These valves were obtained from the performance curves in Figs. 7-15 by interpolating between the data for the low and medium values of water supply pressure and between the medium and high settings of the pressure adjusting spring. It was found that the rise in refrigerant condensing pressure required to obtain a given percentage of maximum water flow rate, i.e. a given fraction of maximum valve opening, varied only a little with water supply pressure, and with the setting of the pressure adjustment spring. Thus, the interpolation involved only small deviations from observed data. In a few cases, extrapolation was necessary because the observed data did not bracket the specified water pressure difference of 25 psi or the specified valve opening pressure of 101.55 psig.

Table 9

Comparison* of Observed Water Flow Rates with Military Standard Requirements

Valve Size	Military Standard Water Flow Requirement	Maximum Observed Water Flow Rate			Percent Max Flow Rate to meet Mil. Std			90% Maximum Observed Water Flow Rate		
		CC of A	Marsh	Penn	CC of A	Marsh	Penn	CC of A	Marsh	Penn
in.	gpm	gpm	gpm	gpm	%	%	%	gpm	gpm	gpm
3/8	6	7.5	10.5	13.4	80	57	45	6.8	9.5	12.1
1/2	10	20.3	20.3	21.5	49	49	47	18.3	18.3	19.4
3/4	25	29.0	36.0	32.5	86	69	77	26.1	32.4	29.3

*Based on water pressure difference of 25 psi between inlet and outlet of valve.

Table 10

Rise in Refrigerant Condensing Pressure Required to Obtain Selected Water Flow Rates*

Valve Size	Military Standard Water Flow Requirement	Rise in Refrig. Condensing Pressure to Attain Required Water Flow Rate			Rise in Refrig. Condensing Pressure to Attain 90% Maximum		
		CC of A	Marsh	Penn	CC of A	Marsh	Penn
in.	gpm	psi	psi	psi	psi	psi	psi
3/8	6	59	32	16	67	62	42
1/2	10	26	40	22	39	69	55
3/4	25	43	52	35	44	70	43

*Based on water pressure difference of 25 psi between inlet and outlet of valve and opening refrigerant condensing pressure of 101.55 psig.

The data in Table 10 indicate that the rise in refrigerant condensing pressure required to produce the specified water flow rates was 26 psi or less for three of the nine valve specimens, and that it ranged from 32 to 59 psi for the other six specimens. Thus, for an opening pressure of 101.55 psig, the operating level of condensing pressure would range from about 118 to 161 psig for the various valve specimens at the required water flow rate.

The rise in refrigerant condensing pressure required to obtain 90 percent of the maximum water flow rate for each valve with a water pressure difference of 25 psi and a valve opening pressure of 101.55 psig is also shown in Table 10. These pressure increments ranged from 39 to 70 psi for the nine valves, indicating that undesirably high operating pressures would occur in attempting to use 90 percent of the maximum capacity of the valves.

7.0 Mechanical Performance of Valve Specimens

Some difficulties with chattering and water leakage developed in two sizes of the Marsh valves during the tests.

Figure 11 shows the results obtained with the 1/2-inch valve manufactured by Marsh Instrument Co. The first test runs were made with a nominal valve opening pressure of 80 psig and with supply water pressures of 30, 70, and 98 psig, followed in turn by a series with minimum valve opening pressure, with the same three supply water pressures. At the test conditions indicated by "A" in Figure 11 with a minimum valve opening pressure and increasing values of refrigerant condensing pressure, severe chattering of the valve occurred which caused a water leak to develop at one edge of the diaphragm under the power element assembly and water was discharged to the atmosphere. The tests of this valve were terminated at this time.

Figure 12 shows the results obtained with two specimens of the 3/4-inch water-regulating valve manufactured by Marsh Instrument Co. Both specimens developed water leaks after severe chattering in similar fashion to that described for the 1/2-inch valve from the same manufacturer.

Test runs at increasing and decreasing refrigerant pressures were made with the 3/4-inch valve submitted by the Quartermaster Research and Engineering Command at the minimum opening refrigerant pressure and at water supply pressures of 30, 70, and 93 psig. While the refrigerant condensing pressure was being decreased during the run at 93 psig supply water pressure, at the conditions indicated by "A" in Figure 12, the valve chattered causing a water leak to develop. No further capacity tests were made of this valve. Another valve of the

same make and model was obtained from a commercial supplier in Washington, D.C. During a test run at 98 psig supply water pressure with the valve set to open at 80 psig refrigerant condensing pressure, this valve chattered and developed a water leak at the conditions indicated by "B" in Figure 12. No further capacity tests were made with this valve. The curve for the second 3/4-inch valve extends above the graph in Figure 12. It essentially leveled off at a water flow rate of 68 gpm at a refrigerant pressure of 250 psig.

It is not known what caused these three valves to chatter and develop water leaks in this manner. The chattering apparently produced dynamic pressures which forced water through the diaphragm seals at an edge and at the valve stem permitting water to leak out at the diaphragm seal between the power element assembly and the valve body. The body screws were checked for tightness before each test, but not the screw and washer on the valve stem. No mechanical damage was evident when the valves were disassembled after testing. The chattering occurred both with and without a booster pump in operation and only at supply water pressures in excess of 80 psig.

8.0 Discussion and Conclusions

The tests described in this report were made to explore the performance characteristics of pressure-actuated valves designed to regulate the flow of cooling water through refrigerant condensers. Commercial valves were used as test specimens, but the tests were not intended to be acceptance tests for commercial products.

Supply water pressures of about 30, 70, and 90 psig and valve opening pressures of minimum, 80, and 160 psig were selected for the tests to bracket the probable range of application and use. A constant water pressure at the valve outlet of 10 psig was selected to simulate the pressure drop of a condenser and associated piping downstream of the valve, although it was recognized that the pressure drop in any given condenser system would increase approximately as the square of the water flow rate. The inlet water pressure was also kept constant at all water flow rates although this condition would only be approximated in actual use.

The investigation indicated that the following characteristics of a pressure-actuated water-regulating valve were of importance in meeting the functional purposes of such a valve:

(1) The maximum water flow rate of the valve versus water pressure drop through the valve.

(2) The sensitivity factor of the valve in the working range of water flow rate. The sensitivity factor determines the total change in refrigerant condensing pressure required to move the valve element from a closed position to that of design water flow rate.

(3) The tendency of the valve to leak water at refrigerant condensing pressures below the opening pressure.

(4) The decrease in sensitivity of the valve near the wide open position.

(5) Hysteresis in the moving valve element for increasing and decreasing condensing pressures.

(6) The effect of supply water pressure and setting of the opening pressure adjustment on sensitivity, hysteresis, and shutoff characteristics.

(7) The mechanical stability of the moving valve element against chattering.

(8) The possibility of overtravel of the moving valve element past the position of optimum orifice alignment.

The series of tests on nine commercial valve specimens in three nominal pipe sizes indicated the following conclusions:

(a) The pipe thread size of a valve is not a good indication of its maximum water flow rate. In the 3/8-inch valves the maximum water flow rate varied in the ratio of about 2 to 1, whereas the maximum water flow rates of the three 1/2-inch valves differed by only about 10 percent.

(b) The sensitivity factor of the valves varied widely. The water flow rate and maximum rise in refrigerant condensing pressure specified in the proposed Military Standard indicated the following sensitivity factors between valve opening and specified water flow:

Nominal Valve Size, in.	Sensitivity Factor $\Delta Q/\Delta P$
3/8	0.24
1/2	0.40
3/4	1.00

Only one of the 3/8-inch valves and two of the 1/2-inch valves attained these average values of the sensitivity factor from valve opening to the specified water flow rate as indicated in Table 10. The test results indicated that the operating level of condensing pressure would range from 118 to 161 psig for the several valves at the water flow rates specified in the proposed Military Standard.

The sensitivity of a valve can be increased by increasing the length or diameter of the cylindrical spring, decreasing the diameter of the spring wire, or increasing the diameter of the power element. Economic considerations, mechanical stability, and range of adjustment need to be taken into account in selecting satisfactory combinations of these variables.

(c) Some valves exhibited a sharp shutoff of the water flow at a condensing pressure close to the opening pressure and others required a pressure 10 psi or more below the opening pressure to stop the water flow. The difference between the opening and closing pressures appeared to be significantly greater for valves with a sliding valve port than for the globe type valve element. A valve without a sharp shutoff of water flow would have to be operated at a higher opening pressure, or else it would permit some water leakage during the time the refrigerating unit was not running.

(d) Six of the nine valve specimens revealed a gradual decrease in sensitivity factor as the maximum water flow rate was approached. This characteristic suggests that it might be desirable to rate water-regulating valves on the basis of 90 percent maximum water flow rate, since the operation conditions of a refrigerating unit at maximum water flow rate would be considerably less efficient because of high condensing pressures.

(e) The difference in refrigerant condensing pressure at 90 percent maximum water flow rate for increasing and decreasing values of condensing pressure ranged from a negligibly small value for some valves to about 10 psi for the type with a sliding valve element. This hysteresis effect could result in excessive water flow rates under fluctuating load conditions or fluctuating condenser water temperatures.

(f) An increase in supply water pressure increased the sensitivity factor of the valve specimens, and increased the decrement of refrigerant condensing pressure below the opening pressure required to shut off water flow, but it did not have a significant effect on the hysteresis of the valves in the operating range of water flow rate.

(g) Examples of water-regulating valves that exhibited chattering of the moving element and overtravel of the moving element beyond the position of best port alignment were found among the test specimens. These are illustrations of design factors that are of importance to the proper functioning of water-regulating valves.

(h) The operating level of condensing pressure could be lowered for any given cooling load by using an oversized water-regulating valve. Table 11 shows the rise in refrigerant condensing pressure that would be required to attain a 6 gpm flow rate through the 1/2-inch valves and a 10 gpm flow rate through the 3/4-inch valves. The results show that the valves manufactured by the Controls Company of America and Penn Controls Inc. would operate with a pressure rise of less than 25 psi. The pressure rises for the Marsh Instrument Company valves are higher than the others, principally because the pressure-flow curves flatten out significantly near the shutoff condition, as shown in Figs. 11 and 12. However, it might be more practical to redesign the spring and power elements to obtain a higher sensitivity factor for the valves and make fuller use of the total flow capacity of the valves.

Table 11

Rise in Refrigerant Condensing Pressure Required to Obtain Selected Water Flow Rates with Oversized Valves

Valve Size in.	Required Water Flow Rate gpm	Rise in Refrig. Condensing Pressure to Attain Required Water Flow Rate, psi		
		CC of A	Marsh	Penn
1/2	6	17	30	15
3/4	10	23	44	15

9.0 Recommendations for Specification Requirements

In the preparation of performance requirements and tests in future Military standards and specifications, this investigation indicates that the following factors should be considered:

1. Valve sizes should be based on desired water flow rate at rating conditions with type and size of pipe connections a secondary consideration based on application.

2. The rating conditions should include:

   (a) Valve opening pressure,
   (b) Maximum rise in condensing pressure above opening pressure to attain required water flow rate,
   (c) Inlet water pressure and temperature,
   (d) Pressure drop across the valve.

3. The water flow capacity of a valve should be based on 90 percent of the maximum flow rate to avoid use of the zone of low sensitivity likely to occur near the wide-open position of the valve.

4. Maximum hysteresis effect for increasing and decreasing refrigerant pressures at rated water flow conditions should be specified. A maximum value of 5 psi is probably reasonable.

5. Maximum difference between opening and closing pressures, at the rating conditions listed in item 2, should be specified. A maximum value of 5 psi is probably reasonable for globe-type valves.

6. The range of conditions under which valve stability, and freedom from chatter, resonance, and water hammer is required.

7. The required working range of water and refrigerant pressures should be specified.

8. Safety requirements for strength and freedom from water leakage under working and test pressures should be included.

9. Sampling and test procedures should be included or referenced.

10. The requirements of the Military Standard should be referenced in the specification, if they are applicable.

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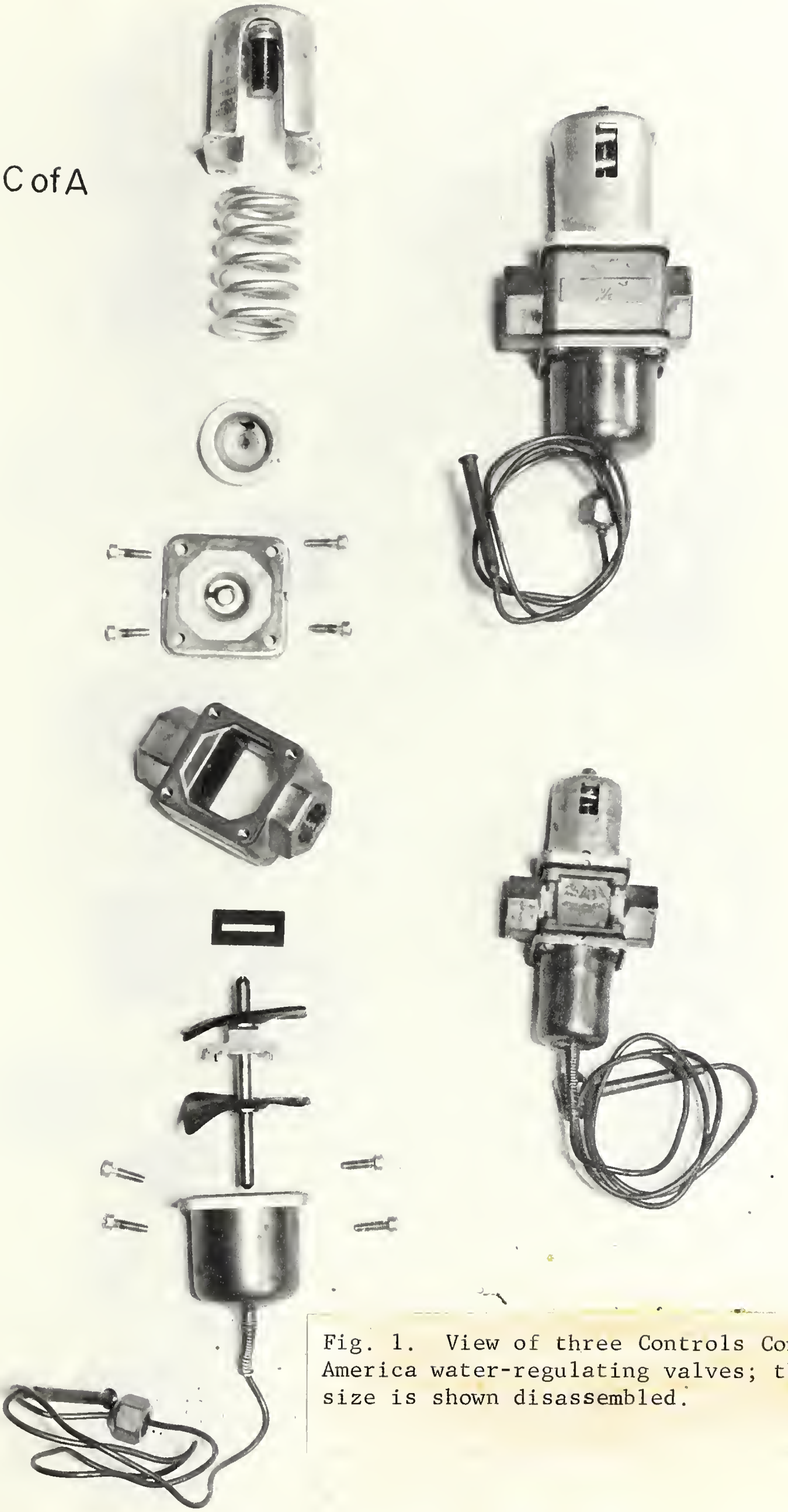

Fig. 1. View of three Controls Company of America water-regulating valves; the 1/2-inch size is shown disassembled.

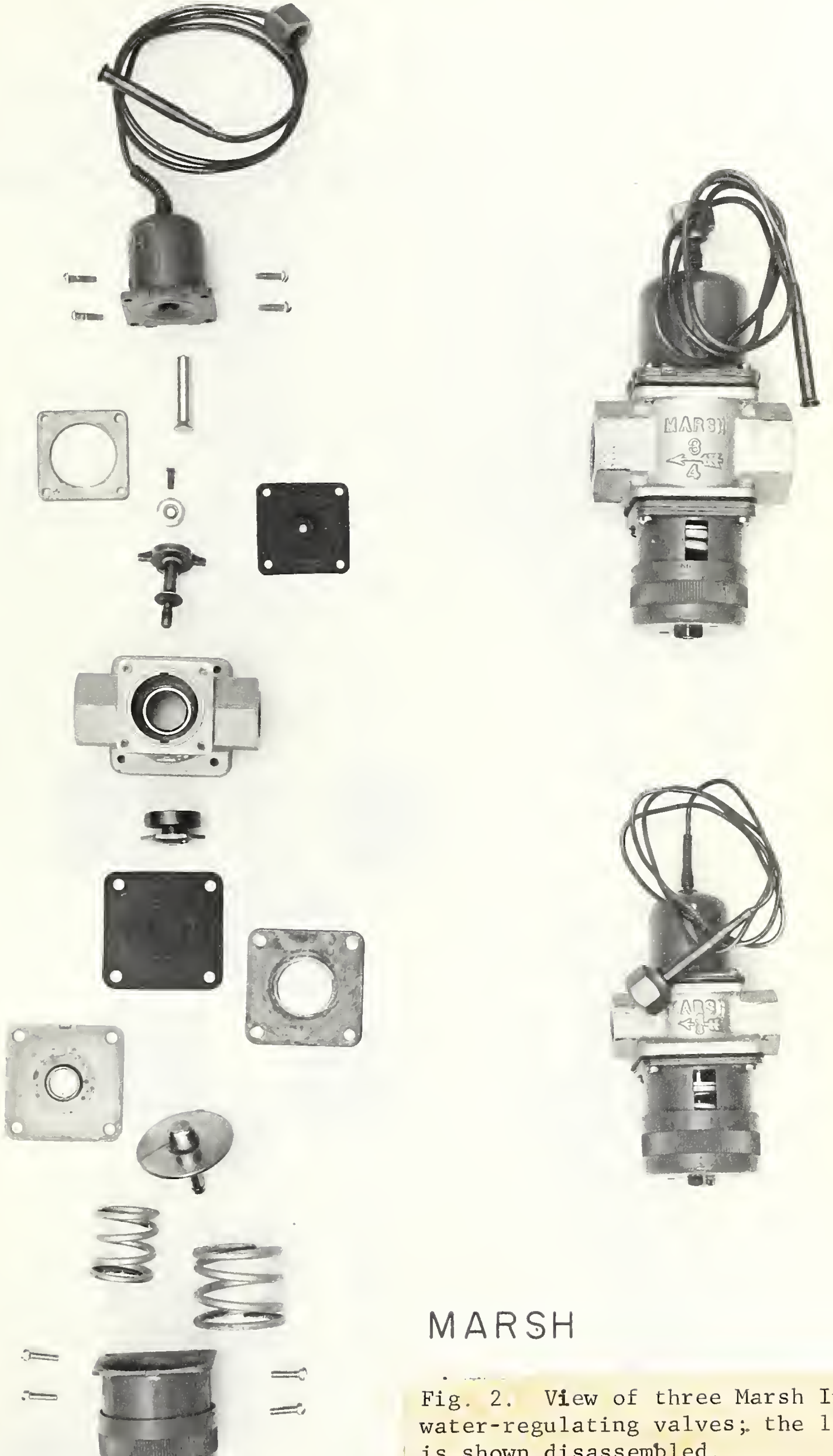

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Fig. 2. View of three Marsh Instrument Co. water-regulating valves; the 1/2-inch size is shown disassembled.

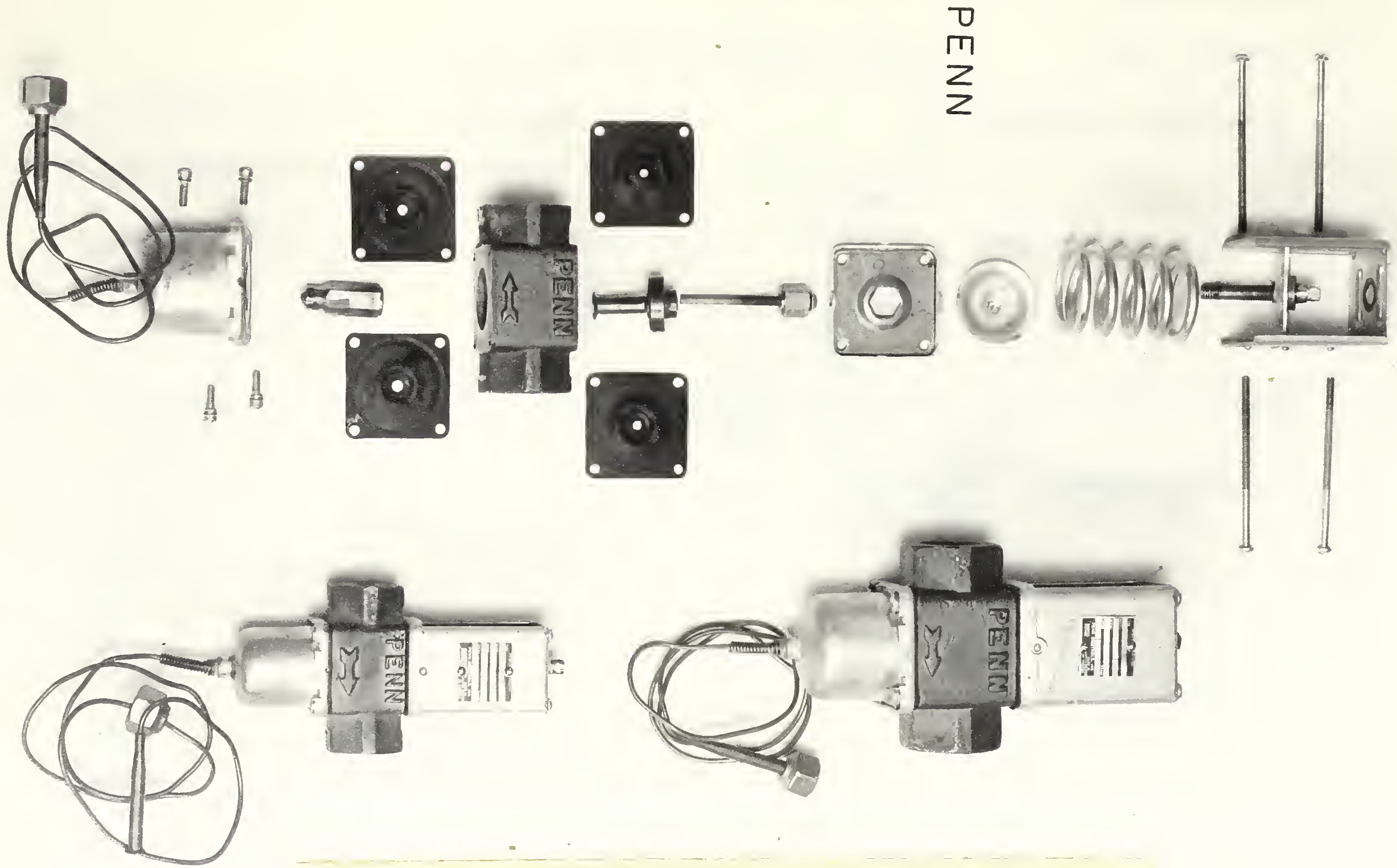

Fig. 3. View of three Penn Controls Co. water-regulating valves; the 1/2-inch size is shown disassembled.

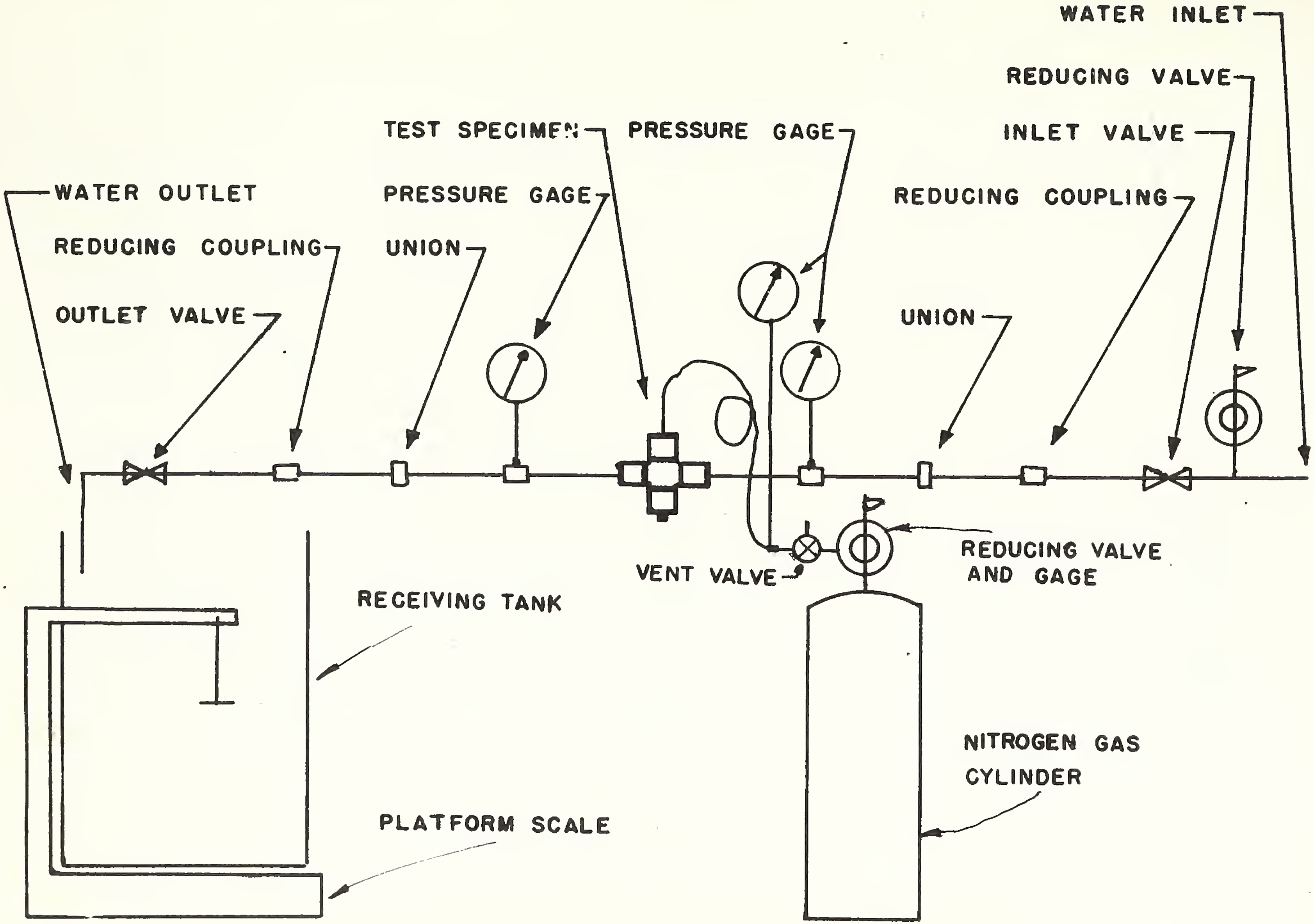

Fig. 4. Diagram of apparatus and piping used for tests of water regulating valves.

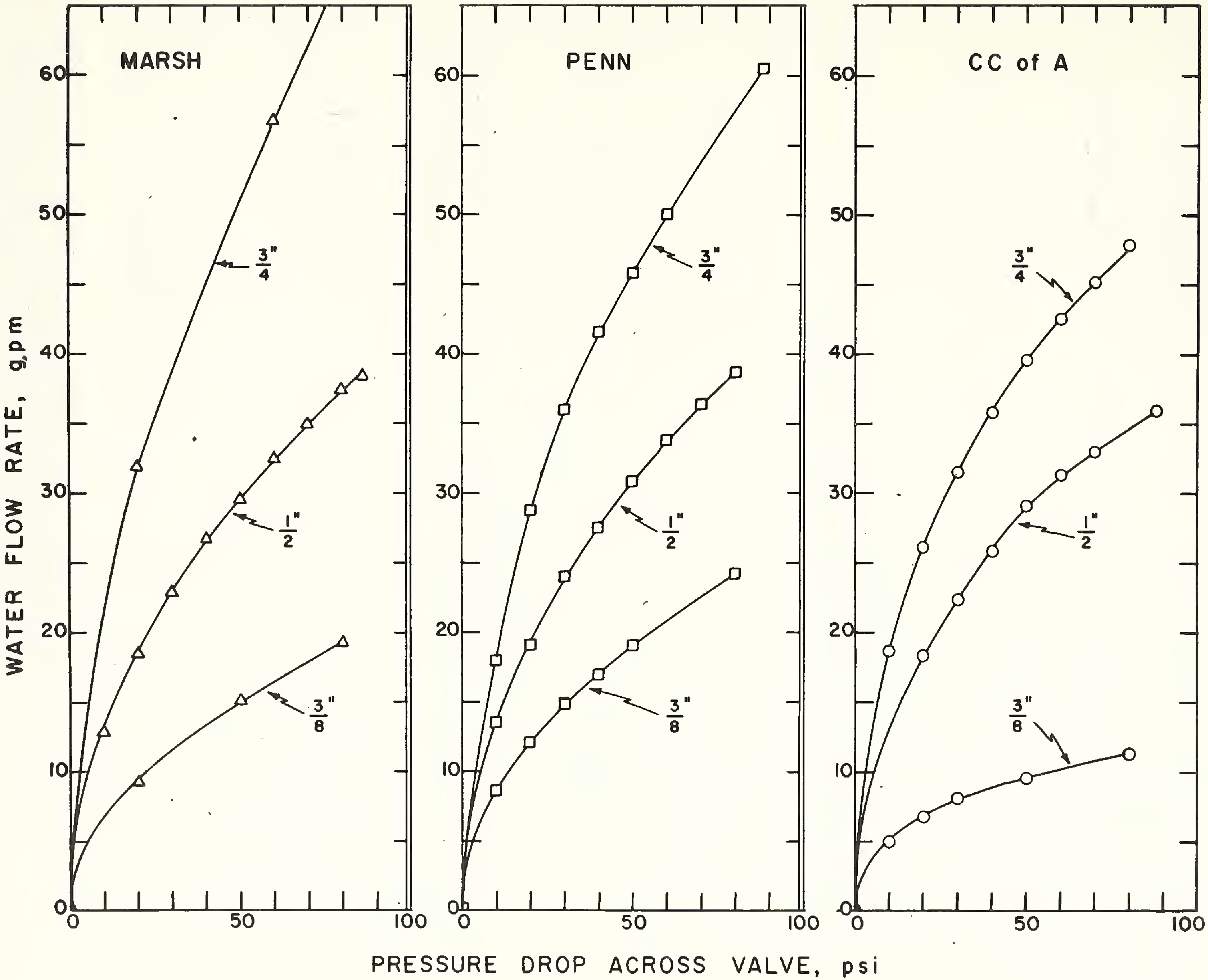

Fig. 5. Maximum water flow rate versus pressure drop for nine valve specimens grouped by manufacturer.

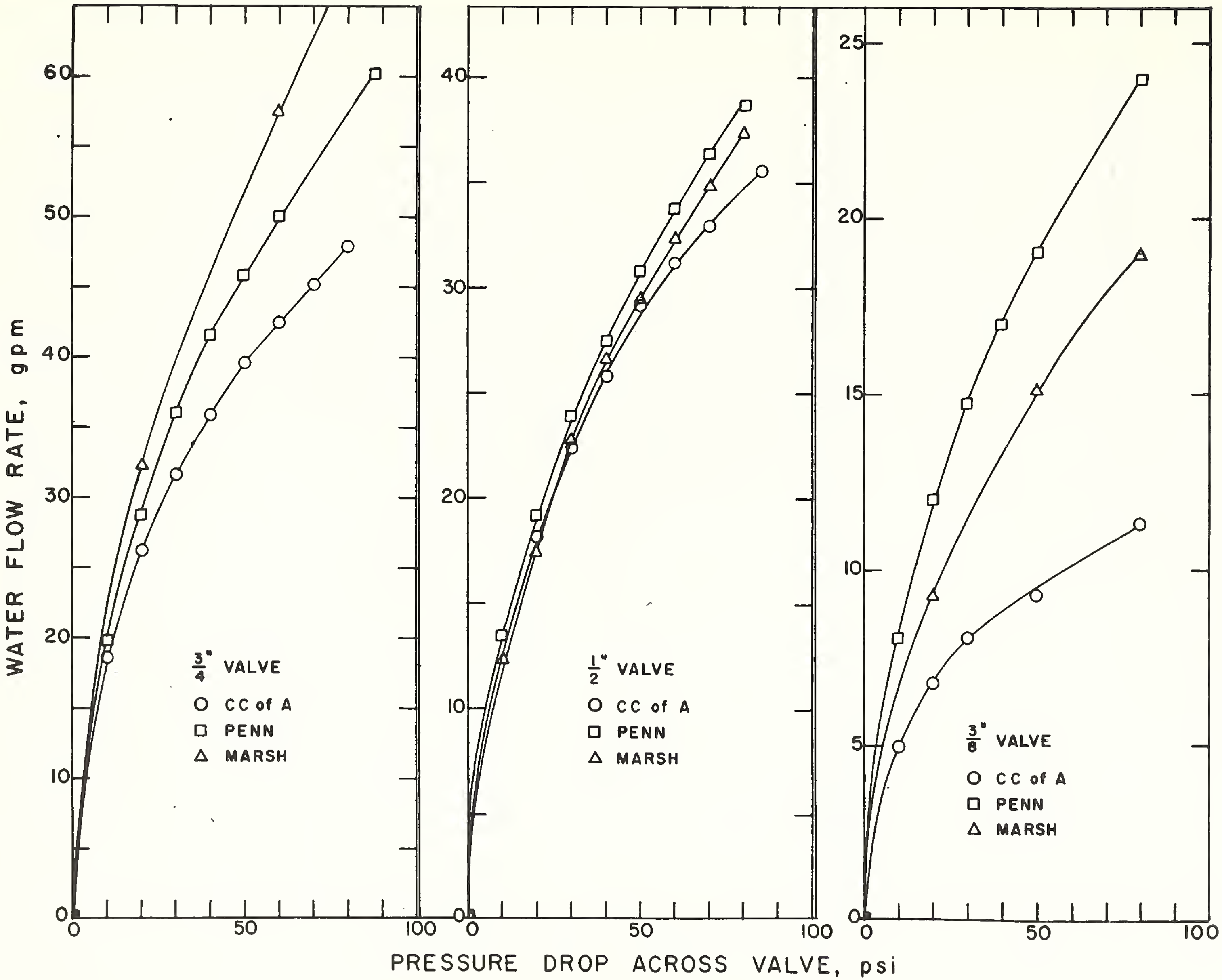

Fig. 6. Maximum water flow rate versus pressure drop for nine valves grouped by valve size.

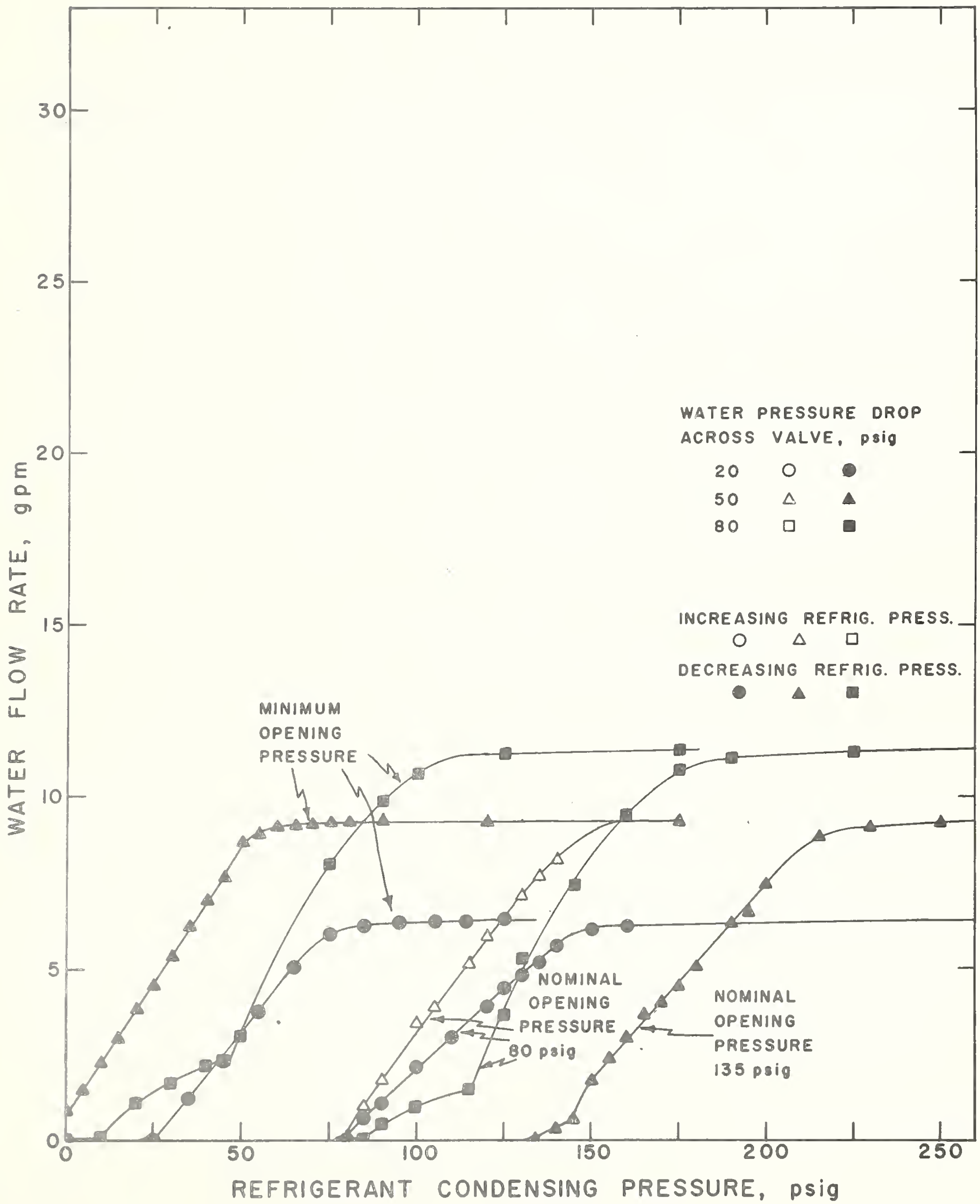

Fig. 7. Water flow rate versus refrigerant condensing pressure for 3/8-inch valve of Controls Company of America.

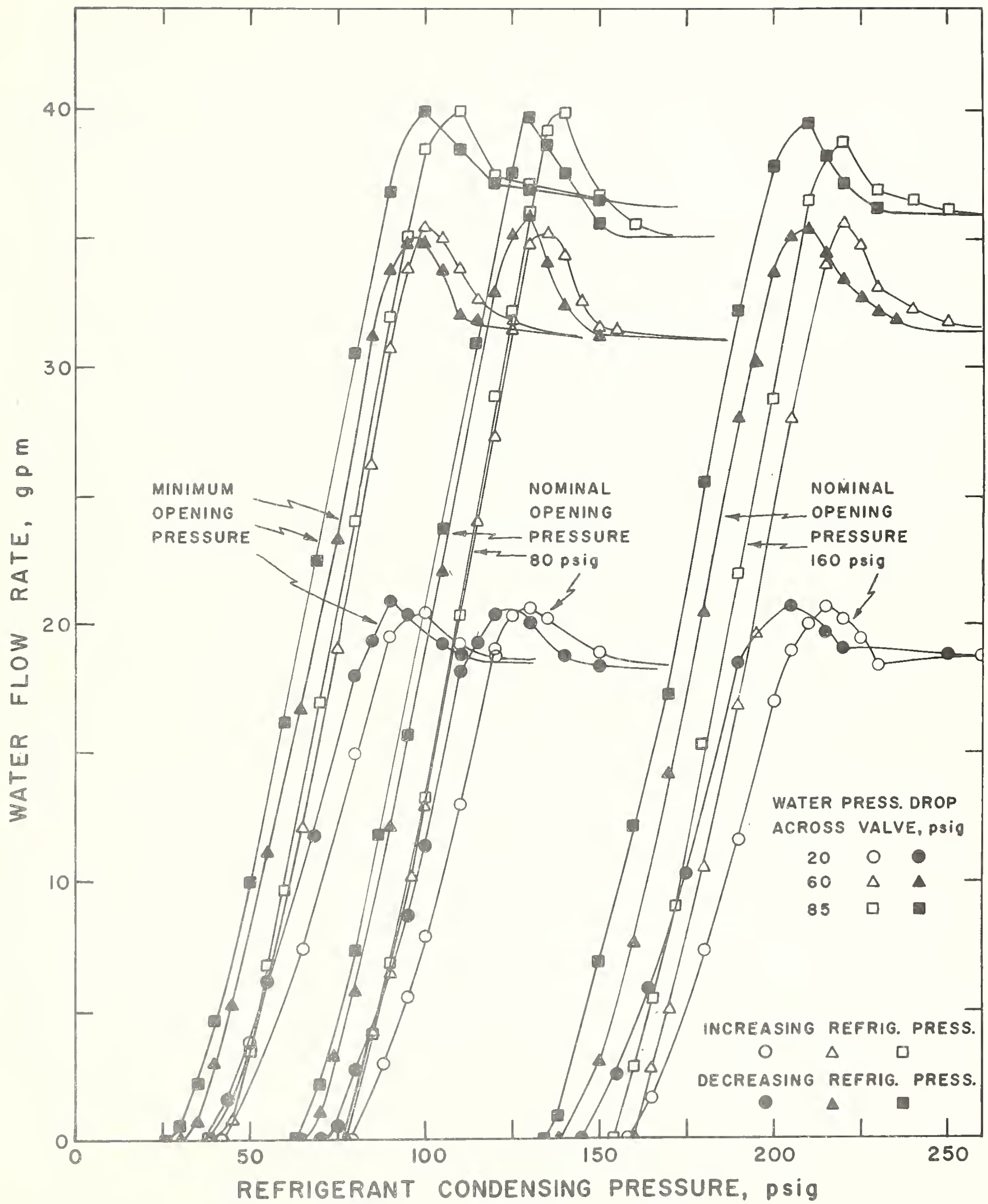

Fig. 8. Water flow rate versus refrigerant condensing pressure for 1/2-inch valve of Controls Company of America.

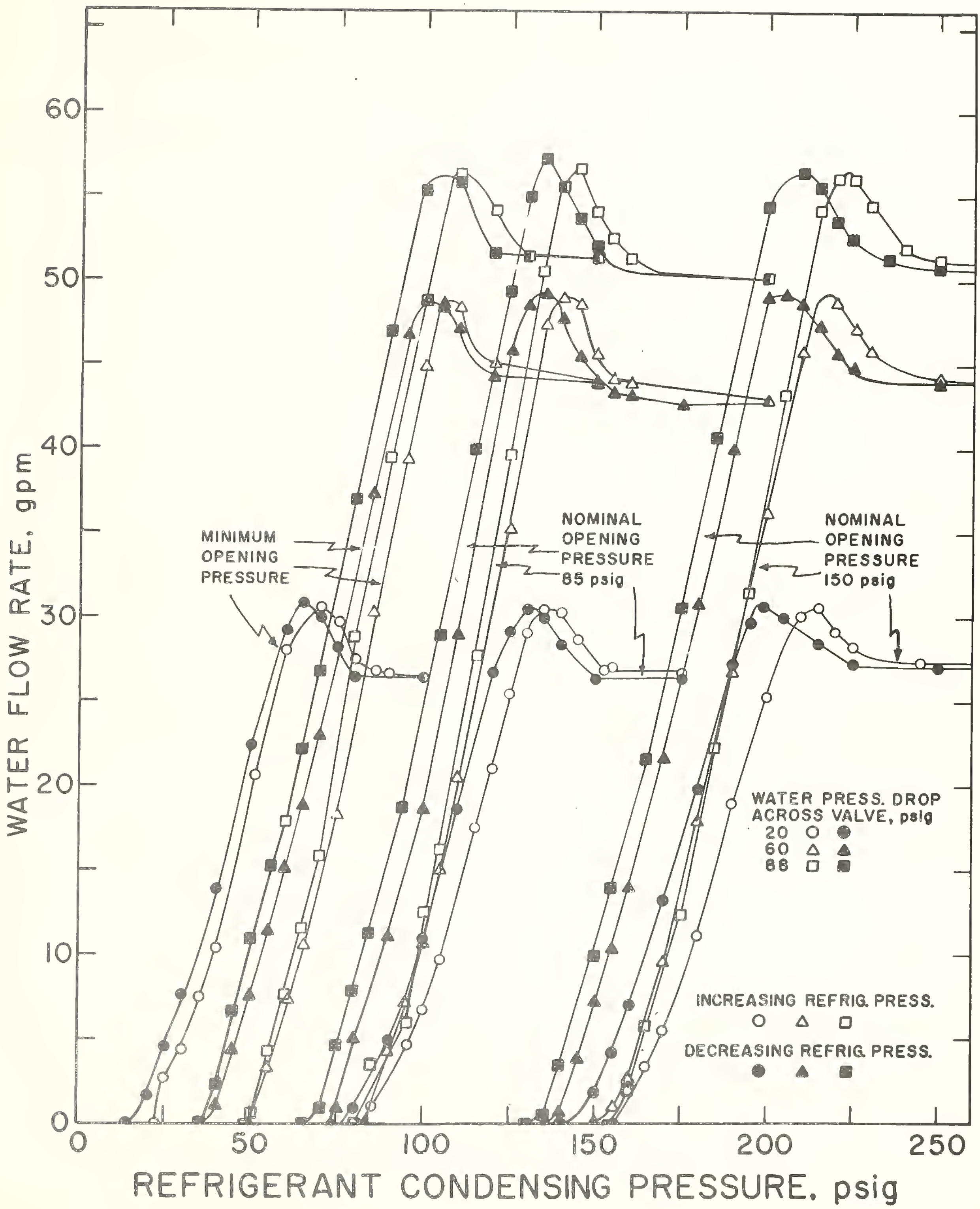

Fig. 9. Water flow rate versus refrigerant condensing pressure for 3/4-inch valve of Controls Company of America.

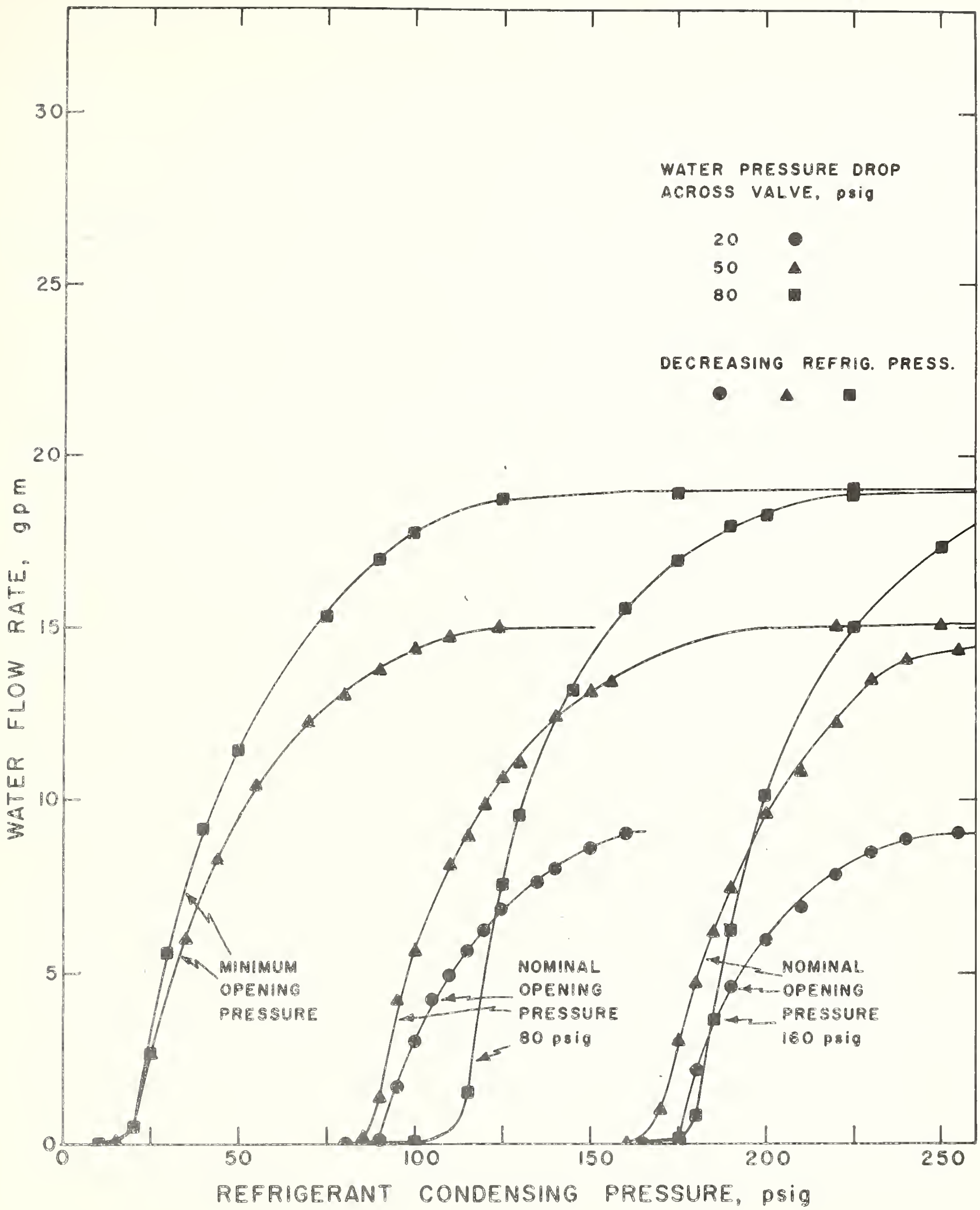

Fig. 10. Water flow rate versus refrigerant condensing pressure for 3/8-inch valve of Marsh Instrument Company

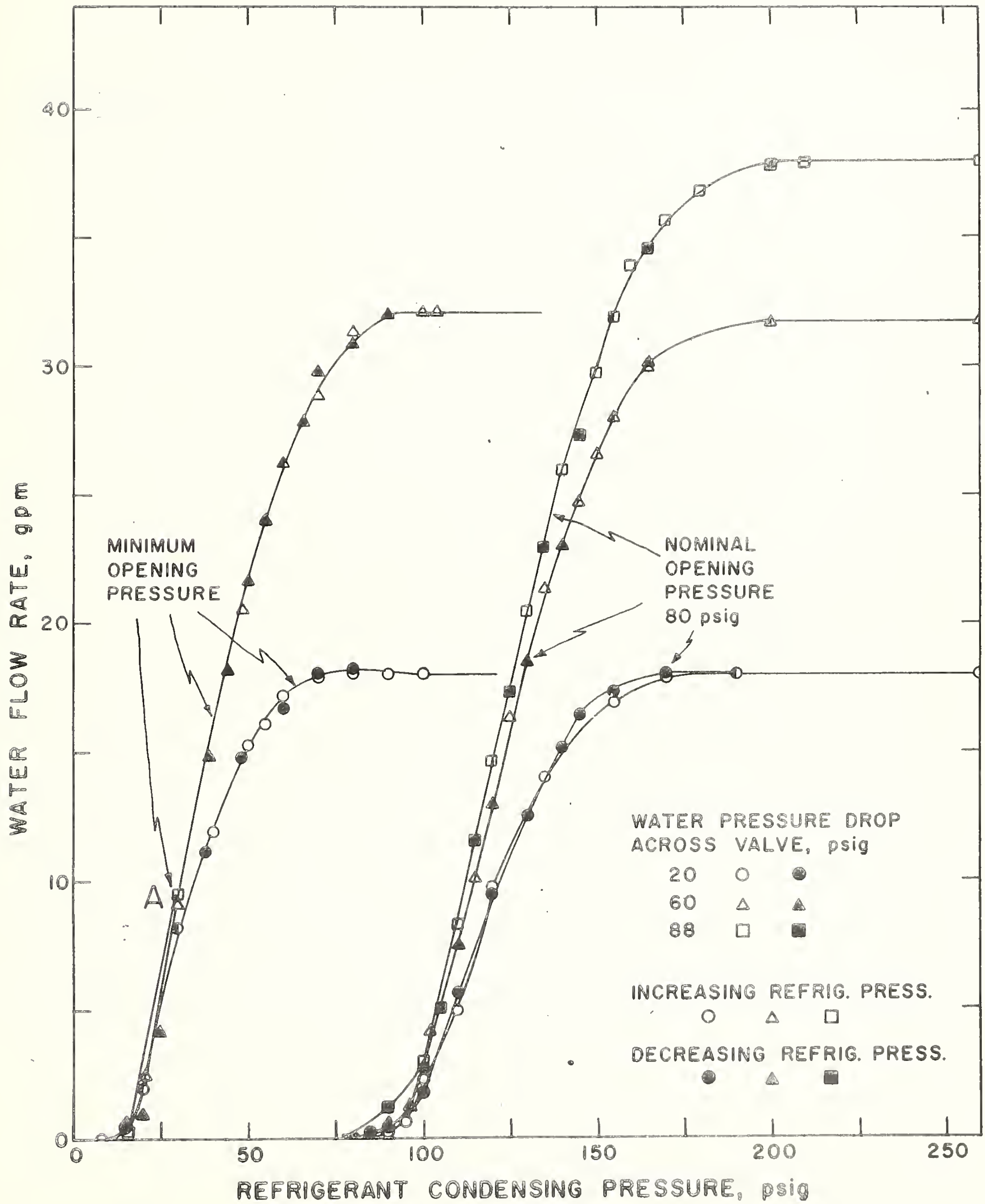

Fig. 11. Water flow rate versus refrigerant condensing pressure for 1/2-inch valve of Marsh Instrument Company

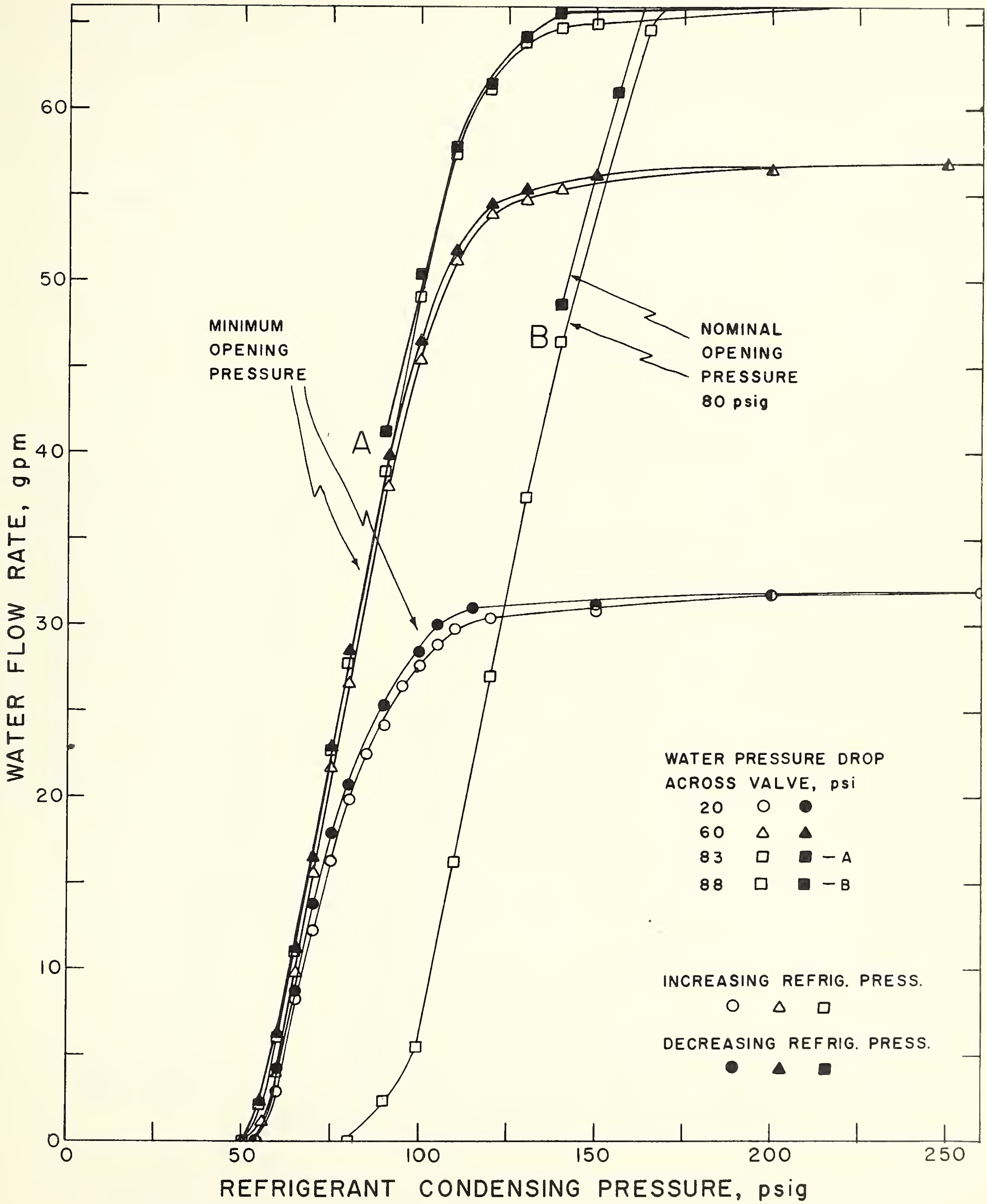

Fig. 12. Water flow rate versus refrigerant condensing pressure for 3/4-inch valve of Marsh Instrument Company

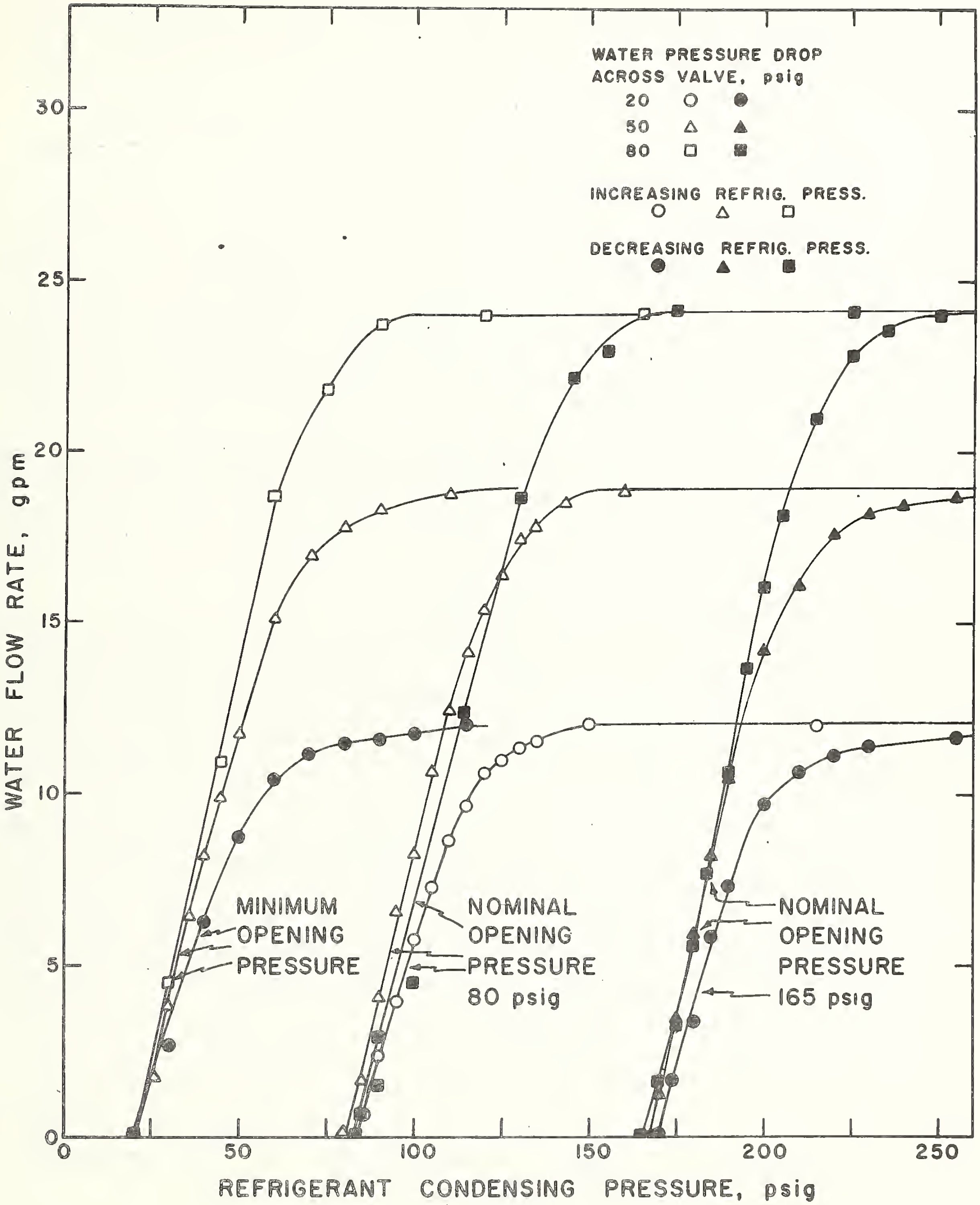

Fig. 13. Water flow rate versus refrigerant condensing pressure for 3/8-inch valve of Penn Controls, Inc.

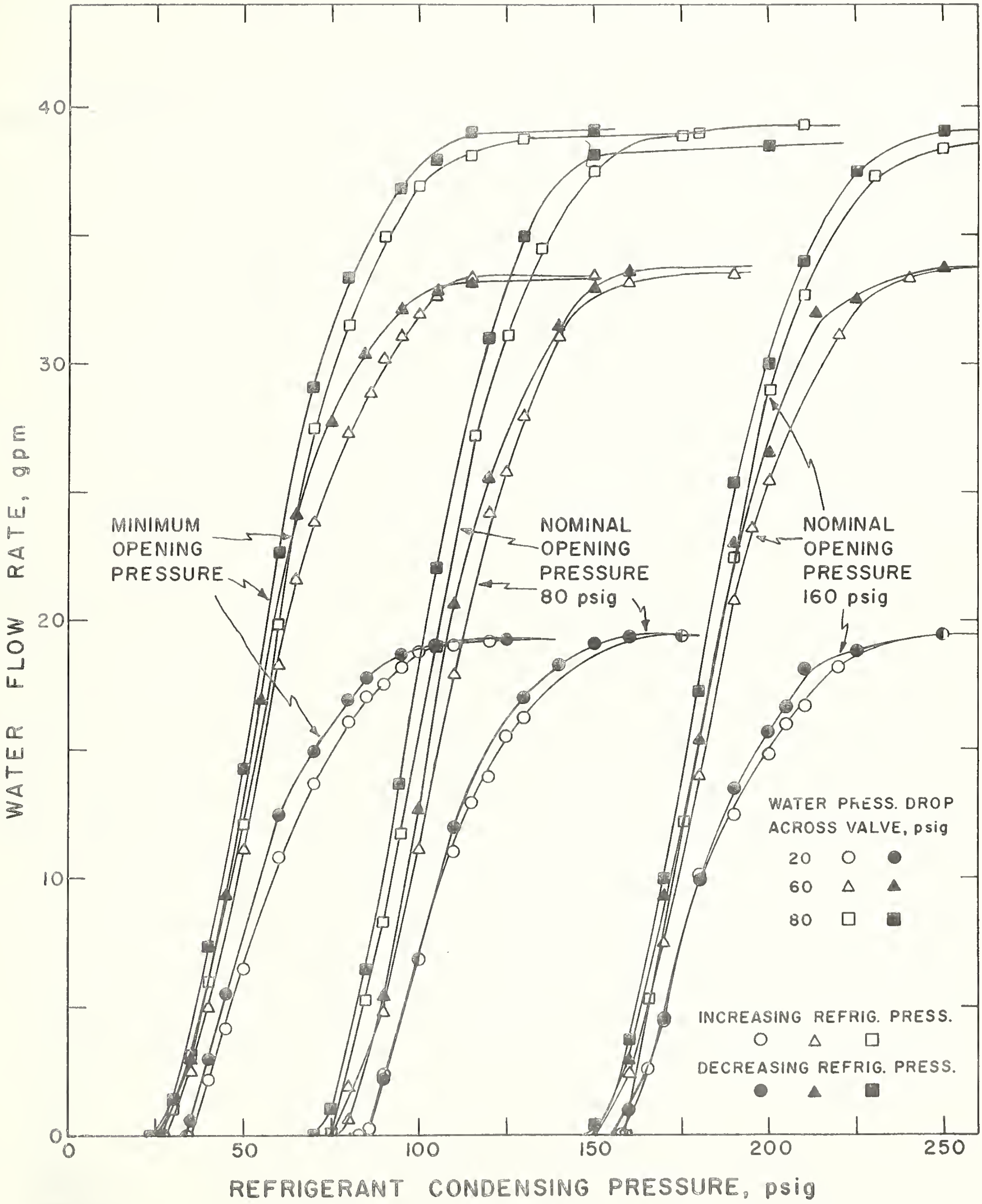

Fig. 14. Water flow rate versus refrigerant condensing pressure for 1/2-inch valve of Penn Controls, Inc.

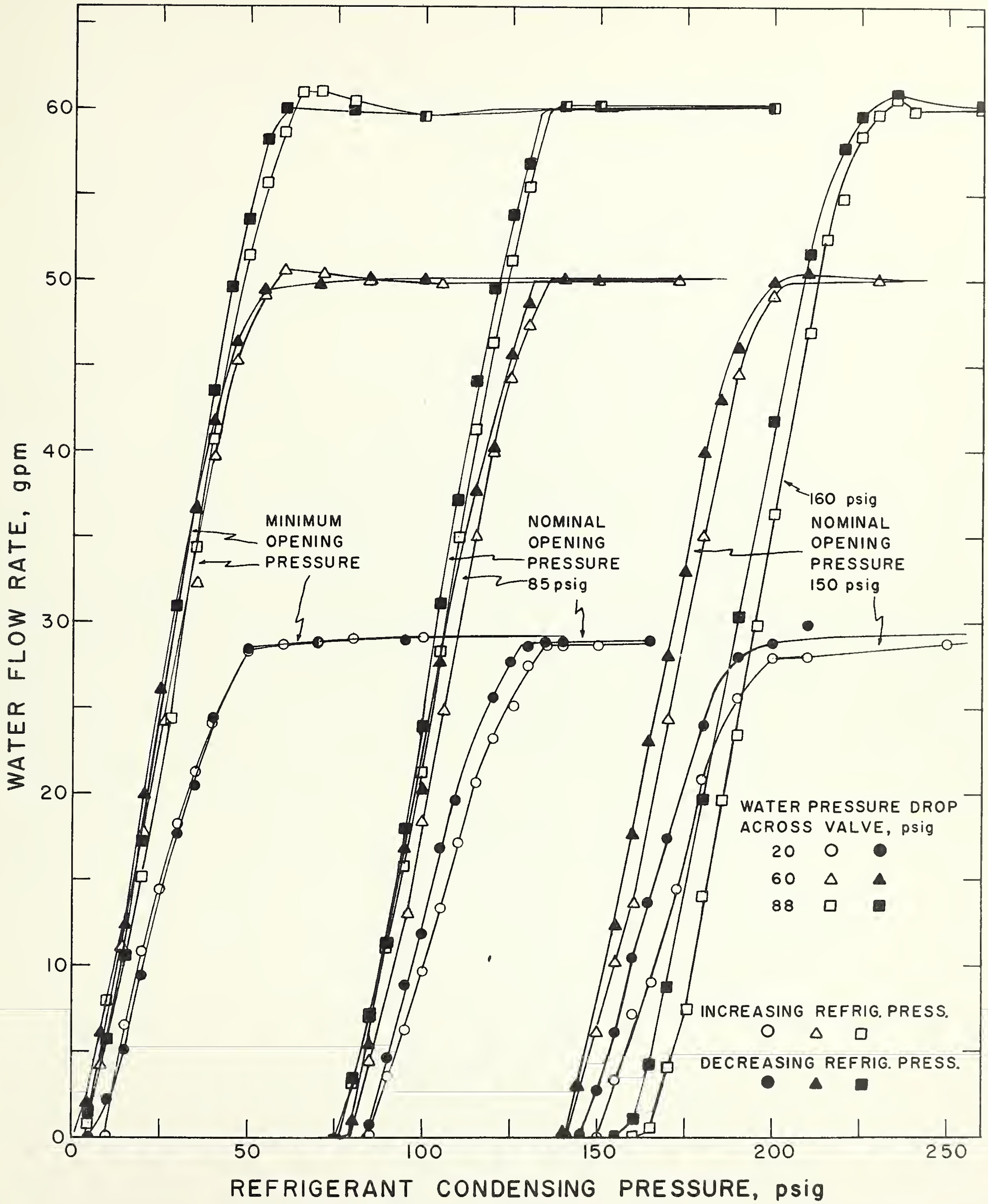

Fig. 15. Water flow rate versus refrigerant condensing pressure for 3/4-inch valve of Penn Controls, Inc.