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STUDIES TO DETERMINE SUITABLE METHODS FOR STARTING AND STOPPING THE PUMPS IN THE GRANBY PUMPING PLANT COLORADO-BIG THOMPSON PROJECT, COLORADO

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CHAPTER I - INTRODUCTION AND SUMMARY

Introduction

1. Description of Granby pumps. Four pumps will be utilized in the Granby pumping plant to raise water from the Granby Reservior to the Shadow Mountain Reservoir for diversion through the Colorado-Big Thompson transmountain tunnel. A plan and elevation of the pumping plant is shown in figure 1 and profile of the piping is shown on figure 2. Water will be supplied to each pump through a 66-inch square suction tunnel approximately 750 feet long. Each discharge line will consist of a 78-inch circular conduit approximately 3,000 feet long. Flow through the latter will be controlled by a hydraulically operated butterfly valve located in close proximity to the pump. An automatic, electrically operated air valve near the outlet of the discharge line will break any back-siphonic action developed by sudden shut-down of a pump.

It is proposed to operate the pumps at two speeds, 300 and 327 r.p.m., the speed depending on the head to be overcome at any particular time. Due to expected fluctuations in the water surface elevation in Granby Reservoir, it will be necessary for the pumps to operate against heads ranging from 96 to 201 feet. Each pump, as proposed, will discharge 240 second-feet of water at 162 feet of head at maximum efficiency and 300 r.p.m., or 180 second-feet at 210 feet of head at nearly maximum efficiency for a speed of 327 r.p.m. The performance curves for this pump are shown on figure 3. The pump impellers will measure about 88 inches in diameter and will be driven by two-speed synchronous motors of approximately 6,000 horsepower each.

2. Purpose of tests. Tests were made to determine whether the undesirable conditions experienced at similar pumping installations, when the pumps are started under load, could be alleviated. These large motors would require five times the normal running current for starting under load at full voltage which would be sufficient to cause dimming of lights on the connected system and, under certain conditions, the rate of increase of power input required by the motors would exceed the rate of response of the generators as limited by penstock or turbine governor capabilities. Four remedial methods - two electrical and two hydraulic - were proposed for these conditions.

In all four cases it was planned to force the water out of the volute by admitting compressed air. Thus on starting, the impeller would rotate in air requiring only a fraction of the power necessary to start it rotating in water. In the first scheme, the impeller would be started in air and brought to rated speed by an auxiliary motor. With the correct speed attained, the main pump motor would be synchronized with the line voltage and the main switch closed. The remainder of the starting procedure then would be to slowly exhaust the entrapped air from the volute allowing gradual submergence of the impeller. At full submergence, the butterfly valve would automatically begin to open slowly admitting water to the discharge line. Thus the load would be applied progressively and the rate of increase of power input would depend largely on the timing of the operation.

The second method of starting a pump electrically would be as in the first case with the exception that the auxiliary motor would be eliminated and the pump started by applying a fraction of the rated voltage to the main motor to bring it up to speed. At rated speed, the motor would be synchronized with the line voltage by closing the field switch at which time full voltage would be applied. With this method, the shock to the electrical system would be reduced to a small quantity since the power input varies with the square of the voltage.

The two hydraulic methods consist of bringing the pump units to operating speed by the use of high-velocity jets impinging on a bucket assembly connected to the pump shaft. The remainder of the starting procedure would be the same as for the other methods. The first method would be to mount a Pelton wheel on the pump shaft in a case separate from that of the pump impeller. With this arrangement there would not be any interference with the impeller action since the buckets would rotate in air under normal operating conditions. The second arrangement would be to mount the buckets on top of the impeller behind the seal rings but inside of the impeller case. This report deals primarily with the latter method of starting.

Summary

3. The starting and shut-down cycles with buckets on the impeller. The procedure developed for starting the pumps by the use of hydraulic jets impinging upon buckets cut in the topside of the impeller was satisfactory as far as reducing the shock to the electrical system was concerned. As shown on figures 18A and E, the increase in power was gradual, and the time required for the power to increase was sufficient to allow the generating apparatus to respond to the increase in load. A limited amount of control can be exercised over this rate of power increase by regulating the rate at which the compressed air is released from the pump volute. However, the buckets in the impeller created a disturbance when the pump was operating under load that reduced the over-all efficiency about four percent. The test data showed about two percent decrease of discharge, figure 22, and a three percent increase of power input, figure 14, at the maximum discharge. This decrease in efficiency in a large unit would be sufficient to make the arrangement impractical.

In the shut-down cycle, where the power was reduced from full load to zero load, figure 18B, the reduction in current was gradual and there were no objectionable hydraulic pressures. The fluctuations in pressure at the intake, as shown by the trace of No. 3 pressure cell (No. 3 p.c.), are characteristic of any pump, and the frequency is related to the apeed and the number of vanes on the impeller. The current is not a reliable index of the power as may be seen by

comparing the current trace of figure 18B with the power trace of figure 19D. It is evident that while the current showed a gradual reduction, the fluctuation in power was considerable due to the location of the port through which the compressed air entered the pump volute, figure 7B.

4. The bypass and the air port. The bypass is necessary regardless of the auxiliary starting device that may be employed. If a device is used whereby buckets are cut into the impeller, the bypass diameter would be controlled by the quantity of water issuing from the jets. It must be large enough to allow the jet water to escape without producing any considerable amount of head to be established in the pump volute. If an auxiliary Pelton wheel is used as a separate unit, a drain will be necessary to remove the jet water and, in addition, a bypass must be provided from the discharge side to the pump intake. If an auxiliary motor or the low-voltage scheme is used with the regular motor, a bypass is necessary although it is desirable to have it smaller in the latter three cases.

The location of the air-supply port directly affects the smoothness of operation and the form of the power curve during the shut-down cycle. With the port located at the discharge side of the pump, there were definite power surges, figure 19D, but with the location changed to the intake, figure 7A, the power curve was smooth as shown on figure 19C. From this it was evident that the air-supply port should be on the intake side and placed to distribute the air evenly in the fluid before it enters the impeller. The exhaust port should be located in the system such that there will not be any trapped air in the piping when the main discharge valve is opened.

5. Electrical power required at starting. There is a decided advantage, from the standpoint of the initial amount of power required, in starting the pump with the water depressed below the impeller. Figure 17 shows that the power required to rotate the impeller of the test pump with air in the volute was about 10 percent of the full-load value whereas, with water in the pump case and with the discharge valve closed, the power required to drive the pump motor was about 53 percent

of full load. Since it is desired to reduce the shock to the electrical system when the motors are started, the use of compressed air will be a material aid regardless of the method of starting.

CHAPTER II - PUMP TESTS

Laboratory Arrangement of Test Equipment

6. The test pump. The test pump(shown in figures 4, 5, 6, 7, and 8) was not considered as a model of the proposed Granby pumps except in the sense that it was of the same general type. This, however, was not of great importance as the experimentation centered about the starting and stopping cycles. Table 1 shows a comparison of the test pump and prototype dimensions giving the scale ratio for each. The general proportions of the two pumps can be studied from figures 1 and 5 and the pumping characteristics of the two are shown on figures 3 and 9.

TABLE 1

Comparison of Test Pump Dimensions with Prototype

Item	Prototype dimensions	Test pump dimensions	Similitude relationship	Scale ratio, N
Area of suction line, sq.ft...	30.25	0.493	A_mN^2	7.84
Diameter at discharge flange, ft.	6.50	0.67	L_mN	9.70
Diameter of impeller, ft.	7.33	1.026	L_mN	7.14
Depth of impeller passage at periphery, ft.		0.208	L_mN	
Capacity of motor, hp.	6,000	15	$P_mN^{3.5}$	5.54
Speed of pump, r.p.m.	300 and 327	875	$S_mN^{-0.5}$	8.50 and 7.15
Capacity of pump at maximum efficiency, sec.-ft.	240 and 180	3.56	$Q_mN^{2.5}$	5.39 and 4.81
Head developed at maximum efficiency, ft.	162 and 210	27.5	L_mN	5.89 and 7.64
Peripheral speed of impeller, ft. per sec.	115.2 and 125.6	47.0	$V_mN^{0.5}$	6.01 and 7.15
Diameter of bypass, ft.		0.167	L_mN	
Diameter of air-exhaust pipe, ft.		0.052	L_mN	
Diameter of air-supply pipe, ft.		0.023	L_mN	
Effective diameter of jet nozzles, ft.		0.0208	L_mN	
Velocity of jets, at 110-pound line pressure, ft. per sec. .		99.4	$V_mN^{0.5}$	

The test equipment (figure 6A) consisted of a tank reservoir, 3 feet in diameter by 12 feet high, to which was connected a transparent pyralin suction pipe and elbow (figure 6B). This in turn was connected to the intake of the single-suction, eight-inch, closed-impeller, vertical test pump. An eight-inch tee and valve were connected to the discharge side of the pump. The valve corresponds to the butterfly valve location in the prototype. An air intake, controlled by a small needle valve; and an air-exhaust line, controlled by a quick-acting cock, were connected into the top of the eight-inch tee (figure 7B).

The top of the pump impeller was altered as shown in figures 5 and 8A by cutting into it thirty buckets, 1/4-inch deep by 1.5 inches in diameter at an angle of 20 degrees with the horizontal. To do this, it was first necessary to sweat on a 3/16-inch brassplate to the top of the impeller to obtain sufficient thickness of metal to cut the buckets. The buckets were cut on a milling machine using a 1/4-inch by 1½-inch keyseat cutter. The four jets which propelled the impeller consisted of 3/8-inch inside-diameter copper tubes inserted through the top cover of the volute at an angle of 20 degrees (figure 5). Short inserts with 1/4-inch bore were soldered into the ends nearest the buckets to form the jet nozzles and 3/4-inch hose connectors were fitted to the opposite ends. Four 3/4-inch garden hoses (figure 6B) were connected from the latter to a main header in which water pressures from 35 to 110 pounds per square inch could be developed.

A two-inch bypass pipe and valve were installed between the eight-inch tee and the transparent suction line as can be observed in figures 6B, 7A, and 7B, the main purpose of which was to relieve pressures created in the volute by the water discharging from the jets and the rotation of the impeller. The bypass served a secondary purpose of allowing a small amount of circulation for cooling during the interval when the pump would be running under load with the butterfly valve closed. This circulation is very essential in the prototype.

The remainder of the apparatus consisted of a return pipe from the pump discharge to the tank reservoir in which was installed a six-inch venturi meter for measuring the discharge (figure 6A).

7. <u>Instrumentation.</u> Three thermomenters were installed in the apparatus to provide an accurate record of the temperatures throughout a test; one in the tank reservoir, one in the eight-inch tee, and the third on the pump motor.

To detect pressure fluctuations, which occurred at various points throughout the operating cycle, four carbon-pile pressure cells (figure 12) were incorporated in the test equipment; one in the tank reservoir (figure 7A), a second in the transparent suction line elbow (figure 6B), a third in the eight-inch tee, and the fourth in the discharge line downstream from the butterfly valve (figure 7B).

The recorded pressure trace transmitted by the electrical current from the cells is based on the pressure-resistance characteristics of a series or pile of carbon disks connected by a piston to a spring brass diaphragm which is actuated by the actual pressures at the point in question. This type of cell was used in two branches of a direct-current bridge to which an oscillograph element was connected as an indicating device. The cell was adjusted and calibrated such that the deflection of the oscillograph trace was a measure of the pressure on the diaphragm.

The electrical measurements on the test pump motor were made with an ammeter, voltmeter, wattmeter, watt-hour meter, and the oscillograph. The diagram of electrical connections for the instruments is shown on figure 13. The oscillograph was used to observe current and voltage to the motor, and the voltage element was used as a timing wave.

The indicating wattmeter gave a means of observing the rate of increase or decrease of power as air was released from or forced into the pump volute. This was of value since the motor power change on the prototype would affect the governor action at the generating station. Since the power factor of the induction motor varied from 18 to 80 percent between no load and full load, the current wave on the oscillograms was of little value in studying power changes. The oscillograph was not equipped with a power element.

A watt-hour meter and stop watch were used to obtain the power where accurate values were required, since the power input as shown by the indicating wattmeter fluctuated considerably. Although the laboratory voltage varied continually, the power input to the motor fluctuated principally because of the roughness in the pump's running performance. Therefore, by timing the number of revolutions of the watt-hour meter over a period of about two minutes, the average power over that period was obtained. The power equation for the connections shown on figure 13, using the watt-hour meter and a stop watch, is as follows:

$$\text{Watts} = \frac{(\text{Revolutions})\ (\text{C.T. ratio})\ (\text{K})\ (3{,}600)}{\text{Seconds}} = \frac{\text{Rev.}}{\text{Sec.}}\ (1{,}200)$$

The power to the three-phase motor was measured by one single-phase wattmeter instead of the usual two wattmeters by assuming the power in the three phases to be equally divided. In order to measure the power in one phase of the three-phase motor, it was necessary to use the line-to-ground voltage instead of the line-to-line voltage, as shown on figure 13.

Because of its value as a timer, it was desirable to use the oscillograph to determine the best place to force the air into the pump volute as far as controlling the rate of power decrease was concerned. In order to obtain a power trace on the oscillograms as shown on figure 19, a slide wire rheostat was mounted directly over the wattmeter in a position very close to the wattmeter indicator and scale. Since the slide wire was connected to an element on the oscillograph it was possible to manually follow the wattmeter indicator with the slide wire and thereby obtain a deflection of the oscillograph element corresponding to the wattmeter reading. It was possible to get the desired results by this method, and a fair degree of accuracy was obtained after a little practice.

Test Procedure and Results

8. Purpose of bypass. It was found by tests that they bypass is essential to successful starting and stopping of the pump for both the electrical and hydraulic methods. In starting, by any of the four methods, the bypass should be open previous to the admission of air into the volute to allow complete draining of the latter. If this is not done, pressures are created in the volute which retard the impeller

speed. This results in an increase of power for starting the motors electrically or the necessity for higher jet pressures in the hydraulic method. The bypass is essential in the hydraulic method to carry off the water imparted to the buckets by the high-pressure jets. The centrifugal pumping action that occurs when the impeller is revolving tends to throw any entrapped water toward the outer radius of the volute case. If a bypass is not provided, the water flowing in from the jets can flow from the eye of the impeller only after sufficient pressure has been developed to force it out. In the absence of a bypass, it is obvious that the hydraulic method of starting approaches a vicious circle. The higher the pressure in the volute due to the trapping of water, the higher the pressure must be maintained on the jets to bring the impeller to speed; and as an increase in jet pressure is naturally accompanied by an increase in discharge, it then becomes necessary to force more water out of the impeller eye.

In the shut-down cycle, the bypass is also essential for all four methods. Without it the admission of air will not force all of the water from the volute. A small portion becomes trapped, and continues to create pressure in the air and water mixture. This in turn requires more power to turn the impeller than would be the case were the water absent. Figure 17 shows that the power required with either air or a mixture of air and water in the case is 10 and 53 percent respectively of the full-load value. In addition, the instability of the air and water mixture produced very noticeable surges which are reflected in the power requirement. The main purpose throughout these tests was to develop a method in which changes in pressures and changes in load would occur as gradually as possible during the starting and stopping cycles.

9. <u>Calibration of bypass.</u> Convinced that the bypass is an essential part of the pumping installation, the next step consisted of determining the most economical size. As previously stated, the bypass consisted of a two-inch inside-diameter pipe with a two-inch gate valve for control. With the pump in operation and the main valve in the discharge line closed, the velocity of flow was measured in the center of the bypass pipe for each turn of the bypass valve. This was done with

a cylindrical pitot tube as shown in figure 7A. The velocity as measured on the center line was assumed as 1.25 times that of the average, and the discharge computed accordingly. From this information the equivalent size of bypass pipe was computed to correspond to each turn of the bypass valve. This calibration, which was used throughout the tests, is plotted on figure 10.

10. Effect of bypass on impeller speed. A series of tests were made to determine the minimum size of bypass, which would relieve the pressure in the volute sufficiently to not deter the speed of the impeller, when propelled by the water jets. With the eight-inch main line valve closed and the bypass valve opened completely, ten turns, a definite water pressure was applied to the jets and the impeller speed measured. The impeller speed was then measured for nine turns, eight turns, etc., down to complete closure of the bypass valve. The above procedure was repeated for line pressures of 110, 70, and 35 pounds per square inch, and the results are plotted on figure 11.

It is evident from figure 11 that the minimum size of bypass to obtain 900 r.p.m. for the jet pressures and discharges used in this model was approximately 2 inches. Jet performance is definitely the governing factor in determining the size of bypass. The size of bypass could have been made somewhat smaller had it been possible to use higher pressures with correspondingly lower discharges on the jets.

The accompanying table shows the magnitude of the total discharge for the four jets, the velocity of the jets, and the pressure on the 1/4-inch jet nozzles for three different line pressures. The jet discharges were measured through a 1-inch orifice meter and the suction elevation was held at 9.0 feet above the center of the suction pipe. The values in the table are independent of the size of bypass pipe.

Pressure above center-line suction pipe, feet of water	Line pressure, lb.per sq.in.	Jet velocity, feet per second	Total discharge of four jets, g.p.m.	Pressure in 1/4-inch nozzles, feet of water
9.0	110	99.4	61	36.7
9.0	70	81.5	50	17.3
9.0	35	57.0	35	7.7

11. Effect of buckets on pump performance. The effect of the impeller buckets on the efficiency of the pump was determined by both hydraulic and electrical measurements. In the hydraulic method, the discharge delivered by the pump was measured through a six-inch venturi meter (figure 6A) for each turn of the eight-inch throttle valve, located immediately downstream from the meter. The measurements, hydraulic and electrical taken simultaneously, were repeated four times with the buckets on the impeller (figure 8A) and four times with the buckets filled with Cerro Bend metal (figure 8B). The readings of each four runs were averaged and the hydraulic results plotted on figure 22. There is a maximum deviation between the two curves of about two percent in the discharge for the larger openings of the throttle valve. This would indicate that the buckets created additional turbulence in the volute which interfered slightly with the normal functioning of the pump. This was further evidenced by the fact that the gage at pressure cell No. 2 (figure 7B) fluctuated over a range of about six feet of water when the buckets were incorporated in the impeller and less than a foot of water after the buckets were eliminated. These fluctuations, at pressure cell No. 2, can be observed on the oscillograph records (figure 18).

The power input measurements, made simultaneously with the water discharge measurements, indicated that the power input to the pump was increased materially by the presence of the buckets. The readings of four runs with the buckets and four runs without the buckets were averaged and the results are plotted on figure 14. The data on figure 14 shows that the loss in power due to the buckets amounts to from 0.5 to 3 percent with a maximum of 3 percent occuring at maximum discharge. This is also the point where hydraulic measurements showed that the buckets caused a two percent reduction in the discharge.

It is apparent from the above results that the over-all efficiency of the pumping unit was reduced appreciably by the buckets. Combining the effects observed by both hydraulic and electrical measurements the total loss in efficiency for the pumping unit amounted to about four percent at the maximum discharge. This loss when applied to the four prototype pumps would be considerable over a period of time and the bucket installation as used on the test pump is not feasible.

Another method of determining the friction losses due to the buckets was to measure the power input at shut-off head with and without the buckets. With the discharge valve closed, power measurements were made with zero, one, two, and three turns open on the bypass valve. Immediately after these values were obtained air was admitted to the pump volute and the power was measured. By subtracting this value from the four previously obtained it was possible to separate the motor and bearing losses from the losses due to water in the pump case for these four positions of the bypass valve. Then, by comparing these water friction losses with and without the buckets, the losses due to the bucket were obtained. The results on figure 15 show that the losses due to the buckets amount to about 1.5 percent.

With air in the pump case, the power input to the motor decreased considerably during a series of consecutive tests which was probably due to variations in the bearing and gland friction. Although motor and water temperatures varied, their effect was less than the errors in observation and the change from 1,470 to 990 watts over a period of about 2 hours, as shown on figure 16, could not be attributed to this source. The power input, with water in the case, decreased during the tests about the same amount as did the input to the motor with air in the volute. Therefore, the difference between these two quantities, which represent the losses due to the water, is fairly independent and constant. To make certain that an accurate value of loss due to the buckets was obtained, fourteen runs with the buckets and ten runs without the buckets were made and the average of each plotted on figure 15. It is believed that fairly accurate determinations of power were made due to the fact that with air in the volute the power input average, with buckets on the impeller, checked the power input without the buckets within 0.8 percent.

The hydraulic starting arrangement of housing a small Pelton wheel in a separate unit on the same shaft does appear feasible. Provisions must be made to remove the discharge from the Pelton wheel and drain the case after the jets are closed, thus allowing the buckets to revolve in air during normal operation of the pump.

12. The starting cycle. In an electric motor the starting current is several times the full-load current and with the large motors on the Granby pumps, the starting current would be sufficiently large to momentarily reduce the ordinary line voltage. To minimize this condition some of the previously mentioned starting devices were used in conjunction with the pump motor. The laboratory studies chiefly concerned the method whereby the rotor was brought to operating speed by the force of hydraulic jets impinging on the buckets cut in the topside of the impeller.

The procedure for bringing the pump from a standstill to speed under full load was as follows: With the discharge valve closed and the bypass valve open, compressed air was admitted to the discharge side of the pump until the water level in the intake had depressed to a point approximately eight inches below the impeller. Then with the use of hydraulic jets impinging on the impeller buckets synchronous speed was attained with the impeller rotating in air. This was followed by closing the motor switch, closing the valves in the supply to the jets, and gradually releasing the air at the discharge side, thereby allowing the water to enter the pump slowly and the pressure in the discharge side to gradually build up to shut-off head. The discharge valve was then opened slowly and the bypass valve closed. Thus the motor operated at full load, completing the starting cycle.

13. The starting-cycle oscillogram. A graphic record, figure 18A, shows the magnitude of the current, voltage, and the related hydraulic pressures during the starting cycle. It will be noted that the rotor was brought to speed in air by the hydraulic jets and the motor switch closed before the record was started. At this point in the cycle, the current consumed was 20 amperes; the head on the intake line was 9 feet of water, which remained constant; and the air pressure in the volute was approximately the same. The pressure in the discharge line (No. 1 pressure cell) represents the static head maintained by the closed valve.

As the air was released from the pump volute and the water entered, a very rapid fluctuation in the hydraulic pressures occurred at the intake. The frequency corresponded to the product of the revolutions per second and the number of blades of the impeller. The reduction in pressure at the intake (No. 3 pressure cell) was simultaneous with the increase in pressure at the discharge side (No. 2 pressure cell). The corresponding increase in pressure in the discharge line was due to opening the discharge valve before the shut-off head had attained its maximum value. In the prototype, this would not occur because enough time could be allowed between the releasing of the air and the opening of the valve to allow the head to stabilize, whereas in the test pump, the capacity of the oscillograph camera limited the amount of time that could be allowed for each step in the cycle. When the main valve was completely open, the pressure upstream dropped and then increased as the bypass was closed. An inspection of the current trace on the oscillogram shows that the increase in current from 20 to 44 amperes was sufficiently gradual to allow the generating apparatus to respond to the change in load. Comparing figures 18C and D, the ordinary starting current as compared to the magnetizing current, which was the first current used with this starting device, was approximately five and one-half times as great. This would be an appreciable reduction in the prototype current where the motors will be 6,000 horsepower each.

Figure 18E is an oscillogram of the motor demand during a starting cycle where the diameter of the bypass was decreased to 1.30 inches and the rate of air release was slower. In this instance the load increased more slowly requiring 9.1 as compared to 2.5 seconds for the former. This indicated that the size of the bypass and the rate at which the air was released determined, within certain limits, the rate at which the load was applied to the motor and was signified by the corresponding increase in power on the oscillogram.

14. <u>The shut-down cycle.</u> The following steps, which were taken to reduce the power gradually from full load to zero, are applicable to the stopping of any vertical pump regardless of the starting devices

employed, as the shut-down cycle is independent of the device used for attaining synchronous speed. The discharge valve was slowly closed and the bypass valve opened. Air was forced into the volute, causing the water to be depressed allowing the impeller to rotate in air. Thus the power taken by the motor was reduced to ten percent of the full-load value and the fluctuation in the line voltage was very small when the motor switch was opened to complete the cycle.

15. The shut-down-cycle oscillogram. The shut-down-cycle oscillogram, figure 18B, is divided into three steps: (1) the normal operation, (2) closing the discharge valve, and (3) forcing air into the volute. At normal operation under full load, the current consumed was 44 amperes, the pressure in the discharge line 27.5 feet of water (No. 1 pressure cell), the pressure at the discharge side of the pump 33 feet of water (No. 2 pressure cell), the pressure at the intake 2 feet of water (No. 3 pressure cell), and the head on the intake was 9.0 feet of water (No. 4 pressure cell).

As the discharge valve was closed and the bypass valve opened, the current consumed gradually decreased to 31 amperes when the discharge valve was completely closed, then it remained constant until air was forced into the volute. In the meantime, the hydraulic pressure at the discharge valve increased to 42.5 feet of water, the pressure in the line became negative for 0.6 of a second, then increased to the static head of 6.5 feet of water. The pressure at the intake increased to its static value of 8.0 feet of water, and the head on the intake line remained constant. When the compressed air entered the discharge side and depressed the water, the pressure at the intake fluctuated with the frequency of the impeller vanes and for an instant reached a maximum value of 16 feet of water. As the air continued to enter the volute, the water level gradually lowered, the shut-off head decreased to the static pressure at the discharge side, and the current consumption dropped to a minimum value of 20 amperes for the impeller rotating in air.

16. The relation of the bypass to the shut-down cycle. During the course of the studies it was found that the diameter of the bypass

affected the power curve during the shut-down cycle. For instance, a bypass with a large diameter caused the power to decrease much more rapidly than did a smaller bypass (figure 19). Without a bypass the power consumed with the impeller rotating in an air and water mixture was 50 to 60 percent of the full-load value, whereas with a bypass it was only 10 percent of the full-load value. Without a bypass, the air did not force the water out of the discharge side and a mixture of air and water remained trapped around the outside of the impeller and in the line upstream from the discharge valve. As a result, there was sufficient trapped water to cause the pump to create approximately 60 percent of the shut-off head. When a small-diameter bypass was provided, from the discharge side to the intake the trapped water and air flowed slowly out of the discharge side and the corresponding decrease in head and power were gradual. If the diameter of the bypass was large the water left the pump quickly and the corresponding decrease in power was rapid. The rate of decrease in power is shown on figure 19 for the bypass diameters of 1.30 inches and 0.67 inch. The curves also show the results obtained with different locations of the air ports, which are discussed in the following paragraph.

17. The location of the air-supply ports. The location of the air-supply ports affected the rate at which the load decreased and the manner in which the power decreased. This can be seen by comparing the power-time curves of figures 19A and C with figures 19B and D. The former curves show the decrease in power related to time for bypass diameters of 1.30 inches and 0.67 inch with the air-supply port located at the intake (figure 7A) whereas the latter curves show the results obtained with the port on the discharge side (figure 7B). In figure 19B the drop in power was quite rapid, 1.8 seconds, with the air-supply port at the discharge side of the pump, and it was much slower, 6.2 seconds, and more gradual with the port located in the intake, figure 19A. The difference in the results for the two locations became more pronounced as the diameter of the bypass was made smaller (figures 19C and D). The total time required for the power to decrease 4,500 watts and become steady at 1,500 watts was in both cases satisfactory, 33.9

seconds for the former and 28.6 seconds for the latter, but when the port was located on the discharge side, the decrease in power was not gradual and would cause undesirable power surges in the prototype. In the test pump, these fluctuations in power, which had a definite period, were gradually damped to a steady demand at 1,500 watts. This was due to the fact that the air and water did not mix when the compressed air was supplied to the discharge side. As a result, large concentrations of air escaped intermittently through the bypass or impeller and suddenly relieved the shut-off head which again built up to some lower value than previously obtained. The process continued until all of the water was drained from the pump. This fluctuation in pressure caused the corresponding fluctuation in power shown graphically on figure 19D.

With the air-supply port at the pump intake, as shown in figure 7A, the power decreased steadily and the total time for the reduction to occur increased. This improvement was due to the thorough mixing of the air and water by the impeller before it entered the discharge side and circulated back through the bypass, thus no large concentrations of air could escape rapidly and cause the head to fluctuate.

As a result of this change, it was evident that the most desirable location for the air port was somewhere on the intake side so another test was made with the compressed air entering at the center of the impeller hub. The data obtained showed no appreciable difference between the power time curves as compared to those of the previous arrangement. However, from visual observations, it appeared that the mixing of the air and water was more uniform and there was little or no vibration of the pump due to the mixing. This indicated that the air was dispersed throughout the fluid, making the mass more uniform when it entered the impeller. If this location of the air port presents mechanical difficulties, the same smoothness of operation can be obtained by locating several ports on equal spaces around the periphery of the intake. A summary of the data for the three conditions is shown on figure 20.

The exhaust port should be located as near the discharge valve as possible or in a place such that there will not be air trapped in the system when the main valve is opened. The rate at which the load is applied to the motor depends upon the opening of the exhaust valve and the diameter of the bypass. For the small-diameter bypass, 0.67 inch, the rate at which the air is released seems to have little effect, but with the 2-inch diameter bypass there is an appreciable difference. A summary of the data is shown on figure 21.

CHAPTER III - CONCLUSIONS

18. Summary.

(a) If the starting device is jet-propelled with buckets on the impeller, the size of the bypass is governed by the discharge from the jets. The buckets reduce the efficiency of the pump approximately four percent.

(b) If the starting device is an auxiliary motor, a Pelton wheel housed in a separate case, or the low-voltage scheme in connection with the regular motor, the diameter of the bypass need not be larger than is necessary to circulate the cooling water.

(c) The power required to drive the impeller at rated speed in air is about 10 percent of the full-load demand, whereas with water in the volute it amounts to about 53 percent.

(d) The starting and shut-down cycles, as proposed in chapter II, sections 12 and 14, reduce the shock to the electrical system sufficiently to make the method practical.

(e) The compressed-air-supply port to the pump should be located on the intake side whereas the air-exhaust port should be on the discharge side at a point where all of the air in the system can be exhausted.

(f) The rate at which the air is exhausted from the pump during the starting cycle is a major factor in controlling the rate at which the load is applied to the motor.

(g) The rate at which the air enters the pump, and the diameter of the bypass control the rate at which the load on the motor reduces during the shut-down cycle.

(h) The test data indicated that the diameter of the bypass was the most important factor controlling the rate at which the power decreased during the shut-down cycle.

- - -

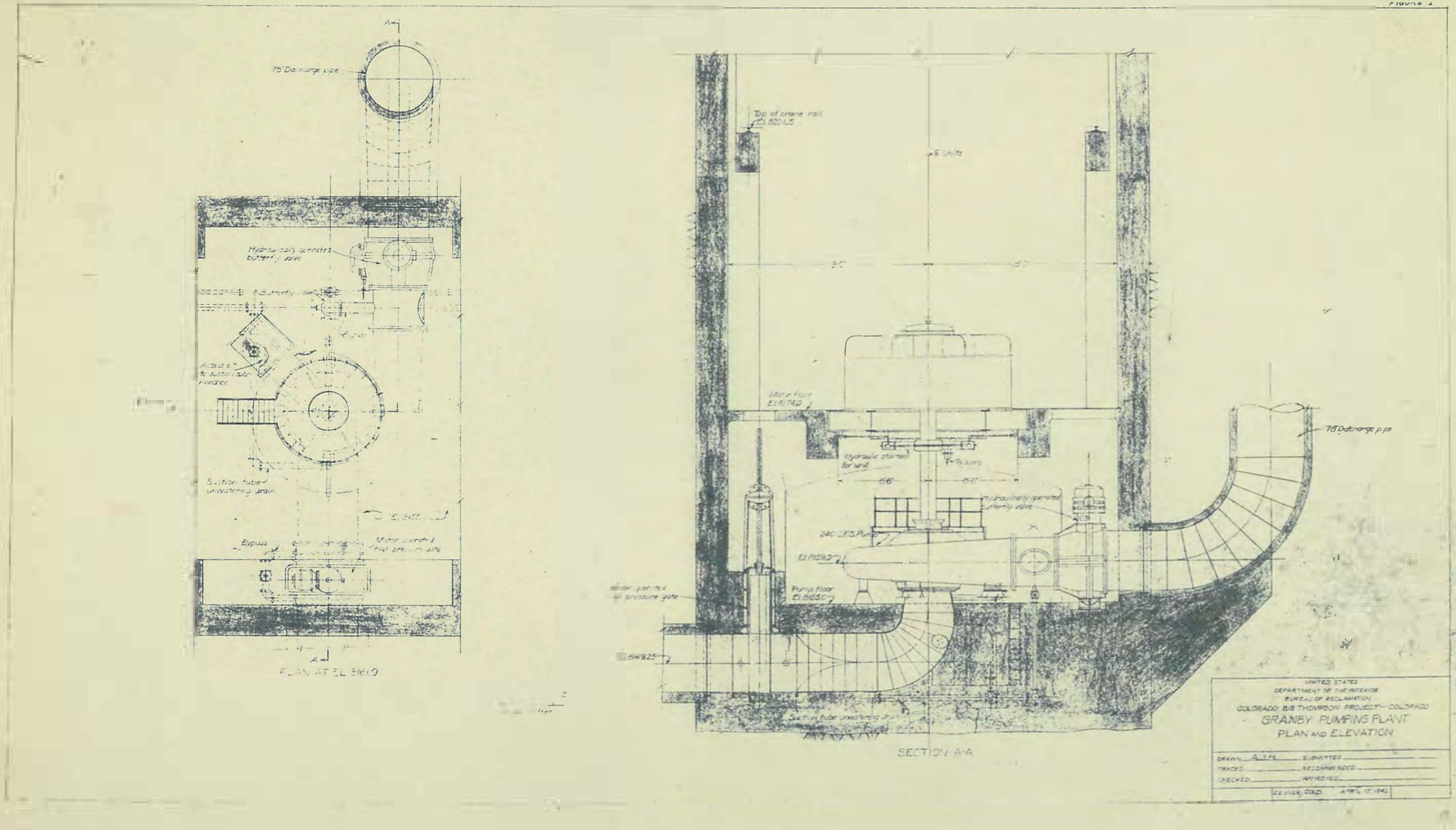
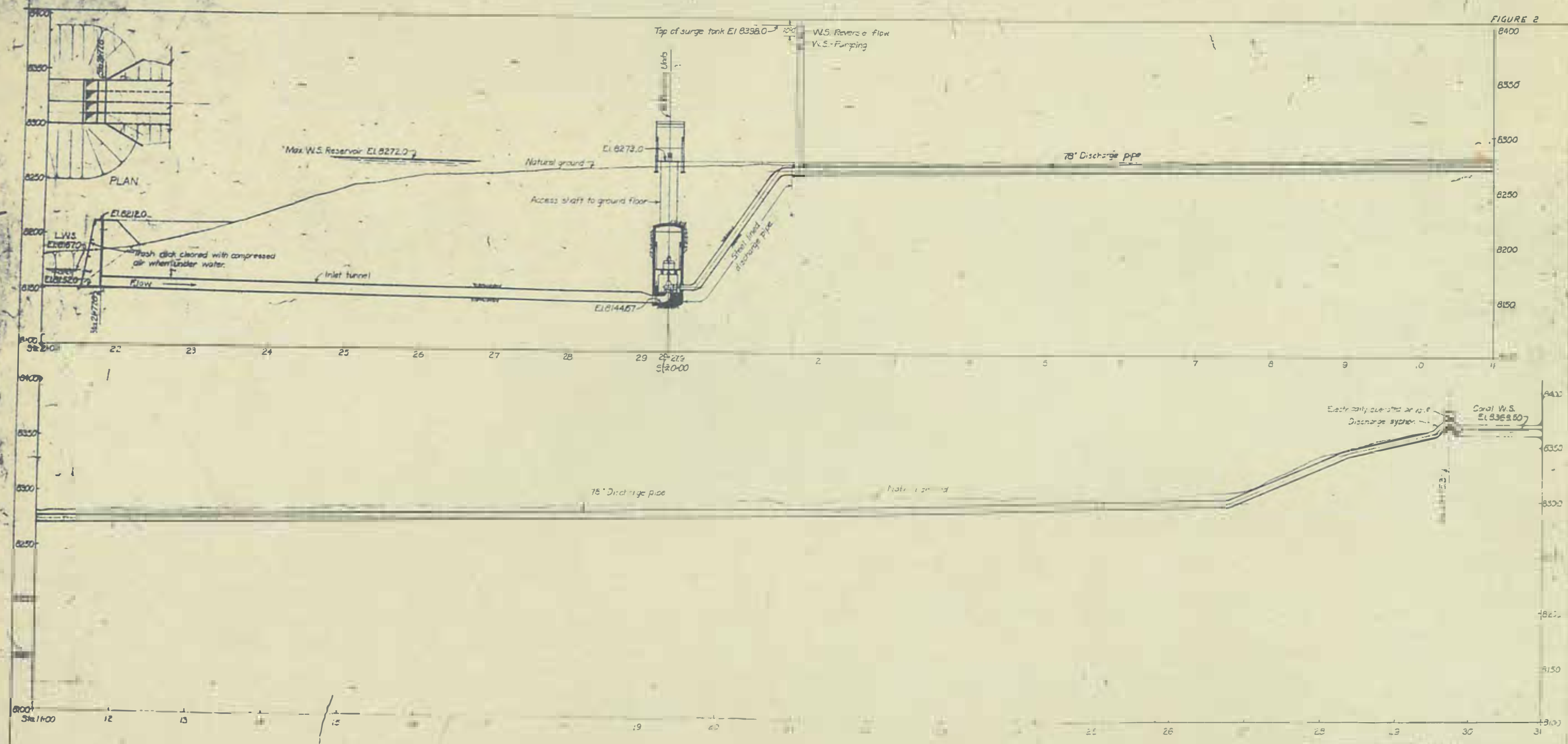

78" Discharge pipe
Hydraulically operated butterfly valve
Suction tube unwatering drain
Bypass
PLAN AT EL. 8160
Top of crane rail
Motor floor
Hydraulic starter for unit
To sump
240 C.F.S. Pump
Pump floor
78" Discharge pipe
SECTION A-A
UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
COLORADO BIG THOMPSON PROJECT—COLORADO
GRANBY PUMPING PLANT
PLAN AND ELEVATION
DRAWN A.J.H. SUBMITTED
TRACED RECOMMENDED
CHECKED APPROVED
DENVER, COLO. APRIL 17, 1942

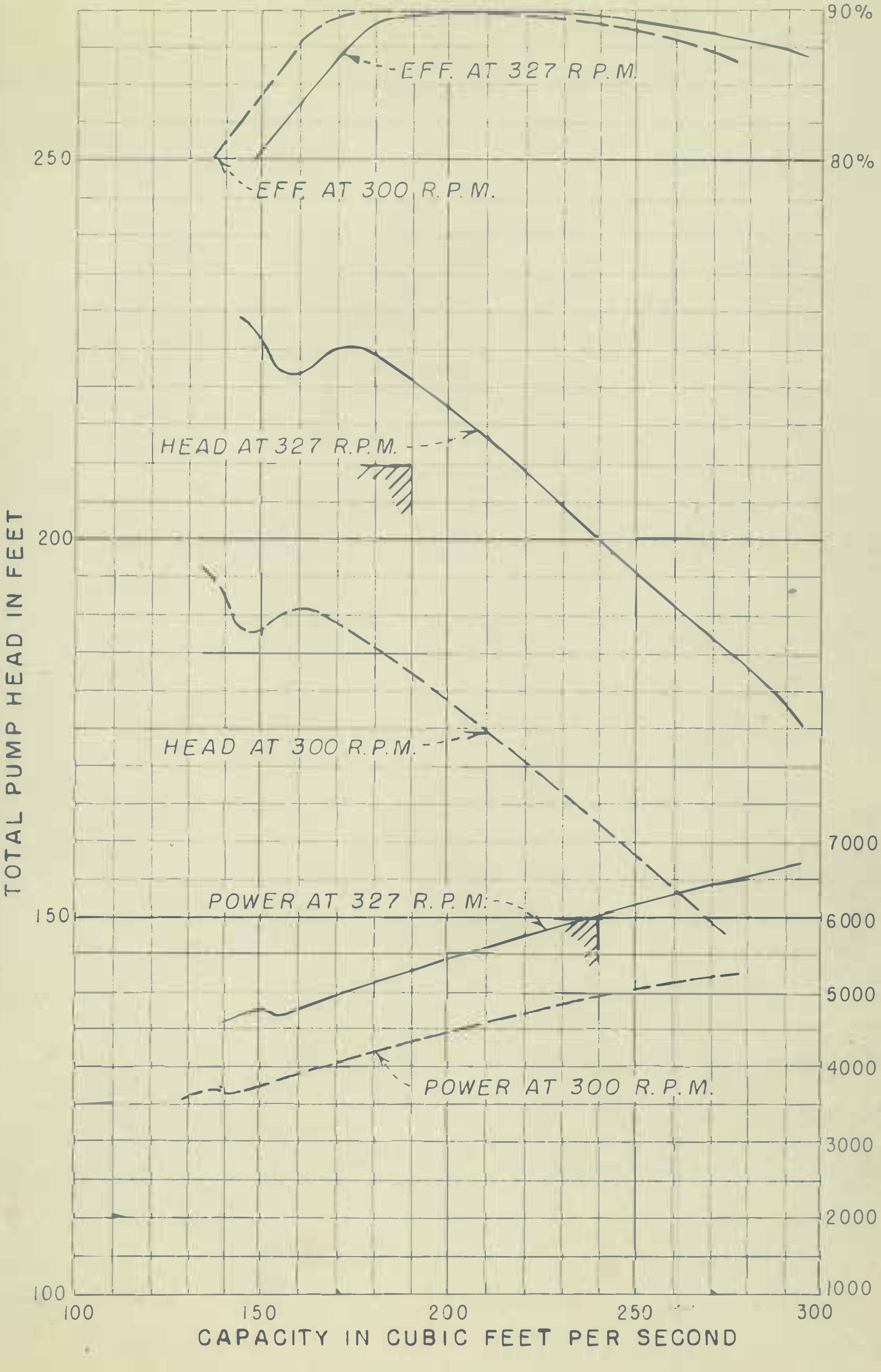
FIGURE 2



DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
COLORADO-BIG THOMPSON PROJECT — COLORADO

GRANBY PUMPING PLANT
SECTION OF PROFILE SHOWING INTAKE
TUNNEL, PUMPING PLANT & DISCHARGE PIPE

DRAWN A.J.H. SUBMITTED
TRACED RECOMMENDED
CHECKED APPROVED
DENVER, COLO.



GRANBY DAM PUMPING PLANT
PROPOSED PROTOTYPE PUMP CHARACTERISTICS

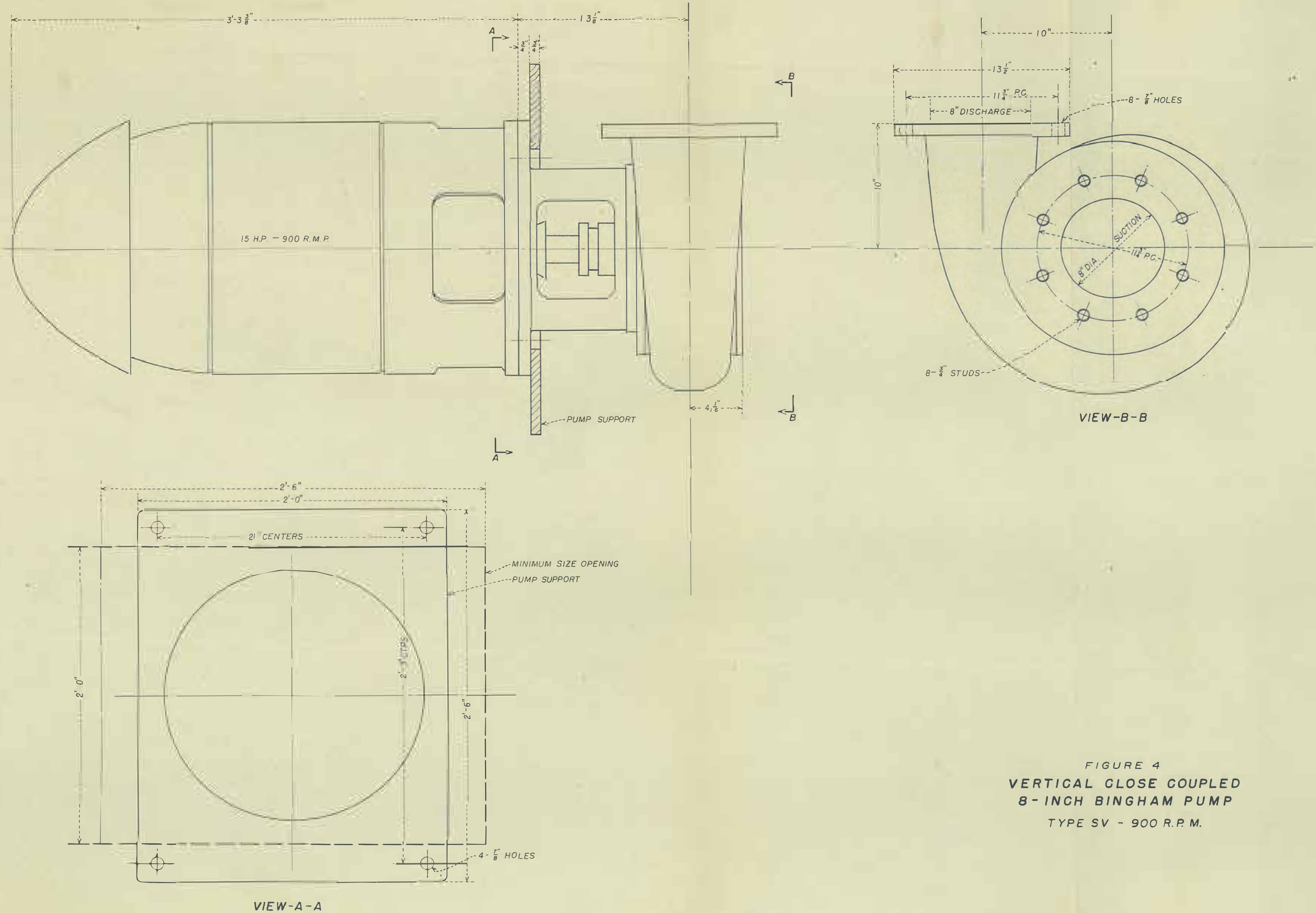
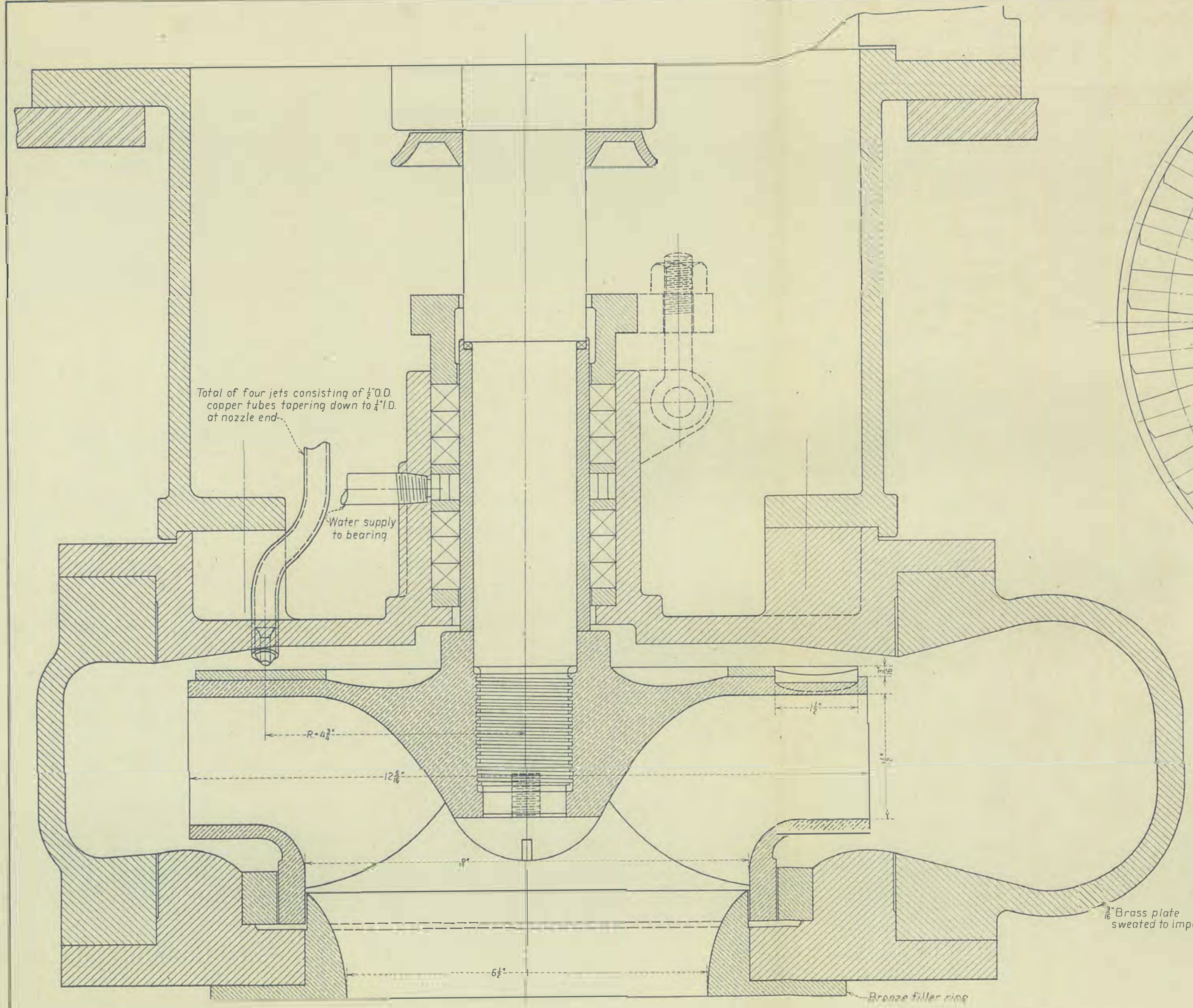


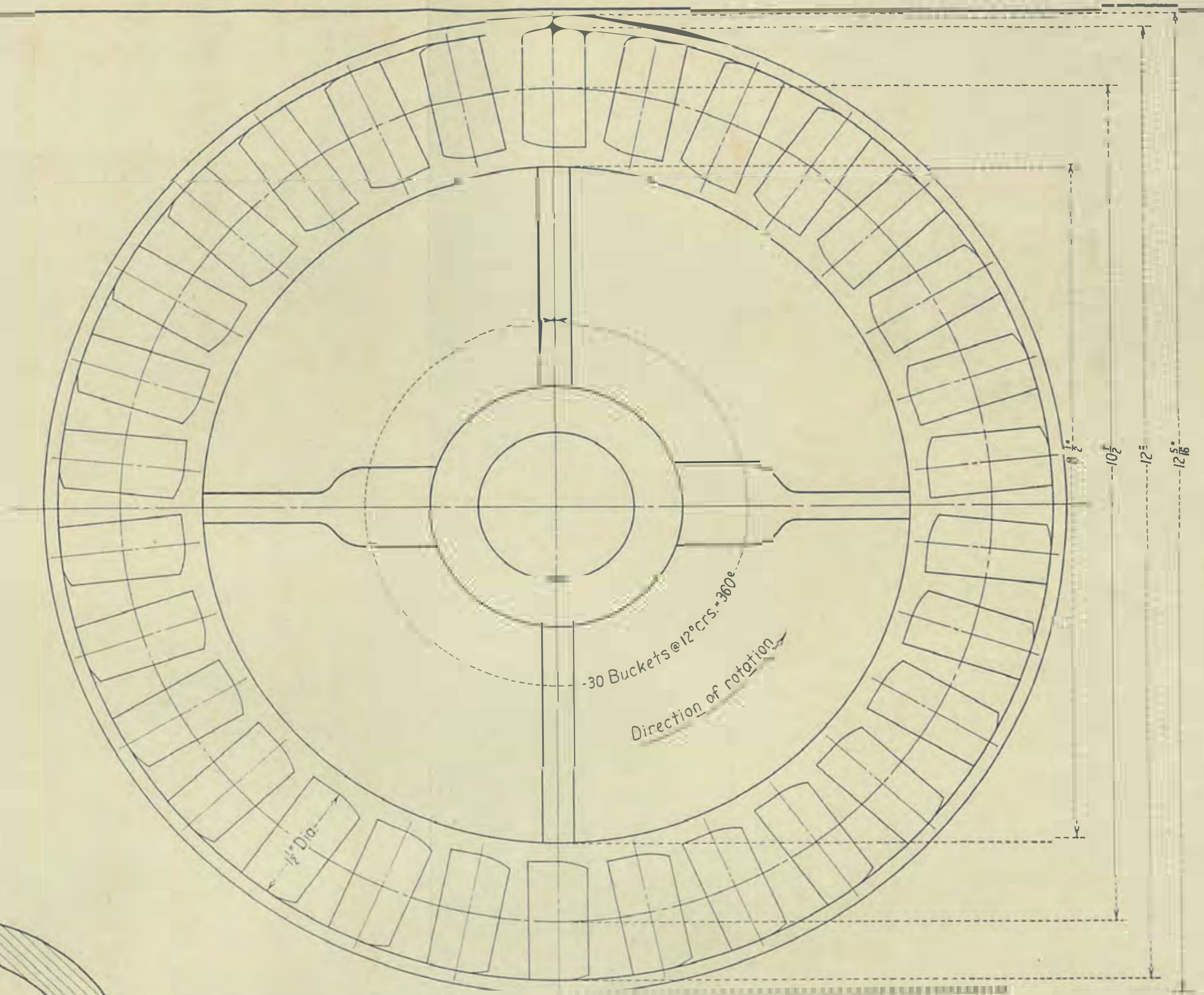
FIGURE 4
VERTICAL CLOSE COUPLED
8-INCH BINGHAM PUMP
TYPE SV - 900 R.P.M.

FIGURE 4

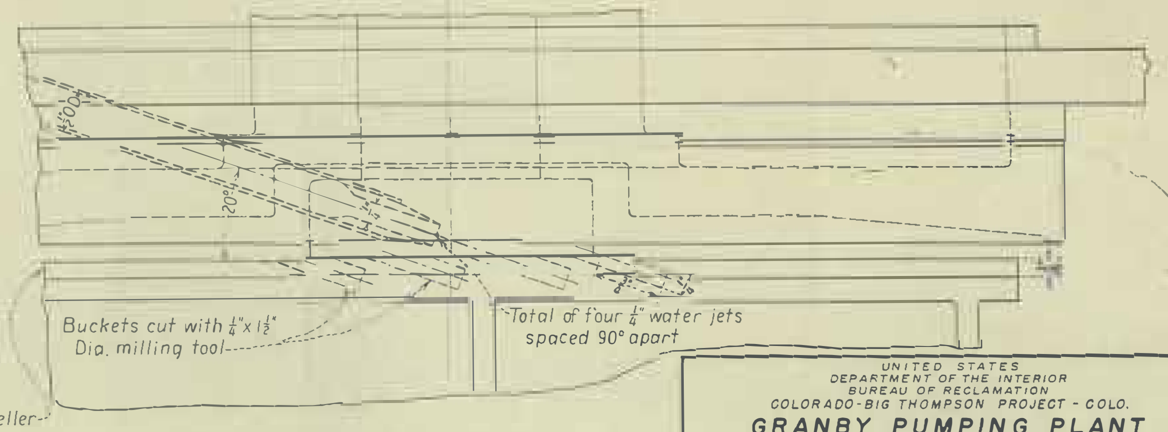
FIGURE 5



VERTICAL CROSS SECTION OF PUMP



TOP VIEW OF IMPELLER



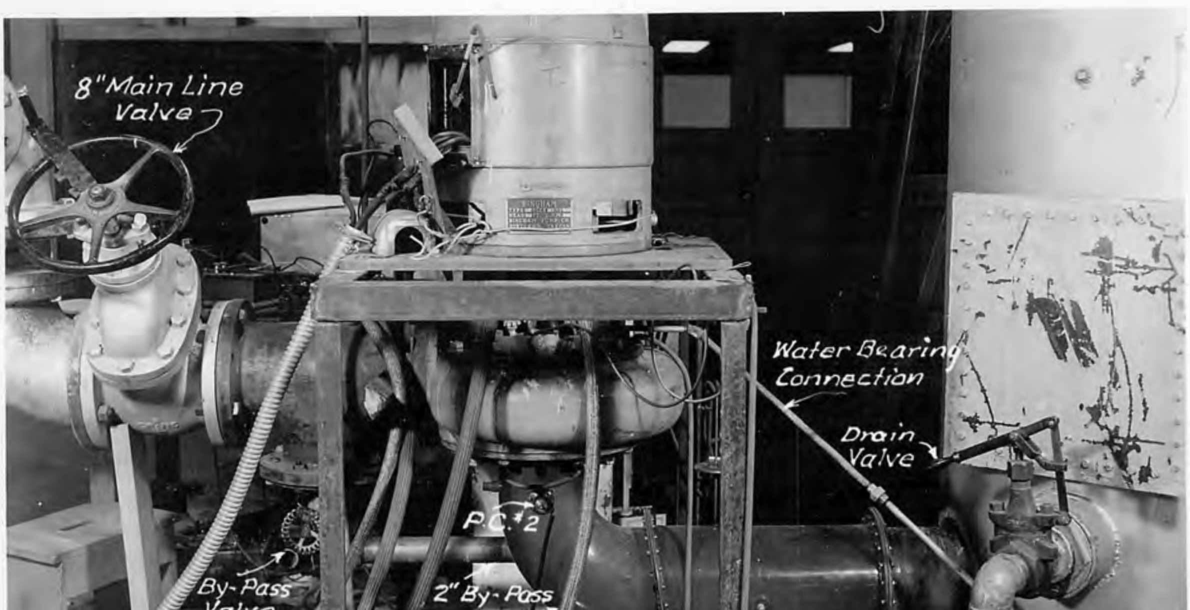
SIDE VIEW OF IMPELLER

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
COLORADO-BIG THOMPSON PROJECT - COLO.

GRANBY PUMPING PLANT
REMODELED BINGHAM 8-INCH VERTICAL CLOSE COUPLED PUMP
TYPE SV - 15 H.P. - 900 R.P.M.
AS USED FOR MODEL PUMP TESTS

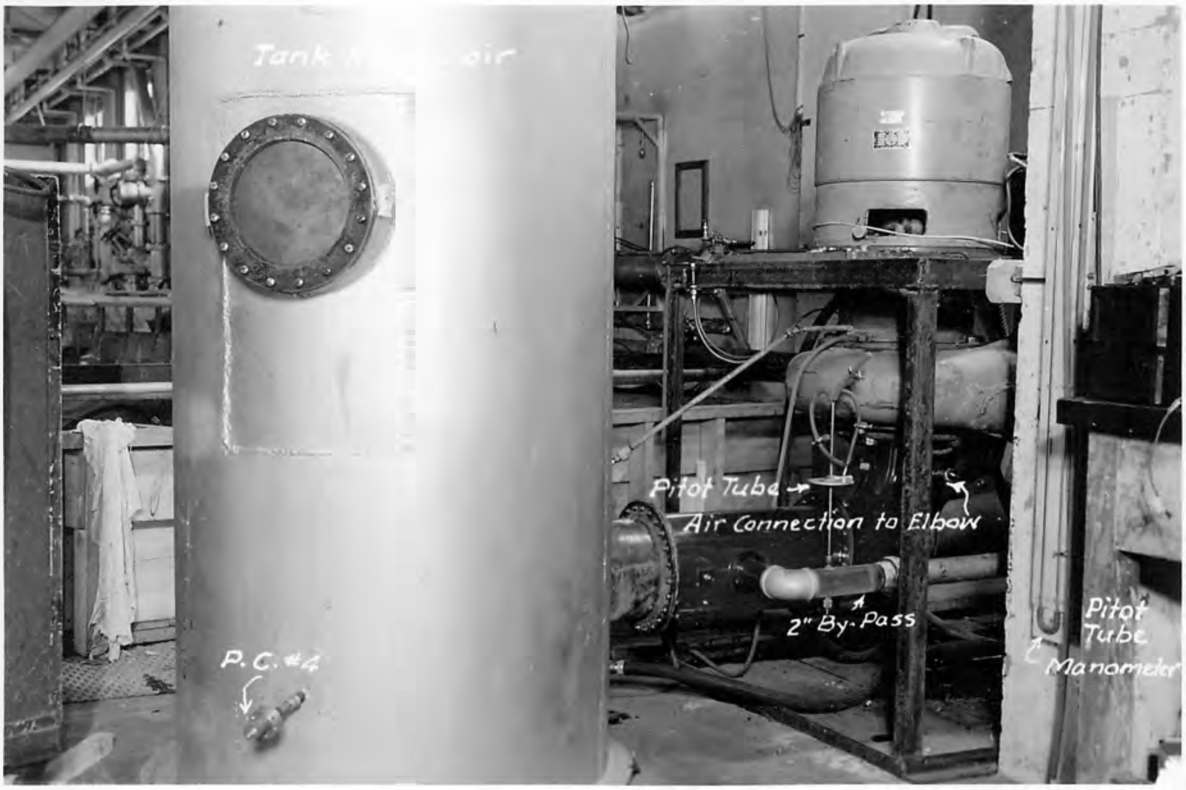
DRAWN J.N.B. SUBMITTED
TRACED C.F.F. RECOMMENDED
CHECKED APPROVED
DENVER, COLORADO APRIL 7, 1942
245-D-1721

A. TEST ARRANGEMENT.

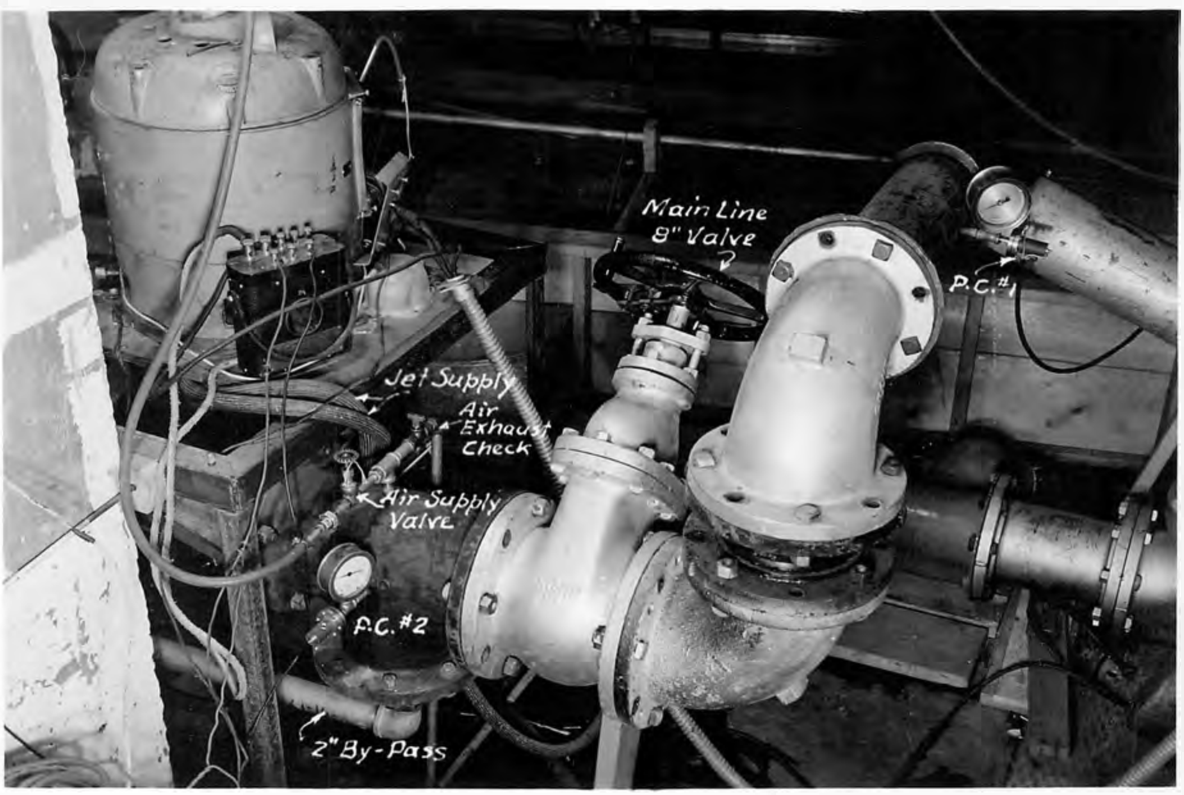


B. PUMP SHOWING TRANSPARENT SUCTION PIPE AND ELBOW.

GRANBY PUMP TEST ARRANGEMENT

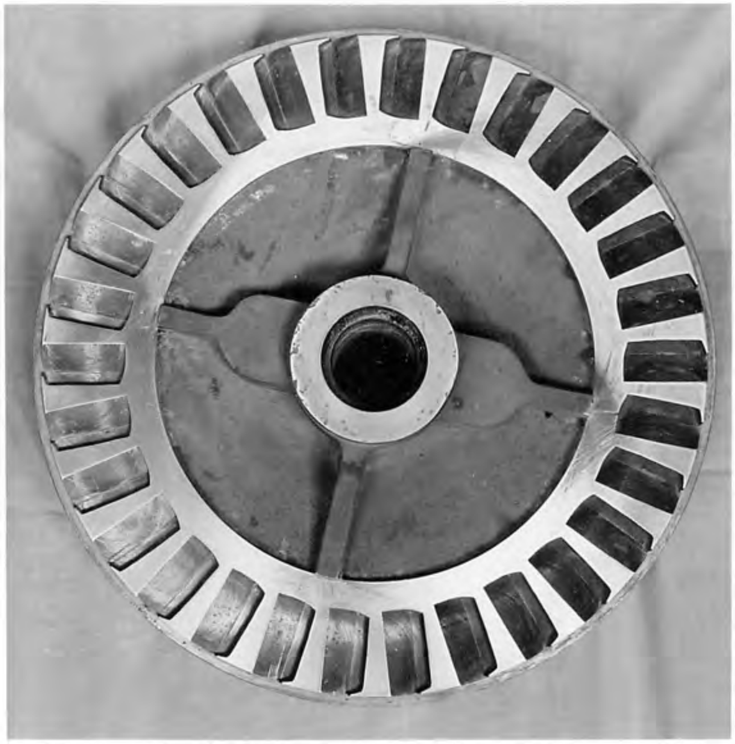


A. PUMP SHOWING BY-PASS AND PITOT TUBE.



B. PUMP SHOWING BY-PASS, AIR SUPPLY AND EXHAUST CONNECTIONS.

DETAILS OF GRANBY PUMP TEST EQUIPMENT



A. TOP OF REMODELED PUMP IMPELLER WITH THIRTY BUCKETS.

B. PUMP IMPELLER WITH BUCKETS FILLED.

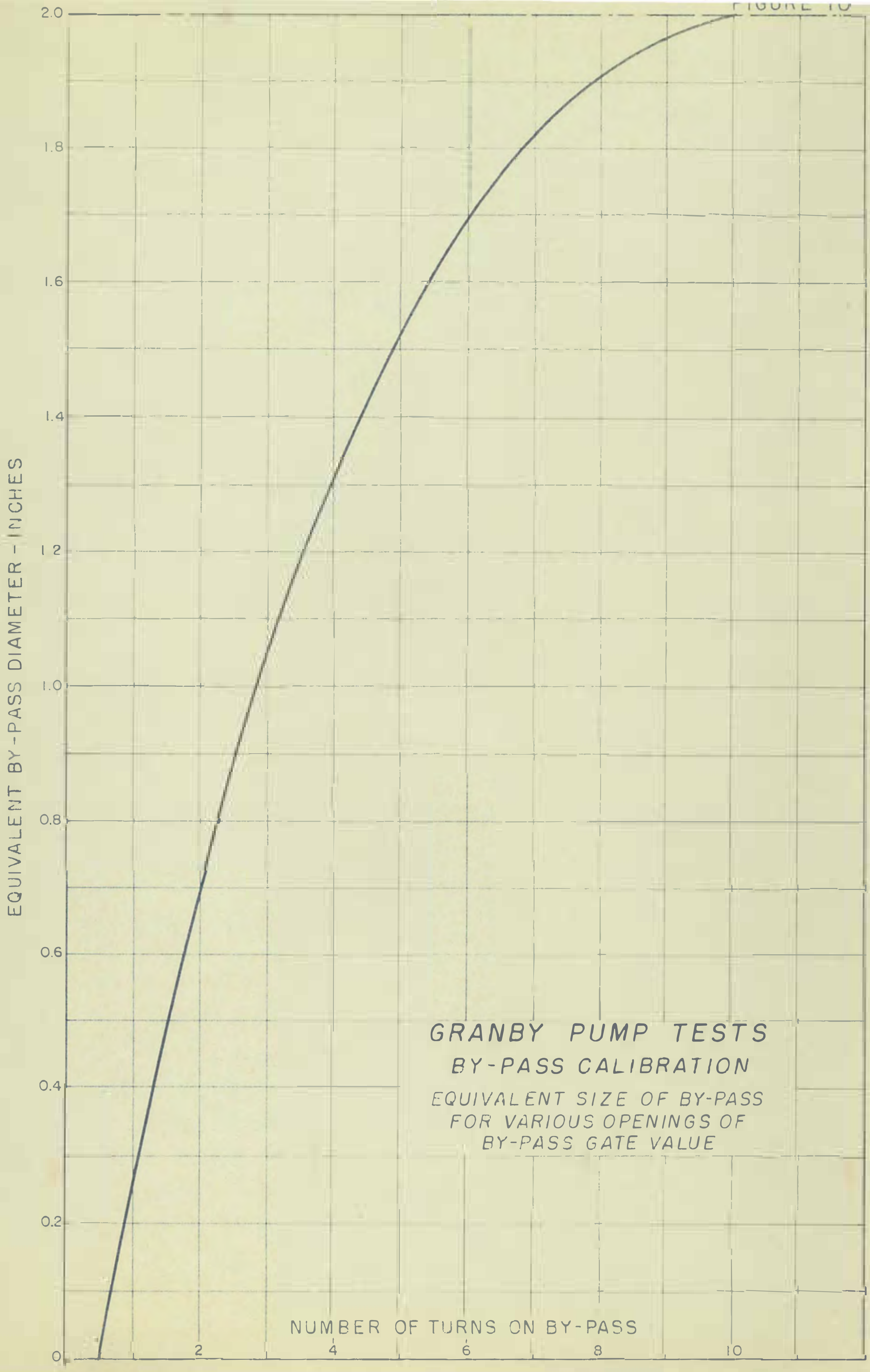
GRANBY TEST PUMP IMPELLER

TYPE SV - SIZE 8" - R. P. M. 900

CHARACTERISTIC CURVE SHEET - BINGHAM PUMP

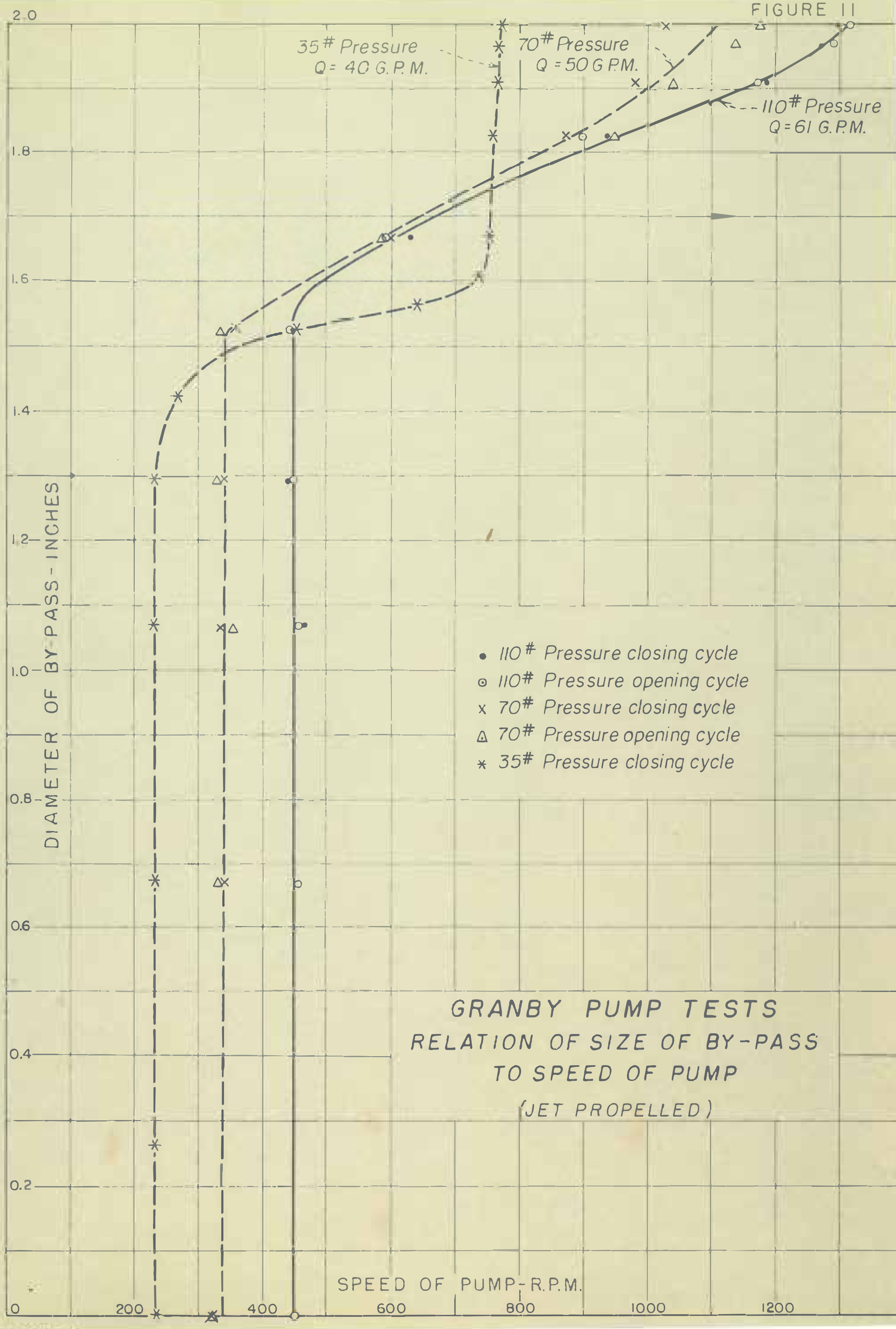
FIGURE 9

FIGURE 10

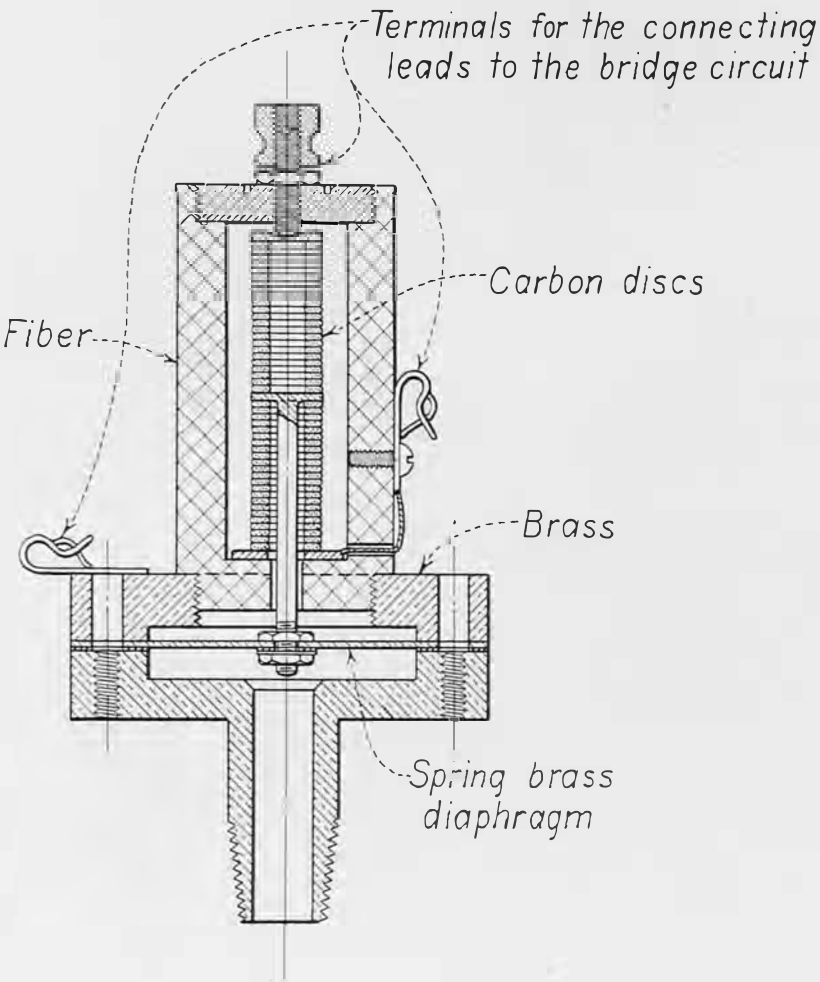


GRANBY PUMP TESTS
BY-PASS CALIBRATION
EQUIVALENT SIZE OF BY-PASS FOR VARIOUS OPENINGS OF BY-PASS GATE VALUE
EQUIVALENT BY-PASS DIAMETER - INCHES
NUMBER OF TURNS ON BY-PASS
0
0.2
0.4
0.6
0.8
1.0
1.2
1.4
1.6
1.8
2.0
2
4
6
8
10

FIGURE 11



GRANBY PUMP TESTS
RELATION OF SIZE OF BY-PASS
TO SPEED OF PUMP
(JET PROPELLED)



SECTION THRU CELL

GENERAL VIEW OF CELL

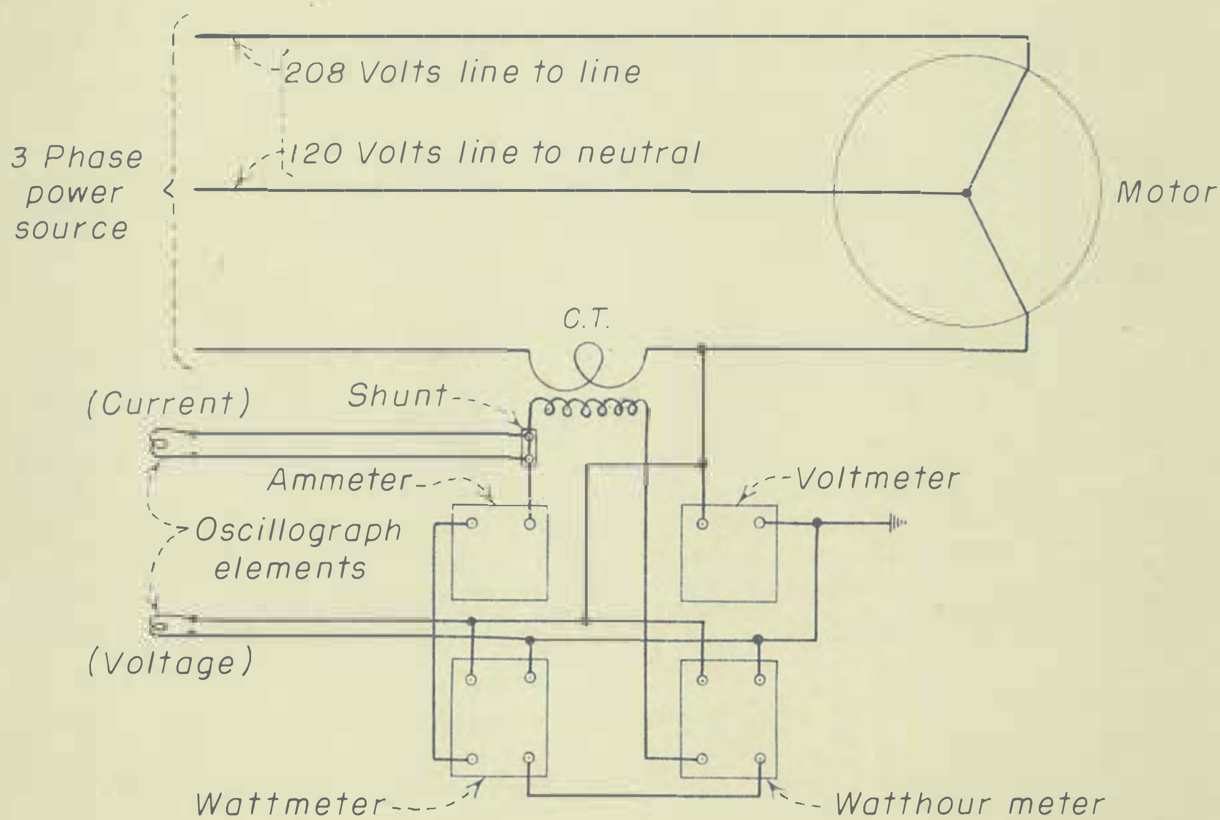
DETAILS OF CARBON-PILE PRESSURE CELL USED IN GRANBY PUMP MODEL TESTS

FIGURE 12

FIGURE 15

GRANBY PUMP MODEL TESTS

DIAGRAM OF ELECTRICAL CONNECTIONS FOR THE INSTRUMENTS



MOTOR———— 15 H.P., 900 R.P.M., 220/440 Volts, 3 phase.

C.T. —————— 800/600/100/50/20/10/5 Ratio current transformer, G.E. No. 6396550.

AMMETER —— 5 Amp., A.C., 250 Amp. scale, Thomson No. 303698.

VOLTMETER — 150/300 Volt range, Weston No. 5949

WATTMETER — 5 Amp., 150 volt, 1ϕ, Weston No. 3700

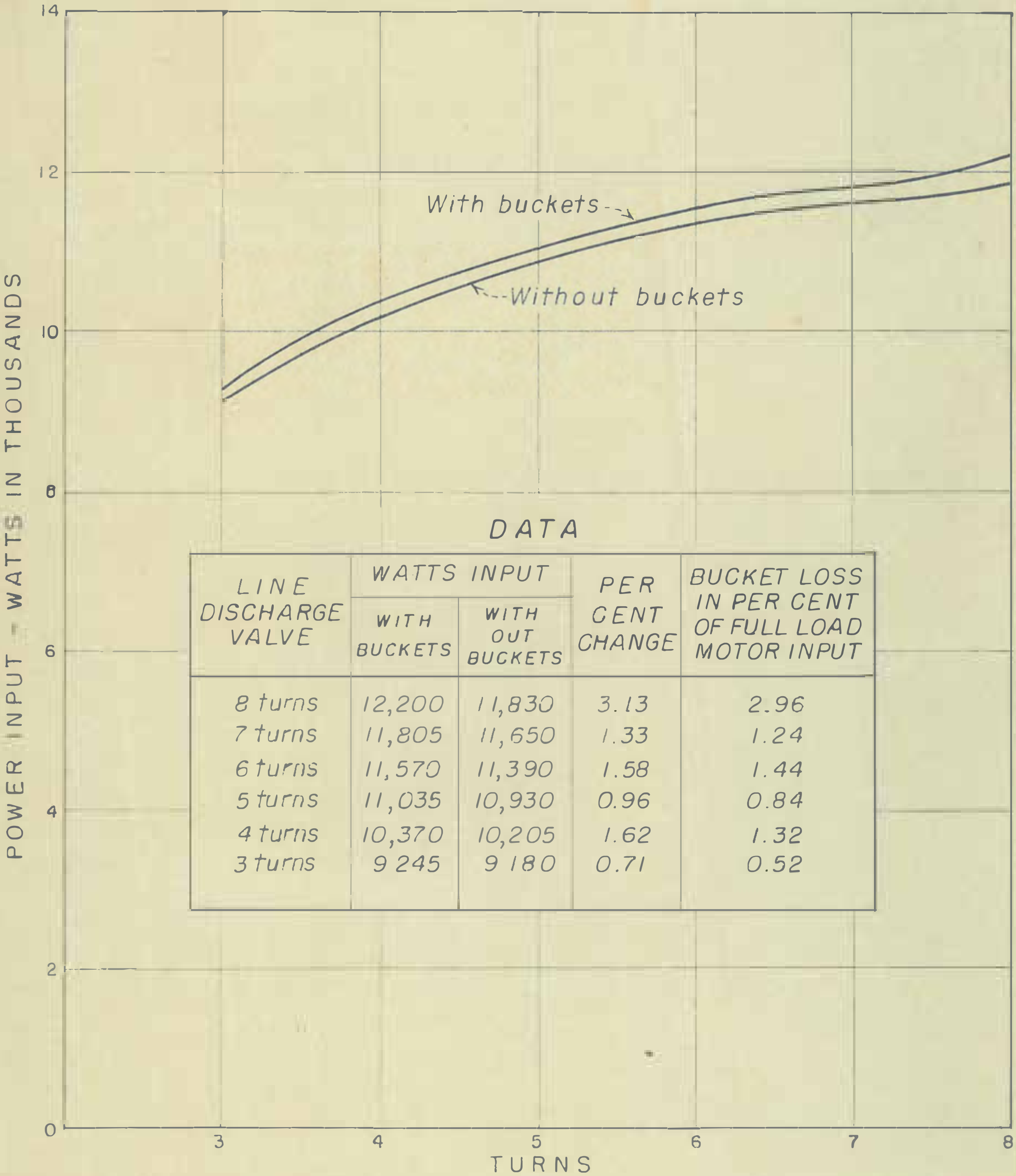
WATTHOUR METER — 1/5 Amp., 150 volt, 1ϕ, K = 1/3 WATT/REV., Sangamo No. 2168925

SHUNT———— 3 Feet of No. 14 A.W.G. solid copper wire

W.A.M. 3-42

GRANBY PUMP MODEL TESTS

MEASUREMENT OF POWER INPUT TO PUMP MOTOR AS THE DISCHARGE VALVE POSITION IS VARIED UNDER CONDITIONS OF WITH AND WITHOUT BUCKETS



DATA

LINE DISCHARGE VALVE	WATTS INPUT		PER CENT CHANGE	BUCKET LOSS IN PER CENT OF FULL LOAD MOTOR INPUT
	WITH BUCKETS	WITH OUT BUCKETS		
8 turns	12,200	11,830	3.13	2.96
7 turns	11,805	11,650	1.33	1.24
6 turns	11,570	11,390	1.58	1.44
5 turns	11,035	10,930	0.96	0.84
4 turns	10,370	10,205	1.62	1.32
3 turns	9 245	9 180	0.71	0.52

MAIN LINE DISCHARGE VALVE

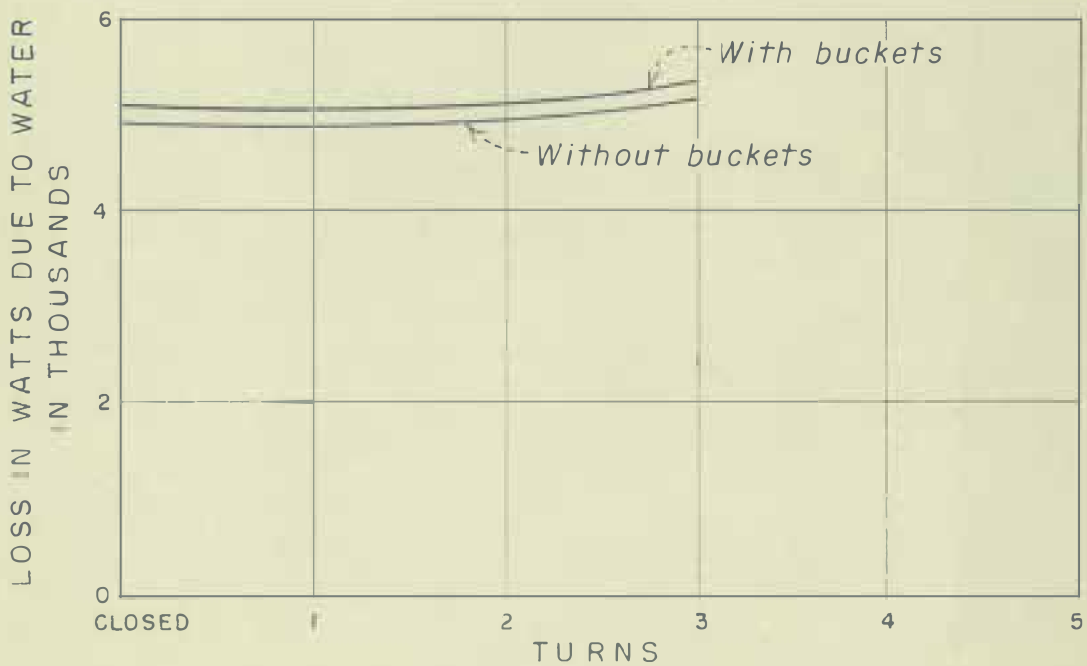
W. A. M. 3-42

GRANBY PUMP MODEL TESTS

MEASUREMENT OF LOSSES DUE TO WATER FRICTION WITH DISCHARGE VALVE CLOSED WITH AND WITHOUT BUCKETS

DATA

CONDITION	BY PASS VALVE	WATTS INPUT		PER CENT CHANGE	BUCKET LOSS IN PER CENT OF FULL LOAD MOTOR INPUT
		WITH BUCKETS	WITH OUT BUCKETS		
Total watts with water in pump case	3 turns	6,554	6,407	2.30	1.26
	2 turns	6,384	6,202	2.94	1.54
	1 turn	6,292	6,116	2.88	1.49
	Closed	6 296	6 124	2.73	1.42
Watts with air in pump case	Air	1 255	1 265	0.79	0.08
Watts loss due to water friction (From above readings)	3 turns	5,299	5,142	3.06	1.26
	2 turns	5,129	4,937	3.89	1.54
	1 turn	5,037	4,851	3.44	1.49
	Closed	5,041	4,864	3.64	1.42



BY-PASS VALVE POSITION

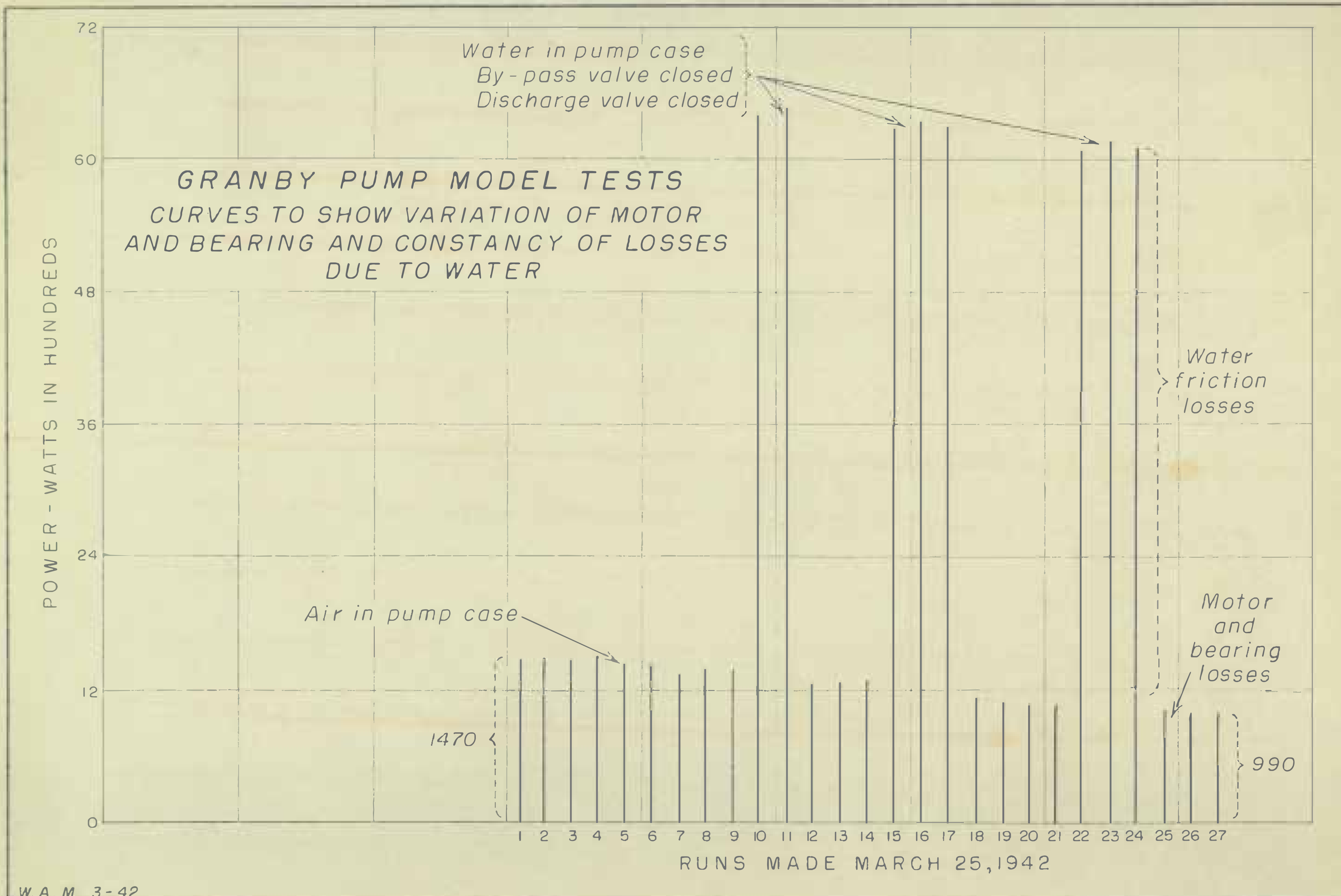
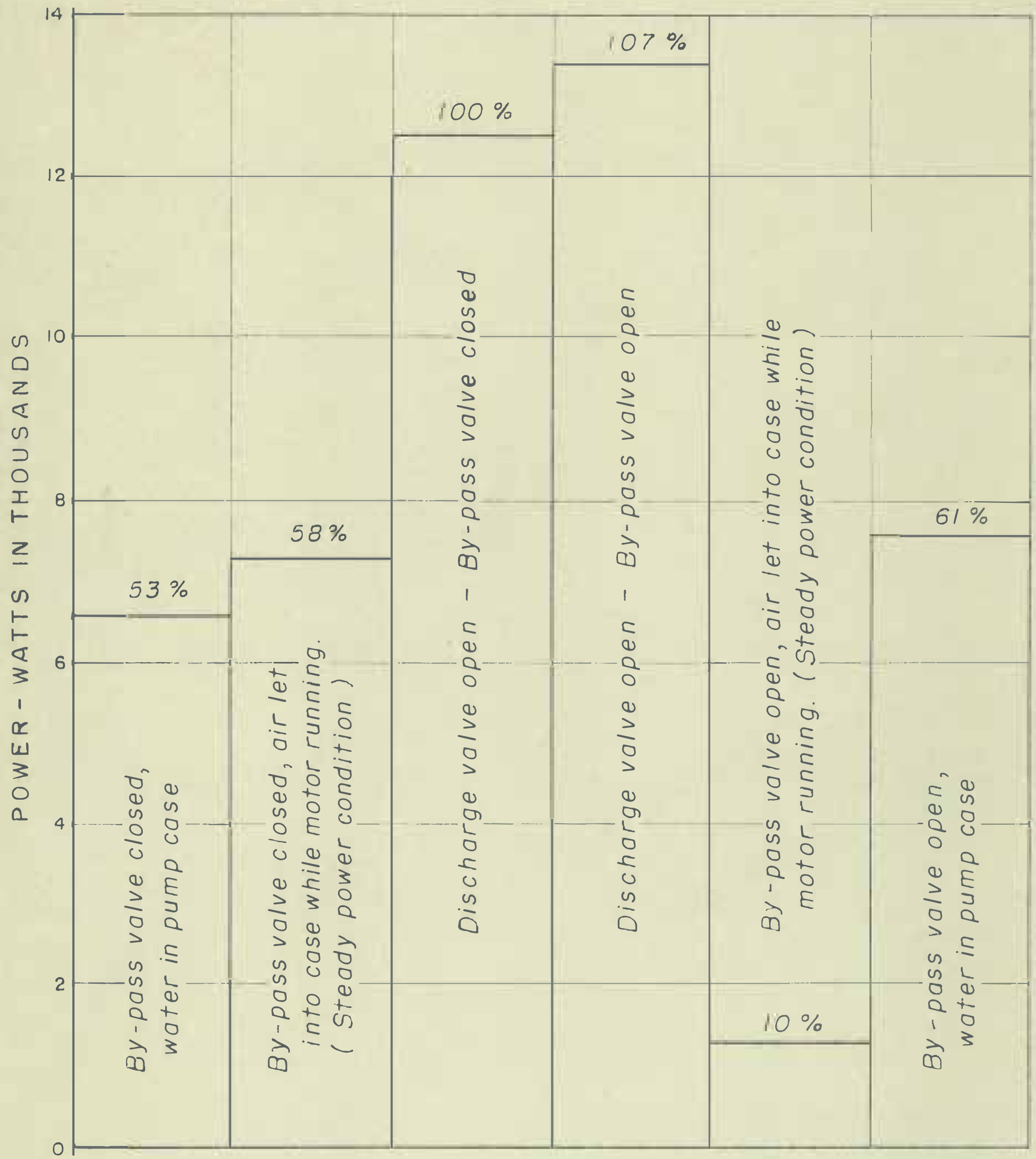

GRANBY PUMP MODEL TESTS
CURVES TO SHOW VARIATION OF MOTOR AND BEARING AND CONSTANCY OF LOSSES DUE TO WATER
POWER - WATTS IN HUNDREDS
72
60
48
36
24
12
0
Water in pump case
By-pass valve closed
Discharge valve closed
Air in pump case
1470
Water friction losses
Motor and bearing losses
990
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27
RUNS MADE MARCH 25, 1942
W. A. M. 3-42

FIGURE 1[illegible]

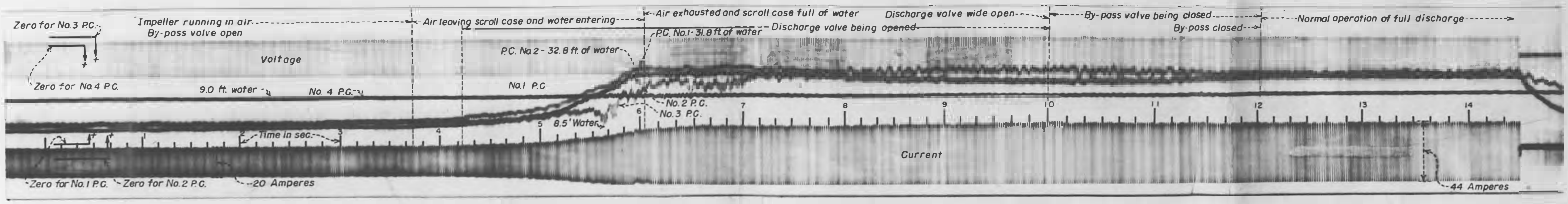
GRANBY PUMP MODEL TESTS

CURVES TO SHOW THE RELATIVE AMOUNT OF POWER TAKEN BY THE MOTOR FOR THE DIFFERENT CONDITIONS STUDIED IN THESE TESTS



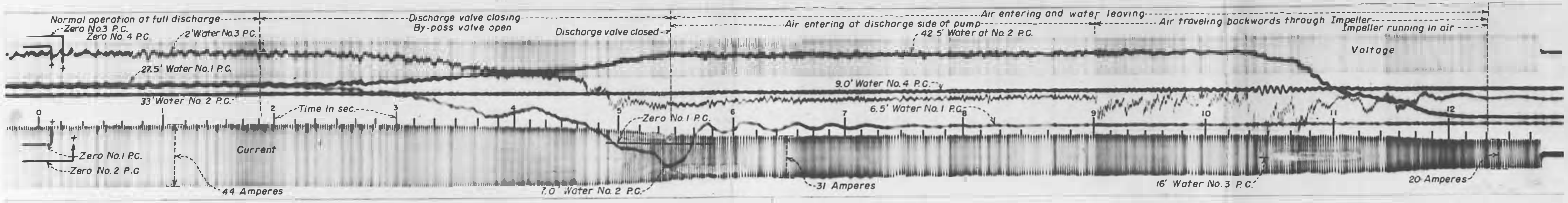
W A M 3-42

Fig 18

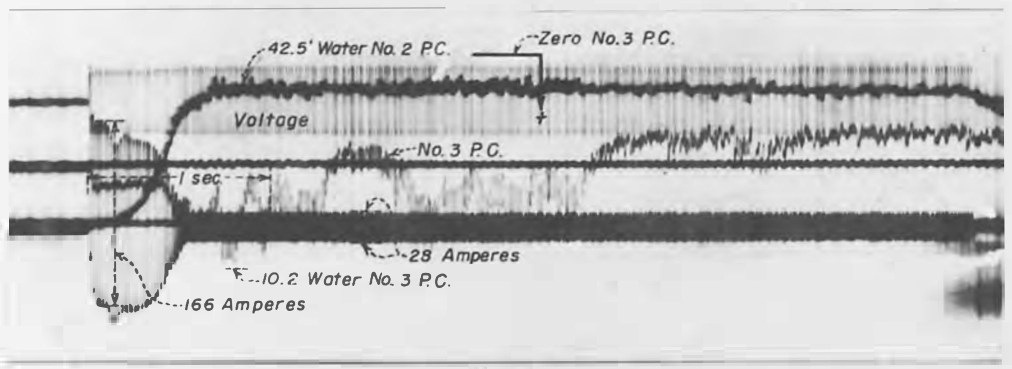


A. STARTING CYCLE

MOTOR BROUGHT TO SPEED BY HYDRAULIC JETS

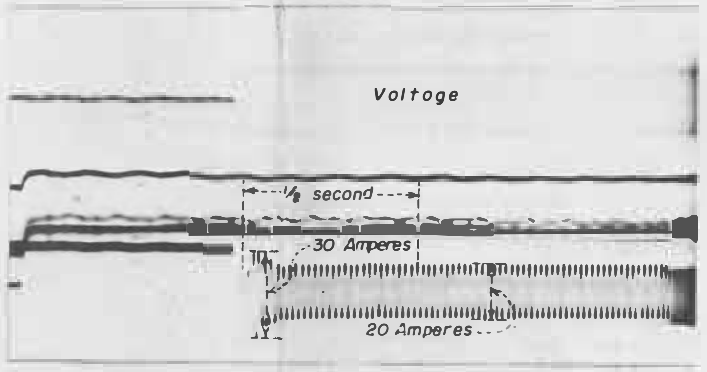


B. SHUT-DOWN CYCLE



C. MOTOR STARTING CURRENT

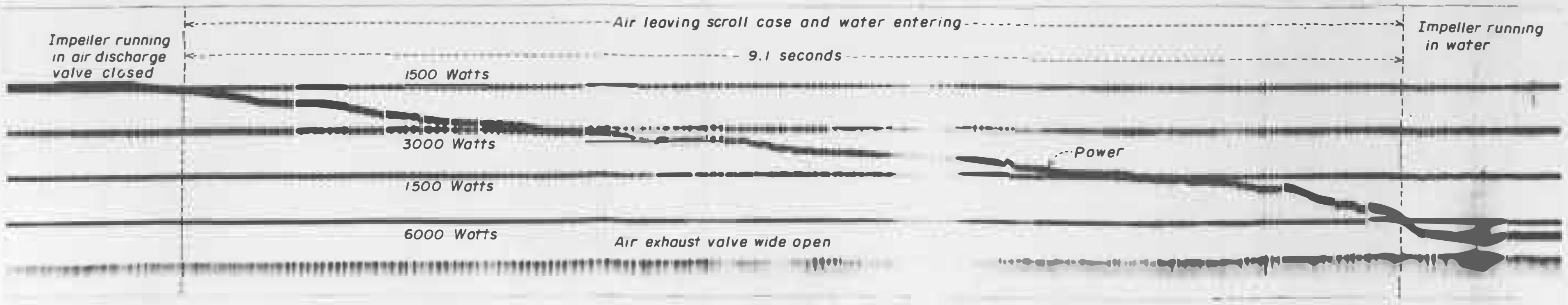
IMPELLER NOT ROTATING WHEN SWITCH WAS CLOSED



D. MAGNETIZING CURRENT

(ZERO POWER)

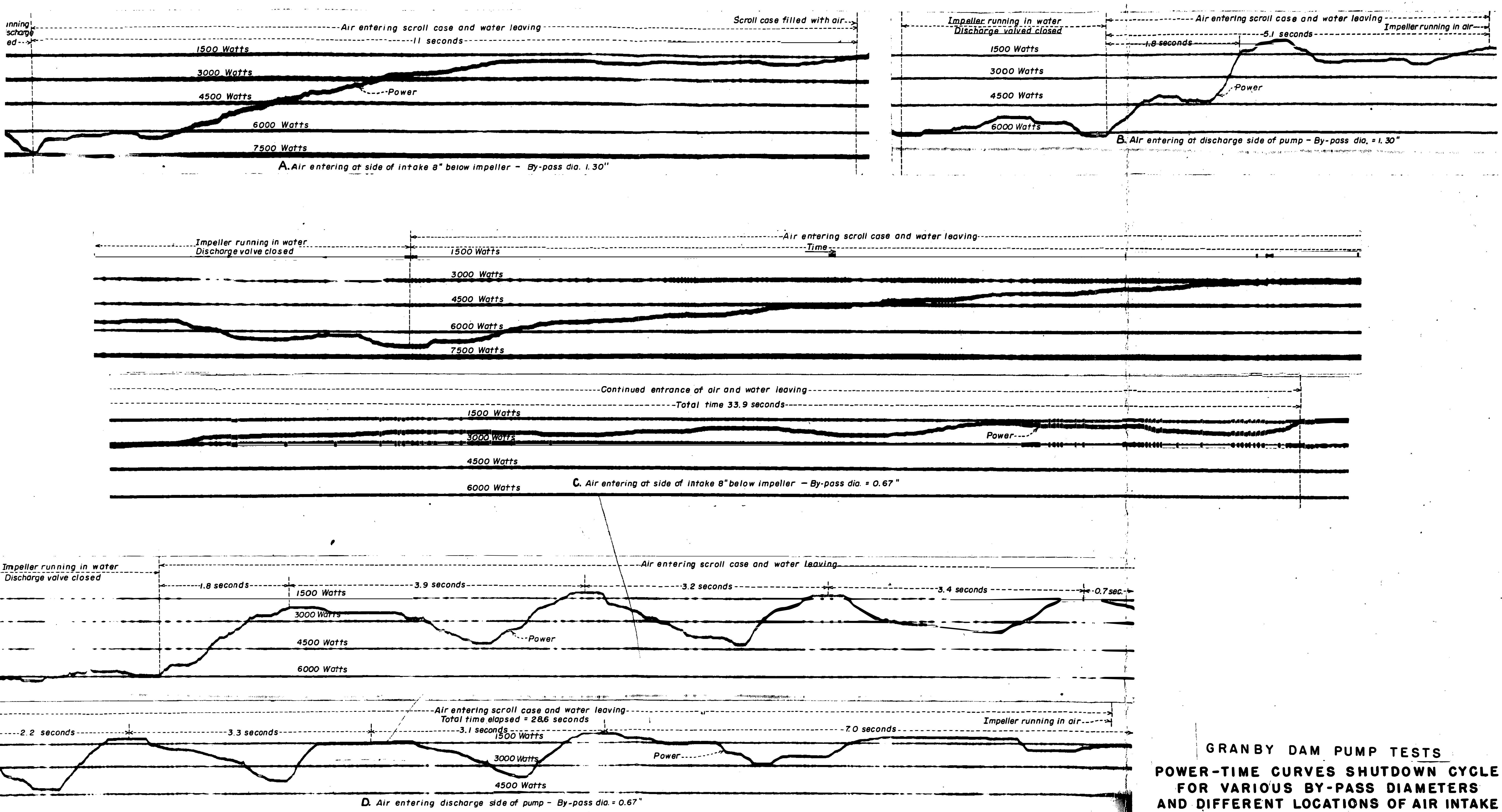
MOTOR RUNNING AT SYNCHRONOUS SPEED WHEN SWITCH CLOSED



E. POWER-TIME CURVE STARTING CYCLE

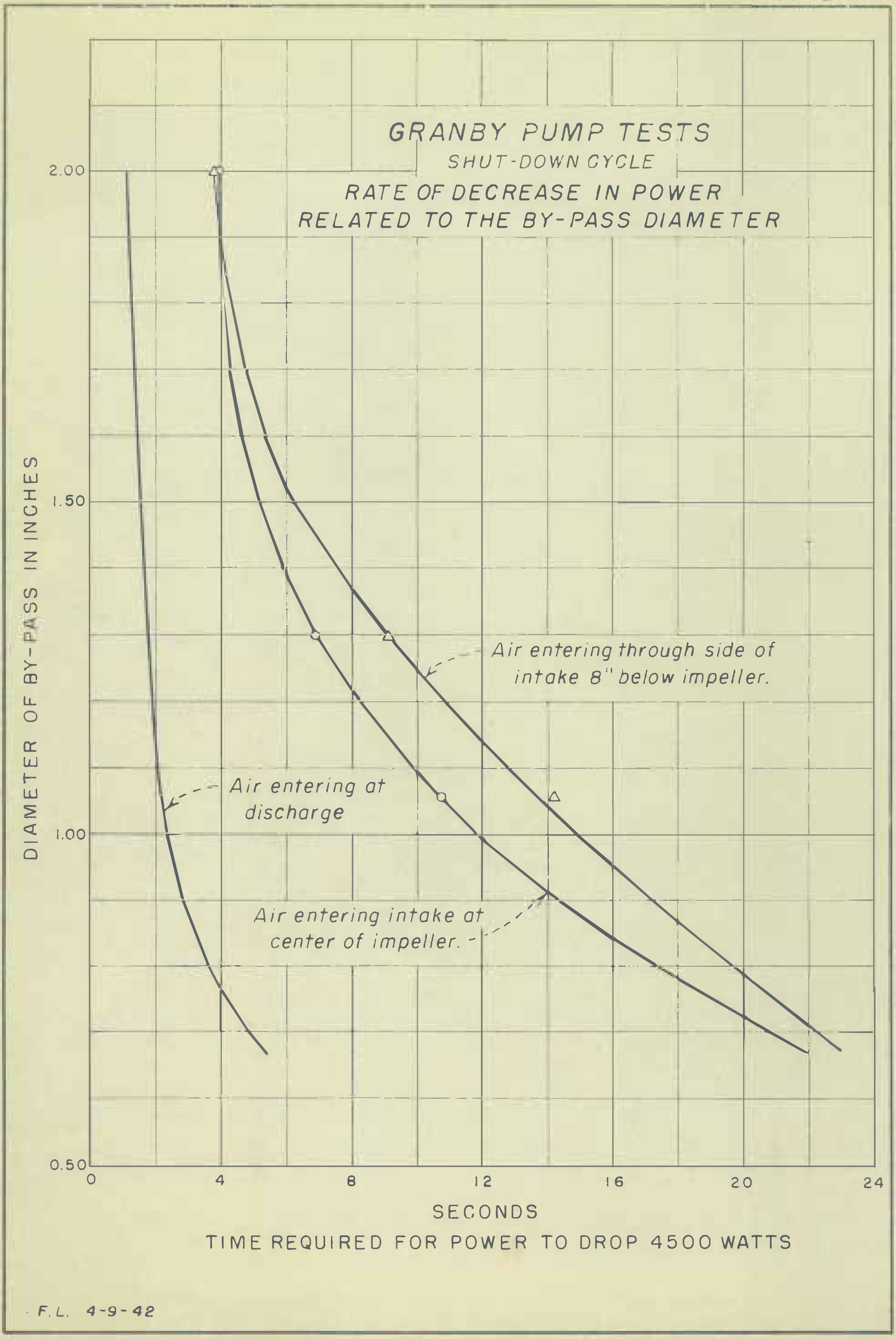
BY-PASS DIAMETER 1.30"

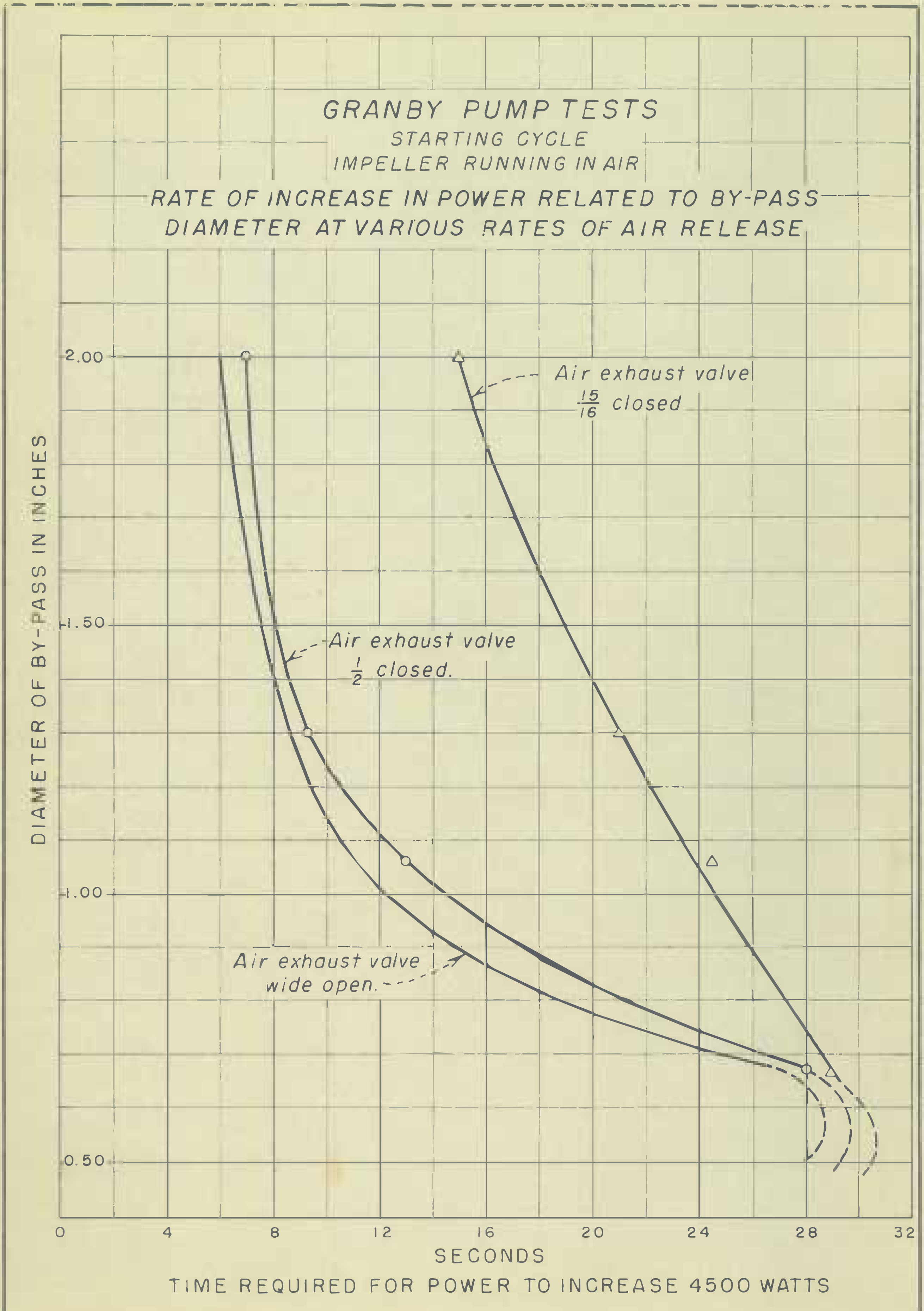
GRANBY DAM PUMP TESTS
OSCILLOGRAMS FOR STARTING AND SHUTDOWN CYCLES



GRANBY DAM PUMP TESTS

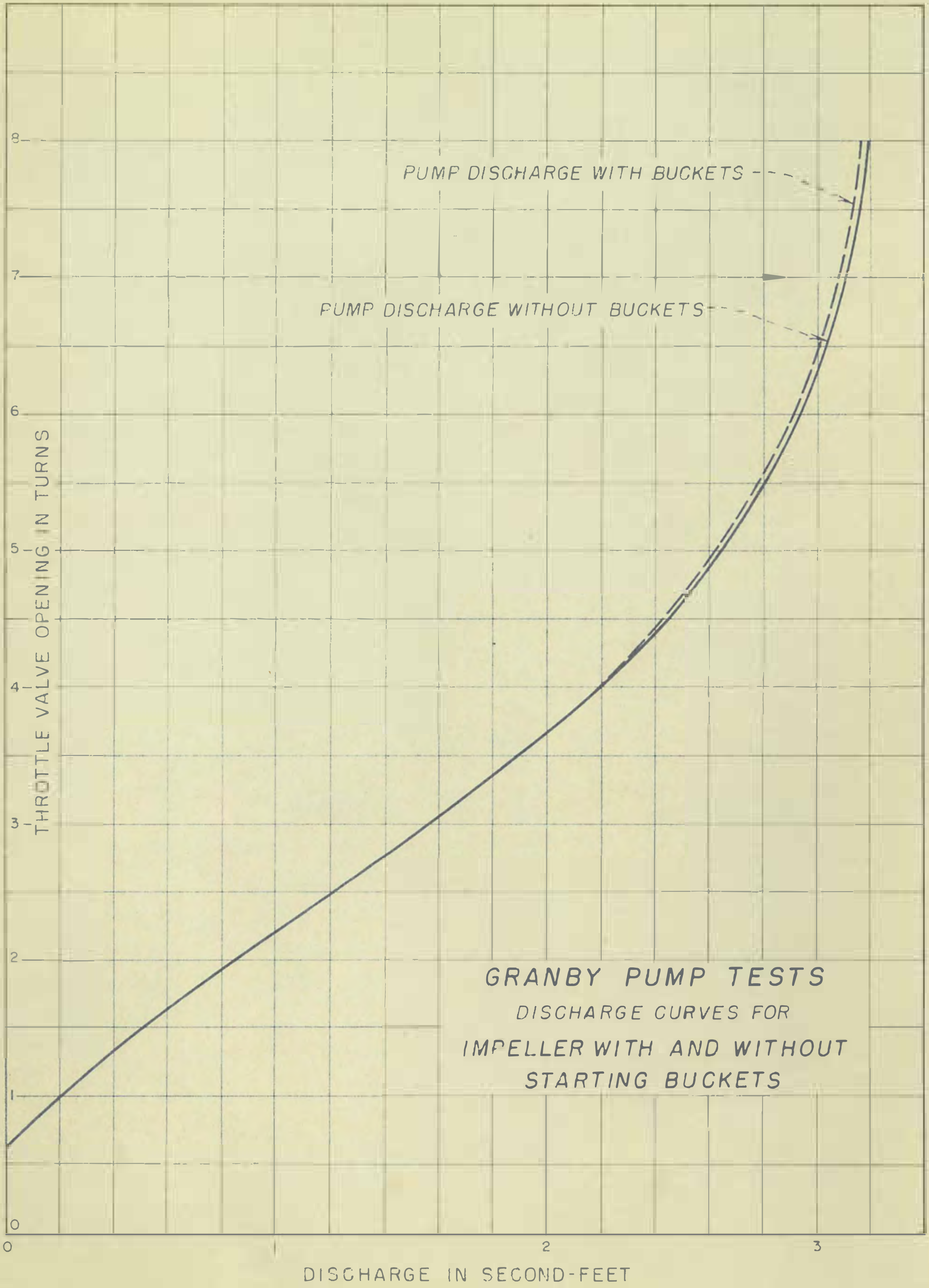
POWER-TIME CURVES SHUTDOWN CYCLE FOR VARIOUS BY-PASS DIAMETERS AND DIFFERENT LOCATIONS OF AIR INTAKE OPENING OF AIR VALVE CONSTANT


GRANBY PUMP TESTS
SHUT-DOWN CYCLE
RATE OF DECREASE IN POWER
RELATED TO THE BY-PASS DIAMETER
DIAMETER OF BY-PASS IN INCHES
2.00
1.50
1.00
0.50
0
4
8
12
16
20
24
SECONDS
TIME REQUIRED FOR POWER TO DROP 4500 WATTS
Air entering through side of intake 8" below impeller.
Air entering at discharge
Air entering intake at center of impeller.
F.L. 4-9-42


GRANBY PUMP TESTS
STARTING CYCLE
IMPELLER RUNNING IN AIR
RATE OF INCREASE IN POWER RELATED TO BY-PASS DIAMETER AT VARIOUS RATES OF AIR RELEASE
Air exhaust valve 15/16 closed
Air exhaust valve 1/2 closed.
Air exhaust valve wide open.
DIAMETER OF BY-PASS IN INCHES
2.00
1.50
1.00
0.50
0
4
8
12
16
20
24
28
32
SECONDS
TIME REQUIRED FOR POWER TO INCREASE 4500 WATTS

F.L. 4-10-42

FIGURE 22



GRANBY PUMP TESTS
DISCHARGE CURVES FOR
IMPELLER WITH AND WITHOUT
STARTING BUCKETS