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UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION

A TOOL OF THE DESIGNING ENGINEER

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and
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THE HYDRAULIC LABORATORY
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INTRODUCTION

This paper presents in very abbreviated form a description of the new Hydraulic Laboratory of the Bureau of Reclamation of the United States Department of the Interior in Denver, Colorado, U.S.A. It is spacious with ample headroom, and the arrangement and type of equipment is unique but very practical for the type of work under investigation.

In addition to a description of the laboratory, two specific hydraulic model studies will be briefed in an attempt to illustrate how the Hydraulic Laboratory serves the designer as a tool, especially when confronted with problems which are difficult of solution analytically. The first consists of a hydraulic machinery problem in connection with the design of the Granby Pumping Plant on the Colorado-Big-Thompson Project in the State of Colorado, and the second deals with a tidal estuary problem in connection with the design of the Antioch Steam Plant on the Central Valley Project in the State of California. These two examples constitute only two of many unusual problems which have been assigned to the hydraulic laboratory, in the past 17 years, only after the analytical approach has failed to yield satisfactory results. Oft times the value of the model study can be realized only after much study and experimentation, as the limitations and possibilities are not clear at the outset. A hydraulic model does not offer an automatic solution to any problem, rather its value lies in the ability and foresight of the experimenter and the facilities and equipment with which he has to work.

The Laboratory

The Hydraulic Laboratory of the Bureau of Reclamation is novel in that it is contained on a single ground floor. Except for 10-inch steel columns on 30-foot centers, the laboratory contains 53,000 square feet (4,930 square meters) of unobstructed floor space (excluding offices, shops, and storage). Seventy percent of this area has 26 feet (8 meters) of headroom while the remainder is restricted to 13 feet (4 meters) of headroom. The laboratory is housed in a new factory-type building which was constructed and used during the last war for the manufacture of munitions. Naturally, a great amount of work was required to convert an area this size, having no floor drainage, into a Hydraulic Laboratory. In fact, 1-1/2 years have elapsed since the laboratory was occupied, yet remodeling and installation work still continues. Figure 1 shows the arrangement of the laboratory. The large channels which traverse the laboratory are 9.5 feet wide and vary from 3 to 8 feet deep. These constitute the sump or reservoir for the entire laboratory. Pump pits are located at each end of the main sump channel as shown and will eventually contain three 100-horsepower horizontal centrifugal pumps plus other smaller pumps with a total capacity of approximately 40 second-feet (1.13 cubic meters per second). The smaller channel shown encircling the main portion of the laboratory is a pipe chase which contains the permanent measuring equipment, piping, and valves. The measuring equipment consists of four banks of Venturi meters, ranging from 4 to 14 inches, arranged such that one bank serves a quarter of the main portion of the laboratory. The piping throughout the chase is principally 12-inch standard pipe with tee connections for vertical risers at 15-foot intervals along the line. To connect a model into the system, light temporary piping is employed between the tee connection on the main line and the model.

All valving in the permanent lines is accomplished with hydraulically controlled gate valves, and these are all located under the floor except for some of the valves on the individual Venturi meter lines. To reduce excess bulk and weight, the operating cylinders for these valves were specially designed and manufactured in the laboratory shops. They operate from the city water pressure (approximately 70 psi) and are controlled by 4-way solenoid operated pilot valves located directly on the large valves. The pilot valves are in turn operated electrically from the two central control boards shown on Figure 1. The two control boards will contain specially designed pot-type mercury manometers for indicating discharge through the Venturi meters plus Selsyn-motor-operated indicators, which will show the position and opening of every hydraulically operated valve in the laboratory. The boards will also contain pushbutton controls for operating these valves and pushbuttons and controllers for energizing the main pumps.

In case one pump will not supply the required discharge or sufficient head for a particular model, cross-connections are provided in one pump pit whereby two 100-horsepower pumps can be connected either in parallel or series by merely operating pushbuttons on the control board. Two of the 100-horsepower pumps are driven by 4-speed alternating-current motors while the prime mover on the third pump is an 11-speed slip ring motor.

All meters in the laboratory are calibrated and checked in place at regular intervals by utilizing the volumetric pipette tank shown in the center of the main laboratory, Figure 1. The main tank, which has a volume of approximately 700 cubic feet also contains a similar but smaller tank of about 60 cubic feet capacity. For discharges of 2 second-feet and less the small tank suffices while for discharges greater than 2 second-feet, both the small and large tanks are utilized. Accuracy is maintained with this relatively small capacity tank by the following measures:

1. The small tank or both tanks are always filled to the neck thus improving the accuracy of water surface measurement from which the volume is obtained directly.

2. Water temperatures are recorded as the volume of the large tank changes slightly with temperature.

3. Pneumatically-operated diverting equipment, so that speed of movement of the swing spout is uniform and position of spout can have no influence on the flow through it.

4. Extremely accurate timing device.

5. Special care in determining volumes of the calibration tanks at various water temperatures.

The absence of constant-head tanks is perhaps conspicuous to the experienced hydraulician. Although the laboratory is in possession of a constant-head tank, it is not the intention to install it until necessity demands its use. It has been found that much more flexibility is possible in the laboratory if constant-head tanks are eliminated. Little accuracy is sacrificed by this procedure as long as the powerlines into the building are ample to prevent local voltage drop, the motors on the pumps have a relatively constant speed, and the hydraulic models are limited to a reasonable size. Incidentally, extremely small models are not popular in this laboratory. The one slip-ring motor mentioned previously is not adapted to constant speeds.

One of the most versatile and useful pieces of equipment is a portable pump unit equipped with individual flow-measuring device which can be easily transported and assembled over any sump channel. It consists of a vertical turbine-type pump, a standard section of lightweight pipe containing a flow straightener, and an 8-inch combination orifice-venturi meter developed especially for this application. The meter, Figure 2, has incorporated in it a ring seal which automatically seals the orifice plate in place when the pump is set in motion. To change orifices or utilize the device as a Venturi meter, it is merely necessary to shut down the pump, lift the orifice plate from the slot, replace it with the one desired, and re-start the pump. There are no bolts or clamps to loosen, and it is not necessary to drain the water from the line when changing orifice plates. The unit has the advantage that it can be used as a Venturi meter for the larger discharges, where losses are important; and can be used as an orifice meter for the intermediate and smaller flows where losses are of little concern. Of course, the outstanding advantage of this meter is its portability. The laboratory possesses six 8-inch and six 6-inch pump units of this type. The 8-inch units have a capacity of 5 second-feet each while the 6-inch units are limited to 2 second-feet each.

In addition, the laboratory possesses air testing equipment consisting of two centrifugal blowers with capacities of approximately 3,000 cfm each at 2 psi together with related apparatus. It is also intended to install in the laboratory a pump and turbine testing unit, of approximately 500-horsepower capacity when this equipment is again available for purchase. A large outdoor river sedimentation laboratory, which will be located adjacent to the laboratory building, is in the design stage.

A HYDRAULIC PROBLEM IN CONNECTION WITH THE DESIGN OF THE GRANBY PUMPING PLANT

The Granby Pumping Plant will be located on Granby Reservoir, 8 miles north of Granby, Colorado, on the western slope of the Rocky Mountains. Water from the headwaters of the Colorado River will be stored in the reservoir during periods of high runoff. Then during periods of deficient runoff water from the Granby Reservoir will be pumped up to Shadow Mountain and Grand Lakes, which are joined by a short channel, and from Grand Lake the water will flow through a 13-mile long tunnel under the Continental Divide to the eastern slope of the Rocky Mountains where it will be used for both irrigation and generation of power.

A plan of the Granby Pumping Plant is shown on Figure 3. There will be 94 feet of variation in the water level of Granby Reservoir, from a maximum of elevation 8280 to a minimum 8186 feet above sea level. The plant will consist of three vertical shaft, single stage, centrifugal pumps, each with a capacity of 200 second-feet at the maximum pumping head of 186 feet. Each pump will be driven by a 6,000-hp, 327-rpm, 6,600-volt, 3-phase, 60-cycle synchronous motor.

The suction line to each pump extends from a concrete intake structure, in the bottom of the reservoir, which consists of an 87-inch inside diameter concrete pipe approximately 570 feet long. Because of the large fluctuation of the inlet head, the long suction pipe, and the low barometric pressure at this rather high altitude, it was necessary to install the pumps well below the minimum reservoir level so as to insure freedom from cavitation throughout the wide range of operation. The three pumps will discharge through 87-inch inside diameter steel pipes. These will branch into a single concrete conduit 11 feet in diameter which will conduct the water up to a canal. From here the water will flow into Shadow Mountain Lake. The total length of discharge pipe from pump to canal will be approximately 3,500 feet.

Eventually, the Granby Pumping Plant will be supplied with energy from the power system on the eastern slope of the Rockies through a 115-kv transmission line. This line will extend over much mountainous country, crossing the Continental Divide by way of 11,700-foot Buchanan Pass. As outages due to storms, slides, etc., are likely to occur along this line, it was imperative that the Granby Pumping Plant be designed so that the pumps could also be started and operated on a limited supply of power from the Green Mountain Powerplant, which is located on the western slope, 45 miles distant. A 69-kv line already connects the two 12,000-kva generators at Green Mountain Dam with the site of the Granby Pumping Plant.

The Granby pumps will be subject to discontinuous operation requiring frequent starting and stopping as it is intended that they be operated on secondary power as much as possible. As these large motors will require from five to seven times the normal running power on across-the-line starting, this would cause dimming of lights and interference with equipment on the connected system, and under oertain conditions the rate of increase of power input required by the pump motors could exceed the rate of response of the generators. These in turn are limited by the inertia of the water in the power penstocks and the capabilities of the turbine governors.

The Model

In an attempt to make the most of these limitations an 8-inch model pump was set up in the Hydraulic Laboratory with the necessary valves, piping, and measuring instruments to investigate methods of starting and stopping, Figure 4. The model pump, while not homologous with the Granby unit was sufficiently similar in type. The scale ratio was 1 to 6.3, based on impeller diameters. The instrumentation consisted of Venturi meters to measure the discharge; electrical pressure cells to measure fluctions in pressure; indicating wattmeters, a watthour meter, a voltmeter, and an ammeter to measure power input to the motor; tachometers to indicate speed of the motor and rate at which valves were opened or closed; thermometers of various types to measure air and water temperatures; and an oscillograph which was utilized to record instantaneously some of these variable factors.

First Method for Starting

The first method investigated for starting the pump with a minimum of line disturbance was to bring the unit up to synchronous speed by means of an impulse water wheel. A 3/16-inch brass plate was attached to the top of the model impeller and 30 buckets were cut at a 20 degree angle by a 1/2- by 1-1/2-inch diameter keyseat cutter as shown on Figure 5. Four 1/4-inch nozzles were arranged to impinge jets of water along the inner curve of the buckets as shown. The arrangement of piping and valves is shown on Figure 6. Before applying the jets, however, the water was forced out of the pump case by first closing the main discharge valve and then admitting compressed air to the pump case through a port in the suction tube. Thus, on starting, the impeller rotating in air was brought up to synchronous speed by the high-velocity jets. It was found that a by-pass connection between the discharge side and the suction tube of the pump was necessary to equalize the pressure while bringing the impeller up to speed. Without it, water from the jets accumulated, due to rotation of the impeller, and built up a pressure on the discharge side, thus requiring considerably greater pressure on the jets to bring the pump up to speed. Upon reaching synchronous speed, the pump motor was energized and the jets were shut off. The air in the pump case was then gradually released to the atmosphere through a port on the discharge side of the pump, thereby allowing the water to submerge the impeller slowly and in this way gradually build up the pressure to shut-off head. The main discharge valve was then opened at a constant rate until the pump motor was operating at full load, the valve on the by-pass line was then closed; thus the starting cycle was completed.

From the electrical standpoint the test was a success, as in no part of the entire starting operation was there a sudden surge in power. In fact, the power requirement characteristic, up through full load, could be regulated to order by controlling the rate of release of air from the pump case, then controlling the speed with which the main discharge valve was opened.

A few things were also learned concerning the hydraulic operation:

1. A by-pass is essential between the discharge and suction side of the pump not only to carry away the water imparted to the buckets by the high-pressure jets, but it provides a necessary means of cooling the <u>water</u> by circulation during the period when the discharge valve is <u>closed</u> and the impeller is turning in water. It also aids in controlling the rate of power decrease in the shut-down cycle. The size of the by-pass in this case is governed by the discharge from the starting jets. The by-pass should be provided with a valve so that circulation through it can be discontinued after the pump is operating at full load, otherwise a loss in efficiency is involved.

2. The compressed air intake port functioned best when located midway in the suction tube of the pump.

3. The air exhaust port should be located on the discharge side of the pump, preferably in a dome on top of the pipe where water will interfere least with the escaping air.

4. The shut-down cycle, which is practically the reverse of the starting cycle, is applicable to any vertical pump, as the shut-down cycle is independent of the device used for attaining synchronous speed. In shutting down, the by-pass valve is opened while the main discharge valve is closed slowly, then compressed air is forced into the suction tube producing gradual unwatering of the impeller. When unwatering is complete the motor switch is opened terminating this cycle. The revolving impeller traps water on the discharge side of the pump during the unwatering operation of the shut-down cycle unless a means of escape is provided. The by-pass serves this purpose.

The effect of the impeller buckets on the efficiency of the pump was next determined by both hydraulic and electrical measurements. Tests were first made with the buckets in place and then repeated with the buckets filled with typemetal. The buckets reduced the overall efficiency of the pump unit about 4 percent at full load and correspondingly less for partial loads, this loss being largely due to the effect of water being circulated in the buckets themselves. The bucket loss in three large prototype pumps would be considerable over a period of time; thus the design of a common impulse wheel and impeller was abandoned.

Second Method of Starting

A second method proposed for bringing the unit up to synchronous speed was to mount a separate impulse water wheel directly on the main shaft under the motor. With this scheme, four 7/8-inch diameter jets (prototype dimensions) discharging under a pressure of 300 psi, impinging on the buckets of an enclosed Pelton wheel, would develop approximately 165 hp at the normal speed of 327 rpm, which would be ample torque to bring the unit up to synchronous speed, providing the pump impeller was rotating in air. With this plan, the valve on the discharge line would be closed and unwatering of the impeller would be accomplished by forcing compressed air into the pump case. The high-pressure water for the jets would be supplied by a boiler feed pump and the discharge from the water wheel would flow into the station sump, from whence it would be removed by the regular sump pumps. Speed control during the period of synchronization would be accomplished by throttling the high pressure supply and by use of the regular service air brakes. The characteristic of an impulse wheel gives the maximum torque at zero speed; however, the breakaway torque of the unit would be limited to a reasonable value by introducing sufficient oil pressure under the collar of the motor thrust-bearing to float the rotating parts on a film of oil. This method of starting was entirely feasible as the loss due to the buckets revolving in air was computed as 0.02 percent; however, it was decided to investigate the possibility of starting the motor, across the line, with the impeller rotating in air.

<u>Third Method of Starting</u>

The bucket ring was removed from the impeller of the model pump and the same procedure of starting was repeated, except the motor was started across the line after the impeller was unwatered. To be sure that the method is clear the procedure will be repeated:

(1) With main discharge valve closed and by-pass open, compressed air is used to unwater pump impeller,

(2) Motor is started, across the line,

(3) Air is slowly exhausted from pump case until full shut-off head is developed by pump.

(4) Discharge valve is opened slowly at a constant speed and

(5) By-pass valve is then closed.

In stopping the pump the procedure is exactly the reverse. An oscillograph record of this test is shown on Figure 7. Starting at the very top, the upper three lines represent pressures (PC 5 in the suction tube, PC 4 in intake line, and PC 6 in discharge line), the sine wave represents current flowing to motor, and the lower two traces show revolutions of motor and positioning of discharge valve, respectively. The vertical timing lines are one-hundredth of a second apart. The starting and stopping procedures are outlined on the oscillogram for positions of the discharge valve up to 45 percent. The important point which this oscillogram illustrates is that, except for the instant that the motor is started in air, the rate at which power flows to the motor can be controlled readily by the timing of the hydraulic operations, thus there is no need to tolerate any but minor power surges.

The input to the motor in amperes, as shown on Figure 7, is not a reliable indication of the power required, as the power factor for the model pump varied from zero at no load to 0.90 at full load. Actually, the power required to spin the impeller in air amounted to 10 percent of that required at full load. Thus, to start the impeller in air would require from 50 to 70 percent of full load power, depending on this characteristic of the prototype motors. To operate the pump, with water in the case, at full shutoff head required 53 percent of full load power. Thus, once the motor is started in air, the rate of transition from 10 to 100 percent of full load power can be controlled by the timing of the hydraulic operations. This latter method will be followed in starting and stopping the prototype pumps when operating on power from Green Mountain Dam. With power available from the plants on the eastern slope, the unwatering procedure would be eliminated. Thus, with discharge valve closed and by-pass valve open, the motor would be started with impeller submerged, which would require approximately 3.5 times full load power on starting. The discharge valve would then be opened slowly.

<u>Fourth Method of Starting</u>

As a matter of interest a fourth method of starting, which was immediately discarded, was investigated. It was desired at one time to eliminate one of the valves on the main line (see Figure 6), preferably the discharge valve, as the intake valve is essential for unwatering the pump during maintenance. Under this plan, the suction valve would be closed while the pump was started across the line with water in the case. Then the suction valve would be moved slowly to the fully open position. To alleviate low pressures during the cycle, an air vent (Figure 6), was provided immediately downstream from the intake valve. It was found that pressure surges and power surges went hand in hand; one created the other and vice versa. When the air vent was closed, cavitation occurred in the model as the intake valve was opened. This was not entirely unexpected but this condition could not be tolerated in the prototype.

On repeating the test with the air vent open, the low pressures were relieved, but the unstable mixture of air and water now flowed through the pump and up the discharge line, resulting in rampaging pressure fluctuations with corresponding power surges. The comment of the experimenter was simply this, "From a hydraulic standpoint, throttling with an intake valve on a pump line is fundamentally unsound as low pressures are created unnecessarily. In the case of the Granby pumps, where surges in pressure are directly affected by the power requirements of the motors, neither cavitation nor venting should be tolerated even for the few minutes that it takes to open or close a valve."

A TIDAL ESTUARY PROBLEM IN CONNECTION WITH THE DESIGN OF THE ANTIOCH STEAM PLANT

A steam-electric plant is proposed on the San Joaquin River near Antioch, California to stabilize the hydroelectric power at peak loads which will be generated at Shasta and Keswick Dams. The Shasta and Keswick Dam powerplants are scheduled to generate a maximum of 450,000 kilowatts and the tentative capacity of the steam-electric plant is 150,000 with provision for an additional 150,000 kilowatts should an increase in capacity be justified by future demands. A general plan of the steam plant is shown on Figure 8.

The San Joaquin River in the vicinity of Antioch is a tidal estuary in which the flow reverses direction every six hours. This water may be brackish or fresh depending on the natural flow of fresh water in the river. Cooling water for the steam condensers will be pumped from the San Joaquin River and, after it has served its purpose, the same water, although heated in the process, will be discharged back into the river. For the ultimate installation of 300,000 kilowatts, computations indicated that a flow of 1,000 second-feet would be required for single-pass condensers allowing a rise of 6.5° F above river water temperature; or 480 second-feet would be needed for double-pass condensers permitting a temperature rise of 16.5°. For the maximum installation, 12 pumps of 85 second-feet each were proposed for the single-pass condensers or 12 pumps of 40 second-feet would be used for the double-pass condensers.

It was desired to know whether recirculation of the heated condenser water would occur during flood tide, and if so, to what extent, as heating of the condenser intake water would be reflected in a reduction in efficiency of the powerplant. With the outlet located down the river from the intake, Figure 8, recirculation could occur only during flood or slack tide; therefore, the study made could be limited to this portion of the tide cycle. The most useful tool in investigating a problem of this type was considered the hydraulic model.

The Model

A model was constructed on a 1:100 undistorted scale. It measured approximately 30 by 14 feet yet included only a small portion of the river adjacent to the site of the steam plant. The pumphouse and outlet channel were constructed to scale, while the intake and outlet conduits were slightly larger than the scale would indicate to compensate for unproportional friction losses in the model. Thermometers were placed in transparent sections of the intake conduits for observing any temperature rise which might accompany recirculation of the heated water issuing from the outlet channel. The intake discharge was throttled and measured through two calibrated gate valves and then wasted into the laboratory sump.

A circular tank was provided for mixing hot and cold water to obtain the desired temperature for the water flowing through the outlet conduit. This included a motor-driven stirring device which maintained a constant temperature throughout the tank. The tempered water then passed into the two discharge conduits, which also included thermometers, and then discharged into the trapezoidal outlet channel leading to the river, Figure 9A. Flow through the outlet conduits was also controlled by calibrated gate valves. Although intake water was wasted directly into the sump, compensation for heat lost in this way was accounted for by raising the temperature of the outlet water a corresponding amount.

The model riverbed, which included approximately one-half the width of the river, was first molded in sand with the aid of metal templates made from navigation charts. The shallower portion on the Antioch side, including the pump intake channel and the outlet channel, was capped with a thin shell of concrete to stabilize this portion of the bed. The deeper portion of the channel remained of sand.

As the most extreme condition of heating of the intake water will occur when the tide is flowing up the river with a corresponding small flow of the river proper, the model was constructed to duplicate only the flow-tide condition. A water supply was provided at the downstream end of the model to produce flow up the river and an adjustable weir was provided at the upstream end for regulating the tide level.

To obtain the correct tidal prism flowing upstream in the model required a rather laborious method. Commencing with historical data which in this case was the tide-stage velocity curve for the San Joaquin River at Antioch, flood tide was duplicated on the model by regulating tide-height with respect to time by the adjustable weir. Simultaneously the tidal flow in the upstream direction was regulated to produce corresponding tide-velocities with respect to time. Many repetitions were necessary before a flow curve was developed which was in step with the correct velocity, tide-stage, and time. The tidal prism in the model, however, bore no definite relation to the field data as only a portion of the river width was utilized in the model.

Model Reproduction

The recirculation problem was considered in two parts; first, a fluid flow study in which all factors in the model bear a definite relation to the prototype, and secondly, a thermodynamic study in which loss of heat and rate of stratification occur at practically the same rate in model and prototype and are not necessarily to scale. Interpretation of model results appeared less certain in the latter case than in the former.

In the first part of the study the model-prototype relationships follow the accepted laws of hydraulic similitude for a scale ratio of 1 to 100. From the thermodynamic portion of the study, the application of these laws was questionable so it was apparent that it would be necessary to treat the flow and the heat studies as separate problems. The model was initially operated to observe fluid motion, and secondly, to study heat flow.

Original Design (Test 1)

The original design, Figure 9A, was first operated to locate predominating currents and determine the extent of recirculation. Two studies were made, and this is true of all following designs: namely, 1,000 second-feet (0.010 second-foot, model) of flow from condensers with 6.5° F temperature rise above intake water temperature, and 480 second-feet (0.0048 second-foot, model) discharge from condensers with 16.5° rise above intake water temperature. Each study was begun at slack tide, extended through the flow-tide portion of the cycle, and concluded at the following slack-tide period. True flow similitude was adhered to throughout these tests, such that time and velocities in the model were one-tenth of those values in the prototype and flow was in the ratio of 1:100,000. It was planned to pump cool water from the river bottom to the condensers and discharge the heated water near the surface.

The flow pattern for the original design is pictorialized for flow tide on Figure 10. Dye was used to color the heated condenser water, and the time indicated on the figure represents prototype minutes after slack tide. During slack tide, Figure 10A, the heated water spread out on the surface over a considerable area. Twenty minutes later the slow tide velocity had forced the condenser discharge water toward the shore, Figure 10B. As the tide velocity increased, the flow pattern assumed the forms shown in Photographs C, D, and E, during which time the heated water passed directly over the pump intake bay on its course upstream. Photograph F shows the pattern as the heated water commenced to spread again at the following slack-tide period.

During each run, temperature readings were taken at various depths with accurately calibrated thermometers for each intersection point of the grid system, Figure 11. The thermometers were mounted in horizontal racks and were read in a totally submerged position. These readings were recorded during the period of maximum velocity upstream, which was considered the most adverse condition. The grid system merely served as a convenient means of recording and photographing results. The grid lines as shown were 100 feet apart in each direction, or one foot apart in the model.

Cross-sections on Figure 11 show the measured temperature rise above tide-water temperature, at various points throughout the grid system produced by 480 second-feet of condenser water, heated approximately 16.5 degrees discharging into the river. In this case the intake water was heated 2.1 degrees above tide-water temperature by recirculation. The pump bay as located in the original design formed a dead, slowly rotating pool of water in which heat tended to accumulate with time. This is evidenced by the manner in which the colored water remained in the pump bay after slack tide, Figure 10F.

A second test was made with the discharge of 1,000 second-feet which was heated 6.5 degrees in its passage through the condensers. Recirculation, in this case, heated the intake water 1.4 degrees above the tide-water temperature. The BTU content discharged into the river was slightly greater in the former case, and this is evidenced by a larger temperature rise.

<u>Discharge Conduits Extended 500 Feet (Test 2)</u>

In an attempt to obtain better mixing action with the cooler water in the river, the condenser discharge conduit was extended 500 feet farther out into the river as shown in Figure 9B. The photograph on Figure 12A records the heated flow pattern taken 300 minutes after the beginning of the flow-tide cycle. A comparison of flow results with those of the original design (Test 1) shows that the heated water moved about 100 feet farther out into the river with the outlet extension; however, heat accumulated in the pump bay as in the former case. Confetti floating on the water surface of the pump bay, Figure 12A, illustrates the type of slow, rotating motion experienced at this location. The intake-water temperature increased 0.8 degrees above the tidewater temperature for the 1,000 second-foot discharge and 1.5 degrees for the 480 second-foot discharge. The amount of recirculation in this case was less than for the original design.

<u>Dike on Downstream Side of Pump Bay (Test 3)</u>

In a second attempt to reduce recirculation, the outlet-pipe extension was removed and a dike, which extended above high water, was placed along the downstream side of the pump bay as shown on Figure 9C in anticipation that the heated water would be directed away from the intake. One photograph of the heated-water pattern taken 300 minutes after the beginning of flow-tide is shown on Figure 12B. The dike did direct the warm water farther out into the river at the intake, however, the predominating flow of the tide again forced it toward the shore immediately after passing the extremity of the dike. A small portion of this heated water was attracted to the pump bay by the slowly rotating motion of the water therein which gradually produced an accumulation of heat comparable to that in the previous designs. The temperature rise in the intake for a condenser discharge of 1,000 second-feet was 1.4 degrees and for 480 second-feet the temperature rise was 1.9 degrees.

Split-outlet Channels (Test 4)

It was desired to develop a design in which recirculation would be absent or minimized as much as practically possible. The pumphouse was relocated as shown on Figure 9D and a split-outlet channel with automatic diverter gate was next utilized to control the direction of flow of the heated condenser water as it entered the river. The heated water was diverted downstream at ebb tide and up river during flood tide, as shown by the photographs on Figure 13. The diverter gate could be controlled by a velocity device which would actuate a relay when flow reversed in the river or the same result could be obtained by the use of a clock. Should the gate ever fail to operate, no material damage could result.

For both the 1,000 and 480 second-feet discharge no temperature rise of the intake water was recorded, giving the result desired. In this design, heating of the intake water can occur only at slack tide which will exist but for a few minutes.

Analysis of Heat Losses

The heat in the condenser discharge water can be dissipated in three ways: (1) heat loss to the surrounding atmosphere by convection and evaporation, (2) heat loss to the riverbed by conduction, and (3) heat transfer to the cooler river water by conduction and mixing. In the first two cases the heat loss to the atmosphere and the riverbed is primarily a function of the contact areas. In the last case volumes are the governing factor.

The rate of dissipation and stratification of heat in model and the prototype is the same. Assuming that the initial temperatures in model and prototype are the same, the heat supply to the model is proportional to the quantity of heated condenser water entering the river per unit of time in the ratio of 1/100,000. The heated-water pattern, as it moves from the outlet to intake in the model, occupies a surface area equal to 1/10,000 of that of the prototype. As the heat losses, due to convection and evaporation at the surface and conduction to the riverbed, are proportion to the areas exposed, the model losses per unit of time will be 1/10,000 of those in the prototype. The relative losses to atmosphere and riverbed are therefore proportionately $\frac{100,000}{10,000}$ or ten times larger in the model. Thus the temperature drop experienced in the model, due to the above losses, was ten times too great.

As the degree of heating of the intake water was of primary interest, the surface area between the outlet and pump intake will be considered. This involves an area equal to approximately 200,000 square feet, prototype. From Figure 11, the condenser discharge water was 80.2 degrees with a corresponding air temperature of 68 degrees, producing a differential at the outlet of 12.2 degrees. The differential between air and water at the pump intake was -1.7 degrees.

Surface convection from air to the water produces a gain of heat approximating 2.0 BTU per hour per square foot of surface per degree (Fahrenheit) of differential, which amounts to 800,000 BTU per hour for the prototype. Evaporation, on the other hand, is responsible for a loss of heat from water to air of 21.8 BTU per square foot of surface, which is equal to 4,367,000 BTU per hour. The net loss of heat by the water, from these two causes, is, therefore, 3,567,000 BTU per hour. The total amount of heat entering the river from the condenser outlets, using river-water temperature as base, equals 1,768,400,000 BTU per hour. The heat loss to the atmosphere is 0.20 percent of the total.

The heat conduction from the heated water to the riverbed should be approximately 8.0 BTU per square foot of contact surface per degree of differential per hour for the first hour after reversal of flow in the river, and one-half this amount thereafter. The loss of heat from water to streambed is about 3,520,000 BTU for the first hour of flood tide, which is 0.20 percent of the total heat entering the river. The combined heat loss to the atmosphere and riverbed is, therefore, approximately 0.40 percent of the total, leaving 99.60 percent of the heat to be transferred to the adjacent cooler river water by mixing. It is now apparent that the convection, evaporation, and riverbed conduction losses, which were not to scale, are of little importance and could be neglected in these computations. Interpretations of model temperatures would be complicated were it not for the fact that all of the above losses are small in comparison to the heat transferred to the river water by mixing action. Proof of the unimportance of the above makes it possible to deal primarily with the third loss which is basically a hydraulic flow problem.

The transfer of heat from the warmer to the cooler water is produced by a mixing action which, if exactly similar in model and prototype, is governed by the laws of hydraulic similitude. In salinity studies which involve the flow of fresh and salt water, it has been customary to regard the scale ratio of densities as 1:1. Experiments by the U. S. Waterways Experiment Station have shown that the source of error from this procedure is negligible as the density flow of salt water with fresh water, observed from models, duplicates the prototype action closely. This is known as a conservative concentration which can be altered locally only by the process of diffusion and advection. It so happens that heat content in an aqueous solution falls under the same category and the action is similar. In both cases, however, it is essential that turbulent flow exist in the model to obtain similar flow conditions with those of the prototype. It can be stated, however, that from observation the flow in the model approached the type of turbulent flow desired.

The average maximum prototype velocity in the river for flow tide approximates 2.0 feet per second; so the velocity near the shore is about 1.5 feet per second. The average cross-section of the colored area, projected to prototype, measured normal to the shore line, is about 2,250 square feet. The total discharge flowing through the colored area is equal to 3,375 second-feet (prototype). Of this flow, 480 second-feet or 14.2 percent enters the river at 80.2 degrees while 85.8 percent of this total discharge is river water at a temperature of 63.8 degrees. With thorough mixing, the resulting temperature of these two masses of water should be 66.13 degrees or 2.33 degrees above the temperature of the undisturbed river water.

A more accurate figure is obtained by repeating the computation and deducting the heat loss to the atmosphere and riverbed. In this case the temperature increase above river water temperature, at the pump intake, would be 2.32 degrees. Referring to Figure 11, the temperature rise at pump intake for the above conditions was found to be 2.1 degrees from the model. The agreement is within the experimental error expected in a study of this kind. The tests previously described therefore are not merely comparative but represent true prototype results, within limits. It is felt that the prototype intake temperatures rise as stated for the various designs will not differ more than 25 percent from those obtained on the model for any given set of conditions. It should be repeated that the tests included herein were made for the most adverse conditions which can prevail at the prototype site. Whether the extent of heating due to recirculation, as indicated by the model tests, is sufficient to be objectionable depends on a further study of the plant equipment.

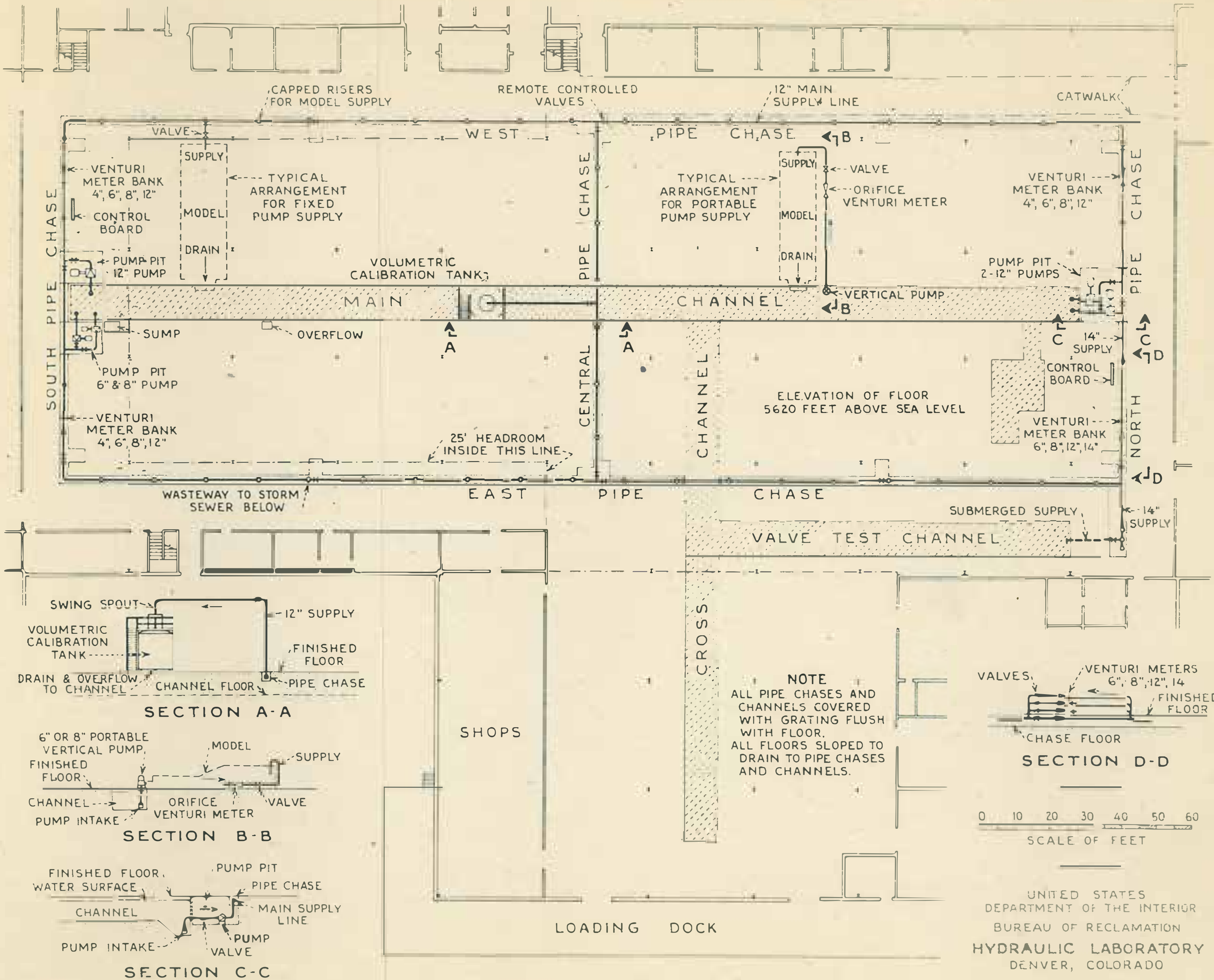


FIGURE 1. - PLAN OF NEW HYDRAULIC LABORATORY IN DENVER, COLORADO

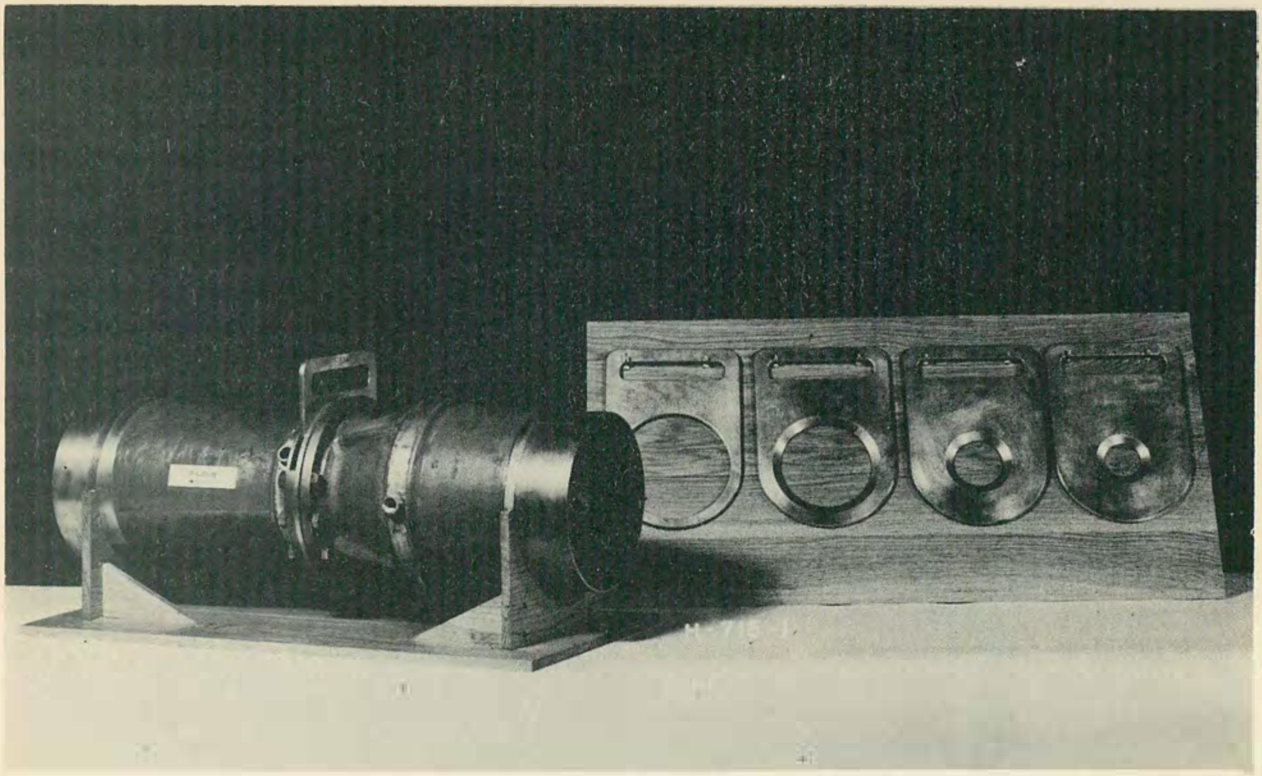


FIGURE 2 - PORTABLE ORIFICE-VENTURI METER

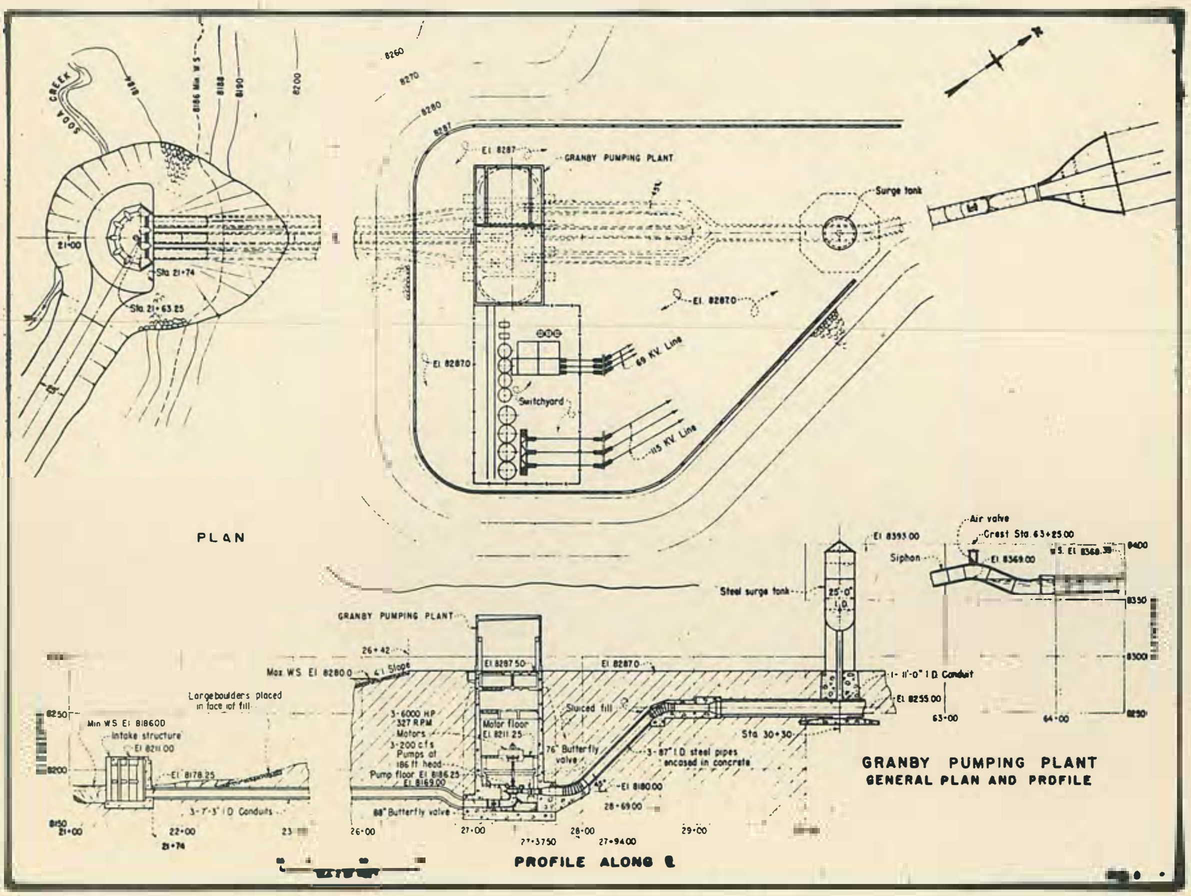


FIGURE 3 - PLAN AND PROFILE OF THE GRANBY PUMPING PLANT

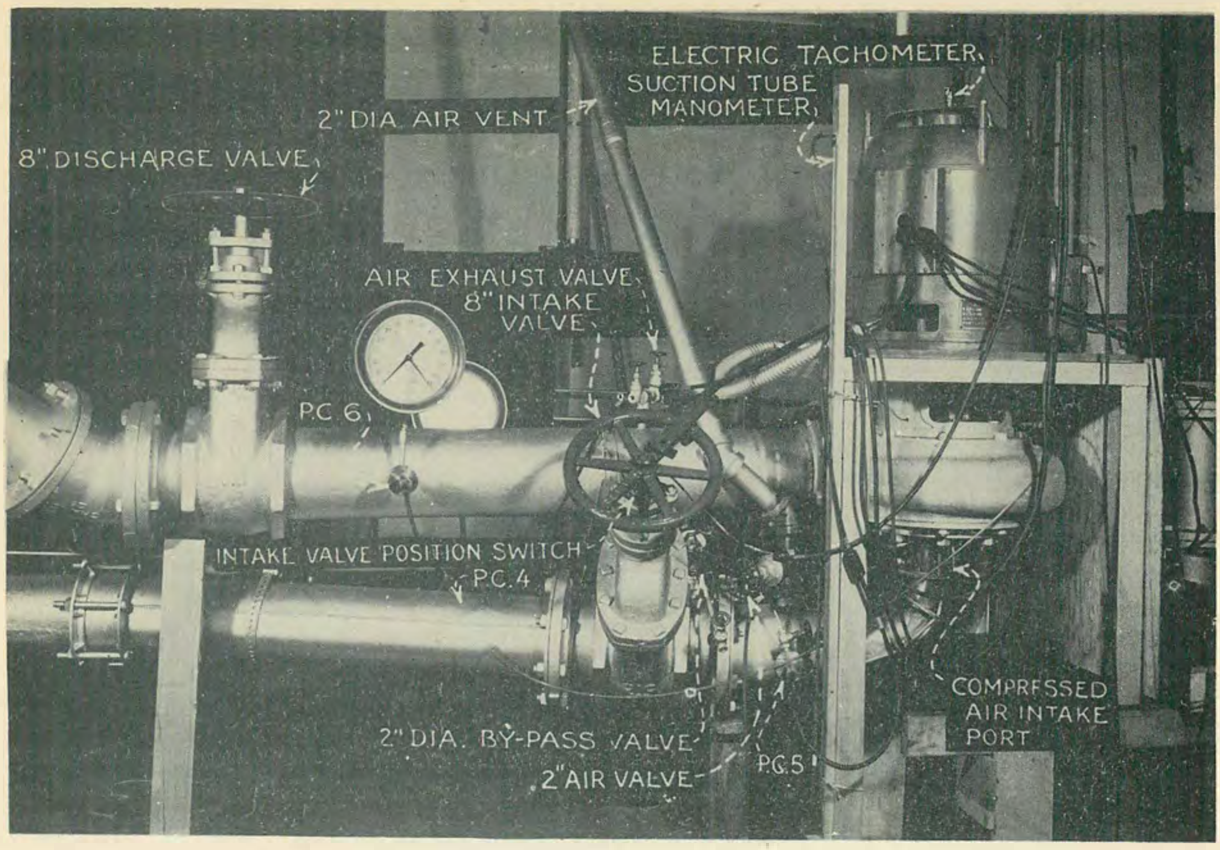


FIGURE 4 - GRANBY PUMP MODEL

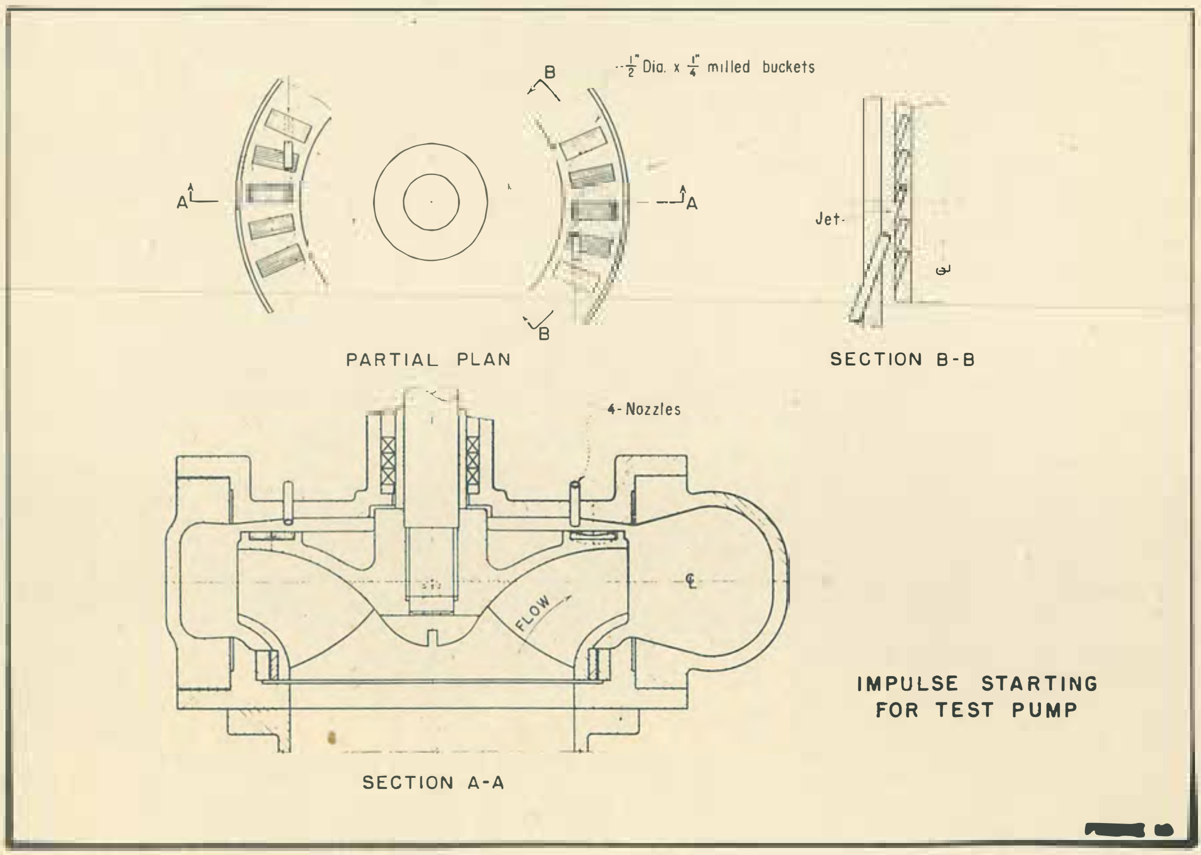


FIGURE 5 - IMPULSE METHOD FOR STARTING PUMP

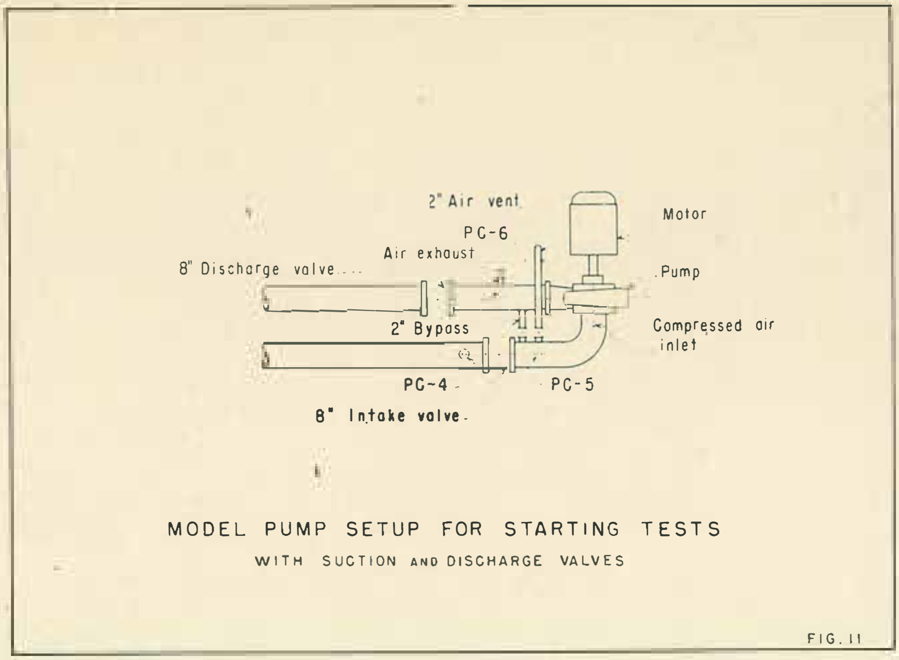


FIGURE 6 - CONNECTIONS TO MODEL PUMP

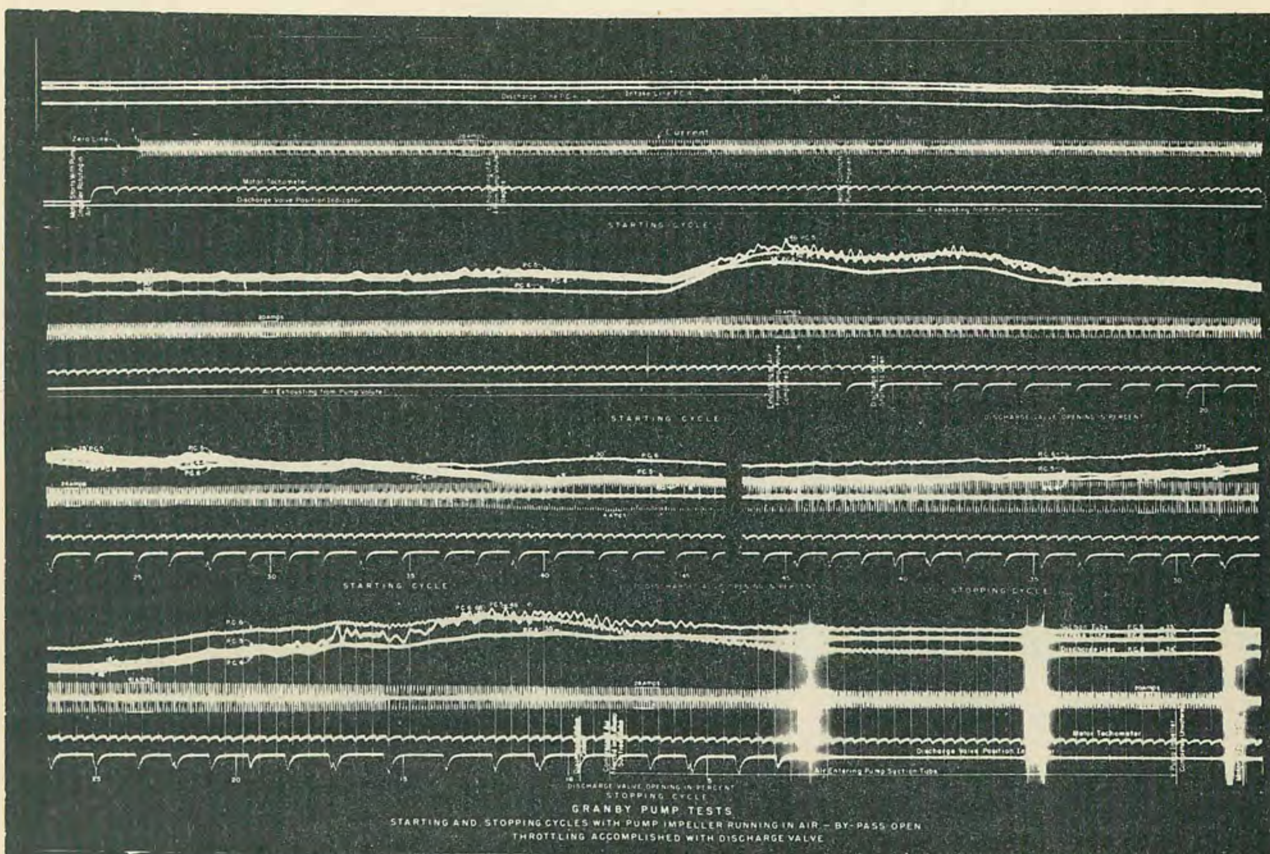


FIGURE 7 - OSCILLOGRAM FOR STARTING AND STOPPING CYCLES

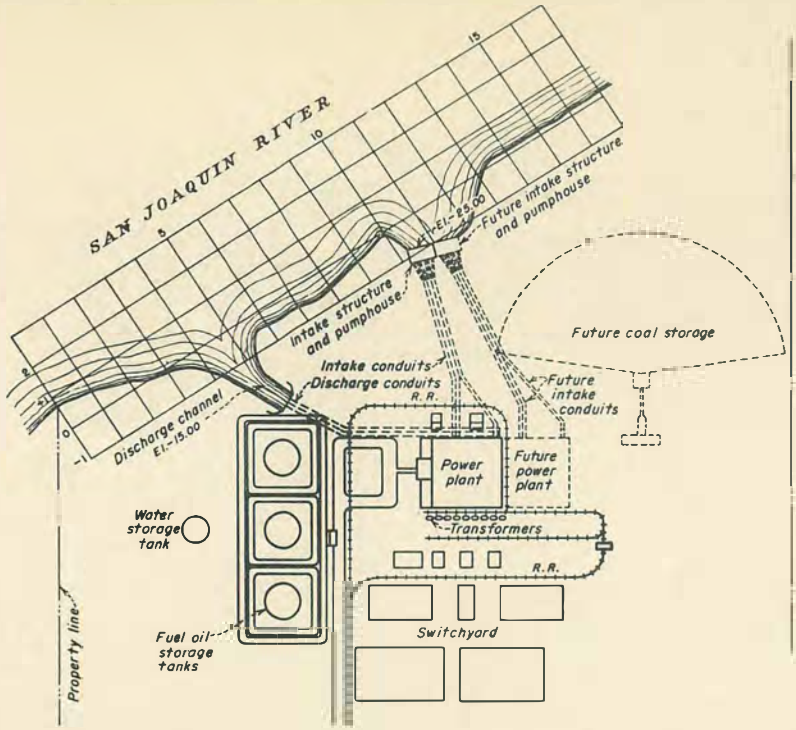


FIGURE 8 - PLAN OF ANTIOCH STEAM PLANT

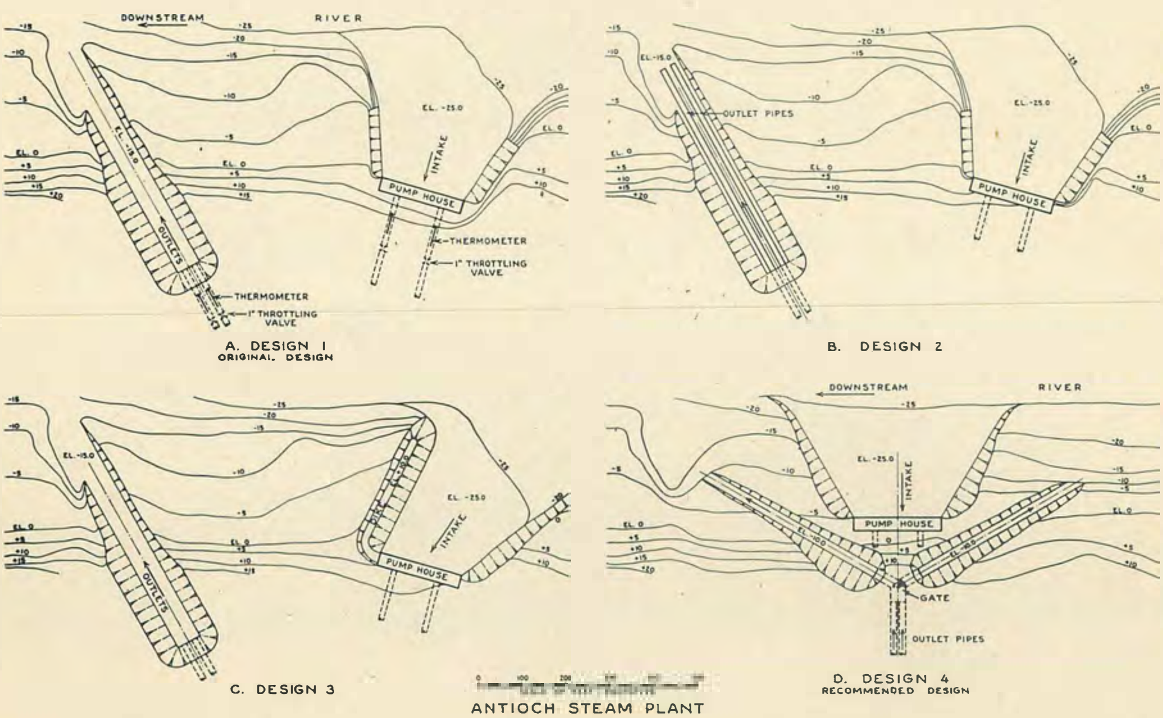
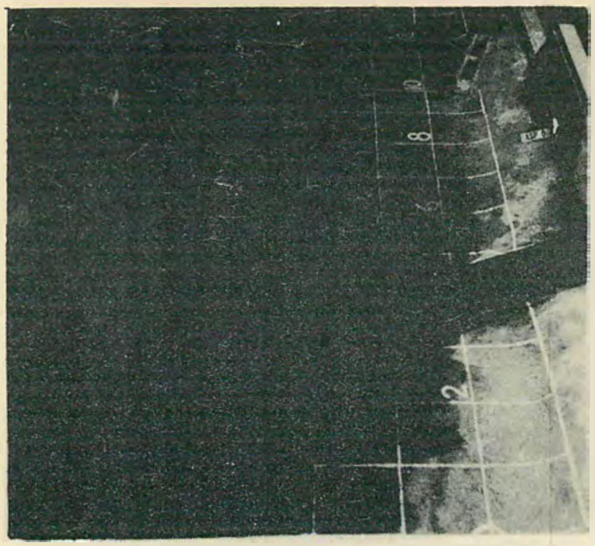
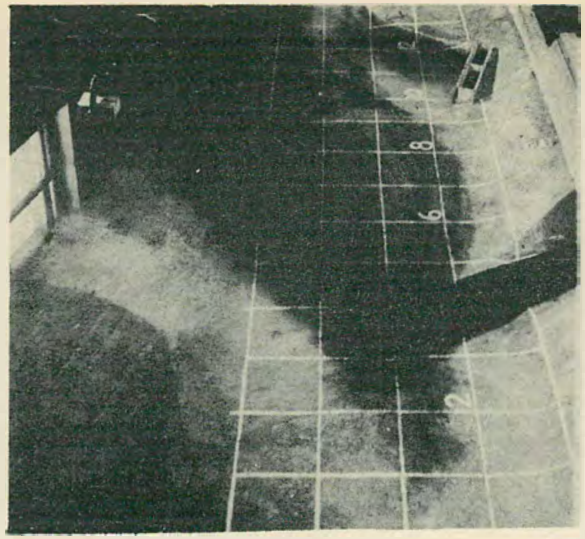


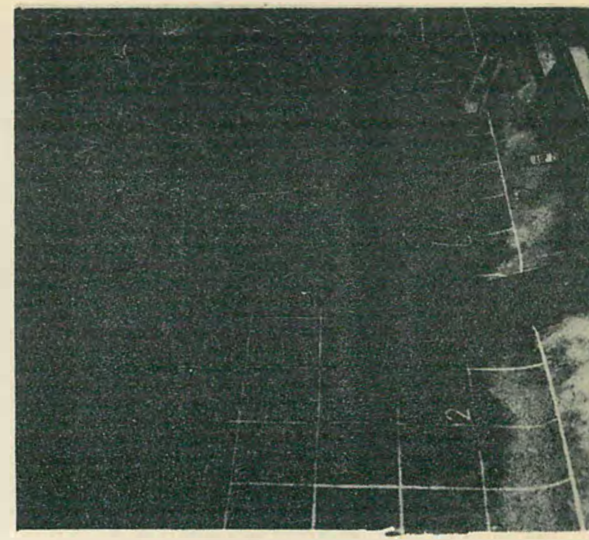
FIGURE 9 - SCHEMES INVESTIGATED TO PREVENT RECIRCULATION OF COOLING WATER



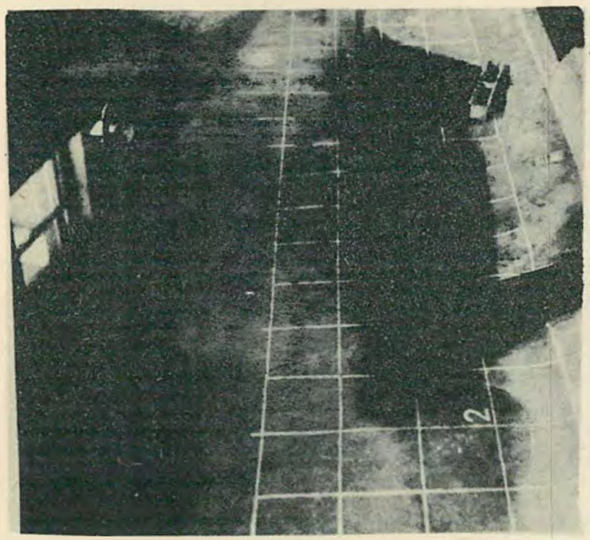
A - 0 Minutes - Beginning of Flow



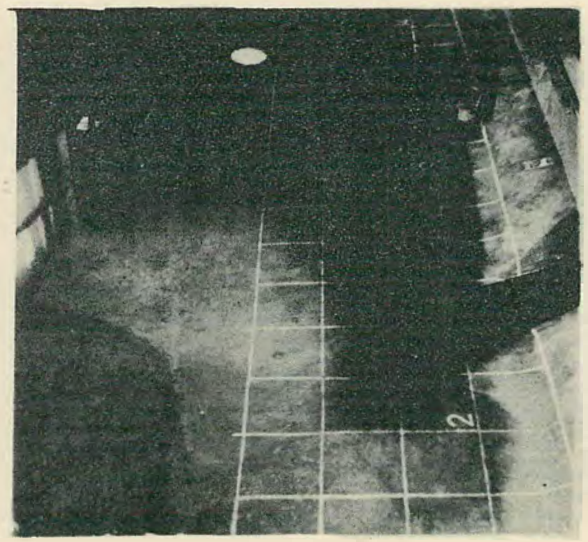
B - 20 Minutes



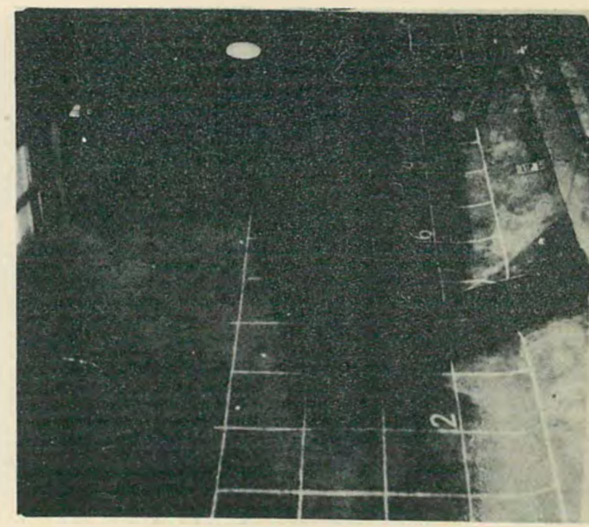
C - 120 Minutes



D - 200 Minutes - Maximum Velocity Upstream



E - 300 Minutes



F - 380 Minutes - Point at which Current Reverses to Downstream

FIGURE 10 PATH OF WARM WATER CURRENTS PRODUCED BY DISCHARGE CONDENSER WATER DURING FLOW TIDE

ORIGINAL DESIGN - TEST 1

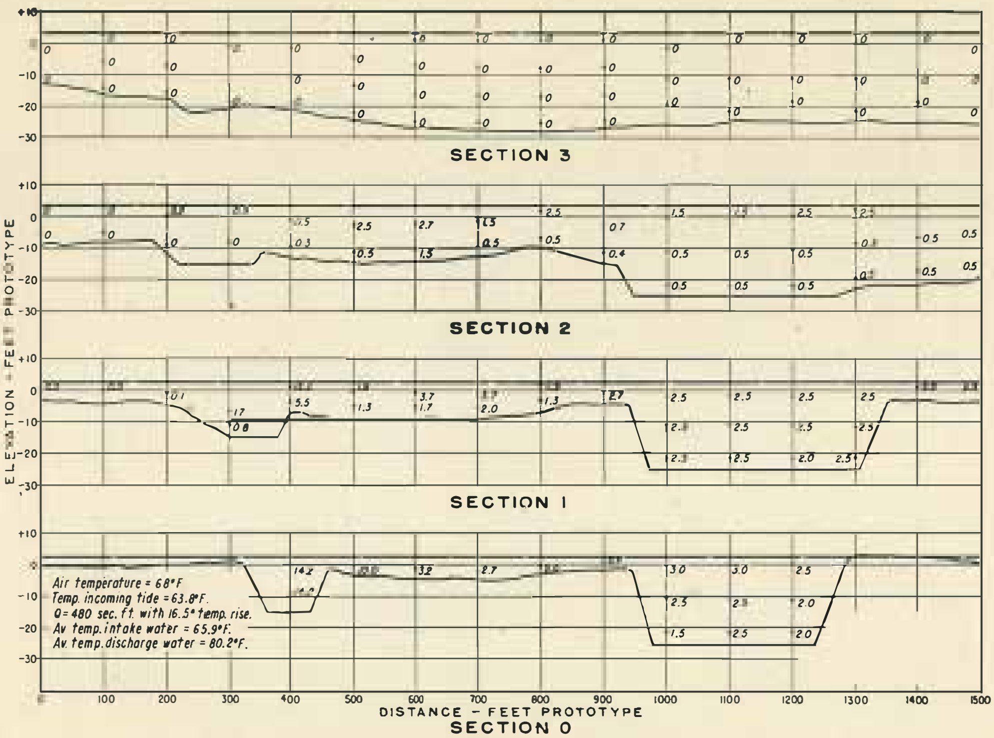
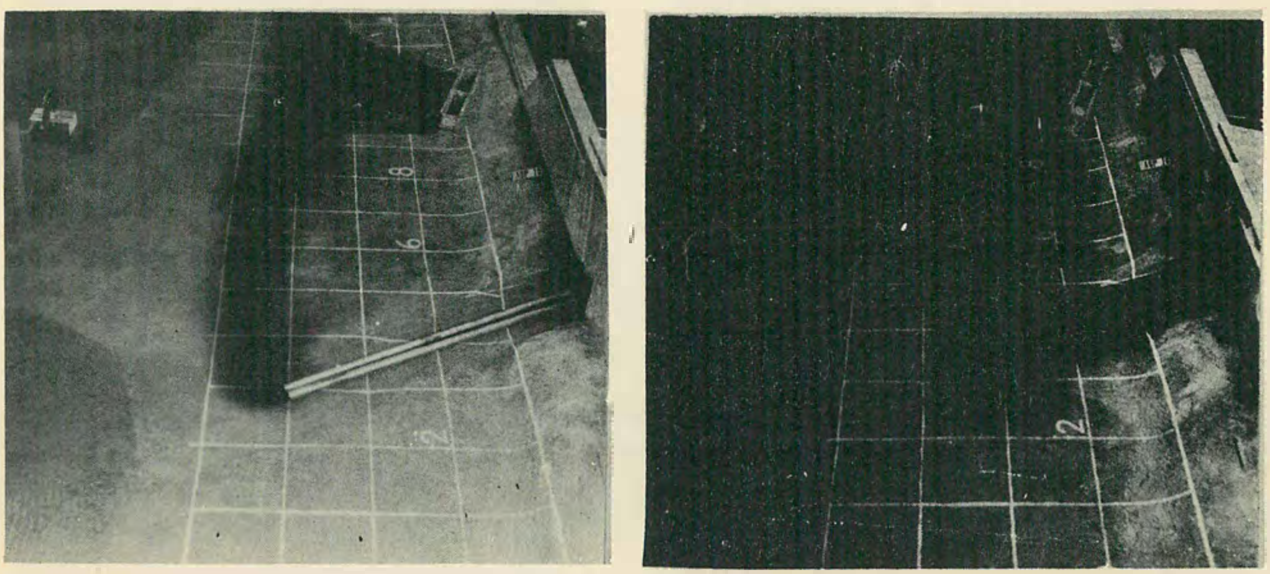


FIGURE 11 - PROFILES SHOWING TEMPERATURE RISE IN RIVER DUE TO RECIRCULATION (ORIGINAL DESIGN)



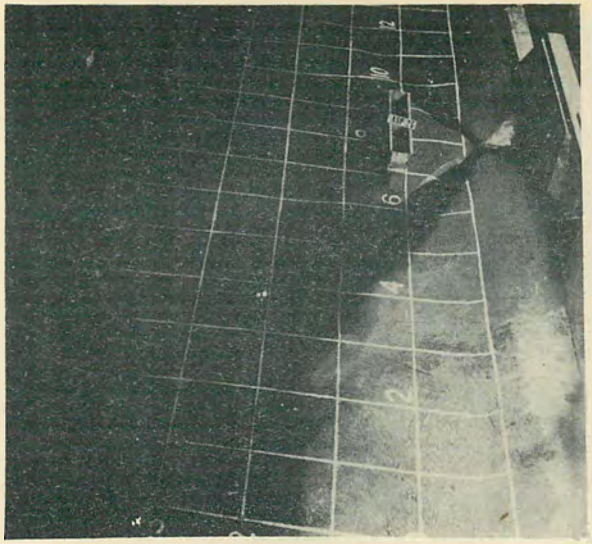
E - 300 Minutes

A - Condenser outlets extended 500 feet (test 2)

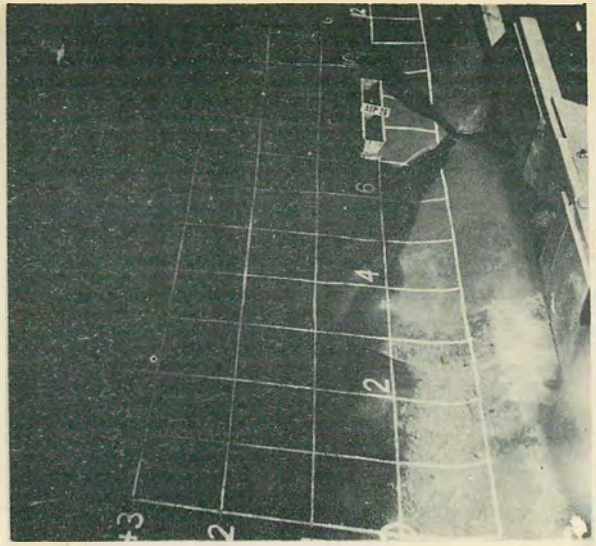
E - 300 Minutes

B - Pump bay protected by dike (test 3)

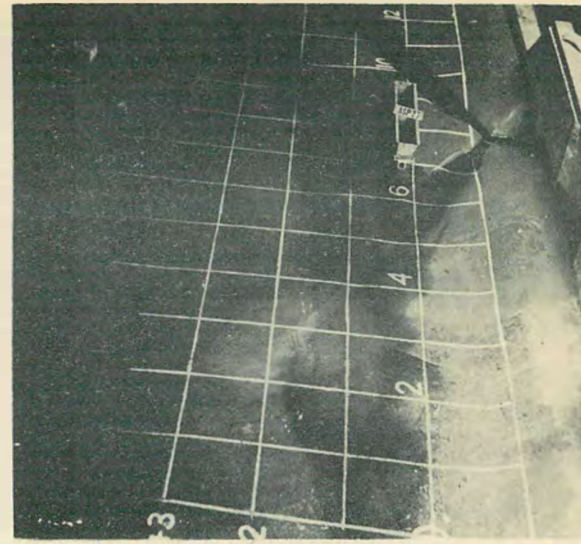
FIGURE 12 - PATH OF WARM WATER CURRENTS PRODUCED BY DISCHARGE CONDENSER WATER DURING FLOW TIDE



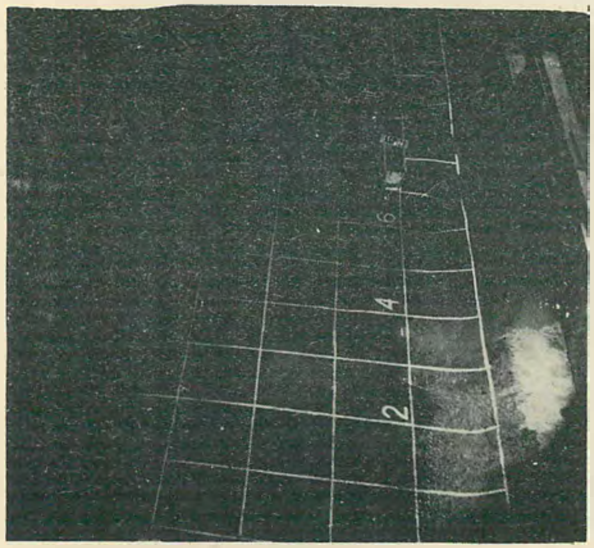
A - Ebb Tide - Flow is Downstream



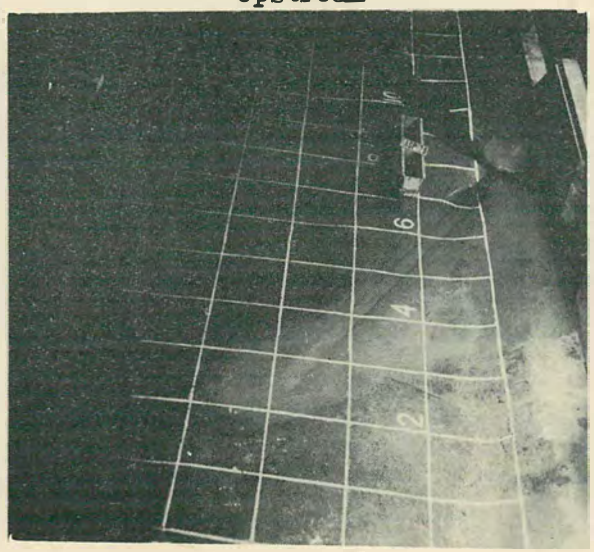
B - Slack Tide - Beginning of Flow Upstream



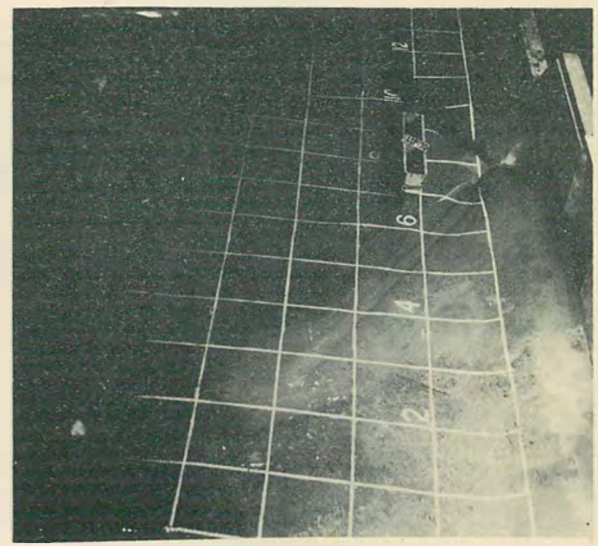
C - 20 Minutes



D - 60 Minutes



E - 200 Minutes - Maximum Velocity Upstream



F - Slack Tide - Point at which Current Reverses to Downstream

FIGURE 13 PATH OF WARM WATER CURRENTS PRODUCED BY DISCHARGE CONDENSER WATER

SPLIT OUTLET CHANNELS - TEST 4