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Final Report

A MODEL STUDY OF CITY OF  
DETROIT BOOSTER PUMPING STATION

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TABLE OF CONTENTS

	Page
SUMMARY	iii
OBJECTIVE	iv
1.0 THE MODEL	1
1.1 MODEL CONSTRUCTION	1
1.2 MODEL MEASUREMENTS	2
1.3 MODEL THEORY: APPLICABILITY OF MODEL RESULTS	3
2.0 MODEL PERFORMANCE AND DESIGN MODIFICATION	7
2.1 HEAD LOSSES	7
2.2 PUMP INTAKES	13
2.3 BEHAVIOR OF DESIGN AT PARTIAL FLOW	19
APPENDIX I	22

### SUMMARY

This report describes the setting up and operation of a model of the City of Detroit's raw water booster pumping station. As a result of this model study, it is recommended:

- 1) That a system of 8—21-in.-radius turning vanes be installed at the downturn where the vertical line to the suction pit joins the main 14-ft line.
- 2) That a twin system of 9—21-in.-radius turning vanes be installed at the junction of the vertical line and the suction pit.
- 3) That two baffles in the form of 24-in. square grids, 5 ft long, be installed in the suction pit upstream of Pumps Nos. 3 and 4.
- 4) That a false floor, 4 ft above the existing floor, be installed in the pit under the intakes of Nos. 1 and 2, and that vertical turning vanes be installed between Pumps Nos. 1 and 3, and between Nos. 2 and 4.
- 5) That a vertical partition wall, 8 to 12 in. thick, and 5 ft wide, be installed at the end of each limb of the suction pit, as shown in Fig. 3.
- 6) That a system of six radial equally spaced vanes be placed in each pump intake. Although the intake itself forms part of the pump structure and is strictly speaking beyond the scope of this study, the vane system mentioned was tested for the sake of completeness: it improved the intake flow so materially that mention is made of it here as an addendum to the main results of the study.

The proposals above are shown in detail in Fig. 3.

The proposed alterations would improve the performance of the station in two ways: (1) by reducing the head loss through the station from 3.38 times the velocity head in the main line to 1.40 times this velocity head; (2) by substantially improving the flow conditions at the pump intakes, although some irregularities still remain.

The primary purpose of the model study was the reduction of the head loss; however, the fact of the model's availability for testing prompted a further study aimed at improving pump intake conditions, with the results noted above.

OBJECTIVE

The objective of this study was to reduce the head loss through the pumping station and to improve the intake conditions of the pumps.

## 1.0 THE MODEL

This section contains a description of the model—construction, material, etc.,—and of the detailed measurements that were taken on each run. There is also a discussion of the theoretical considerations affecting model operation, and of the reliability of predictions made about prototype performance. It is hoped that this discussion will serve not only as an assessment of the merits of this particular model, but as a guide in the consideration of making model studies of other schemes in the future.

### 1.1 MODEL CONSTRUCTION

The model was made to a 1:21 scale from Lucite, a clear plastic. The walls of the pipes and pit were  $1/4$  in. thick, and flanges for connections were either  $1/4$  in. or 1 in. thick. Flanges were fixed to the body of the model with plastic cement, and bolted together. In some cases tightening of the bolts led to cracking of the cement, but the cracks were easily repaired and in general it was found that the flanges could be bolted firmly with a screw driver and an 8-in. wrench without putting excessive strain on the cemented joints.

The main 14-ft-diameter pipe was represented by an extruded 8-in. plastic pipe, available ready made. Other pipe sections were fabricated from  $1/4$ -in. sheet, two half cylinders, or half-cones, being placed together and cemented along the longitudinal joints. These joints gave a little trouble with leakage, but were easily repaired with plastic cement.

The flanged joints were sealed with thin rubber gaskets and, at first, jointing compound. When the model was first tried out with water as the working fluid, these joints, particularly those on the suction pit, gave a great deal of trouble, and it was necessary to place bolts at 3-in. centers to make the joints tight.

The strength of Lucite is not very high and although the pipe sections had a high enough thickness/diameter ratio to make them hold pressure satisfactorily, the suction pit, being of a rectangular 11 in. by 9 in. section, was rather weak structurally. In fact, one wall of the pit developed a bad crack when the model was first filled with water under a head of about 8 ft. Because of this fact and of the difficulties in sealing the joints, it was decided to run the model with air as the working fluid. Pressures in this case would be well below bursting pressure, and the joints could be easily made good with masking tape on the outside of the flange. The model was therefore con-

nected to a centrifugal blower capable of delivering 4000 cfm of air and the full series of tests was run in this way. The reliability of tests performed with air instead of water is discussed in Section 1.3.

The decision to use air instead of water greatly simplified the running of the tests, and serious consideration should be given to using air in any future tests, particularly where:

- a) there are any large box-type sections which are inherently weak structurally.
- b) it is likely that many alterations will have to be made to the interior of the model, necessitating the continual breaking and remaking of joints. When air is used, no jointing compound is necessary and it is a clean, simple job to make and remake the joints.

Figure 1 is a general arrangement drawing of the model showing some of the leading dimensions, and the location of the pressure tapings. A detail of a typical tapping is also shown: there were 41 in all, connected through a manifold to a water manometer graduated at intervals of .01 in. Also shown are the location of the sections, on the discharge lines of Pumps Nos. 2 and 4, where pitot tube traverses were made to measure the discharge in these lines. The total discharge through the system was also measured by a pitot tube traverse in the 12-in. supply line; so that it was possible to compare the discharges in lines 2 and 4, and also to compare their sum with the total discharge.

Plates 1 and 2 are photographs of the model as set up with its air supply. The elbow in the 8-in. supply line is a sharper bend than the corresponding bend in the prototype, but tests showed that straightening vanes downstream of the elbow did not improve the flow characteristics, so it was assumed that the elbow did not materially affect the flow through the model.

The sheet-metal diffuser downstream of the model reduced the outlet velocity substantially. This made the model less of a nuisance in the laboratory, and reduced the mean pressures in the model to values which could conveniently be measured with manometers having a high sensitivity but a small pressure range. Thus, the range of pressures in the model was about 5 inches of water; the presence of the diffuser meant that the pressures varied, not from 0 to + 5 in., but from -3 in. to + 2 in., making possible the use of a 0-3 in. inclined tube manometer with a sensitivity greater than that of a 0-5 in. manometer.

## 1.2 MODEL MEASUREMENTS

The pitot tube traverse in the 12-in. supply line was made with a standard differential pitot tube giving velocity head directly in inches of

water. In the pump supply lines, which were around 5-6 in. in diameter at the measuring sections, a standard tube would have been rather large compared with the pipe area, so it was decided to use a total-head tube consisting of a 1/4-in. diameter copper tube with a sealed end and a small hole in the wall of the tube, very close to the end. This tube was laid along a diameter of the pipe, enabling a traverse across the diameter to be made. There was little interference with the flow, and velocity readings very close to the wall could be made. Static pressure was obtained by linear interpolation between the readings obtained from the manometer tapings, e.g., in line No. 2, interpolation was made between the pressure at No. 22 and the mean of the pressures at No. 23 and No. 24.

At all measuring sections traverses were made across two perpendicular diameters, at points which divided the pipe into equal areas. Hence the mean velocity head was simply the arithmetic mean of all the pitot-tube readings.

Inclined tube watermanometers were used for all readings, as follows:

Pitot tube traverse, 12-in. supply line:	range 0-1 in. graduations .01 in.
Pitot tube traverse, pump lines:	range 0-3 in. graduations .01 in.
Static pressure readings:	range 0-3 in. graduations .01 in.

These gave sufficient accuracy. Flow conditions in the pit were investigated by means of a set of woolen tufts, 1 in. long, attached to light rods mounted across the pit, in the right-hand limb of the pit, looking downstream. They are shown in Fig. 1, and in Plates 3, 4, and 5, and indicate clearly the presence of any irregularities in the flow. Conditions in the pump intakes were explored with another rod with a tuft attached to its end. This rod could be inserted through a hole in the side of each of pump intakes Nos. 1 and 3, and the end moved round to cover most of the cross section.

Another method of examining the flow was to feed confetti through the system. Although the inertia of the confetti made it a poor flow indicator at abrupt changes in direction, it gave a satisfactory, if short-lived, picture of such features as stationary vortices.

### 1.3 MODEL THEORY: APPLICABILITY OF MODEL RESULTS

In a closed conduit system such as the pumping station considered here, the general nature of the flow depends on the form of the solid boundaries, and a parameter called the Reynolds number, defined thus:

$$N_R = \frac{vL}{\nu} \quad (1)$$

where  $v$  is a characteristic velocity of the system (in this case, the mean velocity in the main line)  $L$  is a characteristic length (the diameter of the

main line) and  $\nu$  is the kinematic viscosity of the flowing fluid.  $N_R$  is a dimensionless number, hence its value will be independent of the particular system of units chosen.

The term "general nature of the flow" is more specific than it appears to be. It includes, for instance, such numerical coefficients as the head-loss coefficient; the ratio of the head-loss coefficient in any part of the system to the velocity head. We have:

$$\text{head loss, } h_L = C_L \frac{v^2}{2g} \quad (2)$$

where  $C_L$  is the loss coefficient. The remarks above imply that for a piping system of given shape,  $C_L$  is a function of  $N_R$  alone and is not influenced by any other characteristic of the flow or the fluid.

The Reynolds number also determines the more general qualities of the flow, such as the turbulence. For a specified Reynolds number the size, shape, and location of the eddies will be fixed, as will be the ratio of eddy velocity to mean forward velocity.

It follows that to make sure that events in the model faithfully reflect those in the prototype, it is necessary only to make the model  $N_R$  equal to the prototype  $N_R$ , whether or not the same fluid is used in model and prototype. When this requirement is met, a state of "dynamical similarity" is said to exist between model and prototype.

The practical difficulty in realizing this condition is that model velocities would have to be much higher than prototype velocities. The way out of the difficulty lies in examining more closely the nature of the Reynolds number. It is essentially an inverse measure of the effect of viscosity on the flow—at very high values of  $N_R$  the mechanism of turbulence and of energy dissipation is determined largely by the inertia of the fluid, with comparatively little restraint from viscosity. This is particularly true when pipe lengths are short and there are many abrupt changes in direction. On the other hand, viscous effects tend to become important in long, straight, smooth reaches of pipe where wall shear influences the flow strongly, and the wall shear is itself dependent on viscosity. On the other hand, if the pipe is rough, inertia effects tend to become more important.

The points made above are exemplified in the standard  $f$ - $N_R$  curves for circular pipe (e.g., in Vennard Elementary Fluid Mechanics Fig. 86, p. 195); "f" is the Darcy coefficient, a loss coefficient similar to  $C_L$  as defined above. It is seen that for rough pipe the curves flatten out at high values of  $N_R$ —i.e., inertia effects have taken over and the flow pattern remains substantially fixed, independent of  $N_R$ .

This argument suggests that dynamical similarity can be achieved simply by making the model  $N_R$  high enough, even if it does not approach the



prototype  $N_R$ . Checking this question numerically, we have in the prototype:

Main line velocity 10 ft/sec  
 diameter 14 ft  
 $\nu(\text{water}) 1.2 \times 10^{-5} \text{ ft}^2/\text{sec}$

Hence

$$(N_R)_p = \frac{10 \times 14}{1.2 \times 10^{-5}} = 1.17 \times 10^7 \quad (3)$$

In the model:

max. possible velocity 200 ft/sec  
 diameter 2/3 ft  
 $\nu(\text{air}) 1.58 \times 10^{-4}$

Hence

$$\text{max. } (N_R)_m = \frac{200 \times 2/3}{1.58 \times 10^{-4}} = 8.45 \times 10^5 \quad (4)$$

This value is about one-fourteenth of the prototype  $N_R$ , but is high enough to make it likely that model and prototype flow will be similar. Using as a guide the  $f-N_R$  curves referred to above, it will be found that for a rough pipe the curve has flattened out well before  $N_R$  reaches the value of Equation (4).

To clear up the question, a series of runs was made at different velocities and Reynolds' numbers and the loss coefficients in various parts of the model were measured, and plotted against  $N_R$  (Fig. 2). The highest value of  $N_R$  at which a curve flattened out was around  $3.5 \times 10^5$ , corresponding to a velocity in the main line of 83 ft/sec, so it was decided to use a velocity of about 110 ft/sec in all tests, giving an  $N_R$  of  $4.65 \times 10^5$ .

The above considerations, and the evidence of Fig. 2, make it possible to be quite confident that model events faithfully reflect prototype events, even though a different fluid is being used.

However, it must still be realized that the model time scale is very different from the prototype time scale, simply because the velocity and length scales are different. If the suffix "r" indicates the ratio of prototype quantity to corresponding model quantity, we can write

$$T_r = \frac{L_r}{v_r} \quad (5)$$

and this statement is generally true, whether or not the model and prototype are dynamically similar. Since  $L_r = 21$ , and  $v_r \approx 1/10$ , then

$$T_r \approx \frac{21}{1/10} \approx 200 \quad (6)$$

This means that events in the prototype will take 200 times as long to happen as corresponding events in the model, e.g., if a fluid particle takes 1/10 second to get through the model station, then a fluid particle in the prototype will take 20 seconds, or 200 times as long, to get through the station. Similarly the oscillations induced by eddies will be 200 times slower in the prototype than in the model. This point will be discussed further in Section 2.2.

The only question remaining is whether the compressibility of the air might not produce effects in the model which will not occur in the prototype. The effects of compressibility depend on the Mach number  $M$ , the ratio of flow velocity to sonic velocity, and make proportional changes that depend on the square of the Mach number. For instance, the relationship between stagnation pressure  $p_s$  and static pressure  $p_o$  is, for incompressible flow:

$$p_s = p_o + \frac{1}{2} \rho v_o^2 \quad (7)$$

and for compressible flow:

$$p_s = p_o + \frac{1}{2} \rho v_o^2 \left( 1 + \frac{1}{4} M^2 + \dots \right) \quad (8)$$

In our case  $M = 110/1100 = 1/10$ , so that compressibility effects will make changes in the flow parameters of the order of one-quarter of one percent. These changes are negligibly small.

## 2.0 MODEL PERFORMANCE AND DESIGN MODIFICATION

This question is discussed under three headings. The first is that of head loss in flow through the station, and the second the flow conditions in the suction pit and pump intakes insofar as they seem likely to affect pump performance. Discussion of both topics is concerned mainly with full flow through the station (all pumps running); a third section is added in which partial flow is considered both from the viewpoint of head loss and that of pump performance.

### 2.1 HEAD LOSSES

As implied in Section 1.3, a general measure of head loss is provided by the loss coefficient  $C_L$ , defined in Equation (2), and if the model  $N_R$  is high enough (as it is in this case) the  $C_L$  in any region of the model should be the same as the  $C_L$  in the corresponding region of the prototype. The losses in various parts of the system will be considered in the following discussion. The coefficient  $C_L$  will be defined in two ways. The "nominal"  $C_L$  is the head loss divided by the velocity head in the main 14-ft line. The "true"  $C_L$  is the head loss divided by the velocity head at the particular section concerned. The significance of each coefficient will be discussed below.

#### The Original Design

The original design was given a test run, and pressure measured at the points indicated in Fig. 1. Measurements on pump lines were made on Pumps Nos. 2 and 4 only. Nominal and true values of  $C_L$  are tabulated below. The following system is used to define the terms used.

Velocities:- Subscript "o" - velocity in main line  
 "p" - velocity at entry to pit from vertical line.  
 "t" - velocity at throat of pump intake.  
 "e" - velocity at elbow, in pump discharge line.  
 "d" - velocity at downstream end, horizontal pump discharge line.

Discharges:- "2" - Pump line No. 2  
 "4" - Pump line No. 4.

The nominal  $C_L$ 's indicate whether the losses in individual parts are a significant fraction of the losses through the station as a whole, and the true  $C_L$ 's are in sufficiently general form to be compared with text book values; if they are sufficiently far above standard values, then there is some hope of

TABLE I  
SUMMARY OF LOSS COEFFICIENTS, ORIGINAL DESIGN

Locality	Nominal $C_L$		Basis for True $C_L$	True $C_L$	
Downturn from main	1.81		$v_o^2/2g$	1.81	
Vertical into pit	0.71		$(v_o - v_p)^2/2g$	8.25	
Pit entry	0.40		$v_p^2/2g$	0.80	
	<u>No. 2</u>	<u>No. 4</u>		<u>No. 2</u>	<u>No. 4</u>
Pit to pump intake	-0.19	0	$v_t^2/2g$	-0.083	0
Vertical pump line	0.26	0.06	$(v_t - v_e)^2/2g$	0.893	0.318
Elbow, pump line	0.27	0.24	$v_e^2/2g$	0.287	0.400
Horizontal pump line	0.13	0.07	$(v_e - v_d)^2/2g$	0.553	0.625
Manifold entry	0	0.106	$(v_d - v_o)^2/2g$	0	0.239
Total	3.38				

reducing them by the installation of turning vanes, etc.

The immediate conclusion from the above results is that over 80% of the loss occurs in the flow from the main into the pit. Losses in the flow from the pit back to the main, although high in terms of the true  $C_L$ , are not a significant part of the total head loss. Study of these parts was therefore mainly deferred until work was done on the flow conditions at the pump intakes (Section 3.2). The various parts of the station are now considered one by one.

#### Downturn from Main

The installation of vanes in a right angle bend should normally reduce the loss coefficient to around 0.2 - 0.3, so vanes would seem to be the obvious remedy. But because of the possibility that these vanes would interfere with the straight-through gravity flow when the station was not pumping, alternative remedies were tried, e.g., the installation of vertical splitters and baffles to reduce the vorticity, and the moving of the main butterfly valve upstream so that when closed it formed a flat surface lying across the junction of the main line and the vertical line. None of these proposed remedies made any appreciable improvement, so they will not be described in detail in this report. Finally it was decided to try vanes, and to reexamine the question of what interference they would cause in the gravity flow and whether the consequent increase in pumping costs might not be offset by the savings made by the vanes when the station was pumping. To this end tests were made with various vane installations, the decrease in pumped-flow  $C_L$  and the increase in gravity-flow  $C_L$  were measured, and a complete economic analysis made based

on the present annual operating cycle. The result of this analysis is summarized in Tables II and III. Table II shows the detailed breakdown of costs—both the savings during that part of the year when pumping is necessary, and the extra costs during the part of the year when only gravity flow is needed. During gravity flow the water is flowing through the main line as well as through the pumping station, which is on a bypass off the main line. Hence increased resistance to flow through the station is not serious, and the extra costs referred to above are small. Thus for all vane arrangements there is a substantial net saving which is a maximum for arrangement "L":— 8/21-in. radius vanes on a line at  $45^\circ$  to the pipe center lines.

Detailed calculations leading to Tables II and III are given in Appendix I.

For arrangement "L," the loss coefficient at the downturn is 0.49. However, when further modifications were made in the vertical line, pit entry, etc., it was found that the downturn  $C_L$  was further reduced from 0.49 to 0.32. Since this approaches the lowest value that can be hoped for in a vane installation at a right-angle elbow, arrangement "L" is taken as the final design recommendation. The last line in Table III gives the corresponding figures in money, and the series is described as series Ll. The net saving is approximately \$5700 p.a. This reduction in pumping flow  $C_L$ , from 0.49 to 0.32, is brought about by features down in the pit structure, below the main line. These features cannot, of course, affect the gravity flow; hence the  $C_L$  for gravity flow remains at 2.95, unaffected by the further change in pumping-flow  $C_L$ .

#### Vertical Line to Pit, and Pit Entry

Table I shows that these two sections have, in the original design, a total nominal  $C_L$  of 1.11. The true  $C_L$  for the vertical line is extremely high, and the true  $C_L$  for the pit entry approximates to the figure for discharge of a pipe into a reservoir. This suggests that guide vanes might substantially lower the  $C_L$  in both cases. Various kinds of straightening guide vanes were tried in the vertical line, without any success, but there was an improvement when diffusing vanes were installed at the pit entry, and a hump was placed in the floor of the pit directly under the vertical line. A sketch of the system is shown in Fig. 4, described as "Old vanes at pit entry." This system reduced the total  $C_L$  for the whole system to around 1.5, with slight variations from this figure depending on particular installations in the pit. A further system of vanes, described as "New vanes at pit entry" in Fig. 4, was also tried and gave a further slight decrease in  $C_L$ . This system is incorporated in the final recommended design not only because of head loss considerations, but also because it gave slightly better flow conditions in the pit and at the pump intakes.

The installations described gave a total nominal  $C_L$  from the station entry to the pit of about 0.81; the remaining 0.6 (approximately) occurred in the pit, pump lines, and exit manifold.

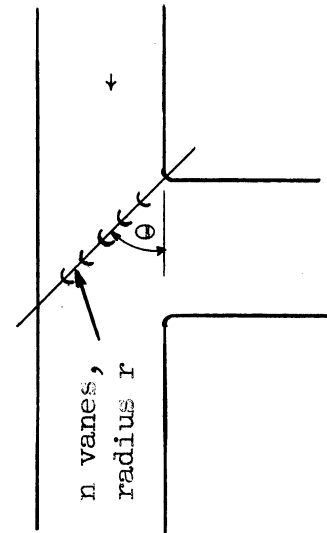
TABLE II  
 ECONOMIC STUDY OF VANE INSTALLATION AT DOWNTURN  
 Sample Calculation - Series F

Period	Length of Period (days)	Avg. Flow During Period % of max.	mgd	cfs	$\frac{V_0^2}{2g}$	Reduction in $C_L$ (same for partial as for full flow)	Head Saved (ft)	Money Saved \$ per year
Pumping	1	91	891	1378	1.245	3.38 - 2.58 = 0.8	1.00	1300
	2	76.55	750	1160	0.882	0.8	0.705	772
	3	67.2	658	1018	0.68	0.8	0.544	523
	4	62.2	610	943	0.584	0.8	0.467	416
	5	60.25	590	912	0.545	0.8	0.436	376
						Total		3387
Gravity	6	58.85	576	891		1.553 - 1.39 = 0.163	0.130	109
	7	57.9	567	877		0.163	0.125	104
	8	56.4	552	855		0.163	0.119	96
	9	53.35	522	808		0.163	0.1065	81
	10	47.9	469	725		0.163	0.0857	59
						Total		449
								Net Saving \$2940 p.a.

TABLE III

ECONOMIC STUDY OF VANE INSTALLATION AT DOWNTURN  
Summary of Results

Series	Description		Pumped Flow			Gravity Flow					Net Saving \$ per yr			
	n	r	$\theta$	$C_L$	Increase in $C_L$ from Original	Saving \$ per yr	$C_L$	Increase in $C_L$ from Original	Increase in $K_1 = K_v$	$K_r = 4.75 + K_v$		$K_0$	Increase in $K_0$	Loss \$ per yr
Original Design				3.38	0	0	0.54	0	0	4.75	1.39	0	0	0
D	5	5"	45°	3.38	0	0	0.88	0.34	0.22	4.97	1.429	0.039	108	-108
E	5	10-1/2"	45°	2.83	0.55	2326	1.55	1.01	0.653	5.40	1.486	0.096	264	2060
F	5	16"	45°	2.58	0.80	3387	2.22	1.68	1.087	5.84	1.553	0.163	449	2940
G	5	21"	45°	2.52	0.86	3640	2.95	2.41	1.56	6.31	1.614	0.224	617	3020
M	8	16"	45°	2.34	1.04	4400	2.54	2.00	1.294	6.04	1.580	0.190	523	3880
L	8	21"	45°	2.06	1.32	5590	2.95	2.41	1.56	6.31	1.614	0.224	617	4970
N	9	16"	45°	2.25	1.13	4780	2.70	2.16	1.40	6.15	1.590	0.200	550	4230
K	5	10-1/2"	60°	2.82	0.56	2370	1.47	0.93	0.602	5.35	1.483	0.093	256	2110
J	5	16"	60°	2.67	0.71	3010	2.26	1.72	1.112	5.86	1.553	0.163	449	2560
LL	8	21"	45°	1.88	1.49	6310	2.95	2.41	1.56	6.31	1.614	0.224	617	5700



Pit, Pump Lines, and Exit Manifold

In discussing these parts of the system it is impossible to isolate the head-loss question for separate consideration, as most of the modifications made were aimed not at reducing head loss but at improving flow conditions through the pumps. The turning vanes described above had reduced the overall nominal  $C_L$  to about 1.5, and in testing modifications to the pit and pump lines it was decided simply to keep track of variations in this overall  $C_L$  to insure that improvements in pump-line flow conditions were not purchased at the expense of overall head loss. No detailed consideration was given to the distribution of losses to the various parts, because, as exemplified above in the case of the downturn vanes, the  $C_L$  for one part may vary depending on what changes are made in other parts of the system. However, there is some point in breaking down the head-loss figure for the final recommended design and checking whether the head loss in any one section is excessive. This is done in Table IV which is in similar form to Table I.

TABLE IV

SUMMARY OF LOSS COEFFICIENTS, FINAL DESIGN

Locality	Nominal $C_L$		Basis for True $C_L$	True $C_L$	
Downturn from main	0.32		$v_o^2/2g$	0.32	
Vertical into pit	0.08		$(v_o - v_p)^2/2g$	0.93	
Pit entry	0.41		$v_p^2/2g$	0.82	
	<u>No. 2</u>	<u>No. 4</u>		<u>No. 2</u>	<u>No. 4</u>
Pit to pump intake	-0.04	0	$v_t^2/2g$	-0.02	0
Vertical pump line	0.28	0.20	$(v_t - v_e)^2/2g$	0.96	1.06
Elbow, pump line	0.28	0.25	$v_e^2/2g$	0.302	0.413
Horizontal pump line	0.07	0.01	$(v_e - v_d)^2/2g$	0.301	0.104
Manifold entry	0	0.13	$(v_d - v_o)^2/2g$	0	0.29
Total	1.40				

It will be seen from Table IV that most of the saving effected by the pit-entry vanes is in the vertical line rather than the pit entry itself. Some of this effect may be more apparent than real due to local variations in pressure, but in any case a substantial saving has been made. Flow into the pump intake again takes place with negligible loss; the apparent negative value is probably due to some impact pressure effect at the pump intake. The true loss coefficients in the pump discharge lines are all rather higher than they should be, but these losses form such a small part of the total loss that it would not be worth trying to reduce them further. In any case, the fittings that will form part of the pump installation may materially alter the situation.



Flow into the manifold takes place with negligible loss; some runs were made with a vertical splitter installed in the main line at the outlet but the losses at full flow were not reduced thereby, and at partial flow the splitter produced severe interference which substantially increased the losses.

It remains to make a final assessment of the savings made by the model study. The overall loss coefficient has been reduced from 3.38 to 1.40; in the prototype, where the velocity head is 1.5 ft, this represents a saving of 3.0 ft head. It has already been pointed out that the reduction in the downturn losses yields a net saving of \$5700 p.a. Savings elsewhere in the system are not reduced by any extra resistance to gravity flow, so the position can be summed up thus:

	Original $C_L$	Final $C_L$	Reduction in $C_L$	Annual Saving (Pump Flow)	Annual Loss (Gravity)	Net Annual Saving
Downturn	1.81	0.32	1.49	\$6310	\$617	\$5700
Remainder of system	1.57	1.08	0.49	\$2070	---	\$2070
Total	<u>3.38</u>	<u>1.40</u>	<u>1.98</u>			<u>\$7800</u>

Hence the net annual saving in pumping costs is \$7800, representing a capital value of \$125,000.

## 2.2 PUMP INTAKES

This part of the project was aimed at improving the flow conditions in the pit and into the pump intakes to insure the best possible operating conditions for the pumps. The main difficulty lay in setting up standards defining good and poor flow conditions, and devising means of measurement by which those standards could be applied to a particular installation. The obvious approach in the first instance is to examine the flow visually by introducing some kind of indicator, such as smoke or confetti, into the flow; this was tried at the beginning, and later a system of woolen tufts, mounted on rods, was placed in the pit. Further significant indices are the flow distribution between different pumps, and the flow distribution within each pump line. The first step is to examine the flow in the model, from the viewpoint of the above remarks.

### Flow Distribution

In the original design, and with practically all other designs tested, the flow divided almost equally between the two sides of the station, i.e., Pumps Nos. 2 and 4 between them shared 50% of the flow. However, the flow in No. 2 was about 27% greater than the flow in No. 4, and the flow distribution

within the horizontal delivery line of Pump No. 2 was very uneven, the ratio of maximum/minimum velocity being about 2.3. The distribution in No. 4 was much more even, the maximum/minimum velocity ratio being about 1.6. Some of the modifications made in the downturn and pit entry altered the ratio  $Q_2/Q_4$  substantially, but the vane system finally adopted yielded a value of 1.25, very little different from the original figure. Modifications in the pit, aimed at improving flow conditions, brought about some variation in this discharge ratio, but as can be seen from the results in Table V, it kept mainly within the range of 1.20-1.30. Although no complete analysis was made of the point, it was concluded from the evidence and from the current knowledge of flow in manifolds that the system is such as to favor flow in Line No. 2 at the expense of Line No. 4.

However, this does not mean that there will be the same difference in flow between No. 2 and No. 4 when the pumps are running in the prototype, because each pump works against a total head of which the head loss through the pump line is only a small part. If we assume that the pumps will equalize the flow we can estimate the difference in the pump heads in this way:

	<u>No. 2</u>	<u>No. 4</u>
If the flow divides in the ratio,	1.25	1.00
then the loss in each line (Table IV) is:	$0.60 (v_0^2/2g)$	$0.60 (v_0^2/2g)$ .
If the flow now divides in the ratio	1.125	1.125
then the loss in each line becomes:	$0.60 \left(\frac{1.125}{1.25}\right)^2 \frac{v_0^2}{2g}$	$0.60 \left(\frac{1.125}{1.00}\right)^2 \frac{v_0^2}{2g}$ ,
i.e.,	$0.486 (v_0^2/2g)$	$0.76 (v_0^2/2g)$ ,
or, in terms of prototype heads:	0.73 ft	1.14 ft .
But each pump must produce a net head		
rise across its line of approximately	50 ft	50 ft ;
hence pump heads are	50.73	51.14 ,
a difference of 0.8%. This should produce a difference of only one or two percent in the pump discharges.		

Since the pump discharges  $Q_2$  and  $Q_4$  are unequal in the model but about equal in the prototype, the character of the flow in the pit, where it divides between the two pump intakes, will be rather different in the model and the prototype. Tests indicated, however, that the difference was not significant; it was found that gradual closure of the delivery valve in Line No. 2, to reduce  $Q_2/Q_4$  to unity and below, did not alter the form or intensity of the turbulence in the pit and the pump intakes.

The value of  $Q_2/Q_4$  was recorded as a significant parameter in each test, not because it was hoped to obtain  $Q_2 = Q_4$ , but because it was still thought desirable to keep  $Q_2/Q_4$  as low as reasonably possible, even though it was in most cases somewhat greater than unity.

The ratio of maximum/minimum velocity within each pump line was

definitely thought to be a significant parameter, which should be as close as possible to unity for satisfactory intake conditions. It is thought to be a definite merit in the final recommended design that in Line No. 2 this ratio is as low as 1.4 (in many cases it was as high as 3 or 4) and in Line No. 4 it is only 1.21. In the pitot-tube traverses of these lines, the velocity readings nearest the wall ( $1/32$  of a pipe diameter from the wall) were not considered in picking the minimum velocity as the velocities so close to the wall would be expected to be low in any case.

#### Turbulence and Vorticity in the Pit

In the original design a very strong vortex formed under the intake of each of Pumps Nos. 1 and 2, with its axis approximately coincident with the pump center line. Much of the remedial work was aimed at cutting and breaking up this vortex in the pit where velocities are still low, rather than relying on straightening vanes in the pump intake bell mouth, where velocities are high.

This vortex could originate either locally or further upstream, being carried down with the flow to the end of the pit. While much vorticity could originate in secondary flows in the downturn and the pit entry, such secondary flows should be broken up considerably by the vane installations placed at these points. However, with the vanes installed the vortices at the end of the pit appeared as distinctly as ever, although with somewhat less intensity. It seems more profitable, therefore, to explore these possible causes of the vorticity:

1) The pit is curved in plan and there will be separation from the inner wall, concentrating the flow on the outer side. When this flow hits the curved end of the pit it naturally tends to sweep round in a circle before going into the intake. The observed sense of rotation of the vortex confirms this view.

2) The pump intakes are in series and there will be marked retardation (and separation) of the flow just downstream of the upstream intakes (3 and 4). This will make the flow more unstable and more susceptible to the development of vortices.

3) The inherent instability in flow to a small intake from a large mass of fluid: this would apply even if the intake were inserted into a lake of still water.

All attempted remedies, therefore, were designed to overcome the problems set by (1) and (2) above.

The best direct way of observing the vorticity was by means of confetti introduced into the air supply. This gave a clear, although short lived, picture of any vortices. A steadier, though less direct, means of ob-

servation was provided by the woolen tufts described in Section 1.2. The single woolen tuft placed in each pump intake gave a clear picture of a further development: the existence of a limited zone of sometimes very strong turbulence, evidenced by a strong high-frequency flutter of the tuft. Reducing the size of this zone and the intensity of the flutter became an object of concern, and in the final recommended design the zone in each pump intake is small, and the intensity not very strong--the tuft's maximum displacement from the vertical was no more than  $15^\circ$ .

Some light was thrown on the origin of this small region of turbulence when its frequency was measured with a Strobotac. The speed given by the Strobotac in rpm was found to be very close to the speed in rpm of the fan supplying the model. This was found to be true at several different speeds. If the fan is responsible, then it might be thought that this turbulence would be removed by sucking air, rather than blowing it, through the model. This is rather doubtful: there is no reason why disturbances should not be propagated upstream from the fan as well as downstream.

Even if this oscillation is a property of the design and not of the model supply blower, the time scale effect mentioned in Section 1.3 will greatly reduce the period of the corresponding prototype oscillation. Maximum model frequency is 30 cps so that the prototype frequency will be around  $1/6$  cps. This will be much less significant in the operation of the pumps than is indicated by the high frequency in the model.

#### Design Modifications in the Pit

This section contains a brief summary of the modifications tried, their significance, and concludes by describing the recommended design. All modifications tried are sketched in Fig. 4, and all of them were placed in each arm of the pit: i.e., any remarks made about Pumps Nos. 2 and 4 apply also to Pumps Nos. 1 and 3. Results of all tests are summarized in Table V.

End partition.--This is recommended by some manufacturers for this kind of pump intake. It was successful in cutting the large vortex under No. 2 intake, breaking it into two vortices more or less in the vertical plane, but gave rise to a very uneven flow distribution within Line No. 2 (see Mod. No. 5, Table V). However, this feature was successfully used with other modifications such as baffles, etc.

Baffles.--These were the most generally successful modification. The idea was to cut down large-scale vorticity and turbulence in the pit, even at the expense of some energy loss. Since the pit velocity head is only, at most, one-fourteenth of the main-line velocity head, this energy loss is small anyway. The baffles could not actively remedy the flow separation from the inner wall of the pit, but could at least hold the flow in position without further separation. Three forms of baffles were used and are shown in Fig. 4. It was

found that the type described as "paper" gave slightly better results than the other two, and the type of prototype baffle recommended (24-in. square openings, 6 ft long) is a compromise between the "pipe" and "paper" types.

In general the model has shown that with this design baffling is essential; no other modification is so successful in keeping separation, turbulence, and vorticity under control.

Reducing clearance under bell mouths.—A standard recommendation in the design of propellor pump intakes is to bring the bell mouth down until it is separated from the floor by one-half the bell mouth diameter. This was tried (Mods. Y and YL, Table V) but without success. The effect was to starve Pump No. 2, even when both the intakes were lowered, to increase the overall  $C_L$  by about 0.7 (\$50,000 in capitalized pumping costs) and to produce very uneven flow distribution within the pump lines. Further, the vortex under No. 2 was as strong as ever. When a false floor was brought up under the intake of No. 2, the  $C_L$  was still on the high side and the flow distribution was still uneven (Mods. W and WL, Table V).

Accelerating flow at end of pit.—Attempts were made to reduce the vortex under No. 2 by constricting and accelerating the flow as it approaches the end of the pit. The first attempt consisted of the false wall sketched in Fig. 4 (Mods. Q-U, Table V). All these arrangements performed quite well except that the flow distribution within No. 2 was poor but for Mod. 5. This latter was worth serious consideration but is not quite as good as, and is probably more expensive than, the final recommended design. The poor flow distribution was probably due to the "piping" of the flow into the intake, producing the asymmetrical flow distribution typical of flow round a bend. Further runs (11, 12, 12a) were made with the original false walls and paper baffles; results were fairly good except for rather poor flow distribution, some unsteadiness in the manometer readings, and the small regions of intense turbulence in the pump intakes. When the false walls were extended right back to the upstream intake, and the flow was raised slightly to make the pit cross-section area half the full area, the throat turbulence in No. 2 became very violent.

One run (No. 16) was made with a false floor underneath No. 2 up to half the height of the pit. The overall  $C_L$  went up to 2.19, and the No. 2 intake was deprived of most of its supply ( $Q_2/Q_4 = 0.36$ ). This was too extreme, and it was decided to try a false floor going up to 1/4 of the height of the pit. This will be discussed in the next section.

Prevention of separation from the inner wall.—In open channel flow there has been some success in making flow round a bend more uniform, and inhibiting separation, by setting up an artificial secondary flow opposing the natural secondary flow. In this case, the concept could be applied by using small vanes which direct the flow outward along the floor and the top of the pit. This should set up a secondary flow which would bring the flow inward

along the horizontal plane halfway up the pit. A scheme of this sort was tried (Mod. 8, Table V), but with no success. The flow which was directed outward returned inward not across the median horizontal plane, but round the end of the pit, accentuating the vortex under No. 2 intake. Turning the vanes inward (Mod. 9) was no better.

It seemed that the most satisfactory way of preventing separation was simply to force all the flow inward by means of vertical turning vanes running the full depth of the pit. These were tried in various positions (Mods. 17-19) but operated best when placed between the pump intakes and associated with a false floor rising to  $1/4$  of the depth of the pit (Mod. 17). The  $C_L$  was only 1.40 and the maximum/minimum velocity ratio for No. 2 was only 1.40 (for No. 4 it was 1.21). This was the best velocity distribution obtained to date; further, the flow in the pump intakes was straight. The zones of turbulence were small, and the turbulence itself was the least violent encountered in the whole study (displacement of tuft  $\pm 15^\circ$  from vertical).

Vertical vortices were still present in the pit and the woolen tufts disclosed some backflow along the inner wall under No. 4 and along both walls under No. 2 (Plates 3-5). However, the backflow was not serious enough to affect conditions in the intakes themselves.

#### Final Recommended Design

The recommended design is therefore that described as Modification No. 17 in Table V, and shown in Fig. 3. The operation might be further improved by altering the angle of the vanes, but the present recommended angle should give satisfactory operation.

The vanes mentioned in recommendation (6) of the summary—6 radial vanes in each pump intake—were observed to make a further substantial reduction in the intensity of the localized turbulent zones in the throat of each intake. When in addition to this the effect of a butterfly valve and a reduction from 72 in. to 60 in. diameter is taken into consideration it seems that most disturbances will have been damped out by the time the flow reaches the pump impeller.

#### 2.3 BEHAVIOR OF DESIGN AT PARTIAL FLOW

The report so far has dealt with full flow, i.e., all pump lines open. It remains to check the performance of the design when some of the pump lines are closed. Four combinations were tried in all—two with two pumps running, and two with three pumps running. Table VI summarizes the performance with regard to flow distribution, loss coefficient, and behavior at the pump intakes.

TABLE VI

BEHAVIOR OF RECOMMENDED DESIGN WITH PARTIAL FLOWS

Run	Pump Line Discharges ( $Q/Q_0$ )				$\frac{U_{max}}{U_{min}}$ in Lines		Overall $C_L$	Behavior at Pump Intakes
	No. 1	No. 2	No. 3	No. 4	No. 2	No. 4		
No. 1 closed	0	0.378	0.331	0.291	1.51	1.66	1.82	Large vortex under No. 1; small turbulent spot in No. 3
No. 3 closed	0.314	0.377	0	0.309	1.58	1.75	1.84	Spot of weak turbulence in No. 1; regular flow in pit
Nos. 1 and 2 closed	0	0	0.509	0.491	-	1.45	2.91	Vertical vortex under No. 3; moderate turbulence in No. 3
Nos. 3 and 4 closed	0.508	0.492	0	0	1.88	-	2.94	Spot of weak turbulence in No. 1; regular flow in pit

The question of head loss is straightforward: there is little to choose between alternative arrangements for a given number of pumps, and the values of  $C_L$  are approximately those that would be expected with the higher velocities through the pump lines. The value of 1.40 for full flow can be split into two parts:

- 0.8 — applying to flow before it splits and goes to the pumps
- 0.6 — applying to flow after it splits and goes to the pumps

The first part should remain approximately the same at partial flow; the second part will be increased in proportion to the squares of the velocities in the pump lines. Hence we can say, approximately:

For 1 pump closed,  $C_L = 0.8 + 0.6 (4/3)^2 = 1.87$ : cf measured 1.82, 1.84  
 For 2 pumps closed,  $C_L = 0.8 + 0.6 (4/2)^2 = 3.2$ : cf measured 2.91, 2.94

Considering the crudeness of the approximation, the agreement is quite good.

The behavior at the pump intakes was substantially sound, although the flow distribution within each pump line was rather uneven. This unevenness, although undesirable, does not weigh very heavily against the fact that the

flow in the intake throats appeared regular and parallel except in the small zones of turbulence. Even in these zones the turbulence was no stronger than in the case of full flow. There is not a great deal to choose between alternative arrangements for a given number of pumps, but the observations suggest slightly better performance when the upstream pump or pumps is closed—i.e., for 3 pumps running, close No. 3 or No. 4; for 2 pumps running, close No. 3 and No. 4.



APPENDIX I

The annual operating cycle can be approximated by breaking the year into ten equal periods of 36.5 days each. The average flow in each period is shown in Column 2 of Table II. The upper part of the table is concerned with the high-flow portion of the year during which pumping is necessary. In this part of the table the columns after No. 2 lead in logical order to figures for the head saved. This is converted into money by using the conversion

$$1 \text{ million gallons raised 1 foot} = 4 \text{ cents.}$$

We finally arrive at a figure for dollars saved per year. Table II relates to the particular case F (5 — 16-in. radius vanes at 45°) and the gross saving is \$3387 per annum.

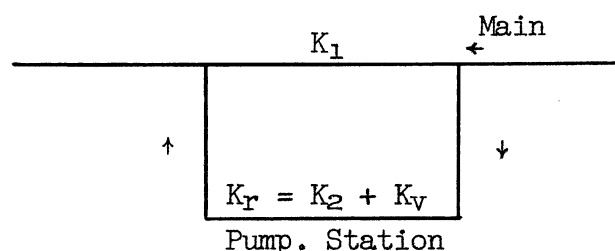
In considering the gravity-flow part of the year, we allow for the fact that the pumping station is in one of two parallel branches, the other one of which (the main line) has a constant loss coefficient. We introduce loss coefficients  $K$  defined by the equation

$$h_L = KQ^2$$

where  $Q$  is the discharge in thousands of cfs. "K" is related to  $C_L$  in this way, for a 14-ft-diameter pipe (which applies to both branches):

$$\begin{aligned} h_L &= C_L \frac{v^2}{2g} \\ &= C_L \cdot \frac{1}{2g} \left( \frac{Q}{.155} \right)^2, \text{ since area of 14-ft} \\ &\quad \text{pipe} = 155 \text{ sq ft} \\ &= 0.647 C_L Q^2 ; \\ \therefore K &= 0.647 C_L . \end{aligned}$$

We now consider the two branches:



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$K_1$  is the (constant) loss coefficient in the main line, estimated at 6.6.  
 $K_2$  is the loss coefficient in the branch line, estimated at 4.75 (without vanes).

$K_v$  is the extra loss coefficient added to the branch line by the vanes.

$$K_R = K_2 + K_v$$

Now if

- $Q_0$  = total flow,  
 $Q_1$  = main flow, and  
 $Q_2$  = branch flow,

we wish to obtain the effective loss coefficient,  $K_0$ , of the two branches combined.

Now

$$\begin{aligned} K_0 &= \frac{h_L}{Q_0^2} \\ &= \frac{K_1 Q_1^2}{Q_0^2} \end{aligned}$$

and

$$\frac{Q_1}{Q_2} = \sqrt{\frac{K_R}{K_1}},$$

$$\therefore \frac{Q_1}{Q_0} = \frac{Q_1}{Q_1 + Q_2} = \frac{1}{1 + \sqrt{K_1/K_R}};$$

$$\therefore K_0 = \frac{K_1}{(1 + \sqrt{K_1/K_R})^2}.$$

The second part of Table II is then found by taking the extra  $C_L$ , and therefore  $K_v$ , introduced by the vanes, obtaining  $K_R$  and hence  $K_0$ . The increase in  $K_0$  over the original  $K_0$  gives the increased  $h_L$  at each discharge, and hence increased cost in dollars. In the particular case dealt with in Table II, this increased cost is \$449 p.a. leaving a net gain of \$3387 - \$449 = \$2938 p.a.

Table III summarizes the results of Table II and applies it to all vane arrangements that were tried. It also gives the values of  $C_L$ ,  $K_v$ , etc., which are applied to the calculations in Table II. All the results of Table III can readily be obtained from the single result of Table II because the gross saving in the pumping cycle is directly proportional to the decrease in  $C_L$ , and the loss in the gravity-flow cycle is directly proportional to the increase in  $K_0$ .

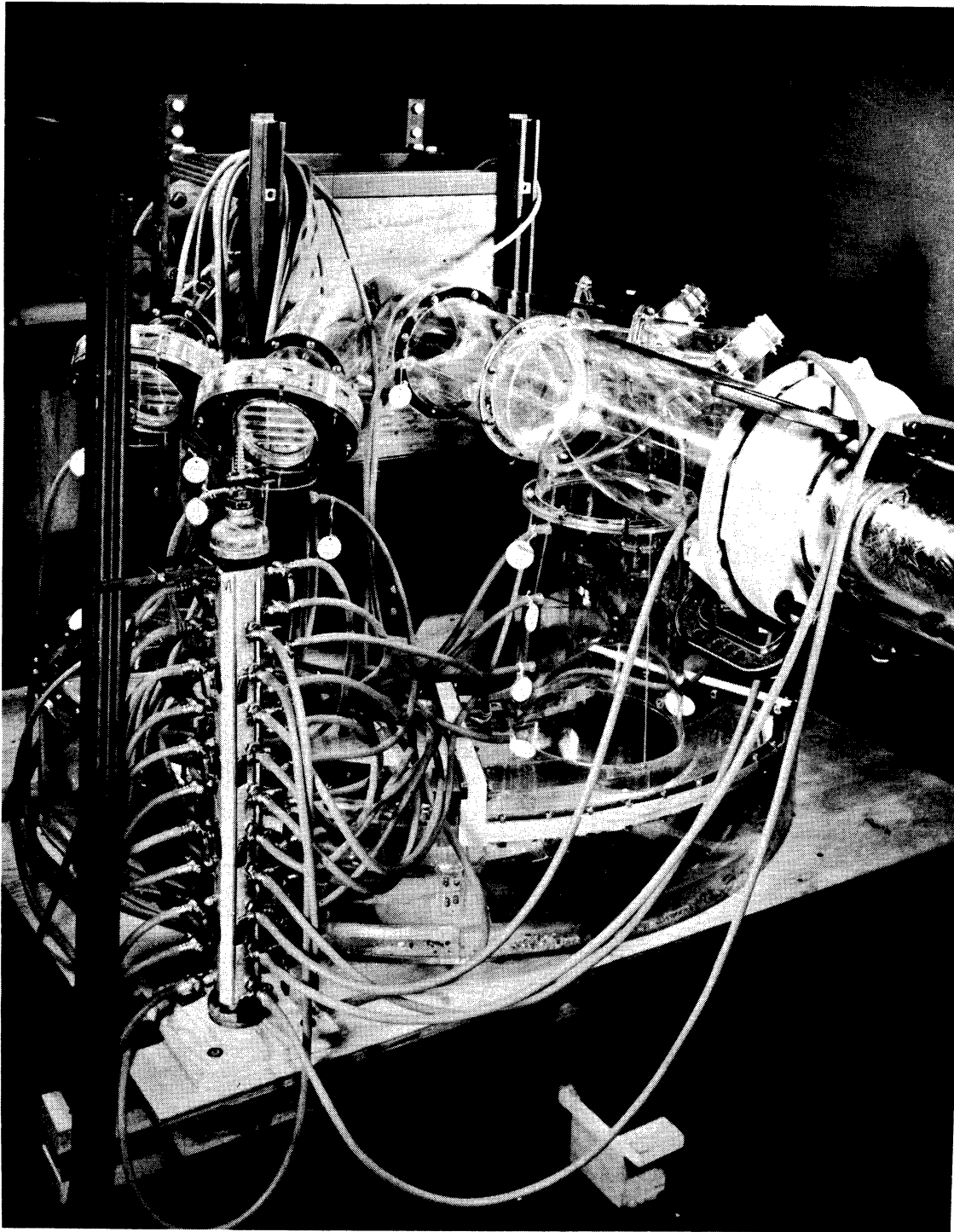


Plate I. View of model from upstream showing manometer connections.

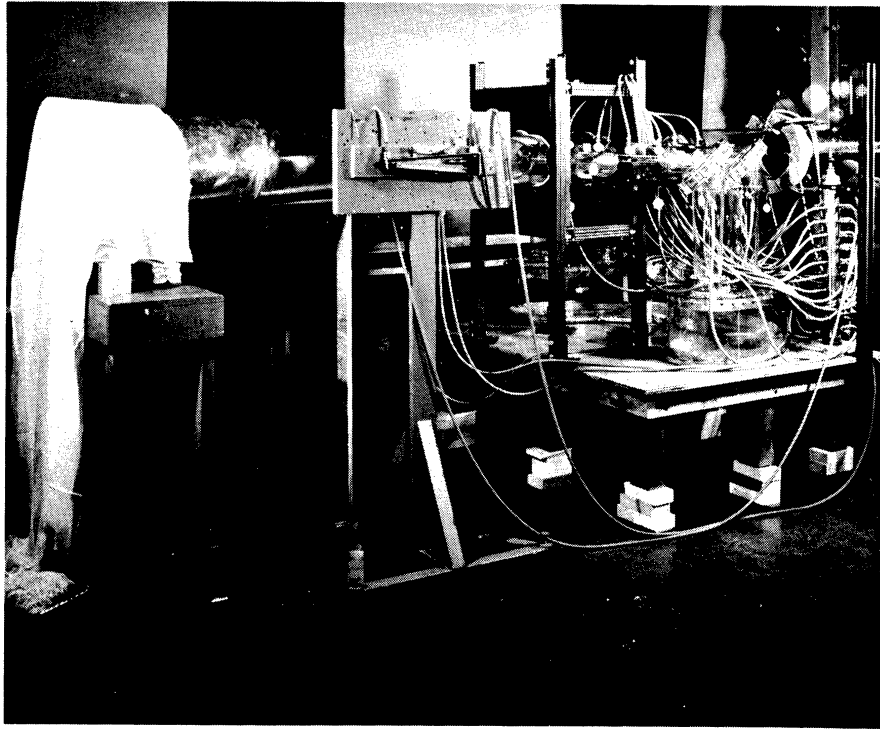


Plate II. View of model from downstream.



Plate III. Recommended design-behavior of woolen tufts upstream of Pump No. 3.

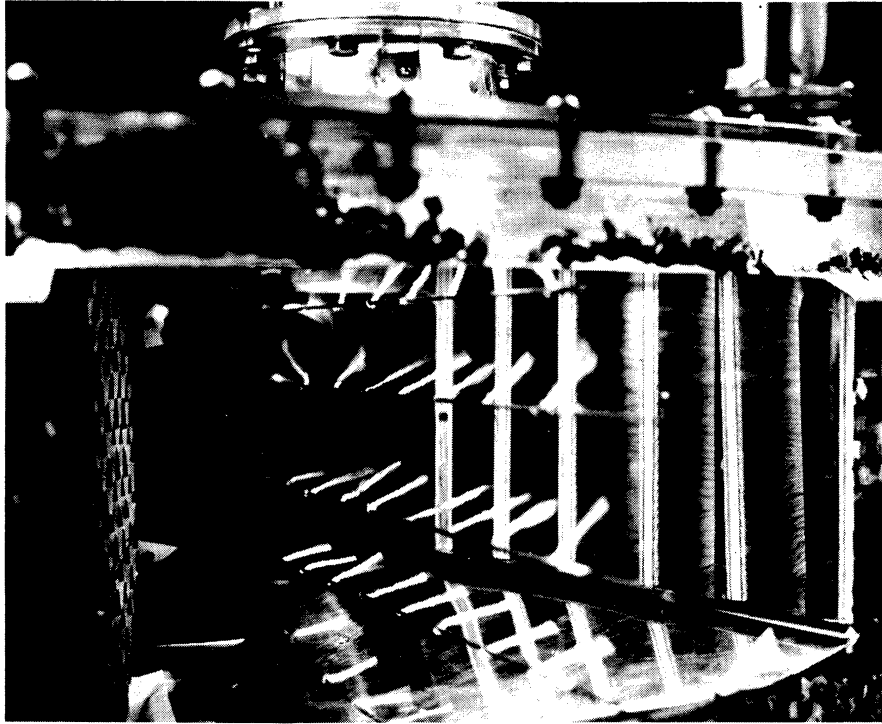


Plate IV. Recommended design-behavior of woolen tufts under Pump No. 3 intake.

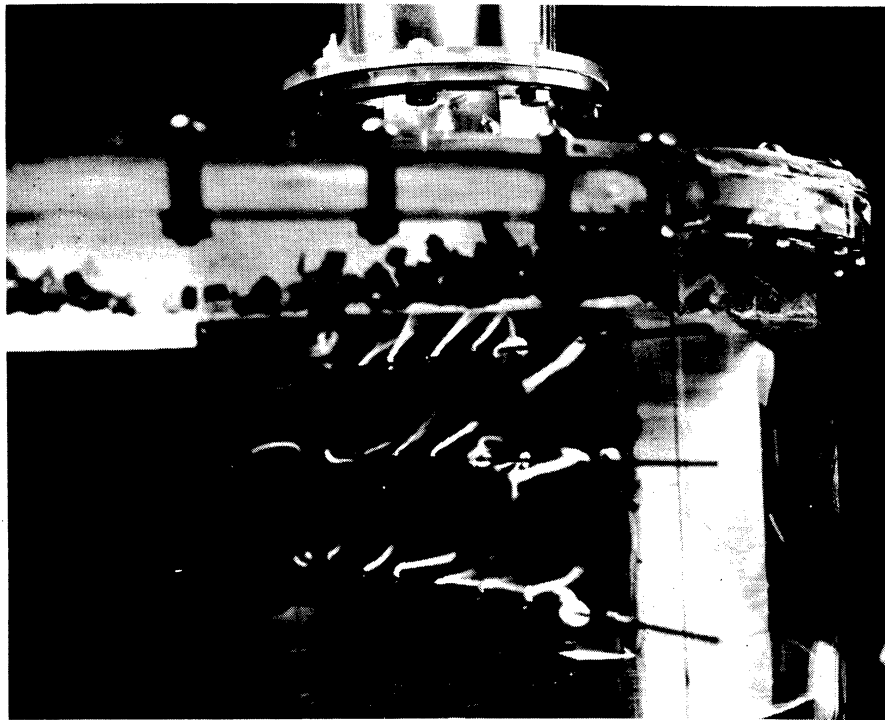
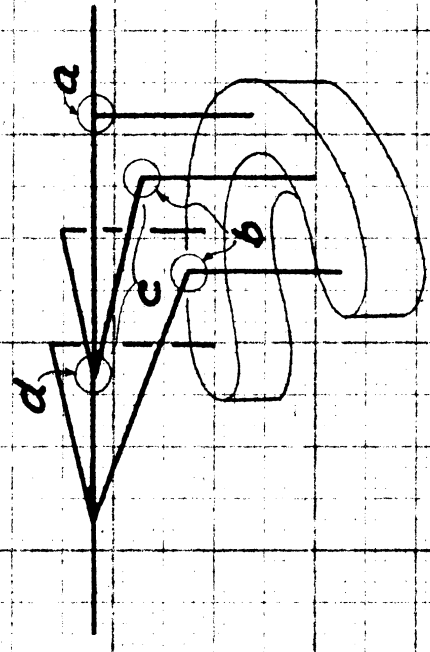
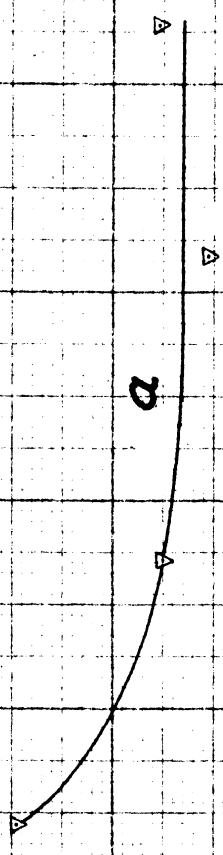


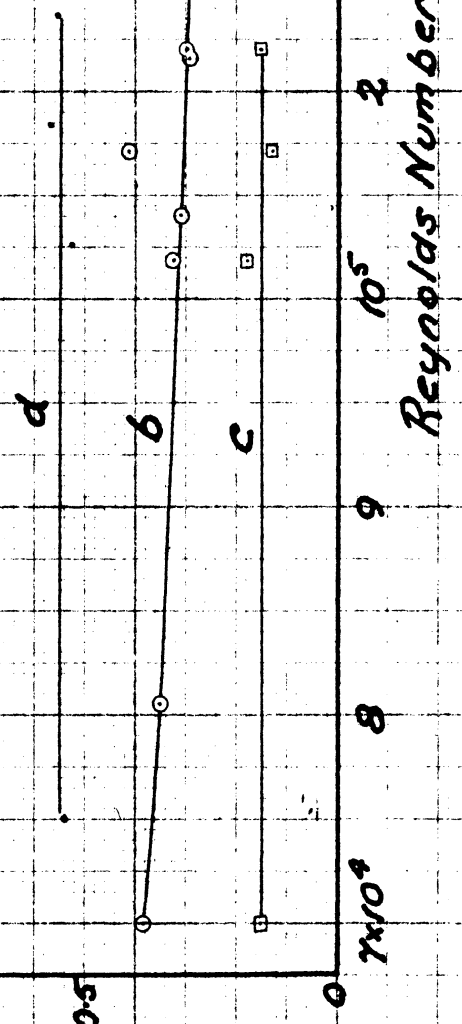
Plate V. Recommended design-behavior of woolen tufts under Pump No. 1 intake.



**BOOSTER STATION MODEL STUDY**

Fig. 2 - Typical  $C_L$ - $Nr$  curves for various parts of the station (Original Design)

$C_L = \frac{\text{Loss in Total Energy}}{\text{Local Velocity head}}$

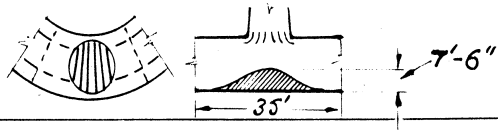
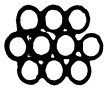
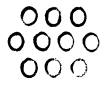
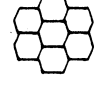


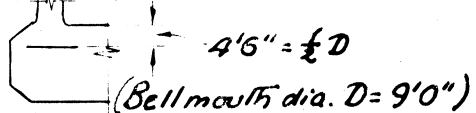
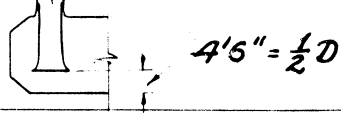

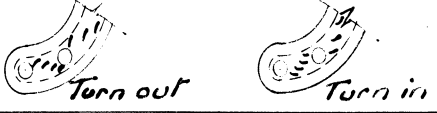

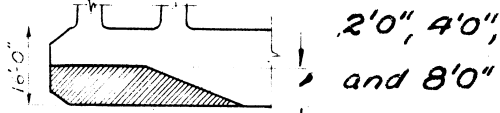


Reynolds Number  $Nr$

7x10<sup>4</sup> 8 9 10<sup>5</sup> 2 3 4 5 6x10<sup>5</sup>

# BOOSTER STATION MODEL STUDY

## FIG. 4 - SKETCHES OF PARTS USED IN MODIFICATIONS

Vaness at Pit Entry	Old	
	New	See Fig. 3
Baffles	Pipe (5'0" wide)	 1'9" outside dia. pipe 2 1/2" wall thickness
	Wood (2'6" wide)	 Perforated wall 16" dia. holes @ 1'9" crs.
	Paper 5'0" wide	 Area of each hole 3 sq. ft. approx.
Pit End	False Walls	 Type 1      Type 2
	End Partition	 See Fig. 3.
Pump Bell Mouth	False Plate	 4'6" = 1/2 D (Bell mouth dia. D = 9'0")
	Extension	 4'6" = 1/2 D
Vertical Splitter		10'6", 12'3", and 17'6" long
Vertical Guide Vanes	Straight (2'10" wide)	 2'-10" spacing
	Stub (2'-0" high)	 Turn out      Turn in
	Inclined	See Fig. 3.
Lateral Wall		 Upper half of suction pit blocked off
False Bottom		 16'0"      2'0", 4'0", and 8'0"





## ADDENDUM TO REPORT 2564-1-F

### "A Model Study of City of Detroit Booster Pumping Station"

Four further runs were made to test two new features: a system of vertical guide vanes at each pump intake, and a new type of baffle installed in the pit. Both features are detailed in the attached sketch, which should be regarded as an addendum to Fig. 4, and results are summarized in the attached table, which is an addendum to Table V. For all of these four runs, the "new" vanes at the pit entry were used, as well as the short partition at the end of the pit, shown in Fig. 4.

The table makes it clear that the baffles, and to a lesser extent the intake guide vanes, were responsible for a large increase in the overall loss coefficient - from 1.40 in the recommended design (No. 17, Table V) to 3.57, which is greater than for the original design at the start of the model study. The conditions in the pit and the pump intakes, briefly described in the table, were if anything worse than in the recommended design. The spots of turbulence in the pump intakes were as numerous, and a little more severe, than in the recommended design and as the table shows, the velocity distribution within pump line No. 2 was much more uneven.

Modification Number.	Overall Loss $C_L$	Flow Distribution $\frac{Q_1/Q_4}{Q_2/Q_4}$	Flow Dist'n within #2 $\frac{Q_{max}}{Q_{min}}$	Observed flow conditions in pit and pump approaches by water level and confetti	Pump intake conditions.
20	1.77	1.1	3.57	Strong whirl & backflow under #1 intake; some backflow at walls under #3 intake & upstream.	Widespread & violent side components & turbulence
21	1.95	0.94	3.62	Regular under #3, vertical lateral vortex upstream, strong backflow under #1	spots of violent turbulence, especially in #3.
22	3.57	0.95	2.0	Slightly irregular at #3 and upstream, strong backflow under #1.	scattered spots of strong turbulence & sideflow.
23	3.51	Not measured	Not measured	Irregular flow upstream; <del>water</del> fairly regular under #1 and #3	Better than 22, but some spots of strong turbulence

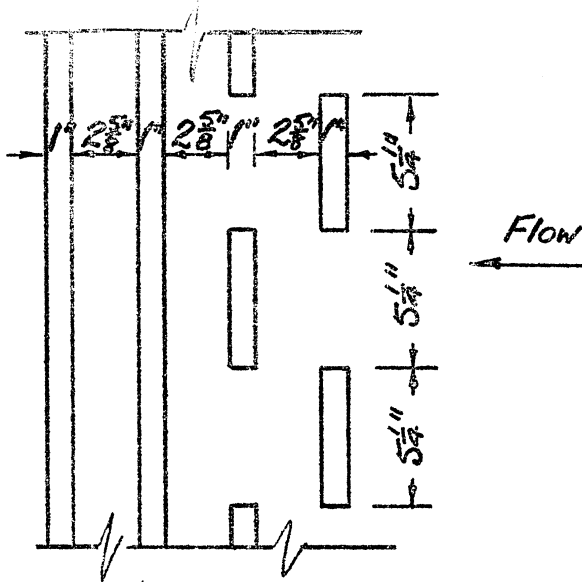
### BOOSTER STATION MODEL STUDY

Supplement to Table I - Modifications

20, 21, 22, 23.

Modification Number.	Vaness at Pit Entry	New Baffles		End Partition	Extended Pump Guide Vanes
		Before pumps	Between pumps		
20	New			✓	✓
21	New		Complete set	✓	✓
22	New	Complete set	Complete set	✓	✓
23	New.	Complete set	Complete set	✓	

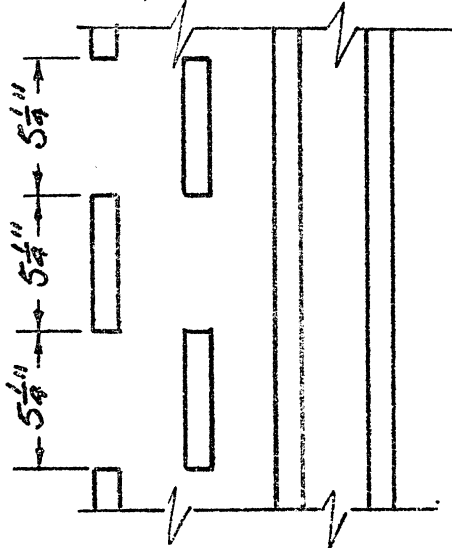
ELEVATION



BAFFLES INSTALLED IN PIT

NOTE: The group of 1 horizontal and 1 vertical set of baffles shown here is described as "1 complete set" in the supplement to Table V

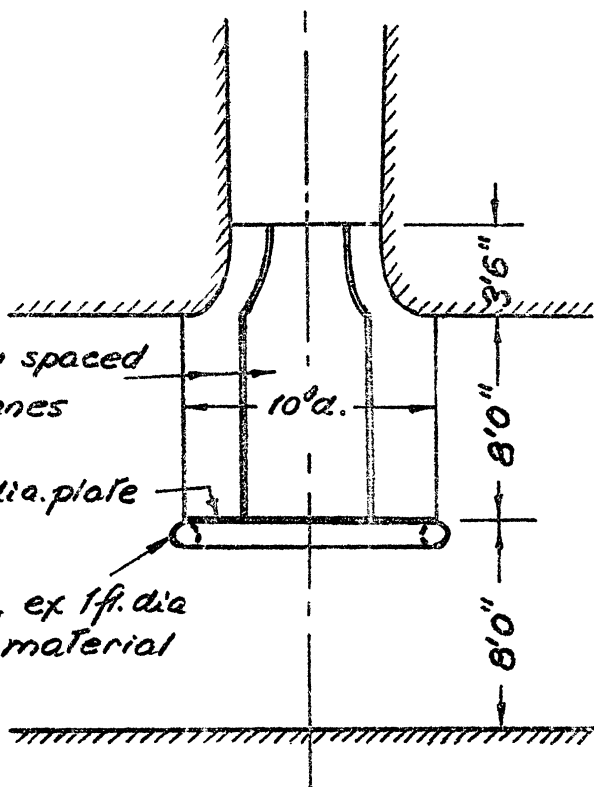
PLAN



Equally spaced radial vanes

10'-0" dia. plate

Torus, ex 1 ft. dia material



VANE SYSTEM AT EACH PUMP INTAKE.

BOOSTER STATION MODEL STUDY.  
Supplement to Fig. 4 - Parts used in modifications 20, 21, 22, 23

Scale: 1/8 in - 1 foot.

All measurements in part: fig. dimensions