A STUDY
of
DESIGN AND OPERATION
of a
LOW SPEED PRECISION WIND INSTRUMENT TEST FACILITY

by

E. J. Plate

and

J. E. Cermak

Interim Report

The study reported in this document was conducted for the Purchasing and Contracting Division, White Sands Missile Range, New Mexico, under Contract Number DA-29-040-ORD-2346.

Colorado State University
Fort Collins, Colorado
November, 1960

CER60EJP58

A STUDY
of
DESIGN AND OPERATION
of a
LOW SPEED PRECISION
WIND INSTRUMENT TEST FACILITY

by

E. J. Plate

and

J. E. Cermak

Interim Report

The study reported in this document was conducted for the Purchasing and Contracting Division, White Sands Missile Range, New Mexico, under Contract Number DA-29-040-ORD-2346.

Colorado State University
Fort Collins, Colorado
November, 1960

CER60EJP58

TABLE OF CONTENTS

TABLE OF CONTENTS - Continued

A STUDY OF DESIGN AND OPERATION OF A LOW SPEED PRECISION WIND INSTRUMENT TEST FACILITY

1. INTRODUCTION

This report contains an evaluation of the engineering and design features of the Precision Wind Instrument Testing Facility for White Sands Missile Range. Three alternate layouts of the facility are considered and their relative merits and disadvantages indicated. For each of the alternate layouts a number of different drive arrangements are possible which are discussed in detail. The discussion of the alternate layouts is based on engineering principles pertinent to wind tunnel design, on the "Specifications For Precision Wind Instrument Testing Facility (low speed) of April 22, 1960", furnished by the Purchasing and Contracting Division, White Sands Missile Range, New Mexico,[*] which form part of the contract, and on prices as given by manufactures of component parts. The best suited instruments for the requirements of the facility are described.

---

[*] Hereafter referred to as the Specifications.

2. LAYOUT OF FACILITY

In the following, the three alternate layouts are presented and briefly discussed. All three satisfy paragraph 3.2.3 to 3.3.3 of the Specifications.

2.1 Alternate 1 (plan A)

Alternate 1 represents the design on which the bid for the present contract was based. The facility is completely enclosed by the building, with the understanding that not many other activities will be conducted in the building. The area between the return duct and the test section is to be used for storing carbon dioxide cylinders required for complying with paragraph 3.2.10 (c) of the Specifications, as well as for placing parts of the drive-control system. This leaves a working area which extends over the whole length of the building and has a minimum width of about 96". The working area is large enough to ensure convenient handling of all models and instruments which can be installed in the test section.

The dimensions of the facility are chosen so that the requirement of no more divergence of the ducts than 6^0 as well as the requirement of a 5:1 ratio in the areas from the transition to the test section are met. As can be seen from plan A*/, this yields a layout which just fits into the building. It is necessary, however, that all sections be either round or square, thus, the large elbow will have outside diameters of at least 10' x 10'. Consequently, the opening in the wall of 9'4" x 9'4" is not large enough to allow the elbow and other large parts to be brought as units into the building. Therefore, the large parts must either be disassembled or else the opening of the building must be increased to about 150" x 150". In the former case the Contractor reserves the right of charging to the Government all extra costs involved in design, construction, and assembly of these parts of the tunnel which have to be disassembled before entering the building.

---

*/ For plans A, B, and C refer to Appendix B.

The disadvantages of this layout are the limitations on the useable space in the building, and the complicated way of exchanging screens. If this layout is used, the test section transition (part 13 in plan A) must be removed whenever the screens are to be cleaned or exchanged in accordance with Specifications 3.2.6 (b). It is therefore planned to put the transition on casters, so that it can be rolled into the working area. Since screen cleaning is not a very frequent operation, the minor inconvenience of loosening the bolts which connect transition and test section appears to be acceptable.

Certain modifications of the basic design of alternate 1 are necessary if Dynaspede and synchronous motors are put on the same shaft. If the location for the drive is chosen between the two small bends, then Fig. 1 is applicable. If the location is chosen behind the second small elbow, then Fig. 2 gives a reasonable solution which meets all the necessary requirements for a properly designed tunnel. Both solutions involve more labor in layout and construction as well as in design, since more complicated geometrical shapes of ducts have to be built. Another disadvantage of these two solutions is that the corner vanes are not installed in rectangular ducts, therefore, it is not known in advance whether they will perform properly.

A solution which avoids these disadvantages and still allows inclusion of a drive arrangement as mentioned above is shown in Fig. 3. In this design a section of the tunnel is expanded more rapidly than at 6^0, thus length is gained where the walls can be parallel, and where the drive can be located. In order to prevent separation in the rapidly expanding section, a screen is placed across it. This design has been used to advantage in some applications (Ref. 1). The small disadvantage of slightly larger pressure loss can be of no serious consequence.

2.2 Alternate 2 (plan B)

Alternate 2 has been designed in order to show a design which uses a minimum in space of the building. An added feature is the convenience of very low noise level in the building, since the drive section is outside of the building. Since naturally the noise level inside the tunnel does not change with this arrangement, no performance gain can be hoped for.

For this alternate, additional features must be added to safeguard the performance of the tunnel. The influence of the outside temperature changes would be felt in the test section unless all tunnel parts which are outside of the building are insulated. Furthermore, for protection of duct work and drive against climatic influences, an outer shell would have to be built around the facility.

If this design appears more acceptable than alternate 1, the Contractor would purchase and install the insulation, while leaving to the Government the design and construction of the outer shell, as well as construction of the large holes in the wall through which the facility enters and leaves the building.

The dimensions of the facility for this layout are essentially the same as that for alternate 1. The major disadvantage of this design is the added cost for insulation and outer shell--factors which do not influence the performance.

The main advantages of this alternate are the reduction of the noise, the better accessibility to the screens, and the saving of space inside the building.

2.3 Alternate 3 (plan C)

Alternate 3 represents a solution which is very suitable to changes if it is found at a later date that an increase in test section size or length is desirable. It has no restrictions on where to put the drive section, and it permits the selection of any reasonable fan or propeller size. In its utilization of building space, it is a compromise between alternates 1 and 2. It requires a little less insulation and outer shell construction, and the addition of only one hole in the wall.

The drive section can be placed inside or outside the building, as indicated in plan C. It can be placed either immediately downstream from the first part of the diffuser section or between the two small elbows, or downstream from the second small elbow. The location of the drive section should be determined by the predominancy of certain requirements, which will be discussed below. Basically, the alternate 3 allows placement of the drive at almost any desired location, while the two other alternates are more restricted.

2.4 Selection of Most Suitable Layout

The design or modification thereof, should be selected on the basis of the following table, where the best performing alternate has a number 1. The second best is indicated by the number 2 and the poorest by the number 3.

Design Feature.	Layout		
	Plan A	Plan B	Plan C
Uniformity of flow in test section	1(2)	2	1
Velocity control	1	1	1
Noise in building	3	1	2(1)
Vibrations	1	1	1(2)
Saving of space	3	1	2
Cost (lowest = 1)	1	2(3)	3(2)
Turbulence reduction	2	2	1
Temperature control	1	2	2
Accessibility	3	1	1(2)

The table should not be taken as an absolute indication of the performance of each tunnel, but should serve as a guide only. According to the emphasis placed on the individual features mentioned, the desired design should be chosen. It is recommended to use plan A, if no other use of the building is planned and the costs are to be held low. Otherwise the layout of plan C is the most versatile solution.

3. DESIGN OF DUCT WORK

In this section the most important criteria and considerations for the design of an efficient wind tunnel are discussed. It contains an adaptation of the design principles expounded in references (2),(3) and (4) to the particular requirements of the present facility. No details of the structural design are given, since these will become evident in the construction drawings.

3.1 Return Duct

Principally, the design features of the different duct sections are the same in all alternate layouts considered above. The air jet which leaves the test section is slowly expanded until the cross section of the drive section is obtained. Care must be taken to avoid separation of the air flow at the duct walls. According to reference (3) this requirement is met if the angle of divergence of the expanding sections does not exceed 6^0, or the gain in width of the section should be not more than 12 inches per ten feet of duct. This requirement will be met for the expanding transitions from square to round and from round to square by requiring that the maximum angle in these transitions does not exceed 6^0.

3.2 Corners and Turning Vanes

The drive section is investigated in Section 4. For convenience of construction the elbow will be made of 14 gage sheet metal and will have circular cross sections. This is in contrast to the rest of the duct work which will be made out of 3/4" marine grade ply board.

In order to keep at a minimum level the disturbances caused by the deflection of the air jet, turning vanes are arranged in the corners. The vanes are designed on the basis of data given by Silberman in reference (5).

Experience has indicated that a vane design as shown in Fig. 4, will work well for all rectangular corners within reasonable limits of air speed. An air foil shaped set of vanes would show a slightly better performance as far as pressure losses are concerned. The small reduction of pressure losses gained by introducing air foil profiles is not considered important enough to merit the additional construction and design costs involved, especially since no significant gain in uniformity of flow and turbulence reduction can be hoped for.

The vanes in the small elbows will be welded into the corners. The uniformity of the flow through the vanes and the direction of the air jet after leaving the vanes will be carefully checked at some significant test section velocities before the vanes are permanently attached to the walls.

The vanes in the large corners will be welded to steel frames. The frames can be moved slightly so that a satisfactory flow pattern can be obtained during performance tests. In case of later need for cooling of the air flow, one of these vane arrangements can be removed and replaced by hollow vanes through which a cooling brine is pumped.

3.3 Access Doors

Access doors will be provided to all sections of the return duct with the exception of the drive section. For access to the drive section, there will be a man hole in the second of the small elbows which can be closed by a cover. The cover will be tightened by means of screws with wing nuts.

3.4 Screens

The air jet leaving the second larger corner is guided through four screens. The screens consist of stainless steel mesh with wire diameter of 0.0075 inches and 24 x 24 meshes per square inch mounted under tension in clear lumber 2" x 6" frames. The screen characteristics are well known from reference (6). The frames are aligned with the flanges of the transition to the test section and of the second large corner by means of heavy bolts. The screens have the double purpose of producing a uniform flow field and of breaking down the turbulence of the flow. They do not create turbulence by eddy shedding as long as the Reynolds number of the screen defined by $\frac{Vd}{\nu}$ is below 40 (See Refs. (4), (7(and (8). In this parameter d is the diameter of the mesh wire, V the velocity of the air flow, and ν the kinematic viscosity. With the screens to be used, the velocity in the test section can go up to about 35 mph without any eddies generated by the screens. At higher velocities, periodic eddies will be shed by the screens, which interact in such a manner that small scale turbulence results which rapidly decays with distance. Only very rough estimates can be made on the basis of references (4) and (6) which show that it is likely that the screens suffice to maintain the turbulent intensity below the maximum permissible level of 0.2 per cent, according to paragraph 3.3.1 of the Specifications. If against expectations the actual turbulence level should exceed the value of 0.2 per cent then one or more of the screens will be replaced by finer meshed screens.

3.5 Transition to Test Section

After passing through the screens, the air jet is rapidly contracted with the purpose of reducing turbulence and size of boundary layers on the walls. The contraction has to have such a shape that no separation occurs. Contractions of this nature have been studied by Rouse and Hassan (9). The contraction of this facility will be designed according to their findings with a longitudinal section which is bounded by cubical parabola arcs.

Construction features of the contracted section are the casters on which it can be rolled in order to permit access to the screens, and the connection between it and the test section on the one side and the screens on the other end. The casters will be permanently mounted to the structure. The connections consist of quick acting clamps which permit disassembly of the screens in very short time.

3.6 Test Section

The test section will be designed according to paragraph 3.2.5 of the Specifications. Access doors extending over the full height of the structure are required. This in turn demands a structural steel support of roof and bottom. These supports will help in making the test section heavy enough to avoid resonant vibrations.

Direct transmission of vibrations through the walls of the facility to the test section will be held at a minimum by providing rubber gaskets between test section and adjacent sections. The possibility of entirely separating test section and adjacent sections by leaving a narrow air gap between them will be investigated during the performance tests.

The test section will be equipped with rails for the carriage. They will extend sufficiently far into the adjacent sections so that the sensing elements of the instruments can be set at any point within the test section.

4. DRIVE SECTION

4.1 General Considerations

The drive section consists of the fan or propeller and the motor. The selection of the most suitable arrangement for motor and fan is based on considerations of performance, size and costs.

The stringent requirements on the performance of the drive under all speed conditions as specified in paragraphs 3.2.4, 3.2.9 and 3.3 of the Specifications necessitate a compromise between different features of possible drive systems in order to arrive at an all around satisfactoty solution. The required performance characteristics are: non-pulsating flow, mean velocity adjustable from 0 to 75 mph, a noise level in the building which does not exceed 75 db, and turbulence levels at all air speeds of below 0.2 per cent of the mean velocity. It is assumed that the Government considers the performance requirements as much more important than the Specifications given in paragraph 3.2.4 on the drive arrangement itself.

Most of the performance characteristics are directly related to the speed of revolution. Thus, for a given discharge rate, a fan with large diameter will require fewer revolutions per minute than a small diameter fan. Also, less noise is generated at lower speeds, and the necessary power input is smaller. However, a large sized fan needs slower speeds of revolution for the minimum discharge rate than a small fan; thus at low speeds the danger of pulsation, due to the low rpm is appreciable. Another problem encountered at low discharges is the small load and rpm requirement on the motor which may cause motor speed fluctuations. Additional measures must be taken to safeguard the motor performance in this range.

The problems of pulsations in the airflow and of noise can best be solved by providing a blower with as many fan blades as possible. From this point of view an aircraft propeller is not too desirable, since it will have at most four blades. On the other hand, an aircraft propeller may require a smaller hub and nacelle, thus, reducing the problem of providing sufficient length for the after body. Since, however, the length of the after body should be governed, if possible, by the hub diameter, and the hub diameter has to be fairly large for avoiding undesirable flow circulation around the blades as well as for accomodation of either motor or pulleys, not much can be gained by using aircraft propellers. Finally, propellers of the small size desired are rarely ever equipped with pitch adjustment, so that a larger one would have to be cut to size and rebalanced, or a custom made propeller would have to be installed. Therefore, the decision was made to restrict the investigations to commercially available, variable-pitch vane-axial fans. These fans are equipped not only with a large number of adjustable pitch blades, but also with straightener vanes behind the rotating parts which ensure a minimum in rotational flow after the air leaves the fan.

The fan size and motor are chosen on the basis of discharge and power requirements.

4.2 Power Requirements

Selection of fan and motor are based on the maximum discharge and power requirements, which occur at a velocity of 75 mph in the test section. The maximum power requirement is then obtained by computing the pressure losses in the facility at maximum discharge and selecting the appropriate fan from the catalogue of the manufacturer. The pressure loss is initially calculated on the basis of the density of air at 68^0F and at sea level, and is later on corrected for the elevation of 3,500 ft by using correction factors given by the manufacturers.

The maximum pressure losses for each section are computed on the basis of the formula

$$\Delta p_i = K_i \ \rho \ \frac{V_i^2}{2}$$

where K_i is the pressure loss coefficient and V_i the mean velocity in section i.

The maximum discharge through the facility is equal to

$$Q_{max} = \frac{75 \times 5280}{3600} \times 4 \times 4 = 1760 \text{ cfs} = 105500 \text{ cfm}$$

For all friction losses smooth boundaries are assumed.

4.2.1 Pressure Loss in Test Section

$$\Delta_p = f \frac{L}{D} \rho \frac{V^2}{2}$$

where
- L = length of test section = 6 ft
- R = hydraulic radius = $\frac{D^2}{4D} = \frac{D}{4}$ = 1 ft
- D = diameter of test section = 4 ft
- ρ = density of air = 0.00234 lb sec²/ft⁴
- V = velocity in test section = $\frac{Q}{D^2}$ = 110 ft/sec
- Re = Reynolds number = $\frac{4RV}{\nu}$ = $2.72 \cdot 10^6$
- ν = kinematic viscosity = $1.62 \cdot 10^{-4}$ ft²/sec
- f = friction coefficient $f(Re)$ found from Ref. 10 page 215 to be 0.010

so that

$$\Delta_p = 0.246 \text{ lb/ft}^2$$

4.2.2 Pressure Loss in Expanding Sections 2, 4, 6, 8 and 9

Very little information is available on expanding sections. Therefore, it is assumed that the pressure loss consists of two parts. The first part is due to the momentum change and is equal to

$$\Delta_{p_1} = K \rho \frac{(V_1 - V_2)^2}{2}$$

where K is a coefficient taken from Fig. 22 in reference 11, p. 418, and is equal to approximately 0.13.

$$V_1 = \text{velocity at small end} = \frac{1760}{16} = 110 \text{ ft/sec}$$

$$V_2 = \text{velocity at large end} = \frac{1760}{81} = 21.7 \text{ ft/sec}$$

so that

$$\Delta_{p_1} = 1.18 \text{ lb/ft}^2$$

The second part is the loss due to friction. It is assumed to be given by

$$\Delta_{p_2} = f \frac{L}{D_m} \rho \frac{{V_m}^2}{2}$$

where

$$V_m = \text{mean velocity} = 1/2 \sqrt{{V_1}^2 + {V_2}^2} = 56 \text{ ft/sec}$$

$$D_m = \text{mean diameter} \sqrt{\frac{Q}{V_m}} = 5.61 \text{ ft}$$

$$R_{e_m} = 1.94 \cdot 10^6 \text{ which yields } f = 0.0105$$

so that

$$\Delta_{p_2} = 0.34 \text{ lb/ft}^2$$

and

$$\Delta_p = 1.52 \text{ lb/ft}^2$$

4.2.3 Pressure Loss in Vane Damper

The pressure-loss coefficient for the vane damper depends to a large extent on the inclination of the blades. For maximum discharge it can be assumed, however, that the damper is fully open. Since the blades are of air foil shape, a pressure loss coefficient of 0.1 is certainly adequate. It is assumed that the damper blades are placed every 6 inches in an opening of 60" x 60", and that the thickness of the blades is 1".

The pressure loss through the damper is then computed according to the formula

$$\Delta_p = 0.1\ A_0\ \rho \frac{V^2}{2}$$

where A_0 is the unobstructed area of the damper which is

$$A_0 = \frac{(60 - 10.1) \cdot 60}{144} = 20.8\ ft^2$$

Thus, the pressure loss becomes:

$$\Delta_p = 0.1 \times 20.8 \times 0.00234 \times 1/2 \left(\frac{1760}{20.8}\right)^2 = 0.83\ lbs/ft^2$$

4.2.4 Pressure Loss in Sections 5, 5a and 7

The pressure losses in section 7 are included in the fan efficiency and therefore, need not be considered separately.

An expression for the corner losses based on experiments is given in Ref. 3, p. 67.

$$\Delta_p = K\ \rho \frac{V^2}{2}$$

where

$$K = 0.10 + \frac{4.55}{(\log R_e)2.58} .$$

The corner ducts are round, thus the velocity in the 6 ft corners is equal to $\dfrac{1760}{\pi \frac{6^2}{4}} = 62.2\ fps = V_1$ and the Reynolds number becomes

$$R_e = \frac{62.2 \cdot 6}{162} \cdot 10^4 = 2.3 \cdot 10^6$$ so that $(\log R_e)^{2.58} = 118$ and

$$K = 0.10 + \frac{4.55}{118} = 0.138$$

The pressure loss is thus given by

$$\Delta_p = 0.138 \times 0.00234 \times 1/2 \times 62.2^2$$
$$= 0.625 \text{ lb/ft}^2 \text{ per corner}$$

4.2.5 Pressure Loss in Large Corners

The same formula is used for sections 10 and 11 as for sections 5 and 5a. These sections are square; therefore the velocity is

$$V = \frac{1760}{9.9} = 21.75 \text{ fps}$$

and the Reynolds number becomes

$$Re = \frac{21.75 \cdot 9}{1.62} \cdot 10^4 = 1.208 \cdot 10^6 \text{ with } (\log Re)^{2.58} = 105.1$$

$$\text{Thus,} \quad K = 0.10 + \frac{4.55}{105.1} = 0.1433$$

and the following pressure loss results:

$$\Delta_p = 0.1433 \quad 0.00234.1/2 \quad 21.75^2$$
$$= 0.079 \text{ lb/ft}^2 \text{ per corner}$$

4.2.6 Pressure Loss Through Screens

Pressure losses across each screen are given by

$$\Delta_p = K_s \, \rho \, \frac{V^2}{2}$$

where $V = \frac{1760}{81} = \_1.75$ fps is the velocity in the section before the screen and K_s is the pressure loss coefficient as taken from Fig. 3 of Ref. 6. The screens chosen are equal to screen B of Ref. 6.

The Reynolds number is given by

$$Re = \frac{21.75 \cdot 0.0075}{1.62 \quad 12} \cdot 10^4 = 84$$

to which a value of K_s of about 0.7 corresponds. Thus

$$\Delta_p = 0.7 \times 0.00234 \times 1/2 \times 21.75^2$$

$$= 0.7 \cdot 2.34 \cdot 1/2 \cdot 10^{-3} \cdot 0.2175^2 \cdot 10^4$$

$$= 0.387 \text{ lb/ft}^2 \text{ per screen}$$

4.2.7 Pressure loss in Transition

Losses in the contraction cone are mainly caused by friction; an approximate formula for the pressure loss coefficient K in the equation $\Delta_p = K \rho \frac{V^2}{2}$ is given in Ref. 3, p. 68, as:

$$K = 0.32 \; f \frac{L}{D_o}$$

where L is the length and D_o the smaller diameter of the contraction. The value for f again is based on an average value of Re, consequently the value taken for computing the losses in the return duct (sections 2, 4, 6, 8 and 9) can be used; i.e., f = 0.0105, and V = 110 fps. Thus:

$$\Delta_p = 0.32 \times 0.0105 \times \frac{108}{48} \times 1/2 \times 110^2 \times 0.00234$$

$$\Delta_p = 0.107 \text{ lbs/ft}^2$$

4.2.8 Summary of Pressure Losses

By adding the pressure losses for all sections, the total pressure loss is obtained.

Test section		0.246 lbs/ft^2
Return duct		1.522 lbs/ft^2
Small corner 1		0.625 lbs/ft^2
Small corner 2		0.625 lbs/ft^2
Large corner 1		0.079 lbs/ft^2
Large corner 2		0.079 lbs/ft^2
Screens: 4 x 0.387		1.548 lbs/ft^2
Transition to test section		0.107 lbs/ft^2
Total without vane damper		4.831 lbs/ft^2
	=	0.93 in. water
Vane damper		0.830 lbs/ft^2
Total with vane damper		5.661 lbs/ft^2
	=	1.09 in. water

On the basis of these pressure losses, an appropriate fan can be chosen.

4.3 Fan Selection

The required fan size can be found by dividing the pressure losses determined above by a density correction factor η for altitude, going into the manufacturers charts for applicable blower at the desired discharge with the corrected pressure. The HP-value of the manufacturer must then be multiplied by the correction factor η to give the maximum HP output of the motor. The corrections are made for an altitude of 3,500 ft; a value of $\eta = 0.875$ is applicable. Thus, the corrected pressures for fan selection are

without damper: $P_T = 0.93/0.875 = 1.06$ in. H_2O

with damper: $P_T = 1.09/0.875 = 1.25$ in. H_2O

Suitable fans, their power requirements, and corresponding noise levels are:

Fan diameter (in.)	With damper				Without damper			
	690 rpm		860 rpm		690 rpm		860 rpm	
	BHP	db	BHP	db	BHP	db	BHP	db
72	24.8	80-82	25.1	82-84	20.8	80-82	21.4	82-84
78	22.9	81-83	23.8	83-85	19.1	81-83	20.3	83.85
84	22.7	82.84	23.6	84-86	18.1	82-84	20.1	84-86

A survey of different fan performances discloses that most of the difficulties in obtaining a stable speed in the test section must be expected at low speeds of revolution of the fan. Since a small fan has the higher revolution at given discharges, the smallest size of the fans listed has been selected.

Among the commercially available fans the one most suited for the present application appears to be a Series 1000 axivane fan made by the Joy Manufacturing Company, with a diameter of 72" and controllable pitch. An after body suitable for the present application will be furnished by the manufacturer. The fan allows a continuous pitch adjustment by remote control. The pitch can be changed while the fan is operating. Characteristics of this fan are shown in Fig. 5. They are based on the drawing of the manufacturer for 690 rpm and have been extrapolated to other rpm values.

Also shown in the figure are characteristics obtained by varying the pitch of the blades while keeping the speed of revolution constant at 690 rpm. The numbers from 2 to 12 correspond to arbitrary settings of the blades.

The third group of curves shown in Fig. 5 represents load lines of the duct. For their computation it has been assumed that the total pressure-loss coefficient remains constant throughout the range of speed and equal to the one corresponding to maximum test section velocity. The two load lines shown indicate the approximate range of load lines which can be obtained with a damper. The lower limit is given by the load line for the duct without damper. This arrangement requires the smallest amount of power to be supplied to the drive. As it is, a 30 HP-motor is required regardless of damper, since the standard motor size which comes closest to the computed requirements is 30 HP. This value includes a factor of safety to account for unknown pressure losses due to instruments and carriage as well as for differences between the assumed and the actual pressure-loss coefficients.

The maximum speed of rotation required by the duct system at optimum pitch setting is approximately 700 to 720 rpm. For a factor of safety the design rpm value is 800 rpm. Maximum discharge through the fan is therefore, under favorable conditions, slightly higher than required by the Specifications and is obtained at 800 rpm.

4.4 Pitch Control

Air speed of the selected fan can be controlled in three different ways. For the high test-section velocities either the rpm or the fan pitch can be changed, depending on the type of drive arrangement preferred. The pitch is adjusted by an axial displacement of a short, non-rotating shaft, which operates an internal lever system in the hub and thus changes the pitch. This shaft passes through the center of the hub and extends into the upstream flow. It is moved by a lever spanned diametrically across the fan housing whose pivot is on one side of the outer housing while its free end extends through the opposite side. The total travel of the free lever end between minimum and maximum pitch setting is about 3.5 inches, and a minimum force of about 450 lbs must be applied for overcoming the resistance of the blades against motion. The photographs of the fan appended to this report show quite clearly the internal mechanism and the external lever system by means of which the pitch is controlled (see Figs. 6 and 7).

The pitch-control lever will be driven through a spindle similar to those shown in the photographs. This 3/4" diameter spindle will be fabricated with twenty threads per inch. In this manner the total travel of the pitch-adjustment lever will correspond to about 70 revolutions of the adjustment shaft. The shaft is driven by a geared motor. It is planned to use an Abart geared motor of 1/6 HP with an output of 120 in. lbs torque at 17.5 rpm. With this arrangement about 8 minutes are needed to bring the pitch from minimum to maximum. This allows a very accurate positioning of the pitch-control lever. The motor is connected to a three phase 220 VAC power source. A switch is provided for interchanging 2 phases, thus allowing reversal of direction of rotation. The pitch position is indicated on the control panel by a synchro driven counter. The counter is made by the Veeder Root Corporation, the synchro is a General Electric Selsyn of appropriate size. This arrangement adds 10 units of the counter per revolution for increase in discharge, and subtracts the same number of units per revolution for decreasing the volume flow. Thus, one reading only corresponds to each position of the pitch-adjusting lever. The main importance of the pitch position indicated lies in the fact that it allows the operator to preset the pitch for obtaining a desired discharge. It cannot, however, be expected that the desired discharge is obtained without further fine adjustment of speed of revolution or pitch.

4.5 Selection of Motor

Among the numerous possible solutions for a continuous variation of speed of revolution the following have been investigated:

a. Dynaspede and Synchronous Motor as required in paragraph 3.2.4

b. Modified Dynaspede

c. DC motor powered by Motor Generator Set

Solutions of the drive problem based on the three listed arrangements are briefly discussed and their merits and disadvantages indicated.

4.5.1 Dynaspede and Synchronous Motor Arrangement

Use of the Dynaspede and synchronous motor would permit varying rpm-values from about 100 rpm to 800 rpm continuously. While the speed of revolution is varied with the Dynaspede control, a pitch limiter should be provided, so that the pitch setting will never exceed that value for which the HP requirement at 800 rpm is equal to 15. This can be effected by placing a switch on the pitch control which breaks the circuit of the pitch-geared motor when the Dynaspede is operating and the critical pitch setting reached. This switch will be short circuited by means of the starter switch of the synchronous motor, so that the full pitch can be used only when the synchronous motor is running. In this manner the Dynaspede motor would be protected against overload without necessitating actual overloads. (Inquiries at the Eaton Company, manufacturers of the Dynaspede, revealed that the eddy current transmission of the Dynaspede can take about 80 - 100 per cent overload of the motor without failing).

A particular feature of the Dynaspede transmission is its water cooling system. The cooling water acts also as a transmission fluid which at large output rpm values and large loads has no significant torque transmission effect. At low loads and low output speeds; i.e., large differences in speeds in the transmission, the hydraulic effect may be the decisive factor, and set a limit to the lower velocities. In this case, an eddy current brake may be required for artificial loadings.

The eddy current flux in the Dynaspede coupling is controlled by a small tacho generator which produces a voltage proportional to the revolution of the output shaft. This generator works well down to rpm values of about 100. Below this value, it is necessary to modify the generator to increase its sensitivity. Information from the manufacturer shows this is a rather inexpensive item.

The synchronous motor would be used only for the range extending from about 10 mph below the upper limit of the Dynaspede up to 75 mph. The appropriate speed of the synchronous motor would be 780 rpm. While this speed is possible with a synchronous motor, it appears to be non-standard, so that either a revised motor or a standard motor with higher speed should be used. If it is desired to use this arrangement, efforts will be made to locate a standard 30 HP 780 rpm synchronous motor.

The arrangement of the two motors and the fan with respect to each other depends to some extent on the layout chosen for the wind tunnel. The most appropriate location for the selected fan in alternates 1 and 2 is downstream from the two small elbows. However, placing the fan, synchronous motor and Dynaspede on a single shaft results in a very long after body with corresponding large pressure losses in the long constricted sections. Instead, separating the two motors is possible by connecting the shaft of the synchronous motor to the Dynaspede through a belt drive Since the 15 HP of the Dynaspede suffice to obtain an air flow at about 50 mph, a small loss of efficiency (about 5 per cent) caused by the belt arrangement should not be serious.

Wind tunnel layout alternate 3 permits the fan and both motors to be located behind each other in sections 4 and 5. Both motors would be arranged in the same manner as for alternates 1 and 2.

The most definite disadvantages of the use of the Dynaspede motor and a synchronous motor are given by the high cost of a synchronous motor, and by the complications involved in the circuits of the controls as well as in the coupling of the motors. Therefore, it appeared desirable to look into other possibilities in which only one motor is needed for the same performance.

4.5.2 Modified Dynaspede

The excellent characteristics of the Dynaspede coupling, which result in stable speed control within the operating range, make it desirable to utilize the available Dynaspede in a modified form. It was found out from the manufacturer that the same Dynaspede coupling can be used with a 30 HP motor which has 1680 rpm instead of the 15 HP motor with 800 rpm. The immediate disadvantages of this arrangement are firstly the poorer performance at low loads and rpm values, and second the need for a speed reduction arrangement between Dynaspede and fan.

The poorer performance at low loads and low rpm is caused by the higher difference of velocities of revolution between input and output shaft in the transmission. Due to the higher velocity gradient the cooling water can transmit more torque, therefore, the uncontrolled hydraulic coupling exceeds the controlled magnetic coupling much earlier than in the previous arrangement. An eddy current brake should therefore be supplied with this Dynaspede arrangement, which will be mounted on the output shaft of the Dynaspede.

In order to reduce the speed between motor and fan from 1680 to 800 rpm either a geared speed reducer or a belt and pulley drive can be used. The speed reducer requires motor and fan to be placed close together, in other words, the motor has to be directly behind the fan. A disadvantage of the gear speed reducer is the additional noise and the lubrication of the enclosed gear box. Its main advantage over the belt drive is a longer life and better efficiency.

If a belt drive is used, then the Dynaspede should be outside of the duct work, so that only the pulleys are located in the housing behind the fan. By placing the Dynaspede on top of the fan, the usually unattractive appearance of a belt drive can be completely avoided. In addition, wiring, cooling and lubrication outlets of the Dynaspede drive are easily accessible, and a highly satisfactory solution is obtained. An added feature of this solution is also the low initial cost. As indicated by the manufacturer, the modifications on the Dynaspede can be made for less than half the cost of a synchronous motor.

If the solution of the modified Dynaspede is chosen, then it should be shipped to the Dynamatic Division of the Eaton Manufacturing Company in Kenosha, Wisconsin. The manufacturer would replace the 15 HP motor by a 30 HP motor, discounting the price for the 15 HP motor from the price of the new motor. The manufacturer also would install the eddy current brake, provide leads for the feed back tacho generator so that it can be used for speed indication, and check all parts and the performance of the modified drive. In this way, a satisfactory job guaranteed by the manufacturer can be realized.

In this arrangement no change in the tacho generator is necessary, since 100 rpm indicated by the generator corresponds to less than 50 rpm of the fan. At this fan speed, and with appropriate pitch setting, it should be possible to obtain stable generation down to a test section velocity of 1 mph.

4.5.3 DC Motor and Motor Generator Set

The third solution investigated consists of a DC motor supplied from a motor-generator set. The DC motor would be mounted directly to the fan axis. Its maximum rpm value would be 800 rpm.

This solution is the most uncomplicated one of the three under consideration. Only leads are necessary to come from the motor generator set to the DC motor. The m-g set could therefore be located wherever desired in the building. By buying a General Electric Speed Variator, which consists of a m-g set complete with all controls and feed back speed stabilizers enclosed in a cupboard and the separate DC motor, no additional wiring is necessary and little labor is required for installation.

Disadvantages of this arrangement (which is referred to as arrangement 2 in plans A and B) are: the abandonment of the Dynaspede, resulting in high costs and possibly less reliable speed control at low velocities. According to manufacturers catalogues, a speed control of from 106 to 850 rpm is possible with a 30 HP DC motor.

4.5.4 Selection of Motor

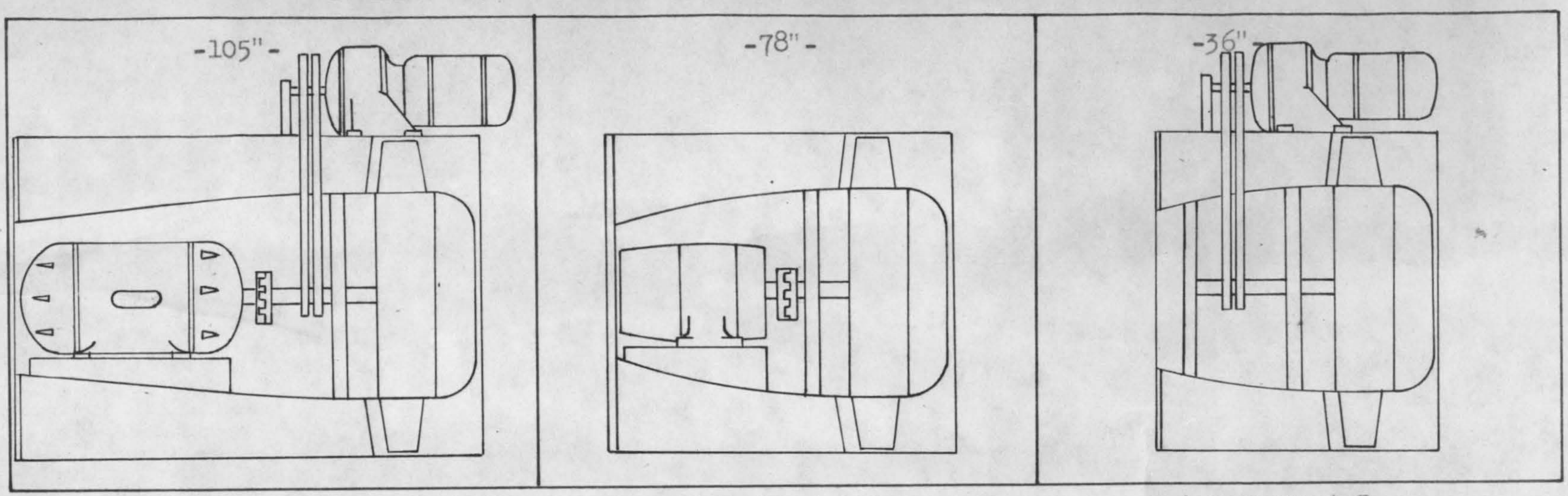
The features of the three motor arrangements investigated are compared in Table 2. Only the most favorable solution for each of the three arrangements have been included in the comparison. The prices quoted include only costs which are different in at least two of the arrangements, common costs pertinent to all three arrangements have been omitted. It has been assumed for price comparison, that the Dynaspede for arrangement 2, available at White Sands, is new.

On the basis of Table 2, it is recommended to use the modified Dynaspede (arrangement 3). It appears to have the only disadvantage of lower drive efficiency, but gains with it the convenience of easy accessibility to motor, transmission and cooling system.

4.6 Vane Damper

If it will be found during the performance tests that difficulties exist at stabilizing the air flow at low speeds, then it is proposed to improve the stability of the flow by the introduction of a vane damper. In this fashion, the pressure across the blower can be regulated to some extent. The damper will be of the opposed blade type. The term "opposed blades" means that every alternate vane rotates clockwise, while the other set of vanes rotates counter-clockwise during adjustment. Thus, instead of producing only a

TABLE 2. COMPARISON OF MOTOR ARRANGEMENTS



Arrangement 1 | Arrangement 2 | Arrangement 3

Feature	Dynaspede and Synchronous Motor	DC Drive and Motor Generator Set	Modified Dynaspede
Drive type	Synchronous Motor direct, Dynaspede Belt	Direct drive of DC motor	Belt drive
Location of drive	Synchronous Motor behind fan, Dynaspede on top	DC Motor behind fan	Dynaspede on top of duct
Speed range	50-800 rpm (with modified tacho-generator)	106-850 rpm (decrease possible at extra cost)	50-800 rpm
Approximate cost	Synchronous Motor, 30 hp, with exiter and starter ≈ 4800, tacho-generator ≈ 80, belts, pulleys and couplings ≈ 250, mechanics labor ≈ 420, electrical labor ≈ 200 Total ≈ 5750 Dollars	30 hp DC drive and MG set with all controls ≈ 9400, couplings ≈ 40, mechanics labor ≈ 60, electrical labor ≈ 30 Total ≈ 9530 Dollars	Replacing motor at factory, including shipment, labor and refund on 15 hp motor ≈ 2600, pulleys and belts ≈ 150, mechanics labor ≈ 160, electrical labor ≈ 80 Total ≈ 2990 Dollars
Drive efficiency	≈ 95% at low loads 100% at high loads	≈ 100% at all loads	≈ 95% at all loads

deflection of the air jet through the damper as in the case of vanes rotating in one direction only, the opposed blades damper breaks down the air stream into a set of narrow jets which remain parallel to each other. Thus, at low velocities, excellent characteristics of speed control are obtained. A sample of the damper characteristics as used by the Contractor is shown in Fig. 8. While it does not allow one to draw a conclusion on the quantitative performance of the proposed damper, it shows that the velocity drop with change in damper angle is quite gradual; a well defined adjustment of velocity should therefore be possible.

The vane damper will be designed by the Contractor and built in the Contractor's shop. The individual vanes consist of air foils of about 6" chord length which can rotate around the center. They are mounted in a channel iron frame, and rotated by a system of mechanical levers.

Since the vane damper is installed near the location of the speed-control panel, the setting as well as the indication of the setting can be accomplished by manually controlled mechanical gears.

5. VELOCITY INDICATION

The setting and maintenance of a desired test velocity is accomplished by manipulating the velocity controls of the control panel according to readings given on the velocity indicators. Thus, the control mechanism consists of two essentially independent parts, the velocity controls and the velocity indicators.

The velocity control principles are described in the previous paragraphs; the proposed arrangements for velocity indication will be treated in this section.

The basic requirements for an indicating system are:

1. Sensitivity towards velocity changes of the order of the acceptable error;
2. Stability, i.e., the indicator should not change its signal with time, so that no calibrations are required, or so that one calibration suffices to define the indicator signal for all times;
3. Convenience in reading, so that the wind tunnel operator can easily detect when an adjustment of the velocity becomes necessary,
4. The instruments or their component parts should be available commercially.

In accordance with these requirements two different instruments have been selected to indicate the velocity in different velocity ranges. For high speeds, the pitot tube equipped with a micromanometer is the most appropriate instrument which satisfies all the requirements. For low speeds, the frequency of eddies shed by a cylinder is an excellent indicator of the velocity. An instrument will therefore be assembled from components which permits measurement of the frequency of these eddies.

5.1 High Speed Indication

The high speed indicating pitot tube is to be manufactured in the Hydraulics Laboratory shop of Colorado State University according to standard specifications (see for example Ref. 12, p. 230). It will be mounted permanently on the carriage in the test section, but not on the instrument carrier of the carriage. The plastic manometer tubing will be taken through the test section wall together with all other cables and tubes. It is connected to the precision manometer (see page 33) over a three-way valve which permits changing the manometer connection from the fixed pitot tube used for velocity indication to the movable pitot tube used for pressure surveys.

Since the manometer can be read accurately within ± one thousandth of an inch of water, the lower limit of the indication depends on the permissible error of measurement. Fig. 9 has been prepared on the basis of this accuracy. It shows the percentage of error which would give a difference in reading of one thousandth of an inch as a function of the velocity. The curve was computed on the basis of the formulas:

$$p = \frac{V^2}{2} = 0.00236 \ \frac{V^2}{2} \text{ (lbs/ft}^2\text{ at } 20^0\text{C and sea level)}$$

$$= 0.000228 \ V^2 \text{ (in. } H_20 \text{ at } 20^0\text{C and sea level)}$$

and

$$(p + \Delta_p) \quad - p = 0.000228 \ (V + \Delta V)^2 - V^2$$

$$= 0.001 \approx 0.000228 \times 2V\Delta V.$$

An error of 5 per cent appears quite acceptable at the low velocities, thus, the pitot tube arrangement is a suitable velocity indicator for velocities above 4.5 mph in the test section.

5.2 Low Speed Indication

For indication of the low velocities from 1 to 4.5 mph the observed fact is used that a cylinder immersed in an air flow sheds periodic eddies whose frequencies n depend only upon the wire diameter d, the flow velocity V and the kinematic viscosity ν of the air, according to the relation

$$n = 0.212 \frac{V}{d} - \frac{4.5}{d^2}$$

for Reynolds numbers between 50 and 150. (See Reference 8). In order to cover the range from 1 to 5 mph within these well defined Reynolds number limits, two different wires are necessary. Their approximate characteristics are shown in Fig. 10. The frequency of the eddy shedding is measured by picking up the periodic velocity fluctuations with a hot wire anemometer mounted in the wake of the cylinder. The output of the hot wire anemometer is fed into an oscilloscope. By establishing an ellipse on the screen of the oscilloscope, the frequency of the hot wire output is matched against the output of an oscillator and thus determined. Initially, the contractor will calibrate this instrument against known velocities determined by a hot wire anemometer calibrated in a calibration tank. After an initial calibration, the instrument will perform as a standard, and no future recalibrations are necessary. Thus, the changes of the characteristics of the hot wire with time do not influence the reading accuracy of the low velocity indicator, since only frequencies are measured.

Preliminary experiments of the Contractor indicate that the amplitude of the voltage fluctuations measured across the hot wire is of the order of magnitude from 0.1 to 10 millivolts. Therefore, the signal is fed into the oscilloscope of the instrument panel through a preamplifier.

The block diagram of the low velocity indicator circuit is shown in Figure 11. Voltage regulator, DC power supply and oscilloscope are used for the instrument panel also and are described on page 39 . In addition, the mentioned preamplifier and an oscillator are needed. According to Fig. 10 the oscillator must have a frequency range from about 10 to 500 cycles, and should permit frequencies to be chosen in the range from 15 to 50 cycles within ± 3 cycles, and in the range from 100 to 500 cycles within ± 10 cycles. An instrument which exceeds these requirements and is therefore well suited for this application is the Hewlett-Packard 200 Interpolation Oscillator. This instrument has a frequency range from 6 to 6,000 cycles at a maximum amplitude of 20 volts.

6. INSTRUMENTATION

The requirements on the instruments as stated in paragraph 3.4 of the Specifications necessitate some interpretations. Also, the requirements on the accuracy of some of the instruments cannot be met unless special instruments are developed by the Contractor or a manufacturer of similar instruments. It was, however, made clear in the letter of submittal of the bid on the present contract, that the Contractor would furnish only such instruments which can be purchased directly from appropriate manufacturers.

The instruments described in the following paragraphs, though not always meeting the requirements of the Specifications, will yield measurements which are comparable to the highest standards now attained in aerodynamic research. When the recommended instrument does not fully meet the Specifications, the difference between the actual and the specified performance will be clearly described.

Generally, when ordering instruments, it will be required that the manufacturers quote accuracies of their instruments which are checked by the National Bureau of Standards. A certificate of this nature shall accompany all instruments, in compliance with paragraph 4.2.4 .

6.1 Instrument Carriage

The instrument carriage will be designed and built according to the requirements stated in paragraph 3.4.1 of the Specifications. It will consist of an airfoil shaped cross member on which the instrument holder travels. The carriage moves longitudinally on rails mounted to the sidewalls of the test section.

The rails for the carriage will be located outside of the test section as indicated in Fig. 12. In this manner it is unnecessary to provide removable rails which would present the continuous problem of readjustment after each removal. The rails will be mounted to the 4" channel which supports the roof. This arrangement does not constrict the test section area and thus does not impede the performance of the tunnel. It has the additional advantage of allowing the instrumentation connections and longitudinal drive members to be hidden in the rail channel.

Each motion of the carriage is powered by a separate DC motor. By regulating the voltage across their fields variable speeds can be obtained, and by interchanging polarities of the field their directions of rotation can be reversed. A schematic diagram of the carriage controls is shown in Fig. 13. The motors will be located so that they do not move with the part that they are driving. Thus, the motor which drives the carriage in the longitudinal direction will be mounted on the outside of the tunnel, while the carriage is connected to it through a system of chains and sprockets. A chain can be used for this application since its backlash does not exceed the permissible error. The motor which controls the transverse movement of the instrument holder is mounted on the carriage platform and operates on a screw which extends over the entire width of the carriage. Turning of the screw moves the instrument holder by means of a nut connected to it in such a manner that it cannot turn. The same type of arrangement controls the up and down motion.

The position of the instruments in the test section is indicated by counters connected through a gear arrangement to the output shaft of the motors by means of a flexible shaft. Screws (or chain and sprockets), gear, motor and counter revolutions will be matched in such a manner that one count read on the counters corresponds to one hundredth of an inch of distance travelled.

The instruments will be mounted to the instrument holder in such a fashion that they are positioned firmly and securely. It will, however, be made possible to remove or exchange instruments with a minimum of effort.

Cables and polyethylene tubing of the instruments, as well as flexible shafts and wires for the drives will be bundled and taken outside of the tunnel in the section downstream from the test section.

6.2 Manometer

The requirements given in paragraph 3.4.3 of the Specifications are interpreted, after consulting with the Contracting Officer, to imply that a manometer should be provided which measures the dynamic air pressure at 3 mph with an accuracy of $\pm$ 10 per cent, and with an accuracy of $\pm <$ 1 per cent at velocities larger than 10 mph. Furthermore, its total range should be wide enough to measure stagnation pressures of velocities up to 100 mph. This accuracy and range should be maintained at all temperatures.

For the maximum velocity the total range required then is approximately 5" of water, while the minimum resolution is given by 0.001" of water as shown in Fig. 9.

A survey of the instruments which satisfy these requirements and are available commercially has revealed two manometers which appear to be suitable, but each one has certain shortcomings. One is a type MM2 manometer built by the Flow Corporation of Cambridge, Massachusetts, and the other is a Meriam Model A-750 manometer.

Both manometers operate on the same principle. They are U-type manometers where one leg is a well and the other is a thin glass tube. Both legs are connected by an inclined glass tube. In the center of the inclined glass tube a hairline is drawn around the tube circumference which serves as zero reference. Initially, both legs of the manometer must be exposed to the same pressure, and the meniscus of the indicating fluid is set at the hairline of the inclined tube, thus yielding a zero reading. If different pressures are applied to the legs, then the meniscus changes position.

It is brought back to the hairline by lifting either the inclined part of the manometer or the well. The distance travelled by the adjusted part is measured accurately and gives an indication of the pressure difference.

For measuring the distance travelled, the Flow Corporation uses a modified micrometer. The Flow Corporation guarantees accurate readings of 0.0002 inches. The shortcoming of the instrument is, however, the limited range of 2 inches. This would accommodate stagnation pressures corresponding to about 60 mph air velocity.

The Meriam Instrument Company uses a precision screw which gives a range of 10 inches with a minimum reading of 0.001 inch.

Both instruments are equipped with reading lenses and temperature calibrated indicating fluid. The costs of the instruments are practically identical (Meriam $577.00, Flow Corp. $585.00.)

Thus, a decision must be made between one instrument with better precision and shorter range than required, and one which has sufficient range but possibly not enough accuracy. The decision of which instrument to provide will be made after discussion with the Contracting Officer.

If most of the testing is done at low velocities which do not exceed 60 mph, then use of the MM-2 manometer is recommended in conjunction with a simple inclined water manometer for velocities higher than 60 mph. This arrangement would require a change of manometers with the danger of careless operators occasionally blowing out the manometer fluid of the MM-2 manometer.

If, on the other hand, it is felt that an accuracy of ±25 per cent at 1 mph is sufficient for dynamic pressures (it should be recalled that the mean velocities will usually be measured with a hot wire anemometer), then the Meriam Instrument Company's manometer should be chosen since no manometer changes would be required.

The manometer will be connected to the precision sensing instruments over a system of valves as shown in Fig. 14. This figure is based on the MM-2 manometer; for the Meriam manometer the manometer valves have to be included in the external switch-board. This can, of course, be done at low cost.

6.3 Pitot Tube and Yaw Probes

The pitot tube and yaw probes will be standard items made in the machine shop from stainless steel as specified in paragraph 3.4.2 and 3.4.5 of the Specifications. They will be connected to the micromanometer as shown in Fig. 14, and will be mounted to the instrument holder of the carriage in such a fashion that they can easily be removed if not needed.

No specific size of the pitot tube is mentioned in the Specifications. Since the size depends to a large extent on the variations with distance of the velocity or pressure to be measured, it is considered advisable to provide a set of 3 pitot tubes with internal diameters of approximately 0.005", 0.01" and 0.05" .

The yaw probes will be made according to Specifications given in Ref. 3. This instrument meets the requirement of paragraph 3.4.5 of the Specifications.

6.4 Temperature Probes and Indication

The temperature probes will be copper constantan thermocouples made from wire certified by the manufacturer of the measuring device. These materials have been selected because they have good temperature response and therefore yield good sensitivity. Copper constantan couples have the advantage of not being subject to corrosion in the atmosphere at standard conditions.

In principle, the curve showing the relation between thermocouple emf and temperature looks slightly different for different thermocouples or different thermocouple wires. Also, thermocouple characteristics vary slightly with time due to aging, corrosion and other influences on the wire. Therefore, with thermocouples an accuracy as the one required in paragraph 3.4.4 of $\pm 0.1^0$F cannot be maintained unless very special measures are taken. For one thing, each thermocouple should be calibrated initially, and the calibration rechecked after certain time intervals. This is a very expensive procedure, unless a temperature standard, like a resistance thermometer, is available. The expense can be reduced somewhat if a reference thermocouple is used only for calibrating and checking the operational thermocouples. The potentiometer to use for this type of operation is a Leeds and Northrup 8686 Portable Millivolt Potentiometer, which has an accuracy of $\pm$ 0.05 per cent of reading or $\pm 6\mu$V, whichever is larger. This should yield an accuracy of temperature measurement of approximately $\pm 0.5^0$F. The instrument is listed at a price of $485.00 in the manufacturer catalogue, and does not include the thermocouples.

The question arises whether an accuracy of $\pm 0.1^0$F required in paragraph 3.4.4 is needed for the application desired. For velocity, pressure, and turbulence measurements the only factors that are affected by the temperature directly are the density and viscosity of the gas. Considering a temperature range between 0^0F and 150^0F as working range for the facility, a change of 1^0F would only produce a maximum change in density of about 0.19 per cent and in viscosity of about 0.36 per cent. Usually, a density and viscosity for computations are taken from Handbooks. Since they vary to some degree with composition of the air (i.e., humidity, carbon dioxide content, etc.,) the Handbook values are certainly accurate only within a few per cent, even if the pressure effect has been considered. Unless other reasons not mentioned in the Specifications prevail, it appears unnecessary to use such an extreme accuracy in the temperature determination if density and viscosity are not measured directly with equal accuracy. Instead, an accuracy of $\pm 1^0$F for calibrated copper constantan couples should suffice, and in this case an instrument can be used which is directly calibrated in degrees Fahrenheit so that a transformation of millivolts into degrees is not necessary. This instrument is a Leeds Northrup 8692 Portable Temperature Potentiometer with a range of from 0 to 300^0F, which costs only \$270.00 and gives a convenient direct reading within the error limits stated above. It is therefore, recommended to use this instrument in conjunction with copper constantan thermocouples and an accuracy of approximately $\pm 1^0$F. Even uncalibrated thermocouples would be sufficient for most cases. If $\pm 0.1^0$F must be maintained for the air temperature, a Mueller bridge and resistance probe must be used which costs about \$1500.00.

The four thermocouples required according to paragraph 3.4.4 will be mounted in the following locations. No. 1 is mounted to the instrument holder and can be used to scan the temperature distribution across the test section. No. 2 is mounted on the carriage top in the proximity of the velocity indicators. It serves for determining the density and viscosity to be used for velocity indication. No. 3 is mounted near the test section entrance in the transition and serves to indicate any changes of the temperature of the ambient air. No. 4 will be provided with holders which permit its mounting to either the instrument holder of the carriage or to the tunnel walls at any desired location.

6.5 Mean Velocity Hot Wire Anemometer

The mean velocity hot wire anemometer will be of the constant resistance type. A different probe will be furnished for the velocity range from zero to approximately 30 mph and for the range above 30 mph. This becomes necessary since the sensitivity decreases with increase in hot wire diameter, and a platinum wire is not sufficiently sensitive for low velocities and at the same time strong enough for the pressures and drags of high velocities. The most appropriate wire sizes for each range will be determined during the performance tests.

In principle, the proposed mean velocity hot wire anemometer forms one branch of a Wheatstone bridge for measuring resistance. A constant current is initially sent through the hot wire which heats it enough to give high sensitivity. The other branch of the bridge is then balanced by adjusting the resistance of the decade box (see Fig. 15). If the wire is immersed in an air flow, then due to forced heat convection it is cooled faster than without flow. Consequently, the resistance of the wire changes and the bridge is unbalanced. By adjusting the current through the wire the bridge is brought back into balance. The current necessary to rebalance the bridge is an indication of the velocity of the air flowing past the hot wire.

The hot wires have to be calibrated before they can be used. This will be done by placing the hot wires close to the velocity indicating sensors, running the tunnel through the ranges of velocities desired and comparing the current necessary to balance the bridge with the velocity determined from the calibration curves of the velocity indicators. If a large range of temperatures of the ambient air is encountered, then a temperature correction may be applied according to the findings of Ref. 13. Under ordinary conditions, no such correction is needed, since the temperature at testing will correspond roughly to the temperature during calibration.

The circuit of the mean velocity hot wire anemometer is shown in Fig. 15. The instruments used are the same as those which have been used successfully by the Contractor, The 0-400 Milliammeter is a General Electric Type DP-2. The decade resistor is a General Radio Type 1431-N, The DC current is controlled by small potentiometers of the rotary type. The galvanometer is a Leeds Northrup Type 2340 d-c Pointer Galvanometer. Sensitivity of the galvanometer can be decreased by decreasing the adjustable shunt resistance. With this arrangement the accuracy requirements of paragraphs 3.4.6 can be met over most of the velocity ranges, provided the hot wires are recalibrated at suitable time intervals.

The power supply for the mean velocity hot wire anemometer must yield steady and ripple free operation. This requirement can be met by using a voltage regulator in series with a power rectifier as a power supply. This solution is indicated in Fig. 11. A less expensive solution is obtained by using a standard heavy duty 12 V battery and providing a simple power rectifier which is connected to the battery through the main panel switch(See Fig. 11) in such a manner that the battery is continuously charged while the hot wires are in the off position.

The latter of the two mentioned solutions is recommended.

6.6 Turbulence Hot Wire Anemometers

An instrument for measuring velocity fluctuations (turbulence) and local turbulent velocity correlations between components in different directions as described in paragraph 3.4.7 of the Specifications is made by the Hubbard Instrument Company in Iowa City, Iowa. This instrument is described in detail in the appended bulletin.

For continuous recording of the hot wire output an oscilloscope is desired. By observing the output of the hot wire it can continuously be checked whether the wire is heated sufficiently to respond to the high frequencies of the velocity fluctuations.

7. SAFETY FEATURES

The facility will be protected against fire by using fire retarding paint and by providing a fire alarm and fire extinguishing system. Temperature sensors will be installed so that bearing temperatures may be monitored. Safety screens will be installed so that personnel and flying debris will not be able to contact the fan.

7.1 Paint

The paint to be used has to be moisture repellent and at the same time fire resistive or fire retarding according to paragraph 3.2.10 (d). In order to achieve this purpose, the steel parts will have 3 coats of paint. The first coat consists of a lead base primer which prevents the steel from rusting. On top of this there will be two coats of fire retarding paint.

The plywood will be protected by 2 coats of fire retardent paint, and one coat of primer for any one of the different fire retarding paints considered.

For all external surface two coats of paint will be applied in the shop of the Contractor, as well as all three coats for the internal surfaces. The last external coat will be applied in the building which will ultimately house the facility. Those surfaces which cannot be treated after assembling the facility will be painted before, and the remainder after assembly is completed.

Two different types of fire retarding paint have been considered. The first type, made by Du Pont, and Flamort Chemical Company, develops ammonia if heated above a certain temperature, which smothers flames and thus prevents the spreading of flames. The second type, made by the Albi Manufacturing Company, puffs up under excessive heat and forms a thick insulating layer between flames and inflammable materials. Both paints are continuously tested by the Underwriters Laboratories. Since both types of paint have about the same rating, as far as inflammability is concerned, the selection between them was made on the basis of the lower cost, and the paint of the Albi Company was chosen. The paint will be applied according to the manufacturer's specifications.

7.2 Fire Alarm System

As required by paragraph 3.2.10 (a), the facility will be equipped with temperature sensitive switches which close, in case of overheating, a number of alarm circuits in addition to actuating a solenoid switch which cuts off the power supply for the drive, instrument panel, and carriage drive. In addition, the same type of switches are provided for avoiding damage due to bearing failure of either motor or fan, or damage due to failure of the water pump or other components of the water cooling system of the Dynaspede drive. A total of seven switches will be provided, of which two are connected to the motor bearings, two to the fan bearings and one to the cooling system of the Dynaspede. The other two switches are controlled by elements installed at the downstream end of the test section and the function of return duct and first large corner. In this manner, fires at all locations in the duct work will actuate alarms at reasonably short times after being energized.

The block diagram of the fire alarm circuits is shown in Fig. 16. According to this diagram, only the case of fire in the ductwork will actuate the audible alarm, while overheated bearings only cut off the main power and produce a visible signal given by a pilot light. Signs on all pilot lights show immediately where the power failure was caused, so that adequate measures can be taken.

The switches will be actuated by a liquid contained in a bulb and connected to the switch through a long capillary tube. Under the influence of temperature rises the liquid expands and the ensuing pressure closes the switch. The switch is a L-49 temperature control made by General Controls Company. It can be set to respond to temperatures between 120°F and 210°F, thus meeting the requirements of paragraph 3.2.10 (a) of the Specifications.

For detecting over-temperature in the duct system, the bulb type temperature sensors will be mounted flush with the ceiling, and separated from the duct interior by a thin metal sheet, so that the flow in the duct is not disturbed by the sensor while the sensitivity of the sensor is maintained.

For detecting over-temperature in a bearing or in the cooling system of the Dynaspede, the sensor will be attached at some convenient location to the outer shell of the housing for the bearing or to the housing of the Dynaspede transmission. The temperature sensitive switches will be set at values obtained either from bearing manufacturers or from the Eaton Company, manufacturer of the Dynaspede, or will be determined during performance tests.

The audible alarm system shall consist of a bell which is started by closing the fire sensitive switches. No additional switch, which would permit the audible alarm to be cut off, will be provided; since only in this way can it be made sure that the alarm is operating, and has not been accidently cut off. The audible alarm can therefore only be switched off by resetting the temperature sensitive switch.

The audible alarm will be an Edwards "Adaptahorn No. 370" with a loudness factor of 94 decibels at 10 ft. distance.

It is considered unlikely that the main electrical power can be cut off by a fire which does not destroy the alarm at the same time, or that a fire can be started while the main electric power is cut off. Therefore, no emergency power circuit will be provided, which would switch the alarm to a battery powered circuit in case of power failure.

7.3 Fire Extinguishing System

According to paragraph 3.2.10 (c) of the Specifications, a fire extinguishing system should be provided which permits the whole duct system of the facility to be flooded with carbon dioxide upon command by the operator.

These requirements, in addition to those on the paint and the alarm system, have obviously been included under the assumption that extreme fire hazard exists during the operation of the tunnel. Generally, this is not the case for the facility itself, unless instruments are tested which contain fire hazardous parts. Therefore, the alarm system indicated in the previous paragraph in conjunction with two portable fire extinguishers should be sufficient to protect the facility. The fire extinguishers would be located in such places where the largest fire hazard exists. Therefore, one unit will be placed near the motor and main power switch, while the other unit will be mounted sufficiently close to the panel area.

If, on the other hand, full protection of the unit is desired, then it is recommended not only to flood the tunnel inside with carbon dioxide, but also the interior of the building, since it seems that the danger of fire is just as great outside of the tunnel as it is inside. The cost of a system which protects the interior of the building has been determined to be $7,183.00 in comparison with the cost of less than $300.00 for the solution suggested above. The prices are based on a cost estimate given by the Walter Kidde Company.

A system which is designed to meet the requirements of paragraph 3.2.10 (c) of the Specifications, also designed and installed by the Walter Kidde Company, would cost a total of $2,664.00 . On this figure the cost estimate of the Contractor has been based. If this system is desired, then it is suggested to provide also hand fire extinguishing equipment for protection of panels and electrical conduits. In this case, the portable units necessary could be furnished for $ 120.00 , since they would be bought in addition to an elaborate extinguishing system.

7.4 Safety Screens

The safety screen required by paragraph 3.2.10 (b) of the Specifications will consist of a coarse mesh galvanized steel screen. It will be placed against the turning vanes in the last corner before the drive section.

REFERENCES

1. Chiccine, Bruce, G., and Abdalla, Kaleel L., "Experimental investigation of expanded duct sections and screens for reducing flow distortions at subsonic flows", NASA Memorandum 1-9-59E, May 1959.

2. Cermak, J. E., "Wind tunnel for the study of turbulence in the atmospheric surface layer", Final Report on Proj. AFCRC-TR-58-287 Colorado State University, Civil Engineering Report No. CER58JEC42, Fort Collins, Colo., November 1958.

3. Pope, A., "Wind Tunnel Testing", John Wiley and Sons, Inc., New York, 1947.

4. Dryden, H. L., and Abbott, I. H., "The design of low turbulence wind tunnels", NACA Report 940, 1950.

5. Silberman, E., "Importance of secondary flow in guide vane bends", Tech. Paper No. 14, Series B, St. Anthony Falls Hydraulic Laboratory, University of Minnesota, 1953.

6. Schubauer, G. B., Spangenberg, W. G., and Klebanoff, "Aerodynamic characteristics of damping screens", NACA TN 2001, 1950.

7. Kovasnay, L. S. G., "Hot wire investigations of the wake behind cylinders at low Reynolds numbers", Proc. Roy. Soc. London Ser. A., Vol. 198, p. 174-90. 1949.

8. Roshko, A., "On the development of turbulent wakes from vortex streets", NACA Report 1191, 1954, 24 pp.

9. Rouse, H., and Hasson, M. M., "Cavitation free inlets and contractions", Mechanical Engineering 71, 213-16, 1949. (See also Ref. 11, p. 417.)

10. Rouse, H., "Elementary Mechanics of Fluids", J. Wiley, New York, 1946.

11. Rouse, H., (Editor) "Engineering Hydraulics", Proceedings of the Fourth Hydraulics Conference, Iowa Institute for Hydraulic Research. John Wiley, New York, 1950.

12. Prandtl, L., and Tietjens, O., "Applied Hydro- and Aerodynamics", Dover, New York. 1934.

13. Earle, Edward N., "Mean velocity profiles for flow over a plane, smooth, heated boundary", unpublished M.S. Thesis, Colorado State University, Department of Civil Engineering, Ft. Collins, May 1960.

APPENDIX A.

APPENDIX A.

LIST OF FIGURES

Figure No.

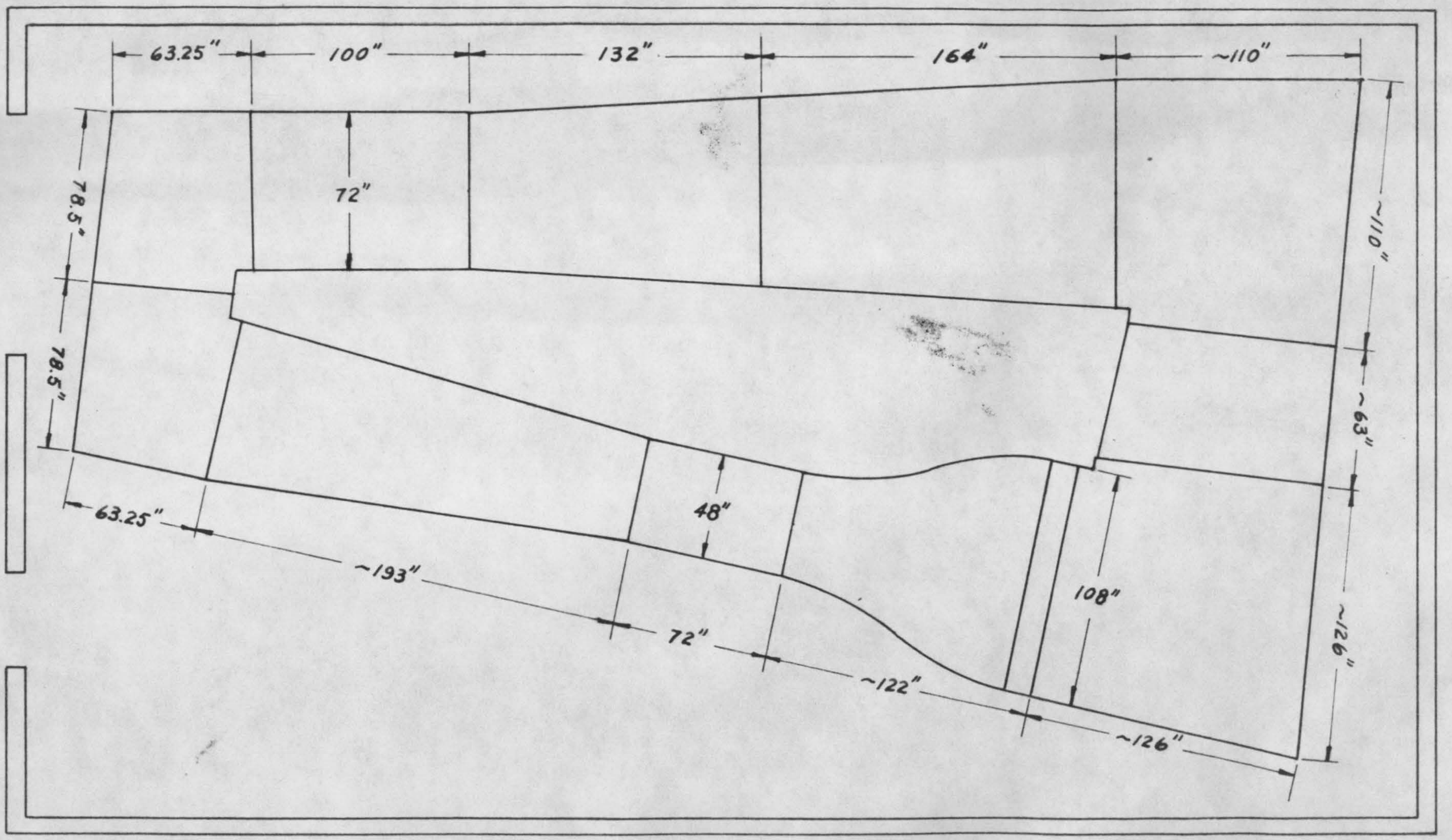


FIG. 1. ALTERNATE LAYOUT 1a

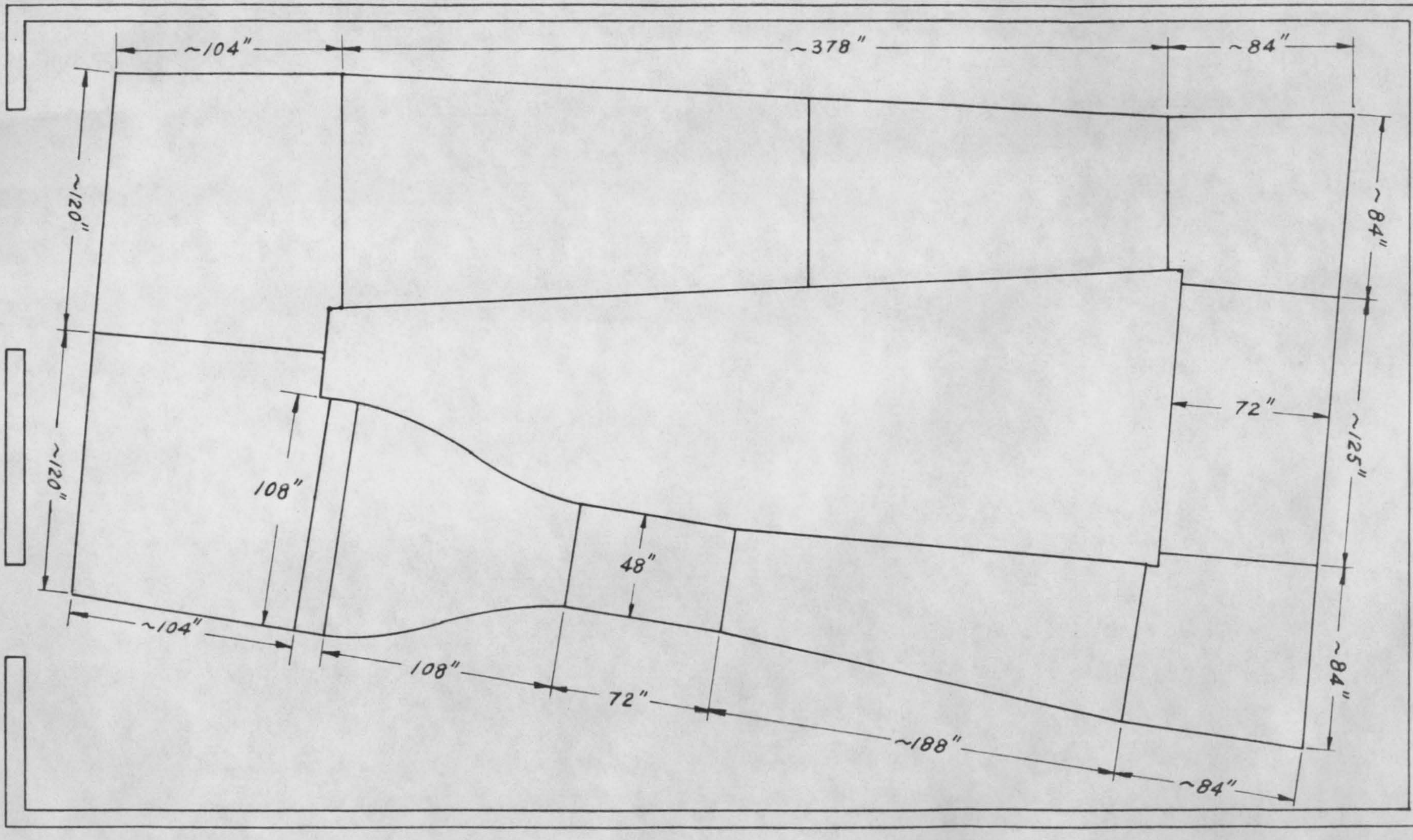


FIG. 2. ALTERNATE LAYOUT 1b

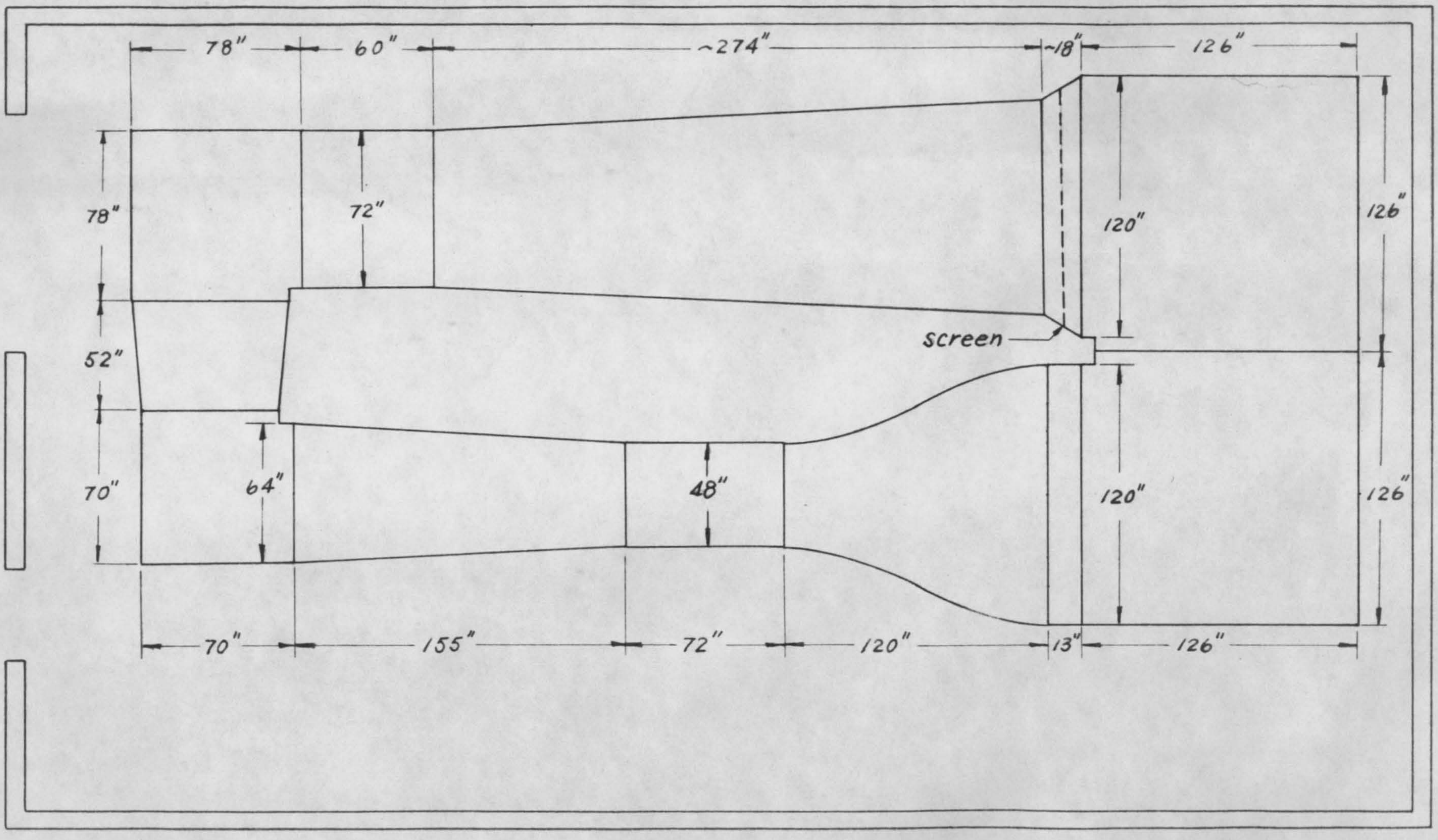


FIG. 3. ALTERNATE LAYOUT 1c : CONTRACTION RATIO 1:6.25

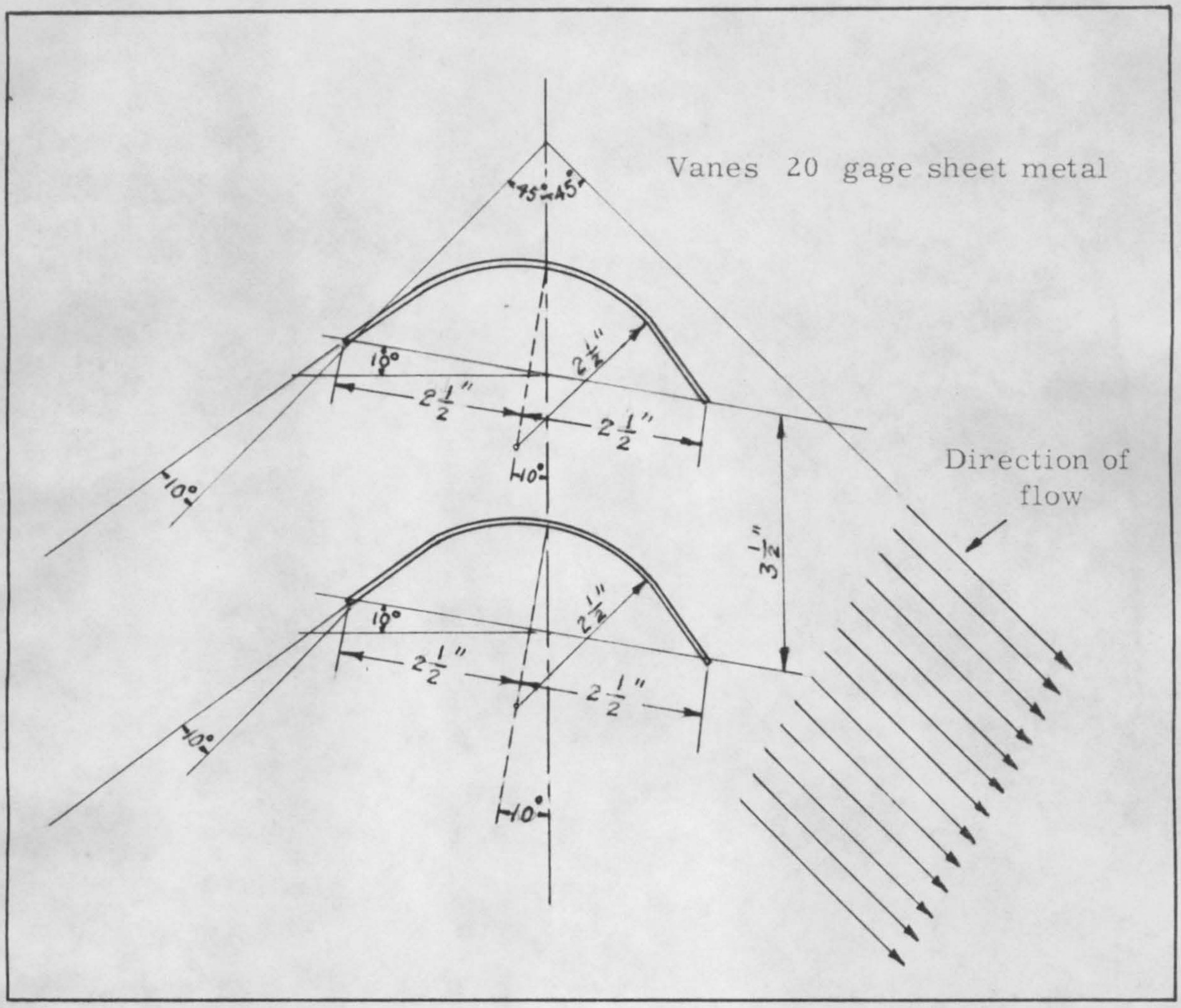
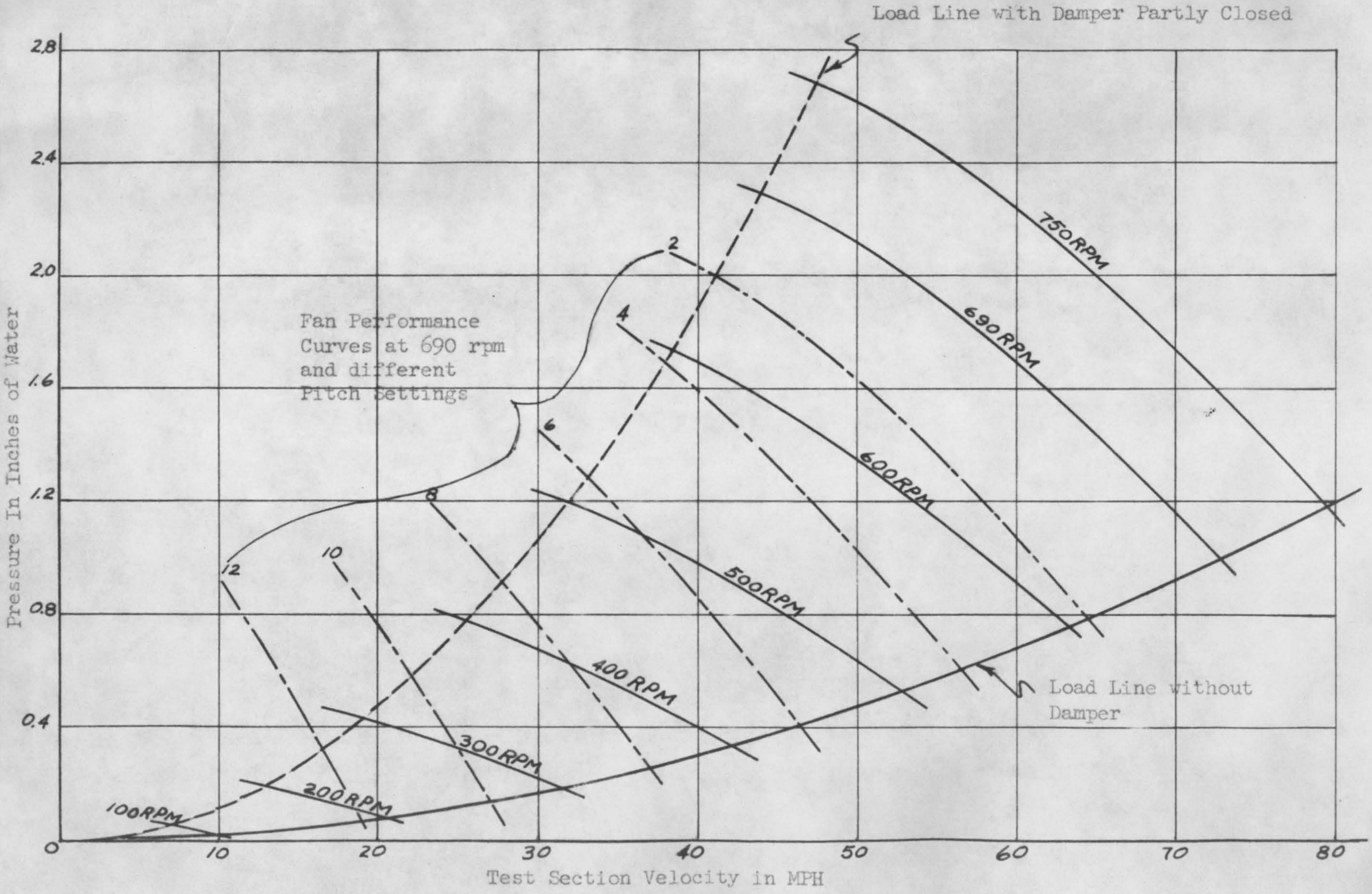


FIG. 4. ARRANGEMENT OF VANES

FIG. 5. TYPICAL CHARACTERISTICS OF A 72" DIAMETER JOY FAN



Based on Joy Co.'s Drawing No. C-1804

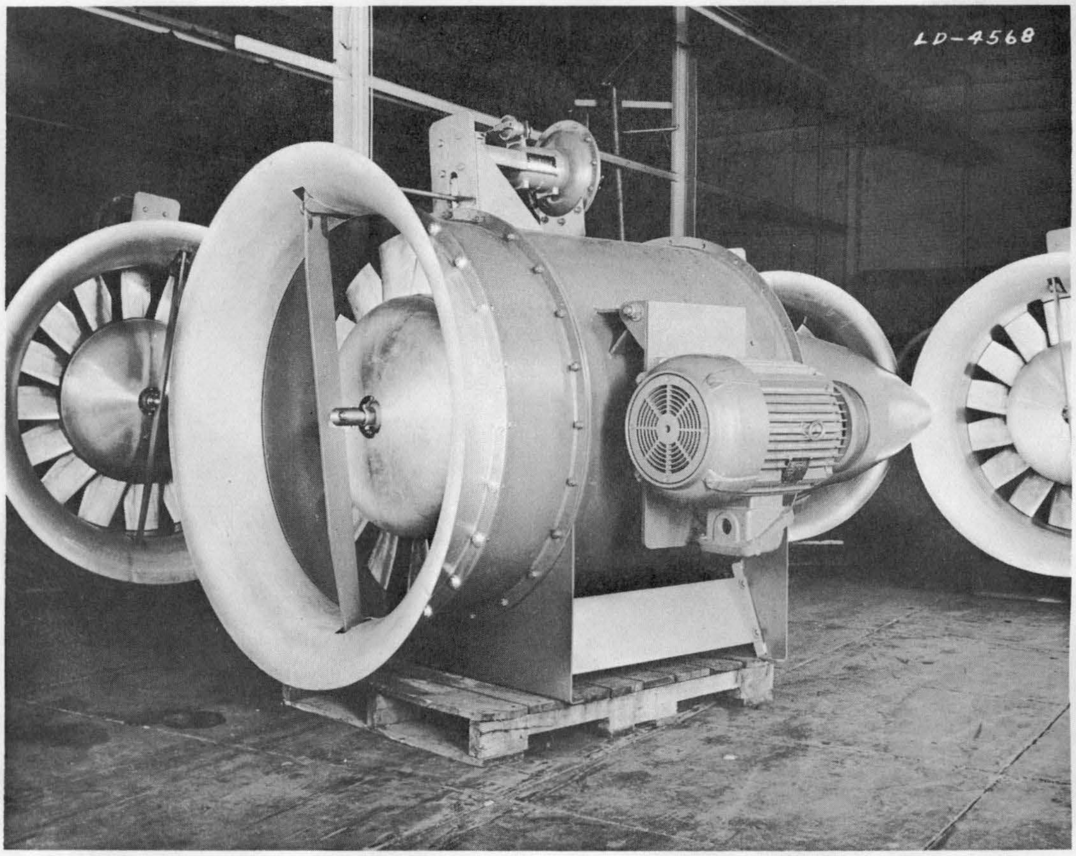


Fig. 6 Joy Series 1000 Variable Pitch Fan Side View (after body can be seen in rear).

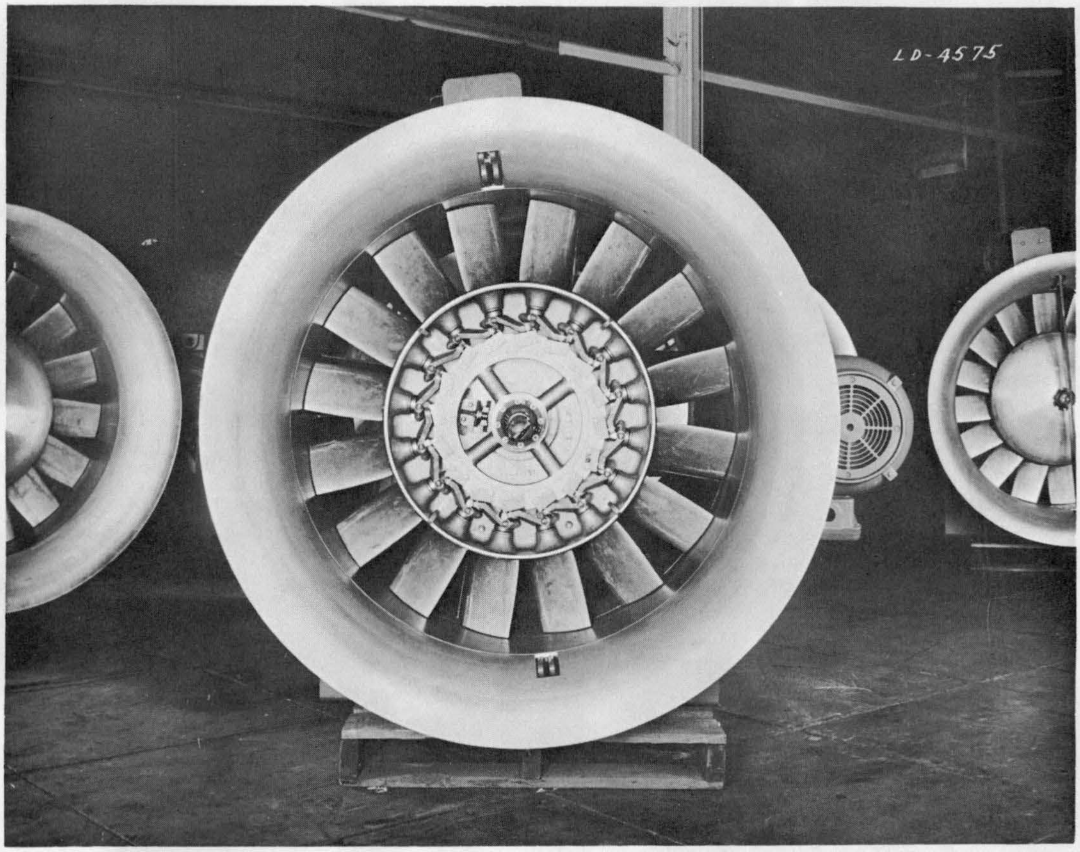


Fig. 7 Joy Series 1000 Variable Pitch Fan Front View with hub removed.

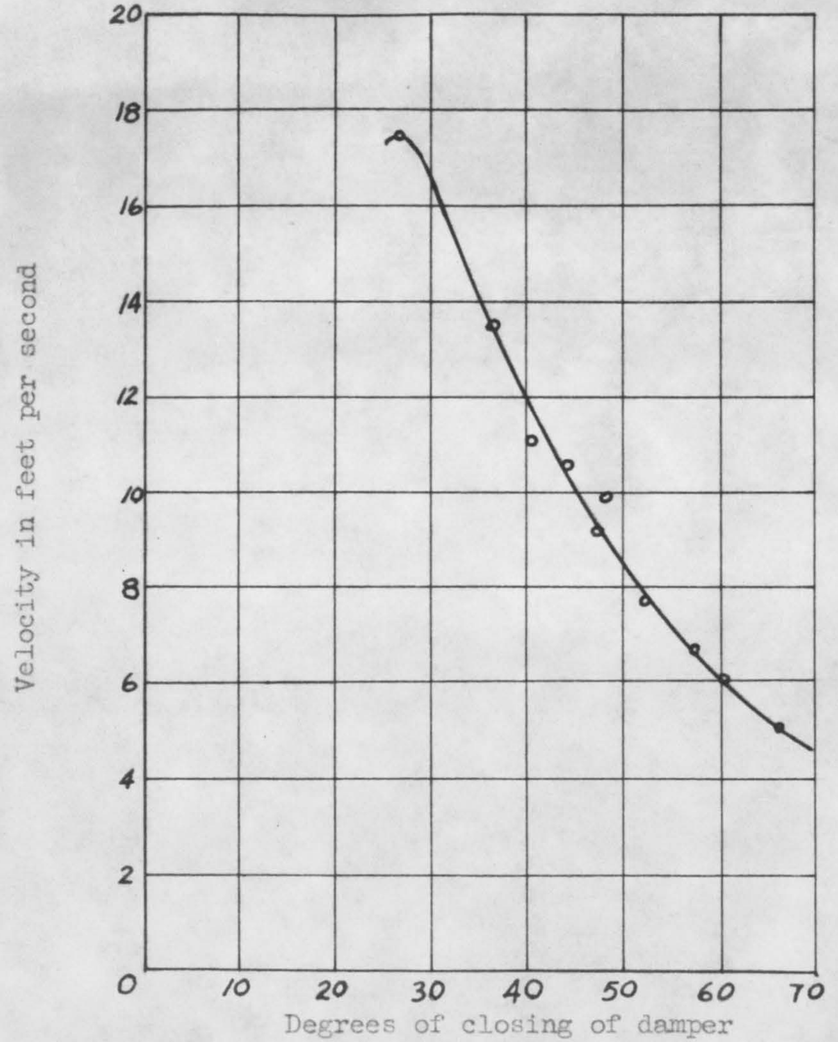


FIG. 8. TYPICAL DAMPER CHARACTERISTICS

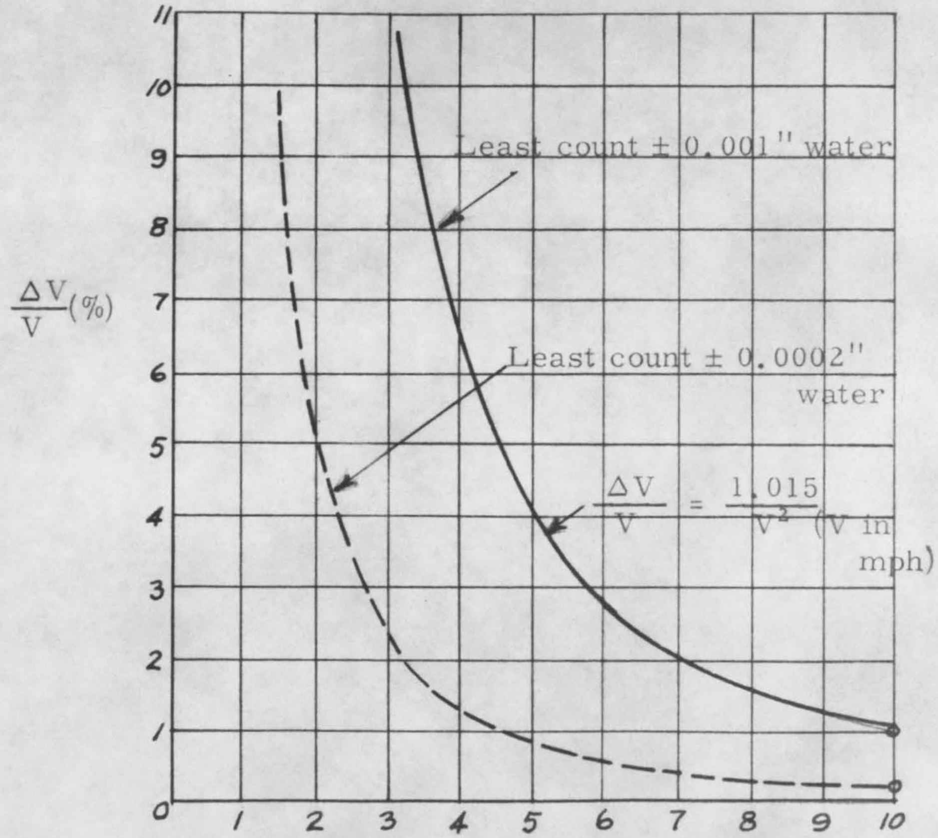


FIG. 9. POSSIBLE READINGS OF VELOCITY WITHIN GIVEN ERROR LIMITS

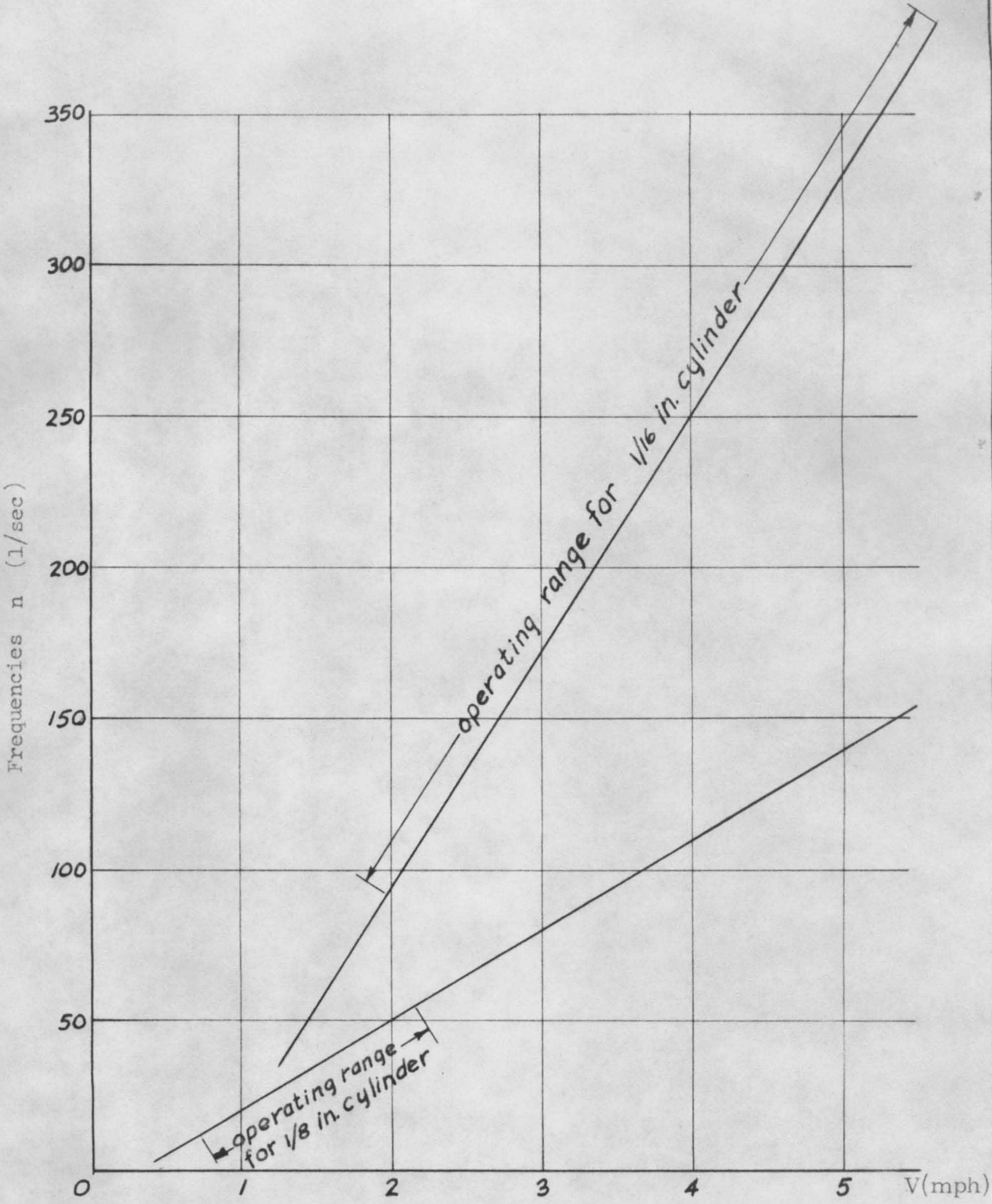
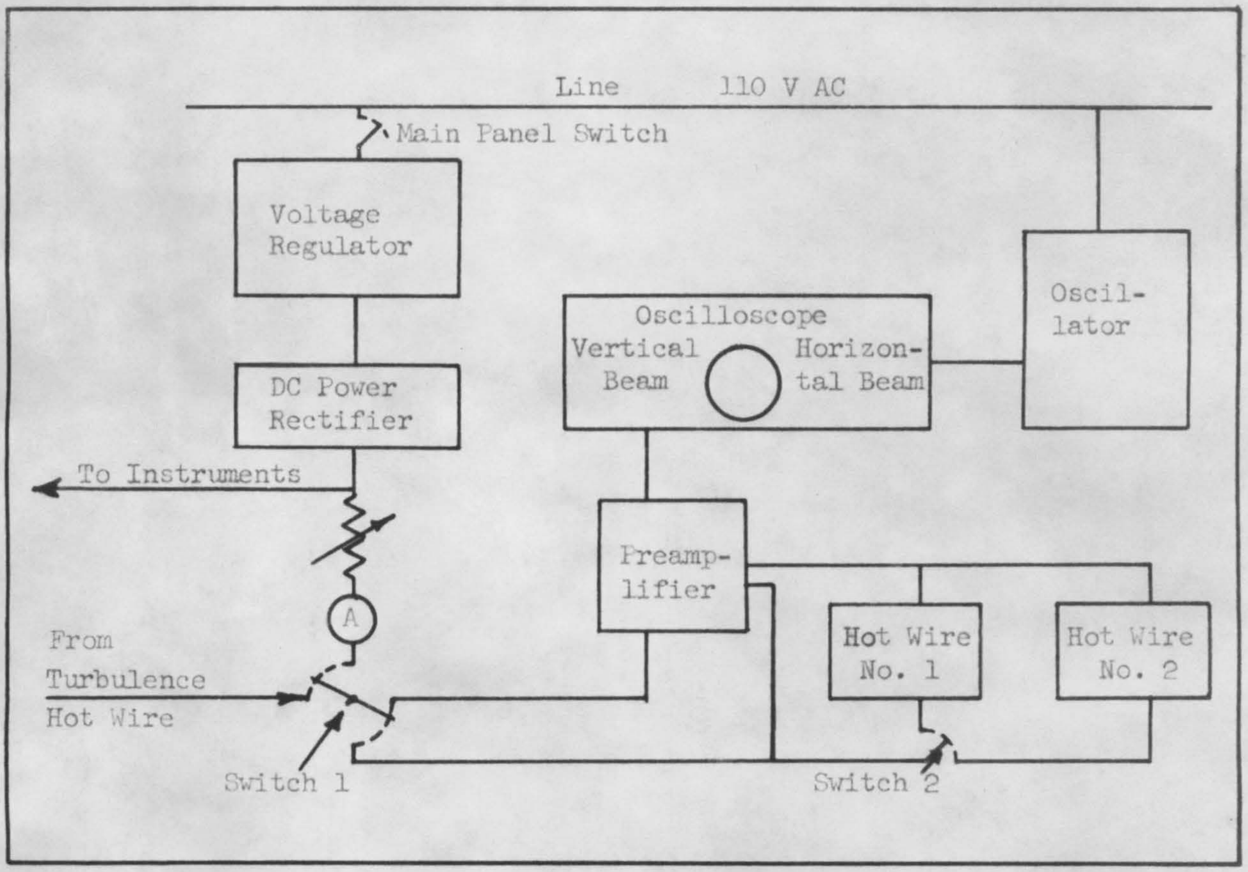


FIG. 10. VELOCITY-FREQUENCY RELATIONS FOR LOW VELOCITY INDICATIONS



<u>Main panel switch</u>: switches on and off voltage regulator, DC power and oscilloscope

<u>Switch 1</u>: switches from turbulence hot wire circuit to low velocity indicator circuit. Putting the switch on low velocity indicator switches on also oscillator and preamplifier (2 position switch)

<u>Switch 2</u>: switches from hot wire no. 1 to hot wire no. 2 (2 position switch)

FIG. 11. BLOCK DIAGRAM OF LOW VELOCITY INDICATOR

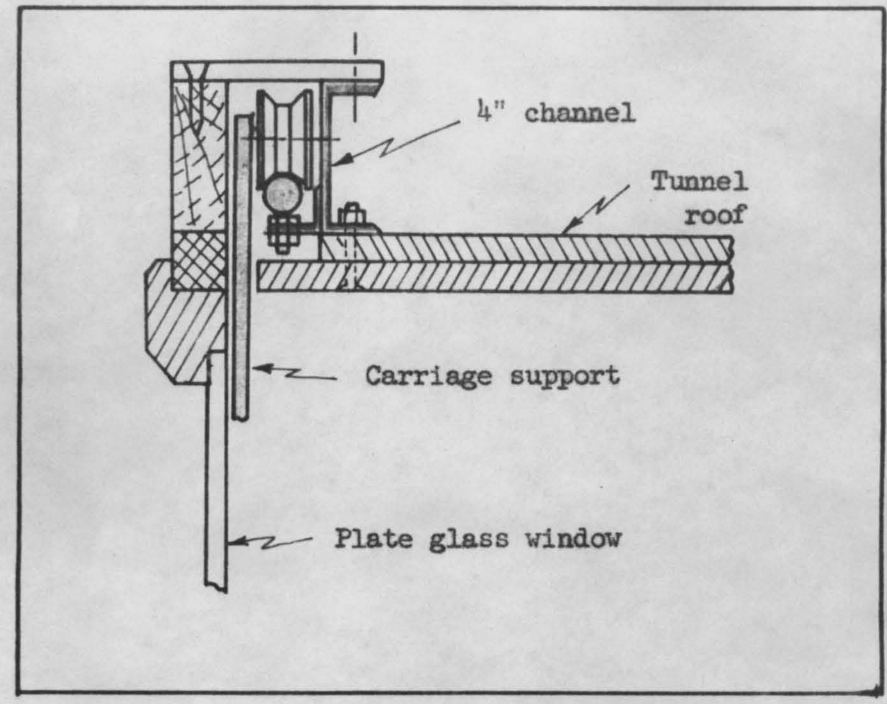


FIG. 12. CARRIAGE SUPPORT WHICH INTERFERES NEITHER WITH AIR FLOW NOR WITH DOORS

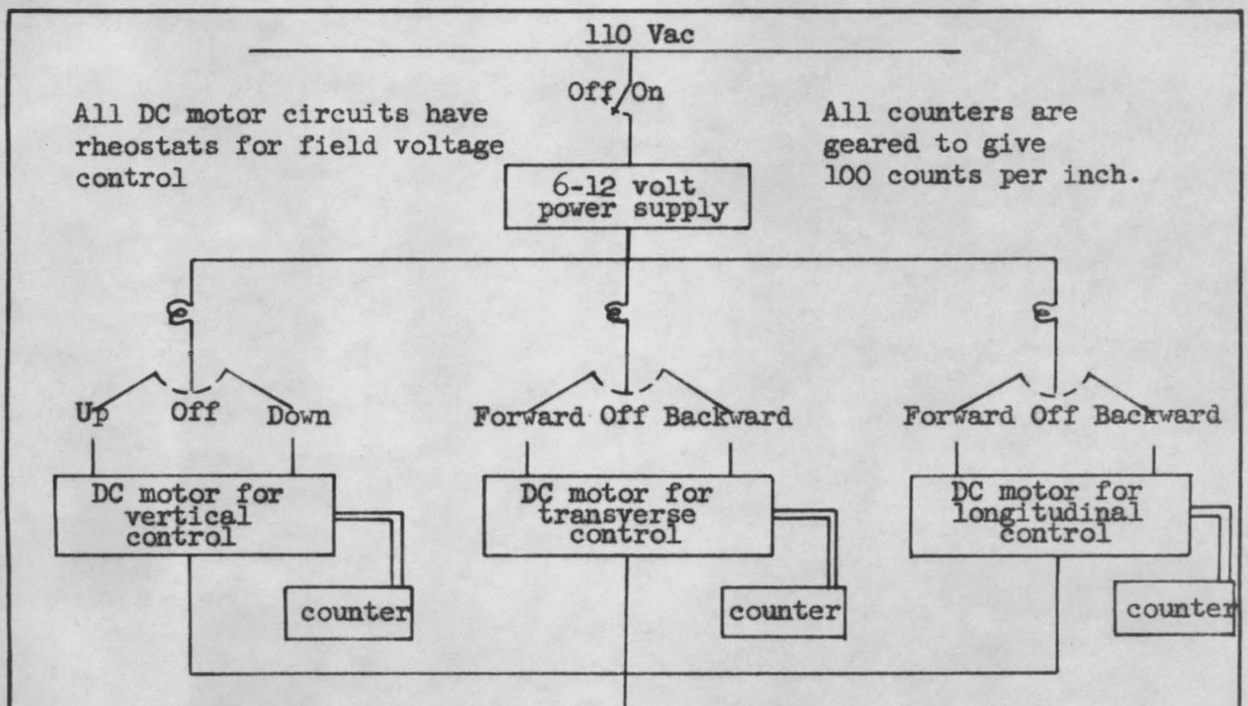
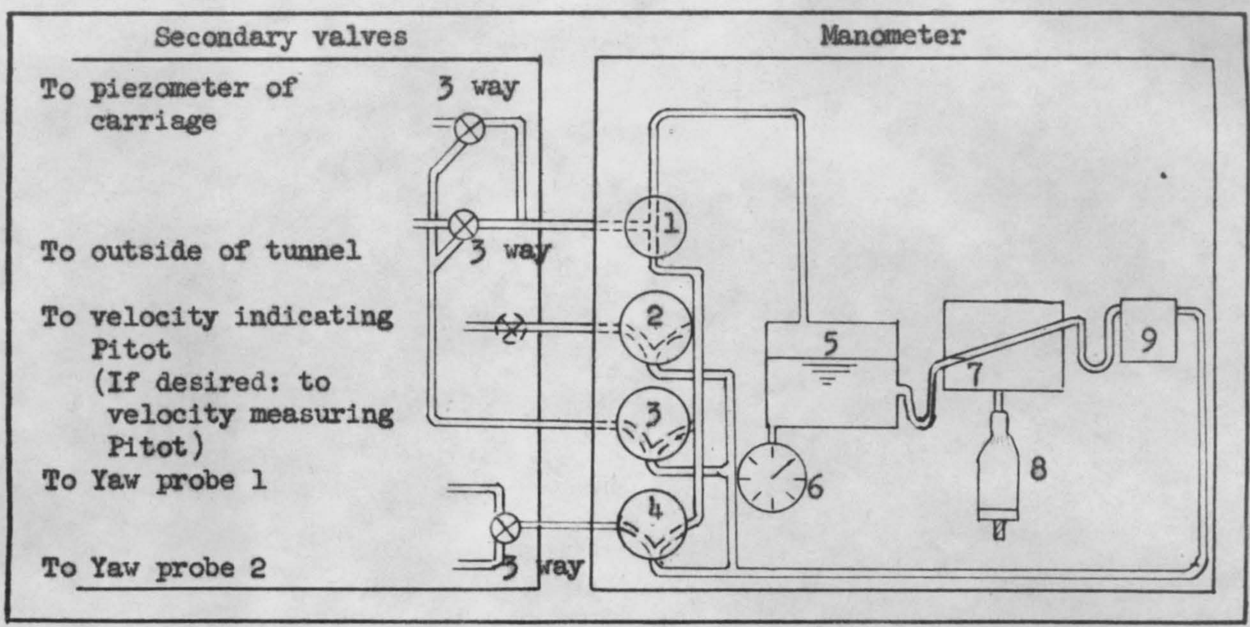


FIG. 13. BLOCK DIAGRAM FOR CARRIAGE CONTROL



1. Static pressure valve
2. Velocity indicating + measuring valve
3. Relative static pressure valve
4. Yaw probes valve
5. Well
6. Thermometer
7. Equilibrium indicator
8. Micrometer
9. Trap

FIG. 14. BLOCK DIAGRAM OF MANOMETER AND MANOMETER CONNECTIONS

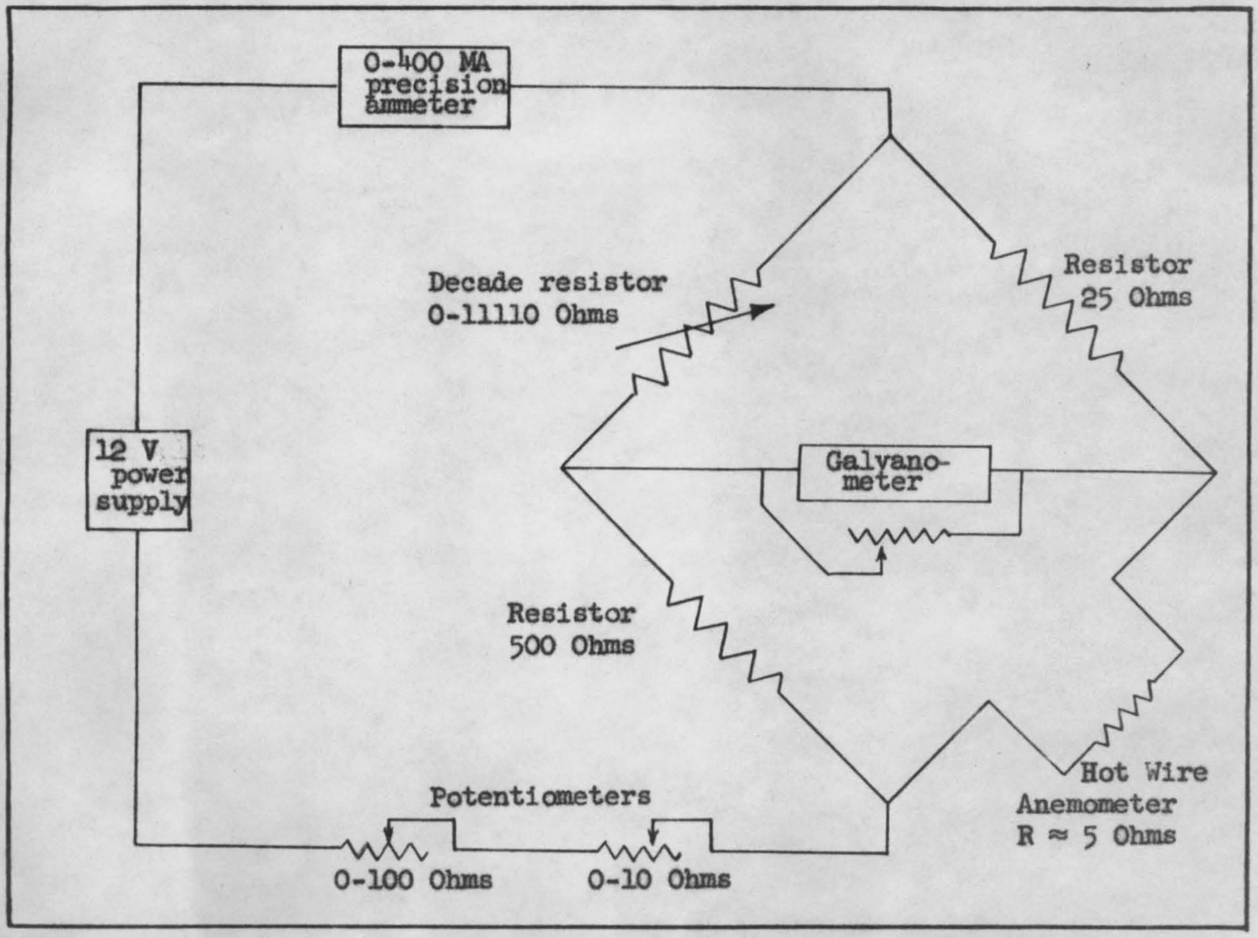


FIG. 15. CONSTANT TEMPERATURE HOT WIRE ANEMOMETER FOR MEAN VELOCITY

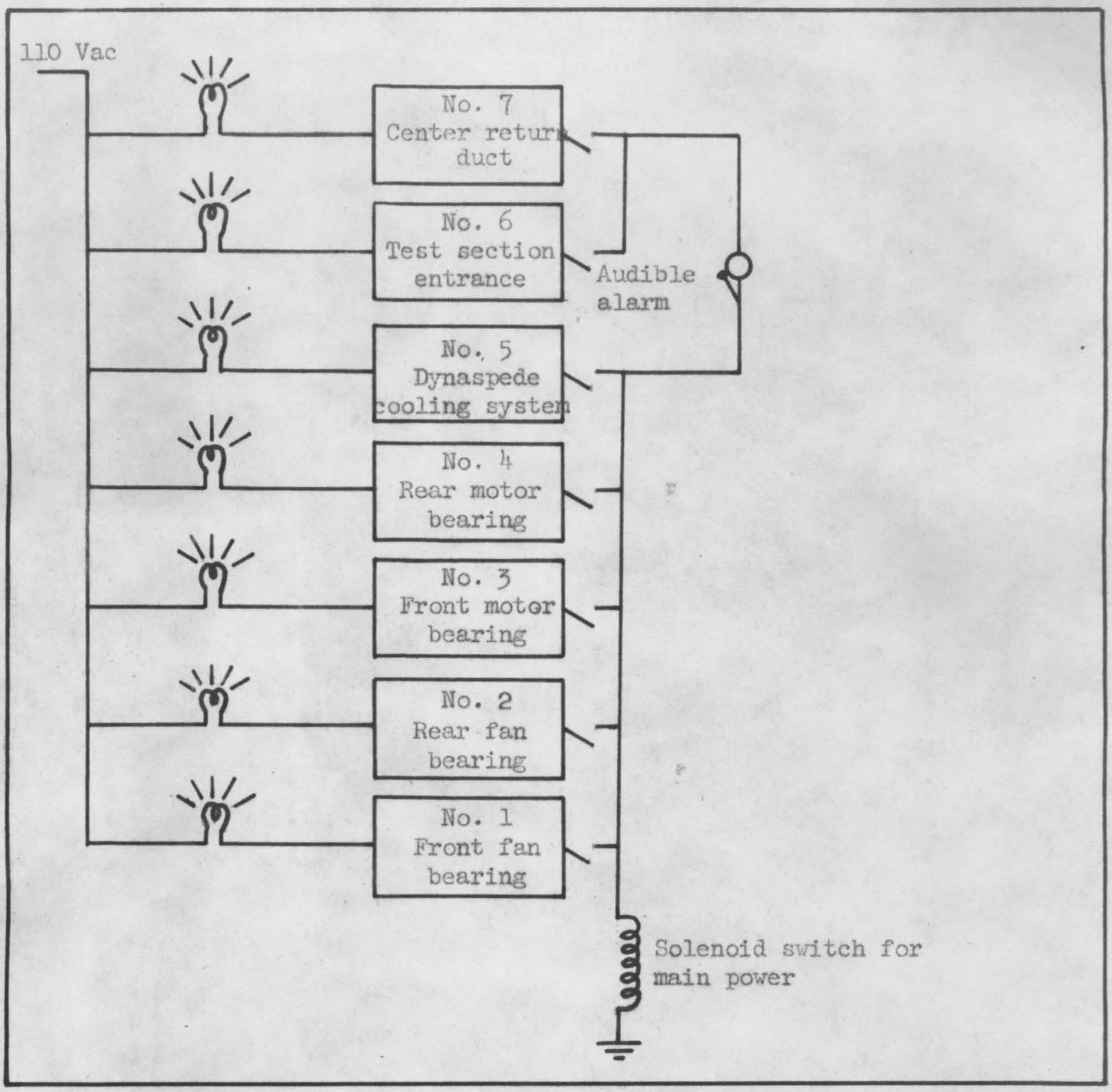


FIG. 16. BLOCK DIAGRAM OF SAFETY PANEL