ASHRAE

TRANSACTIONS

PROCEEDINGS OF THE ASHRAE ANNUAL MEETING
JUNE 23 - 27, 1974 • MONTREAL, QUEBEC, CANADA
INCLUDING TECHNICAL PAPERS AND DISCUSSIONS,
SYMPOSIUM PAPERS AND COMMENTS AND
SOCIETY BUSINESS

1974

VOLUME 80 • PART 2

AMERICAN SOCIETY OF HEATING, REFRIGERATING
AND AIR-CONDITIONING ENGINEERS, INC.
345 EAST 47th STREET, NEW YORK, N.Y. 10017

No. 2307

DUCT TURNING VANES IN 90° ELBOWS

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The design of duct systems requires the evaluation of friction losses in straight duct, and the determination of dynamic losses characteristic of elements other than straight duct. Friction losses for straight circular ducts are given in ASHRAE charts, while their rectangular equivalents are given in the tables. Losses for other elements are indicated in Ch 25 of the 1972 ASHRAE Handbook of Fundamentals. These latter are based on experimental work done by investigators over the past forty years or more.

The element with which this work is concerned is the bend, or elbow. It has long been felt that the best combination for economy in manufacture and efficiency in performance is the elbow having a centre line radius equal to 1½ times the duct width in the plane of the bend. When space does not permit this ratio, the radius is reduced and, conversely, dynamic loss is increased. To improve elbow performance, one or more thin partitions called "splitters" can be installed in the elbow, dividing it into a family of elbows, each part of which has a good R/W (centre line radius to width) ratio.

Frequently elbows must be made without a radius on the inside. If the radius is also eliminated from the outside of the bend, it is easier to construct. To reduce the loss in such an elbow many vane types have been proposed. Some of these have been tested and data published which indicate their performance (1).

OBJECTIVES

The objectives of the investigation described in this paper are:

1. To develop the geometry of one or several types of duct turning vane which will have a high level of performance.
2. To develop data to enable the designer to evaluate, with reasonable accuracy, losses in turn-vane elbows under system conditions.
3. To compare the performance of such vanes with that of several proprietary vanes currently available on the market.

CURRENT PRACTICE

Fig. 1 shows the general arrangement of vanes in a turning vane elbow. The vanes are assembled by using an embossed strip of sheet metal or similar device at the ends of the vanes, fixing them in position relative to one another, and securing them in the elbow on the diagonal line. The effect of turning vanes is to convert the elbow from a single path to multiple paths for air flow, controlling the R/W ratio of each path by selection of vane radius and spacing. The path next to the throat piece (inside piece with respect to the bend) is an exception, since there is no inner radius.

Fig. 2 illustrates the vane and throat piece relationship. The distance, y, is dependent on vane radius and spacing. If the air flow is orderly up to and through the vane assembly, the inside path with no throat radius will have a pressure loss of

$$\Delta p = C_{mitre} \cdot h_v$$

and the remaining paths will have losses of

$$\Delta p = C_{R/\,selected} \cdot h_v$$

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where Δp is the dynamic loss resulting from bending the air stream through 90 deg, C is the loss coefficient ratio, $\frac{\text{pressure loss}}{\text{velocity pressure}}$, and h_v is the velocity pressure.

Consider an elbow fitted with vanes of 4.5 in. radius and 3.25 in. spacing. For these vanes the curved paths will have an R/W ratio of approximately 1.5, and a loss coefficient of 0.19 (2). The inside path, with a sharp inside corner, will have a loss coefficient of 1.50 (2). Suppose the inside path represent 0.2 of the elbow width. By proportion the elbow loss coefficient will be 0.45. However, still assuming that flow through the elbow is orderly, resistance through the path at the throat will reduce the velocity until the resistance through the remaining paths is balanced; the loss coefficient will then be considerably nearer 0.19 than 0.45. This has been borne out by test data and is fortunate, since y can vary considerably.

Consider the air entering a passage between two double-thickness vanes. The walls of the vanes are so located that the sum of the areas of passage at the midpoint of the paths between the vanes is approximately equal to the cross-sectional area of the duct. This has been proposed to eliminate the area expansion from A, the duct cross-section, to 1.4 x A, the elbow diagonal cross-section and the downstream contraction to A again. However, as air approaches the bend in what we like to call "an orderly manner," the air occupying the duct fills the area in the elbow by "expansion." What actually occurs is a change in mean duct velocity: if mean duct velocity is U, then the mean velocity at the diagonal of the elbow will be approximately 0.7U. If we ignore vane thickness and the area outside the vane at the heel of the elbow, air entering the passages between single-thickness vanes will not change velocity within the vane assembly. Entering the passages between double-thickness vanes, and ignoring vane tip thickness, the velocity will change from 0.7 U to U at the center of the passage and back to 0.7 U at the exit from the passage as shown in Fig. 3. The passage from entrance to center, considered as a gradual contraction, is attended by a minor static loss, since this is an efficient and stable process. But the passage from center to exit of the passage, although a gradual expansion, is attended by a considerably higher loss.

Fig. 3 illustrates the geometrical aspects of the passage entering or leaving double-thickness vanes. The trapezoid in the figure indicates a gradual expansion or contraction which, apart from curvature, illustrates the path from center to outside of two vanes. Since the inside radius of each path is just half the outside radius it is probable that the loss will be greater than for a symmetrical passage. Applying the data given in the ASHRAE Handbook of Fundamentals (2) to two double-thickness vanes in current use, the additional loss leaving vanes of 2.0 in. and 4.5 in. radii will be 0.038 and 0.033, respectively, of the velocity head.

From the forgoing it would seem that vane radius and spacing should be kept small in order to reduce the inside corner effect. However, the reduction in loss for large radius double-thickness vanes may offset the increased corner effect of the large Y dimension. Table 1 lists the vane geometries tested in the 24 in. x 24 in. single elbow. The 6 in. radius vane was included because considerable work was done testing vanes with 6 in. radii on the same apparatus used for this project (3). The bend used had a 6 in. inside radius, permitting comparison with the square throat application in this project. Also drawn from that work is the extension of the vane trailing edge by a length equal to 0.25 of the vane chord. There is no readily available means for fastening vanes with less than a 2 in. radius, so proprietary vane No. 1 was used to indicate the performance of small radius vanes.

THE TESTING SYSTEMS

Equipment used throughout the project consisted of a No. 805BL Canadian Blower & Forge centrifugal fan with variable inlet vanes, a 6 ft x 6 ft settling chamber with six 16 x 16 mesh screens and a honeycomb straightener, followed by a short streamlined reducer or nozzle 6 ft x 6 ft to 2 ft x 2 ft. Piezometer rings at the large and small ends of the contraction were provided for the purpose of roughly measuring flow rate in order to set the fan inlet vanes for the desired mean duct velocity. The fan operated at 900 rpm throughout the project.

Four duct arrangements were used. The first, shown in Fig. 4, is essentially the same one used by post-graduate students prior to this investigation. It included a number of sections of 2 ft x 2 ft duct, 4 ft long, constructed of $\frac{3}{4}$ in. plywood, and fitted together with smooth, airtight joints. One of the duct sections contained a sliding pitot rake section with 11 probes, fixed at 2 in. centers in an aerofoil shaped hollow holder and starting 2 in. from the side walls. The holder was round nosed with a thin trailing edge. The rake assembly was movable in a vertical plane to within 1 in. of duct floor and ceiling, at the same time maintaining a smooth side wall. A number of brass static pressure taps with 1/8 in. orifices were mounted in certain sections, designed to receive gauge tubes on the outside of the duct. These were subsequently changed to 1/16 in. diameter holes drilled through the duct wall to improve readings.

The elbow used for the work on 24 in. x 24 in. single 90 deg turns was constructed of galvanized iron having a center-line length of 3 ft. Adapters to fit the wood duct at each side of the elbow were 6 in. long. The throat and heel piece were removable for insertion of the vane assemblies. Joints for these pieces were reasonably smooth and taped for air tightness. Static pressure taps, provided at the entrance and exit of the elbow to obtain static pressure profiles, were made by soldering a stub of $\frac{1}{4}$ in. OD tubing to the exterior of the duct 2 in. upstream from the joint, drilling a 1/16 in. hole in the duct wall through the soldered tube, and sanding it smooth on the inner duct wall.

The second duct arrangement, similar to the first, is shown in Fig.5. This arrangement was necessary to obtain reasonably uniform readings on both sides of the duct at the reading stations. It complies with ASME test standards (4). The pressure readings obtained in the first system proved the need for adopting the second. The elbow is the same as used throughout the program for the 24 in. x 24 in. duct. Two lengths of wood duct downstream from the elbow were replaced by a galvanized duct made of #20 U.S. gauge material for the purpose of providing four lines of pressure taps at 2 ft intervals from the exit of the elbow. The stretchout shown in dotted lines indicates the arrangement for obtaining the friction loss of the duct alone in the test section.

The third duct arrangement, shown in Fig. 6, was adopted to investigate losses in elbows in combination. The drawing indicates obstructions in the space which made it necessary to offset the duct. Straight duct friction losses obtained in the second arrangement (Fig. 5) were used for computing losses due to the elbows. The duct length represented by the second elbow was obtained by moving the upstream reading station from 4.5 to 2.6 diameters from the elbow entrance and removing the section of galvanized duct immediately following the elbow. By this means lengths, materials, and losses pertaining to the straight duct were unchanged. Close-coupled elbows were tested in offset, in U turn, and in change of plane. Space limitations prevented testing elbows two diameters apart, except in offset. One curved throat elbow with R/W = 1.5 was tested for comparison with losses from a vaned elbow. Also, curved elbows were tested in close and separated offset.

The fourth duct arrangement is shown in Fig. 7. All duct in this system beyond the transition from 24 in. x 24 in. to 48 in. x 12 in. is of 20 gauge galvanized iron. Stretchout for obtaining straight duct friction losses is shown in dotted lines.

Room temperature and barometric pressure were recorded for each run made with the first and second system.

Work Performed With the First System

Much time was spent ascertaining that the velocity profiles in the vertical and horizontal planes were symmetrical. The rake section was located 4 diameters from the 6 ft x 6 ft to 2 ft x 2 ft nozzle. Concern regarding velocities close to the duct walls led to the insertion of a 12th pitot probe in the rake assembly; probes were then located 1 in. from the side walls, spaced at 2 in. intervals. The traverses taken with the rake section in this position showed excellent symmetry.

Complete traverses produced 144 readings of velocity pressure. Converting to velocities in fpm and averaging the values obtained was done by program on a Wang calculator Due to the large number of velocity pressure readings, a multitube manometer was connected to the 12 pitot probes. For each velocity and setting of the probe assembly the manometer was photographed. This required a spotlight on the manometer tubes, and heat from the light, located to one side, raised the temperature of the fluid in the tubes, more on the side next to the light than on the distant side, producing uneven and untrue readings. Added to this was the impossibility of reading low pressures accurately.

Use of the multitube manometer was abandoned in favour of a Lambrecht manometer with 200 mm scale adjustable to 2, 5, 10, and 25:1 ratios. One millimeter on the scale at 25:1 represents 0.0016 in. approximately. A manifold of turn cocks was arranged to connect the probes, in turn, to the manometer so that obtaining the readings required only the turning of respective cocks for both velocity pressure and static pressure readings. It was an advantage to use a single instrument, zeroing the scale frequently. Errors due to variation between instruments were found to be a problem when two or more instruments were used.

Calibration of the 6 ft x 6 ft to 2 ft x 2 ft nozzle was done by connecting an Airflow Developments manometer reading to 0.001 in. across the piezometer rings.

Four settings of the fan inlet vanes and a complete traverse of the 2 ft x 2 ft duct plotted against corresponding pressure drops across the nozzle produced the curve in Fig. 8. Subsequently, reading pressures from the graph gave settings for manometer readings to obtain mean duct velocities in thousands of fpm. Traverses taken in the early part of the work established pressure drops across the nozzle which in turn provided mean duct velocities that did not vary more than 3% from the desired velocity, and generally less than 1½% from the setting. Also, taking 12 readings at 1, 3, and 11 in. from the floor of the duct, and weighting them by the number of lines in full traverse that they represented, produced values for mean duct velocities within 1.5% of those found by complete traverse. Subsequent readings taken on these three lines were used to establish the mean duct velocity for each vane geometry at each velocity in the programs for the systems in Fig. 4 and 5.

During the calibration procedure, single-thickness vanes of 2 in. radius, spaced at 1 in. were in the test elbow. It seemed possible that a change in system resistance might change the effectiveness of the nozzle calibration. A block-off at the end of the duct (on one side only) of 1.5 and 3 in. produced increases of approximately 25 and 50% respectively, of the static pressure at the reading stations. Traverses taken showed that the mean duct velocity did not vary more than 1.5%.

To learn what effect vane spacing and vane radius had on pressure loss, the variations listed in Table 1 were selected. A means of fastening vanes in position is essential. A product on the market for this purpose provided such a means for 1½ in. spacing of 2 in. radius vanes, and 3¼ in. spacing of 4½ in. radius vanes. A simple punch and die fitted to a shop bench punch was used to produce fasteners for vanes of other spacings and radii.

Proprietary vanes have their own devices for fastening. Some are very thin and produce low resistance to air flow, while others, ¼ and 5/16 in. high, seem likely to offer considerable resistance. Losses recorded for all vanes include those caused by the turn and the fastening device.

Summary of Work Performed With The First System

It was intended to obtain direct readings of losses between the elbow entrance and exit, but pressure readings at the elbow exit were so erratic that it was necessary to use readings at the rake, one diameter downstream. Even so, the static pressure readings at opposite sides varied widely from one another for most of the vanes tested. It was evident that useful loss coefficients could not be obtained from this system. However, static pressure readings at the entrance and exit of the elbow were recorded starting 1 in. from the duct wall, at 2 in. intervals across the top of the duct. Velocity pressure readings 11 in. from the duct floor were taken at the rake section, one diameter downstream from the elbow exit. Static pressure and velocity profiles for all vanes tested are shown in Fig. 9 to 22.

NOTE Where the term "diameter" is used in this paper, it signifies a 2 ft length of duct. This term is not strictly accurate because a 24 in. x 24 in. duct has a circular equivalent of 26.2 in. diameter, and 48 in. x 12 in. duct is equivalent to a 24.8 in. diameter.

Fig. 9 to 12 show proviles for single-thickness vanes of 2 in. radius spaced 1, $1\frac{1}{2}$, and 2 in. apart. The 90 deg arc vane produced a marked corner effect on the inside of the bend for all spacings, which caused negative pressures to extend about one third of the distance across the elbow. In contrast, when an extension of 0.25 of the vane chord was added to the trailing edge of the vane spaced at $1\frac{1}{2}$ in., negative pressures were eliminated and there was little apparent corner effect.

Fig. 13 to 16 show profiles for single thickness vanes of $4\frac{1}{2}$ in. radius spaced $2\frac{1}{4}$, $3\frac{1}{4}$, and $4\frac{1}{2}$ in. apart. The corner effect was more pronounced than for the 2 in. radius vanes, with negative pressures extending to the midpoint of the elbow for vanes spaced $2\frac{1}{4}$ and $3\frac{1}{4}$ in. apart. At $4\frac{1}{2}$ in. spacing the negative pressures continued to within 10 in. of the outside of the bend. At 5000 fpm the negative pressure at the inside wall was 0.75 in. water. Extending the trailing edge of this vane at $3\frac{1}{4}$ in. spacing had a stabilizing effect on the velocity profiles but only reduced negative pressures by approximately 25%.

Fig. 17 and 18 show profiles for two single thickness vanes, one with 6 in. radius, the other proprietary vane No. 1, which was constructed to a profile proposed by Keiber (5) and was spaced at approximately 0.7 in. The 6 in. radius vane had an extreme corner effect, while the proprietary vane produced very small negative pressures at the inside of the bend, and only slight corner effects on the velocity profiles.

Fig. 19 and 20 are profiles for shop made double-thickness vanes of 2 and $4\frac{1}{2}$ in. radius, respectively. Fig. 21 and 22 are profiles for double-thickness proprietary vanes, Nos 2 and 3, which have radii of 2.63 and 2 in. respectively. The smaller radius double-thickness vanes tested have similar characteristics showing only slight negative pressures at the inside of the bend and relatively little corner effect in velocity profiles. The $4\frac{1}{2}$ in. radius vane produced negative pressures similar to the single thickness vanes of the same radius.

Conclusions From The Testing Program With the First System

1. The design of vanes to effect a 90 deg bend in an air stream without producing negative pressures is critical, since only one of the fourteen vanes tested produced this result, viz. a 2 in. radius single thickness vane with extended trailing edge.
2. In general, small radius double thickness vanes produce lower negative pressures leaving the elbow than large radius double and single-thickness vanes.
3. In general, double-thickness vanes produce more uniform velocity profiles leaving the elbow.
4. Elbow losses cannot be obtained by a direct approach such as the first system.

TABLE 1

Vane Testing Program - 24 in. x 24 in. duct

2 in. radius single-thickness	s = 1 in.
	s = $1\frac{1}{2}$ in.
	s = 2 in.
	s = $1\frac{1}{2}$ in. with $\frac{3}{4}$ in. t.e. extension
$4\frac{1}{2}$ in. radius single-thickness	s = $2\frac{1}{4}$ in.
	s = $3\frac{1}{4}$ in.
	s = $4\frac{1}{2}$ in.
	s = $3\frac{1}{4}$ in. with 1-5/8 in. t.e. extension
6 in. radius single-thickness	s = 3 in.
2 in. radius double-thickness	s = $1\frac{1}{2}$ in.
$4\frac{1}{2}$ in. radius double-thickness	s = $3\frac{1}{4}$ in.
Proprietary vane No. 1 (approx.) single-thickness	s = $\frac{3}{4}$ in. (this vane was first proposed by Keiber and is not a circular arc vane)
Proprietary vane No. 2, double-thickness	s = 2-1/8 in. (r = 2-5/8 in.)
Proprietary vane No. 3, double-thickness	s = 1-7/16 in. (r = 2 in.)

Work Performed With The Second System

Fig. 5 shows the system layout used to re-test all vane geometries in Table 1. Before installing the elbow the duct was set up in a straight line from the fan, with a 3 ft piece included to represent the center-line length of the test elbow. Friction losses for straight duct were recorded over the range of the testing program and plotted on Logarithmic paper to produce the equation,

$$\Delta p = 0.28\ h_v^{\ 0.90},$$

for 16.8 diameters of straight duct. The static taps provided at 4 in. spacing across the duct located 1, 2, 3, and 4 diameters downstream from the elbow exit were used to note changes in flow irregularities (with distance from the elbow exit) which could be detected by changes in wall static pressure.

Fig. 23 gives loss coefficient data for single-thickness vanes of 2 in. radius spaced 1, $1\frac{1}{2}$ and 2 in. apart. The lowest coefficients were obtained with $1\frac{1}{2}$ in. spacing. The trailing edge extension for $1\frac{1}{2}$ in. spacing did not change the loss coefficient significantly as tested in a single elbow. Its superiority can be seen by comparing Fig. 10 and 12.

Fig. 24 presents the loss data obtained when testing single-thickness vanes of $4\frac{1}{2}$ in. radius spaced $2\frac{1}{4}$, $3\frac{1}{4}$, and $4\frac{1}{2}$ in. apart. With respect to loss coefficients, the $2\frac{1}{4}$ in. spacing was best up to 3000 fpm, after which the $3\frac{1}{4}$ in. spacing produced the lowest coefficient. Extension of the trailing edge of the vane was effective in stabilizing the velocity profiles at lower velocities, as shown in Fig. 13 to 16.

The results of testing double-thickness vanes of 2 and $4\frac{1}{2}$ in. radius, and a 6 in. radius single-thickness vane, are given in Fig. 25. The $4\frac{1}{2}$ in. radius vane produced a lower coefficient than the one with a 2 in. radius. This is due in part to the result of the discussion of Fig. 3. The profiles for pressure and velocity leaving the elbow, shown in Fig. 19 and 20, indicate that the 2 in. radius vane is superior in this respect. The 6 in. radius single-thickness vane performance was unstable, as shown by the curve in Fig. 25 and the profiles in Fig. 17.

The three proprietary vanes tested provided the data given in Fig. 26. The 2 in. radius single-thickness extended vane coefficients are also plotted in this figure for comparison. Proprietary vane No. 1 is a single thickness vane with the profile developed by Keiber (5) and spaced approximately 0.7 in. apart. Proprietary vane No. 2 is a double-thickness vane with rounded nose and extended trailing edge; this profile was developed by A.B. Collar (6). Proprietary vane No. 3, a double-thickness vane of 2 in. radius, is similar to the shop-made double thickness vane of the same radius. The fastening device for this vane spaced the vanes in the assembly 1.41 in. apart, reducing the area of the air passage between vanes and accounting, in part, for the high loss coefficients obtained. To judge the effect of double spacing vanes, every second vane in proprietary vane No. 3 was removed and the remaining assembly was tested at 1000, 2000, and 3000 fpm, giving the results shown in the uppermost curve of Fig. 26. Doubling the spacing doubled the loss coefficient. It was impossible to see a trend in the results so no further testing of this arrangement was made.

The sound level in the laboratory was considerable at low air flow volumes because of the nearly closed position of the variable inlet vanes on the fan. As the variable vanes were opened the sound decreased. No sound measurements were taken for the fan or turning vanes. (The only sound observed from the turning vanes was an oppressive, low-pitched whistle produced by the 2 in. radius, double-thickness shop-made vanes in the system in Fig. 5, which occurred at velocities above 3000 fpm. There was no immediate explanation for it.)

Conclusions From Testing Single Elbows In 24 in. x 24 in. Duct

1. The selection of vanes tested was broad enough to enable us to state that small radius single-thickness vanes are best.
2. Loss coefficients obtained for the best vanes are at the bottom end of the values of 0.10 to 0.35 currently given in the ASHRAE Handbook of Fundamentals (2). Study of Fig.12 indicates that there is little room for improvement in the performance of the extended 2 in. radius vane.

3. Carefully made shop vanes compare favourably with the proprietary vanes tested.
4. There is a pronounced drop in loss coefficients with increase in velocity up to 4000 fpm for all double-thickness vanes, proprietary and shop-made. This indicates that the loss coefficient cannot be given as a single factor for all velocities from the curves produced by plotting C_L vs. U. The same problem exists for single-thickness vanes where curves are other than straight lines and coefficients are velocity-head dependent.

System Arrangement For Testing Elbows in Combination

Space conditions made it necessary to arrange the duct system as shown in Fig. 6. Preliminary work established that the system, as arranged, would provide uniform velocities at the reading station upstream from the test elbows.

A traverse at the upstream reading station showed the need for a straightener. This was constructed of thin metal with 2 in. x 2 in. air passages 6 in. long and installed just downstream from the second elbow from the fan. The straightener was adjusted until traverses showed uniformity in flow similar to that obtained in the system of Fig. 5.

Since considerable work has been done on curved elbows by other investigators, they were not included in the single elbow testing program. However, as the investigation proceeded there arose a desire to compare vaned elbows with curved elbows under identical conditions. Two curved elbows with an R/W ratio of 1.5: 1 were constructed and provision made to compensate for the center line length of the elbows when installed, so that the total length between reading stations was unchanged.

Four vane geometries and one curved elbow were tested in the single elbow arrangement of Fig. 7. Δp across the nozzle was used as the only control reference to obtain nominal duct velocities in even thousands of fpm. From the data obtained, loss coefficients were deduced using the duct friction losses determined for 16.8 diameters of straight duct. Loss coefficients are shown in Fig. 27. These varied downward from 0 to 13% for the single-thickness vanes tested, and by 21% for the double-thickness vanes. Variations in performance were due in part to variation in mean duct velocities from the nominal thousands of fpm.

Combinations of elbows tested were close-coupled offset, separated offset, close-coupled U-turn, and close-coupled change-of-plane. Space in the laboratory did not permit testing curved elbows in U-turn and change-of-plane in other than close-coupled formation. A study of Fig. 28 to 31 shows no set pattern of performance from one arrangement to another. It is evident that the static and velocity profiles leaving the first elbow led, in some arrangements to reduced loss in the second elbow (and therefore reduced total elbow loss), and to increased loss in others. For example, the curved elbows in close-coupled offset showed a loss coefficient of 0.19; when separated two diameters the loss coefficient became 0.25. The single-curved elbow loss coefficient was 0.11. For double-thickness vanes the loss coefficient dropped approximately 10% when the elbows in offset were separated two diameters. Single thickness vanes showed no significant change due to separation or arrangement. In both close and separated offset the small single thickness vanes without extension vibrated considerably in the second elbow and the loss coefficients at higher velocities departed widely from the curve drawn. The same vane with extended trailing edge performed well under all conditions.

Conclusions From Testing Elbows in Combination

1. When vaned elbows are installed close to one another, only those which show stable performance under all conditions should be used.
2. No adjustment in estimated loss per elbow should be made from using the sum of single elbow losses since a combination may vary plus or minus 10%.
3. Low loss vanes increase in importance with the number of elbows in a system.

Change Of Aspect Ratio

Only one variation from the 24 in. x 24 in. duct was tested: 48 in. x 12 in. Obstructions in the laboratory made it necessary to arrange the duct as shown in Fig. 7. Although the rake section was located as shown, it was not used since reliable readings could not be obtained so close to the elbow. Mean duct velocities in thousands of fpm were used to calculate C_L. These velocities were obtained by setting the fan inlet vanes to obtain the pressure drops across the nozzle indicated by Fig. 8. Duct sections were constructed of 20 gauge galvanized iron, and the joints were made smooth inside and taped. When obtaining the loss for straight duct, one 1.9 diameter section had to be removed from the test section to prevent back-up pressure from an obstruction near the end of the duct. After testing the shortened straight duct section, Δp was adjusted to represent the total length of 15.1 diameters in the test section. Friction losses for the straight duct were recorded for the range of velocities used in the testing program and plotted on logarithmic paper to produce the equation,

$$\Delta p = 0.37 h_v^{\,0.91}$$

for 15.1 diameters in length.

Five vane geometries were tested in this series. The loss coefficients obtained are presented in Fig. 32. The considerable scatter from some of the curves can be attributed mainly to variations of actual mean duct velocities from the assumed thousands of fpm. A comparison of the loss coefficient curves in Fig. 32 for a 48 in. x 12 in. vaned elbow with the curves in Fig. 23 to 25 for the same vane geometries in 24 in. x 24 in. elbows shows higher coefficients in the former by as much as 29% at velocities of 1000 fpm. At 5000 fpm the coefficients for the two elbows are nearly equal,except in the case of the 4.5 in. radius single-thickness vane spaced 3.25 in. apart. The coefficient for this vane increased from +8% at 1000 fpm to +18% at 5000 fpm. Study of Fig. 10, 12, 13, 14, and 18 shows that the static pressure and velocity profiles for this vane were more irregular than for the others tested in the 48 in. x 12 in. duct. As duct width increases, irregularities have more room to increase and produce corresponding losses.

Correction of Velocities For Aspect Ratio

An attempt was made to account for the disparity of losses in the 24 in. x 24 in. and 48 in. x 12 in. elbows. Since tables for straight duct losses are given in terms of their circular duct diameter for equal friction, it follows that the velocity in a square or rectangular duct will differ from that of the equivalent circular duct. Dynamic losses in rectangular ducts will correspond to the actual duct velocity. (The testing programs in this project were conducted with rectangular duct velocities as shown, apart from boundary layer effect for which no allowance was made.) For example the 24 in. x 24 in. duct has a circular equivalent of 26.2 in. diameter and the 48 in. x 12 in. duct has a circular equivalent of 24.8 in. diameter. This means that when the velocity in a 26.2 in. duct is 1000 fpm, the velocity in a 24 in. x 24 in. duct will be 940 fpm. When the velocity in a 24.8 in. duct is 1000 fpm, the velocity in a 48 in. x 12 in. duct will be 840 fpm.

These rectangular duct velocities for friction losses should be used when comparing dynamic losses. The results of elbow losses calculated on this principle are tabulated below.

Duct size	Circular Duct Velocity	Velocity for friction loss equiv.	h_v	Δp r=2.0 s=1.5 single	Δp r=2.0 s=1.5 single ext.	Δp r=4.5 s=2.25 single	Δp r=4.5 s=3.24 single	Δp r=4.5 s=3.25 double
24 x 24	1000	940	0.055	0.0072	0.0066	0.0082	0.010	0.014
48 x 12		840	0.044	0.0072	0.0069	0.0079	0.0088	0.0135
				0%	+5%	-4%	-12%	-4%
24 x 24	2000	1880	0.22	0.028	0.0265	0.033	0.038	0.0475
48 x 12		1680	0.175	0.026	0.0262	0.030	0.033	0.0498
				-5%	-1%	-10%	-12%	+3%
24 x 24	3000	2820	0.494	0.061	0.059	0.074	0.078	0.091
48 x 12		2520	0.393	0.056	0.056	0.065	0.071	0.097
				-8%	-6%	-13%	-9%	+6%
24 x 24	4000	3760	0.88	0.104	0.106	0.132	0.127	0.146
48 x 12		3360	0.70	0.093	0.093	0.111	0.119	0.152
				-11%	-12%	-16%	-7%	+4%
24 x 24	5000	4700	1.37	0.154	0.165	0.206	0.180	0.219
48 x 12		4200	1.10	0.134	0.137	0.162	0.173	0.207
				-13%	-17%	-21%	-4%	-5%

Note: Δp and h_v are in. water

Three vane geometries tested in the change of aspect ratio program produced loss coefficient curves in the 24 in. x 24 in. duct which were practically independent of velocity head. They can be seen in Fig. 23 and 24. The definite slope of the loss coefficient curves for these vanes in the 48 in. x 12 in. duct can be attributed, in part, to the increased length of the vane fastening device.

Conclusions From the Change of Aspect Ratio Program

The reduction in loss caused by the elbow, shown in the table above, can be exptected as the aspect ratio increases from 1:1, since the "y" dimension to the first vane (Fig.2) will constitute a smaller percentage of the duct width. Conversely, when the aspect ratio diminishes from 1:1, the loss caused by the elbow can be exptected to increase.

Further work is desirable to confirm the trend in losses with change of aspect ratio. Losses should be found for the vane fastening device alone. This may help predict losses for other duct widths and heights.

SUMMARY OF TESTING PROGRAMS

The first objective of this project was the development of the geometry for one or several turning vanes with a high level of performance. Three sufficiently different radii for circular arc vanes were selected to indicate the variation in performance influenced by radius alone. Next vanes were tested for a given radius to find the optimum spacing. The third variation was from single to double-thickness. (Double-thickness vanes do not permit varying the spacing for a given radius.)

Single thickness vane radii adopted for the project were 2, 4.5, and 6 in. Only one vane spacing was tested for the 6 in. vanes.

Vane spacings tested were 1, 1.5, and 2 in. for the 2 in. radius vane. The 1.5 in. spacing was shown to be the best, having a loss coefficient of 0.13 as compared with 0.16 for 1 in. spacing and $\frac{0.24\ h_v^{\ 0.83}}{h_v}$ for vanes 2 in. apart. Vanes of 4.5 in. radius were spaced 2.25, 3.25, and 4.5 in. apart, for which the loss coefficients were 0.15, $\frac{0.20\ h_v^{\ 0.90}}{h_v}$ and 0.19 respectively.

A variation of the 2 and 4.5 in. radius vanes (the extension of the trailing edge by 0.25 of the chord dimension) was applied to the vanes at the intermediate spacings only. In the case of the 2 in. radius vane, the loss coefficient dropped from 0.13 to 0.12. An important characteristic of this vane is the impressive stability produced in the air stream leaving the vanes, illustrated in Fig. 12. In this Fig. the static pressure profiles at the exit from the elbow indicate that there is little "downstream effect" characteristic of the average bend and shown in Fig. 9, 10, and 11. In these three Fig. it is conceivable that there is a reverse flow through the vanes on the inside of the elbow. The elbow entrance static profiles in Fig. 12 show a uniformity and relation to the increasing velocity on the outside paths to effect the longer travel required in that portion of the duct. With this vane, branches near the bend would not be subject to the penalty common to such a location. The performance of the 2 in. radius extended vane was not duplicated by the 4.5 in. radius vane with trailing edge extension.

Double thickness vanes have the highest losses. The estimated loss in the vanes themselves, illustrated in Fig. 3 and discussed earlier in the paper, is borne out by this investigation. Additional losses are present since the loss for the single-thickness 2 in. vane is 0.13 h_v while that for the double-thickness vane ranges from 0.34 h_v at 500 fpm to 0.16 h_v at 5000 fpm. For the 4.5 in. radius single thickness vane, the loss ranges from 0.21 h_v at 500 fpm to 0.14 h_v at 5000 fpm, while the double-thickness vane shows a loss of 0.28 h_v at 500 fpm and 0.16 h_v at 5000 fpm.

The two double-thickness vane geometries tested are widely used in the industry. Larger sheet metal shops have machines for producing these vanes from strip. In such shops, the vaned elbow is found to cost less than the radius elbow. Unfortunately, at the more common velocities these vanes produce losses approximately double those for 2 in. radius single thickness vanes spaced 1.5 in. and for curved elbows with an R/W ratio of 1.5:1.

TABLE 2

Losses in a single 24 in. x 24 in. 90 deg elbow with no throat or heel radius derived from log plotting of Δp vs h_v

Vane Type	Radius	Spacing	Elbow Loss
Single-thickness	2.0 in.	1.0 in.	$0.16\ h_v$
Single-thickness	2.0 in.	1.5 in.	$0.13\ h_v$
Single-thickness	2.0 in.	2.0 in.	$0.24\ h_v^{0.83}$
Single-thickness - trailing edge extended	2.0 in.	1.5 in.	$0.12\ h_v$
Single-thickness	4.5 in.	2.25 in.	$0.15\ h_v$
Single-thickness	4.5 in.	3.25 in.	$0.20\ h_v^{0.90}$
Single-thickness	4.5 in.	4.5 in.	$0.19\ h_v$
Single-thickness - trailing edge extended	4.5 in.	3.25 in.	$0.18\ h_v$
Single-thickness	6.0 in.	3.0 in.	$0.19\ h_v$
Double-thickness	2.0 in.	1.5 in.	$0.30\ h_v^{0.84}$
Double-thickness	4.5 in.	3.25 in.	$0.23\ h_v^{0.90}$
Proprietary vane No. 1 single-thickness (approx.)	0.75 in.	0.69 in.	$0.20\ h_v^{0.96}$
Proprietary vane No. 2 double-thickness (approx.)	2.63 in.	2.13 in.	$0.21\ h_v^{0.83}$
Proprietary vane No. 3 double-thickness	2.0 in.	1.41 in.	$0.48\ h_v^{0.77}$

TABLE 3

Losses in two 24 in. x 24 in. 90 deg elbows in combination derived from log plotting of Δp vs hv

Type and arrangement of Vanes	Close coupled Offset	Offset separated 2 diameters	Close coupled "U" bend	Close coupled Change of Plane
Single-thickness 2.0 in. radius, 1.5 in. spacing trailing edge extended	$0.21\ h_v$	$0.22\ h_v$	$0.20\ h_v$	$0.21\ h_v$
Single-thickness 2.0 in. radius, 1.5 in. spacing	$0.27\ h_v$	$0.27\ h_v$	$0.27\ h_v$	$0.25\ h_v$
Single-thickness 4.5 in. radius, 2.25 in. spacing	$0.27\ h_v$	$.027\ h_v$	$0.25\ h_v$	$0.28\ h_v$
Double-thickness 4.5 in. radius, 3.25 in. spacing	$0.45\ h_v^{0.88}$	$0.40\ h_v^{0.90}$	$0.40\ h_v^{0.88}$	$0.40\ h_v^{0.91}$
Radius elbows, R/W = 1.5	$0.19\ h_v$	$0.25\ h_v$	laboratory space did not permit testing	

TABLE 4

Losses in single 48 in. x 12 in. 90 deg elbow with no throat or heel radius, derived from log plotting of Δp vs h_v

Vane type	Radius	Spacing	Elbow loss
Single-thickness, trailing edge extended	2.0 in.	1.5 in.	$0.15\ h_v$
Single-thickness	2.0 in.	1.5 in.	$0.17\ h_v^{0.91}$
Single-thickness	4.5 in.	2.25 in.	$0.18\ h_v$
Single-thickness	4.5 in.	3.25 in.	$0.23\ h_v^{0.90}$
Double-thickness	4.5 in.	3.25 in.	$0.33\ h_v^{0.87}$

TABLE 5

Losses in Single 90 deg Elbows

h_v = velocity head, in units desired
Δp = dynamic loss due to the bend in units consistent with h_v
C_L = loss coefficient = $\frac{\Delta p}{h_v}$
U = feet per minute x 10^{-4}
r = vane radius, inches
s = vane spacing, inches

Duct Size	Vane Geometry	C_L	Δp
24 x 24	single, r=2.0, s=1.0		$h_v(0.18-0.03U)$
	single, r=2.0, s=1.5		$h_v(0.137-0.05U)$
	single, r=2.0, s=2.0		$h_v(0.223-0.21U)$
	single, r=2.0, s=1.5 extended	0.12	
24 x 24	single, r=4.5, s=2.25	0.15	
	single, r=4.5, s=3.25		$h_v(0.20-0.15U)$
	single, r=4.5, s=4.5		$h_v(0.225-0.1U)$
	single, r=4.5, s=3.25 extended		
24 x 24	single, r=6, s=3.0	0.18	not valid above 4000 fpm
24 x 24	double, r=2.0, s=1.5		$h_v(0.36-0.69U+0.53U^2)$
	r=4.5, s=3.25		$h_v(0.315-066U+0.7U^2)$
24 x 24	Proprietary, single r=0.75, s=0.69 approx.		$h_v(0.198-0.053U)$
	Proprietary, double r=2.63, s=2.13		$h_v(0.212-0.29U+0.17U^2)$
	Proprietary, double r=2.0, s=1.41		$h_v(0.363-0.32U)$
48 x 12	single, r=2.0, s=1.5 extended		$h_v(0.167-0.1U)$
	single, r=2.0, s=1.5		$h_v(0.172-0.12U)$
	single, r=4.5, s=2.25		$h_v(0.182-0.08U)$
	single, r=4.5, s=3.25		$h_v(0.213-0.13U)$
48 x 12	double, r=4.5, s=3.25		$h_v(0.348-0.44U+0.15U^2)$
24 x 24	Radius elbow, R/W=1.5 (centre line radius to duct width)	0.11	

The second and third objectives are closely related. It is no problem to evaluate the losses in elbows for which the coefficient is unchanged at any velocity. Shop-made vanes in common use and proprietary vanes tested in this project do not have a single number loss coefficient. For these the coefficient varies with the velocity up to 4000 fpm or more. Attempting to obtain a mathematical expression for losses in all vanes tested, the loss Δp was plotted against the velocity head, h_v, on logarithmic paper. Tables 2, 3, and 4 give the expressions so derived. In certain cases the data varied from the curve at one end or other of the range, and in such cases the expression is not valid over the whole range tested. Subsequently, algebraic expressions were developed for the loss coefficient curves plotted to rectangular coordinates. These are given in Tables 5 and 6.

Table 7 gives the numerical loss coefficients for four selected shop-made vanes and the three proprietary vanes tested. The equation used for each vane from Table 2 or 5 was the one which produced coefficients nearest to the experimental data for the whole range. This table fulfils the second objective of the project for the vane geometries treated, in that it gives data to enable the designer to evaluate losses in vaned elbows with reasonable accuracy. It also fulfils the third objective of the project, giving comparative data for proprietary vanes. It should be noted that the loss coefficients presented here are applicable only to the sizes of ducts tested. Values will vary for other duct sizes. Some indication of the variation can be obtained from the data tabulated on page 8 and the discussion of that data.

TABLE 6

Losses in two Elbows in Combination

h_v = velocity head, in units desired
Δp = dynamic loss due to the bend in units consistent with h_v
C_L = loss coefficient = $\frac{\Delta p}{h_v}$
U = feet per minute x 10^{-4}
r = vane radius, inches
s = vane spacing, inches

Duct	Arrangement	Vane Geometry	C_L	Δp
		Single thickness		
24x24	Close coupled offset	r=2.0, s=1.5 ext.	0.21	
	Offset separated 2 diam.	"	0.22	
	Close coupled U bend	"	0.20	
	Close coupled change-of-plane	"	0.21	
		Single thickness		
24x24	Close coupled offset	r=2.0, s=1.5	0.27	
	Offset separated 2 diam.	"	0.27	
	Close coupled U bend	"		$h_v(0.311-0.2U+0.1U^2)$
	Close coupled change-of-plane	"	0.25	
		Single thickness		
24x24	Close coupled offset	r=4.5, s=2.25	0.27	
	Offset separated 2 diam.	"	0.27	
	Close coupled U bend	"	0.25	
	Close coupled change-of-plane	"	0.28	
		Double thickness		
24x24	Close coupled offset	r=4.5, s=3.25		$h_v(0.442-0.44U+0.21U^2)$
	Offset separated 2 diam.	"		$h_v(0.447-0.46U+0.2U^2)$
	Close coupled U bend	"		$h_v(0.405-0.4U+0.15U^2)$
	Close coupled change-of-plane	"		$h_v(0.406-0.36U+0.2U^2)$
		Radius elbows		
24x24	Close coupled offset	R/W=1.5	0.19	
	Offset separated 2 diam.	"	0.25	

TABLE 7

NUMERICAL LOSS COEFFICIENTS FOR SELECTED VANES

IN 24 in. x 24 in. 90 deg ELBOWS AND THEIR DERIVATION

U in fpm	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
Single-thickness $r=2.0$ in., $s=1.5$ in. trailing edge extended 0.75 in.	0.12	-	-	-	-	-	-	-	-	0.12
Single-thickness $r=4.5$ in., $s=2.25$ in.	0.15	-	-	-	-	-	-	-	-	0.15
Double-thickness $r=2.0$ in., $s=1.5$ in. $C_L=\frac{0.30h_v^{0.84}}{h_v}$	0.34	0.27	0.24	0.22	0.20	0.19	0.18	0.17	0.17	0.16
Double-thickness $r=4.5$ in., $s=3.25$ in. $C_L=0.315-0.66U\times10^{-4}+0.7U^2\times10^{-8}$	0.28	0.26	0.23	0.21	0.19	0.18	0.17	0.16	0.16	0.16
Proprietary vane, P-1 $C_L=0.198-0.53U\times10^{-5}$	0.20	0.20	0.19	0.19	0.19	0.18	0.18	0.18	0.17	0.17
Proprietary vane P-2 $C_L=\frac{0.21h_v^{0.83}}{h_v}$	0.24	0.19	0.16	0.15	0.14	0.13	0.12	0.12	0.11	0.11
Proprietary vane P-3 $C_L=0.363-0.32U\times10^{-4}$	0.35	0.33	0.31	0.29	0.28	0.26	0.25	0.23	0.22	0.20

By Definition $C_L=\frac{\Delta p}{h_v}$ Therefore $\Delta p = C_L \times h_v$

CONCLUSIONS

Losses in vaned elbows can be reduced to approximately half that of those in current use by adopting of a vane geometry with a performance similar to the best vanes presented here.

In this energy-conscious age, retooling to produce more efficient vanes would be in keeping with national (and international) efforts to conserve energy.

Investigation of vane performance should be continued to determine such factors as vane strength, fastening device losses, and effect of change of duct height at constant duct width, and change of duct width at constant duct height.

REFERENCES

1. The ASHRAE Guide and Data Book states that the loss coefficient, C, for vaned 90 deg elbows is between 0.10 and 0.35 of the velocity head depending on manufacture.

   Modern Air Conditioning, Heating, and Ventilating, 1959 by Carrier, Grant, and Roberts give additional lengths of duct to be added to the center-line elbow length in equivalent diameters as 20 for single-thickness vanes and 10 for double-thickness vanes. The single-thickness vane data is stated to be based on work by Stuart, Warner, and Roberts as recorded in ASHVE Transactions, 1942.

   Fan Engineering, 1970, by Buffalo Forge gives the loss for single thickness-vanes as

35% of the velocity pressure and for double-thickness vanes as 10% of the velocity pressure. These losses are from work done by Loren Wirt as recorded in the General Electric Review, June 1927, and are estimated values.

Stuart, Warner, and Roberts did their investigation with a square 7 in. x 7 in. 90 deg elbow. Wirt's work was done with 3 in. round, 3 in. x 3 in. and 6 in. x 6 in. square, and 1½ in. x 8 in. rectangular duct from which the author proposed loss calculation procedures.

A survey made by the Sheet Metal and Air Conditioning Contractors National Association showed that approximately 30% of the vanes used are manufatured, or proprietary vanes and the remainder are made by the individual sheet metal contractor. If the loss coefficients for proprietary vanes vary widely, the coefficients for shop-made vanes may be little more than a guess, for the designer.

2. ASHRAE Handbook of Fundamentals, 1972 Ch 25, Table 4.

3. Performance of Thin Turning Vanes in Square Conduits, Ahmed & Brundrett, University of Waterloo, May 1967.

4. ASME Power Test Codes, Part 2, Pressure Measurements, Ch 2.

5. Keiber, Ingenieur Archiv. Vol. 3, No. 5, 1932.

6. Some Experiments with Cascades of Aerofoils by A.R. Collar, B.A. B.Sc. of the Aerodynamics Dept., N.P.L. Reports & Memoranda No. 1768, 11 Dec. 1936.

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Wind Tunnel Testing, by Alan Pope, Ch 3, "Instrumentation and Calibration of the Test Section", Wiley, New York, 1961, Copyright 1954.

ACKNOWLEDGEMENTS

A work statement for research in the field of duct turning vanes was distributed to members of ASHRAE TC 4.1 on January 5, 1967. The statement was produced by a subcommittee comprised of D.J. Mosshart of the Limbach Company in Pittsburgh and J.R. Wright of Tennessee Technological University in Cookeville, Tennessee. Due to lack of support it was tabled at the ASHRAE Semiannual Meeting in Columbus February, 1968. Subsequently, the author became interested in the field covered by the work statement and was granted the privilege of funding and performing an investigation based on it.

Grateful acknowledgement is due to the originators of the work statement for their continued interest and assistance by consultation throughout the project and the preparation of this paper. Other members of TC 4.1 gave valuable advice on a number of occasions. (TC 4.1 was broken into TC 5.2 and TC 5.3 in October, 1972)

The investigation was conducted in the fluids laboratory of the University of Waterloo, Waterloo, Ontario. The project was visited by Wm. G. Colborne of the University of Windsor and by D.J. Mosshart both of whom made valuable criticisms of the work.

The care and detail with which the project was conducted would not have been possible without the space, apparatus, instrumentation, and advice from members of the staff which the University of Waterloo made available to the author, who was a member of neither the staff nor the student body.

Finally, the author gratefully acknowledges the construction work for the project performed by employees of S.E. Rozell & Sons Limited, and the painstaking data reading and recording from August 15th to November 3rd, 1969, and from June 5th to July 17th, 1970, by Bryan L. Price of the same company.

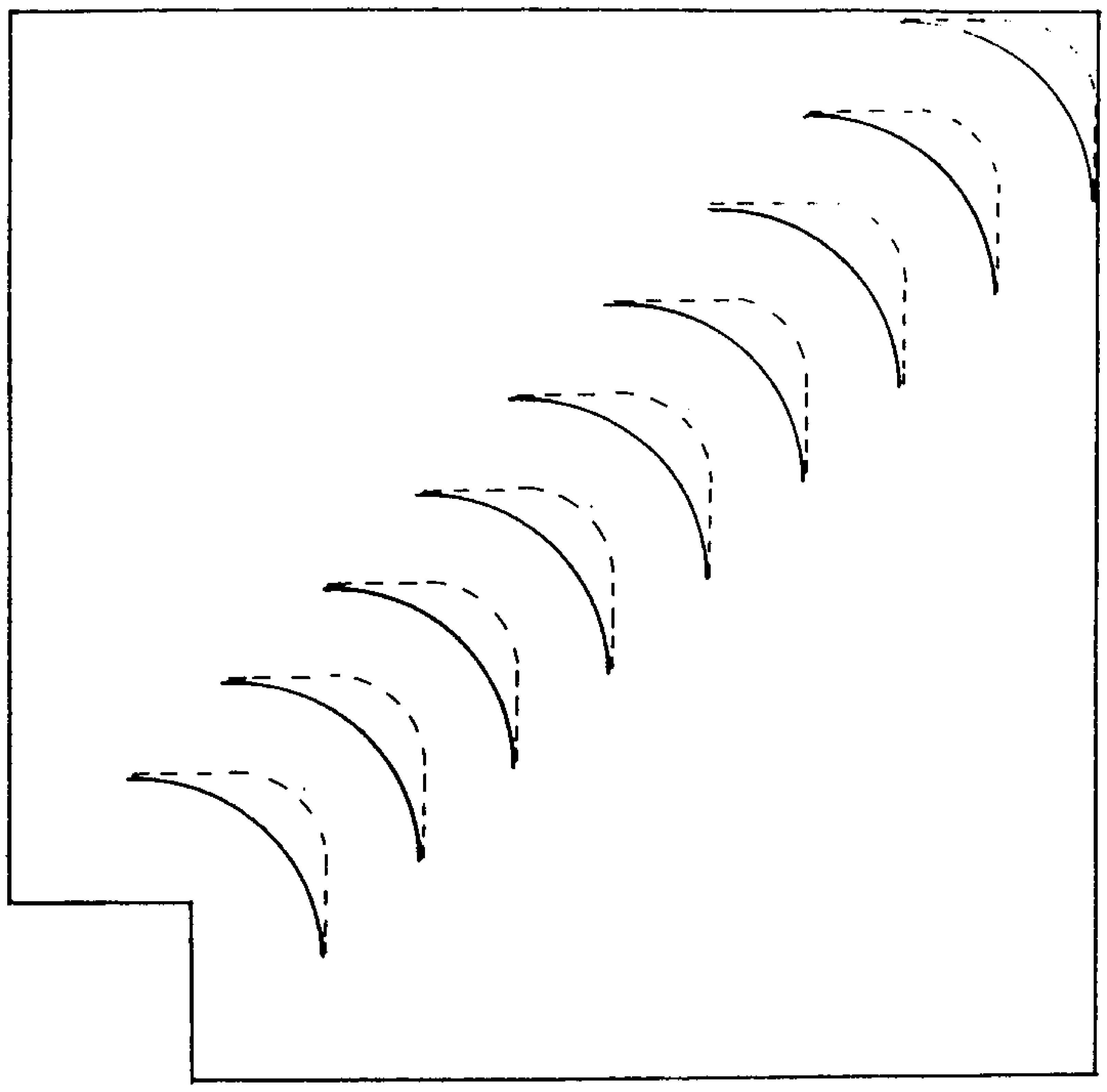
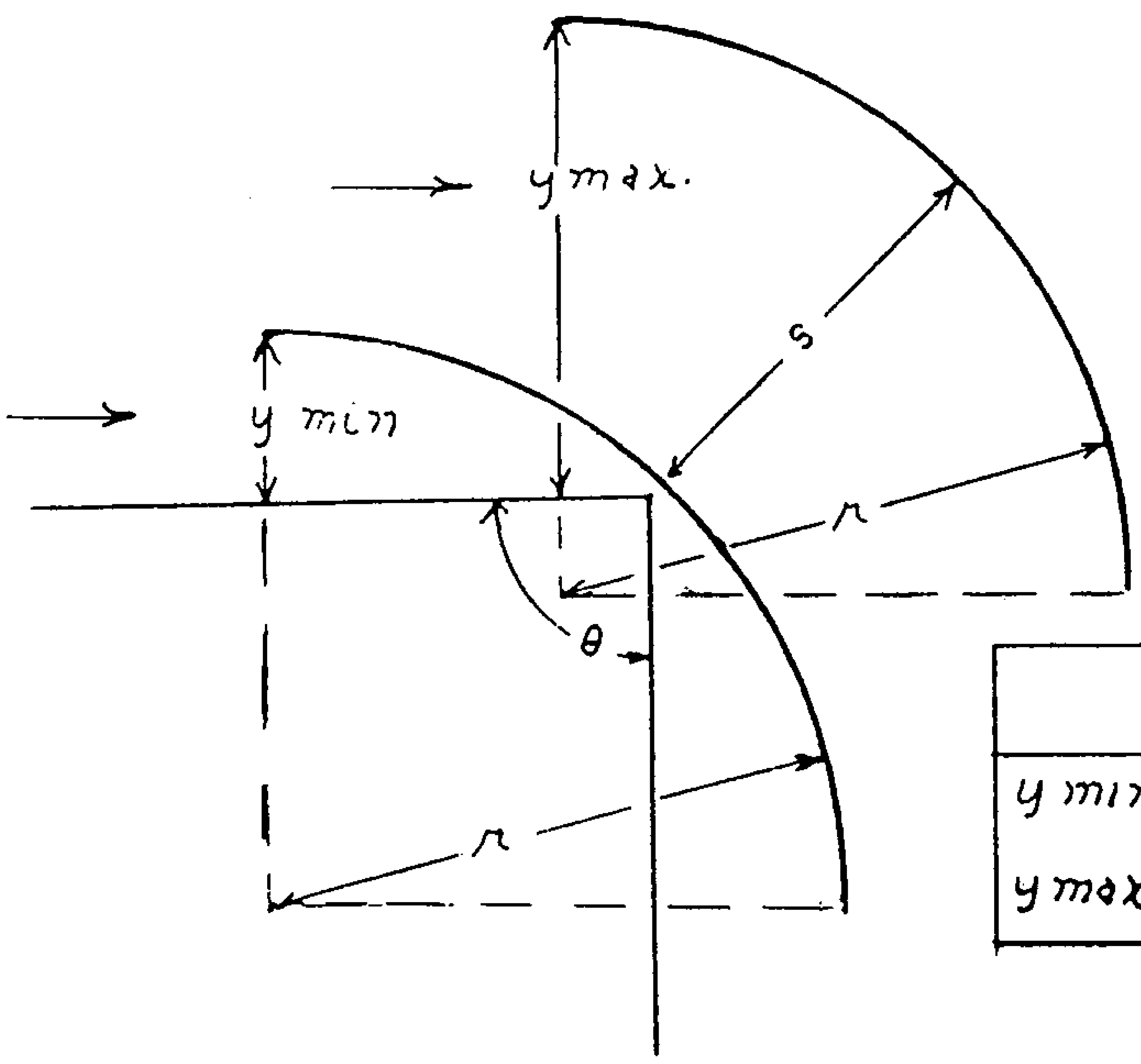
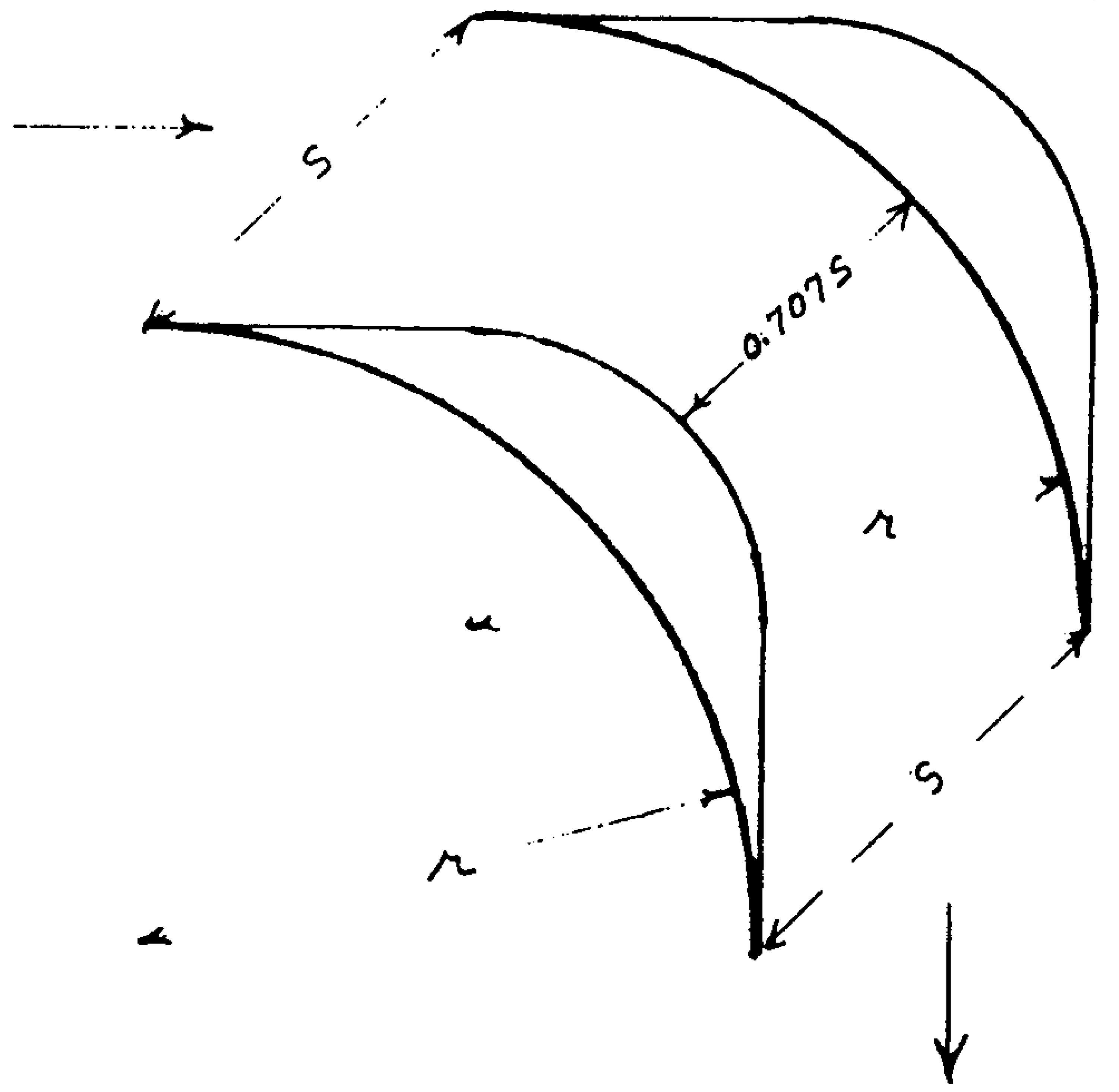


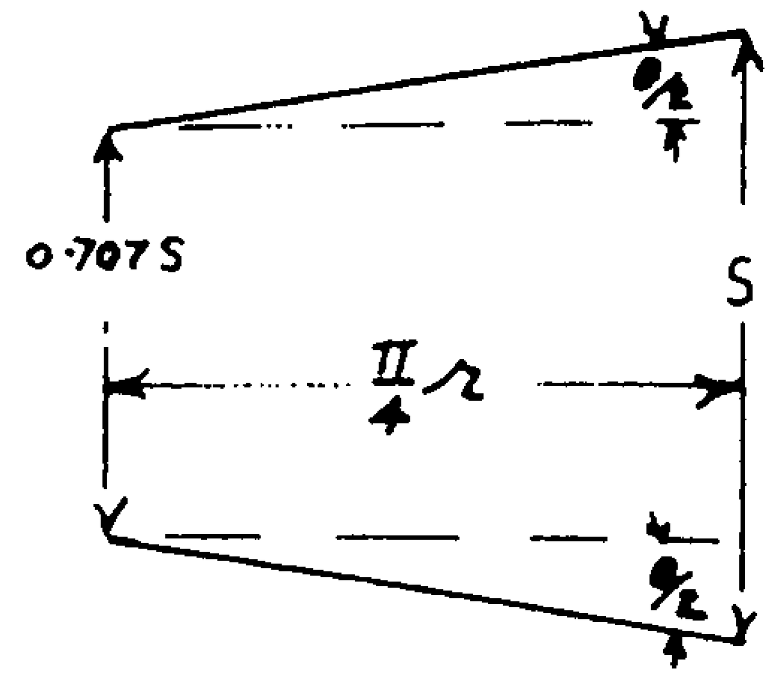
Fig. 1 General arrangement of turning vanes in 90° elbows



	$r = 2''$ $s = 1.5''$	$r = 4.5''$ $s = 3.25''$	$r = 6''$ $s = 3''$
$y\ min. = r - r\cos\frac{\theta}{2}$	0.59"	1.33"	1.75"
$y\ max = r - (r-s)\cos\frac{\theta}{2}$	1.65"	3.62"	3.9"

Fig. 2 Relationship of first vane to elbow throat piece





	s = 1.5, r = 2.0	s = 3.25, r = 4.5
$\tan\frac{\theta}{2}$ =	0.162	0.138
θ =	18.4°	15.3°
$C_{expansion}$ =	0.038	0.033

Fig. 3 Air passage between two double thickness vanes

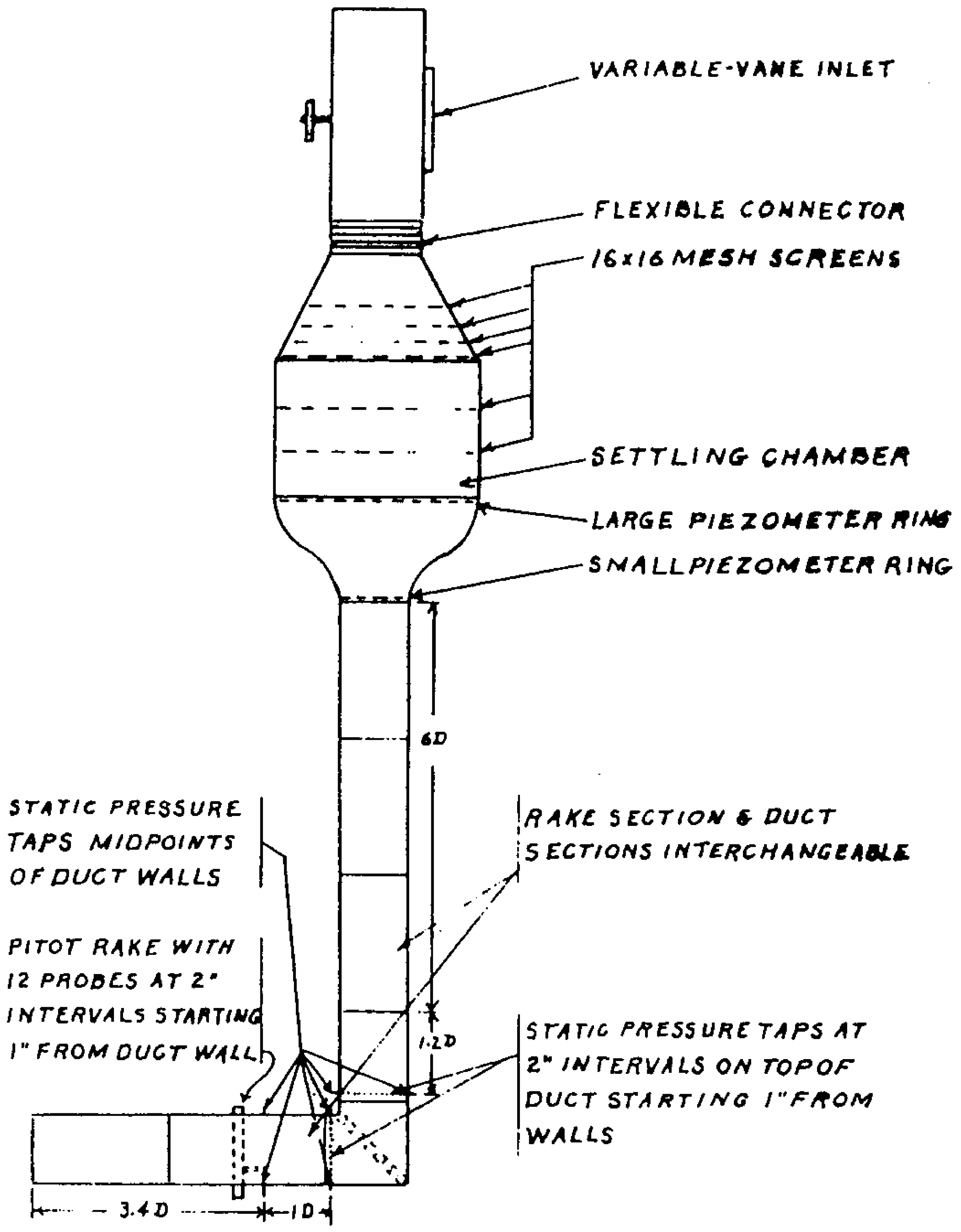


Fig. 4 Testing system with the first duct arrangement for testing turning vanes in a 24" x 24" 90° elbow by direct reading

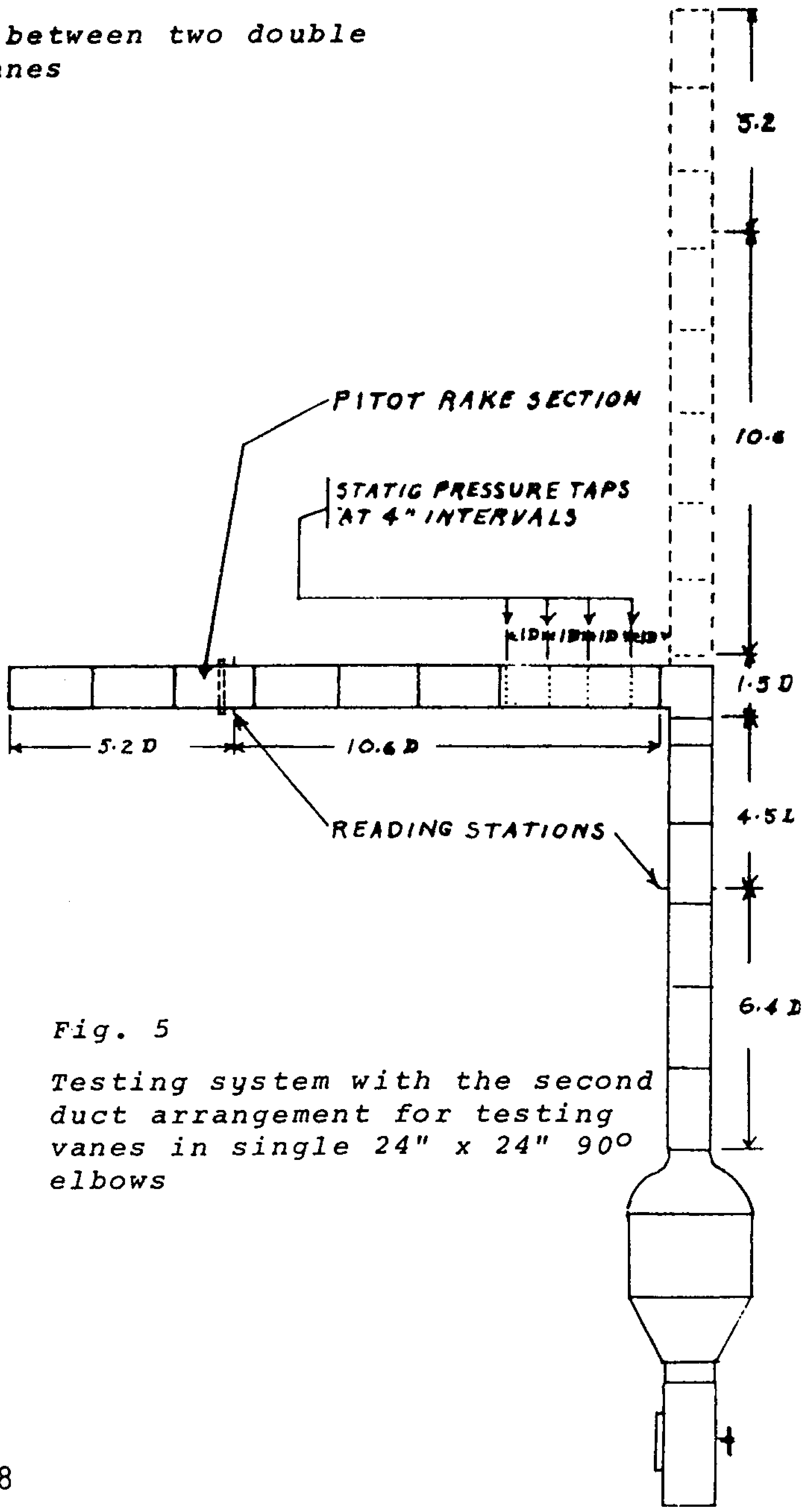
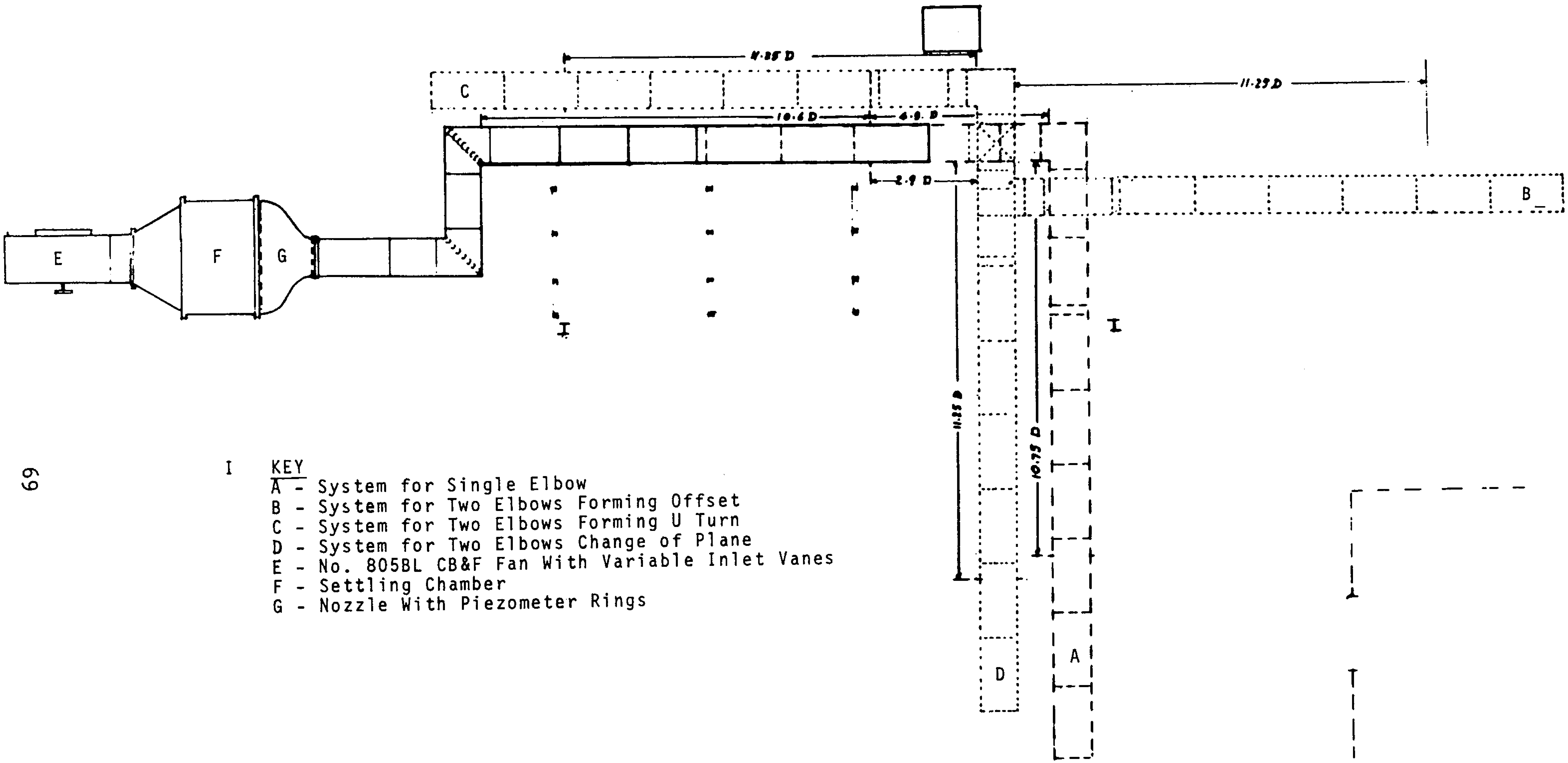


Fig. 5 Testing system with the second duct arrangement for testing vanes in single 24" x 24" 90° elbows



I KEY

A - System for Single Elbow
B - System for Two Elbows Forming Offset
C - System for Two Elbows Forming U Turn
D - System for Two Elbows Change of Plane
E - No. 805BL CB&F Fan With Variable Inlet Vanes
F - Settling Chamber
G - Nozzle With Piezometer Rings

Fig. 6 System for testing two 24" x 24" 90° elbows in combination

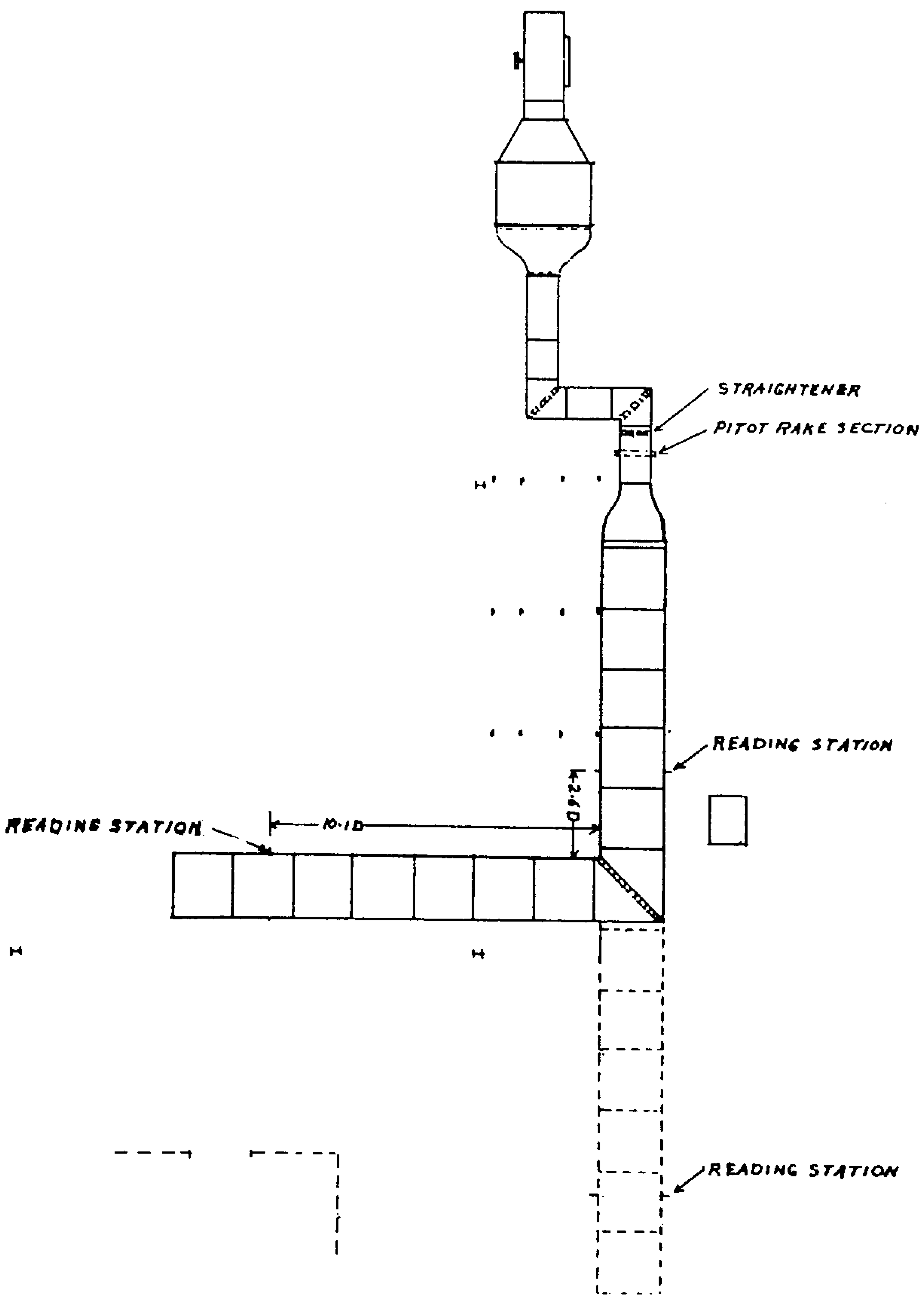


Fig. 7 System for testing turning vanes in a 48" x 12" 90° elbow

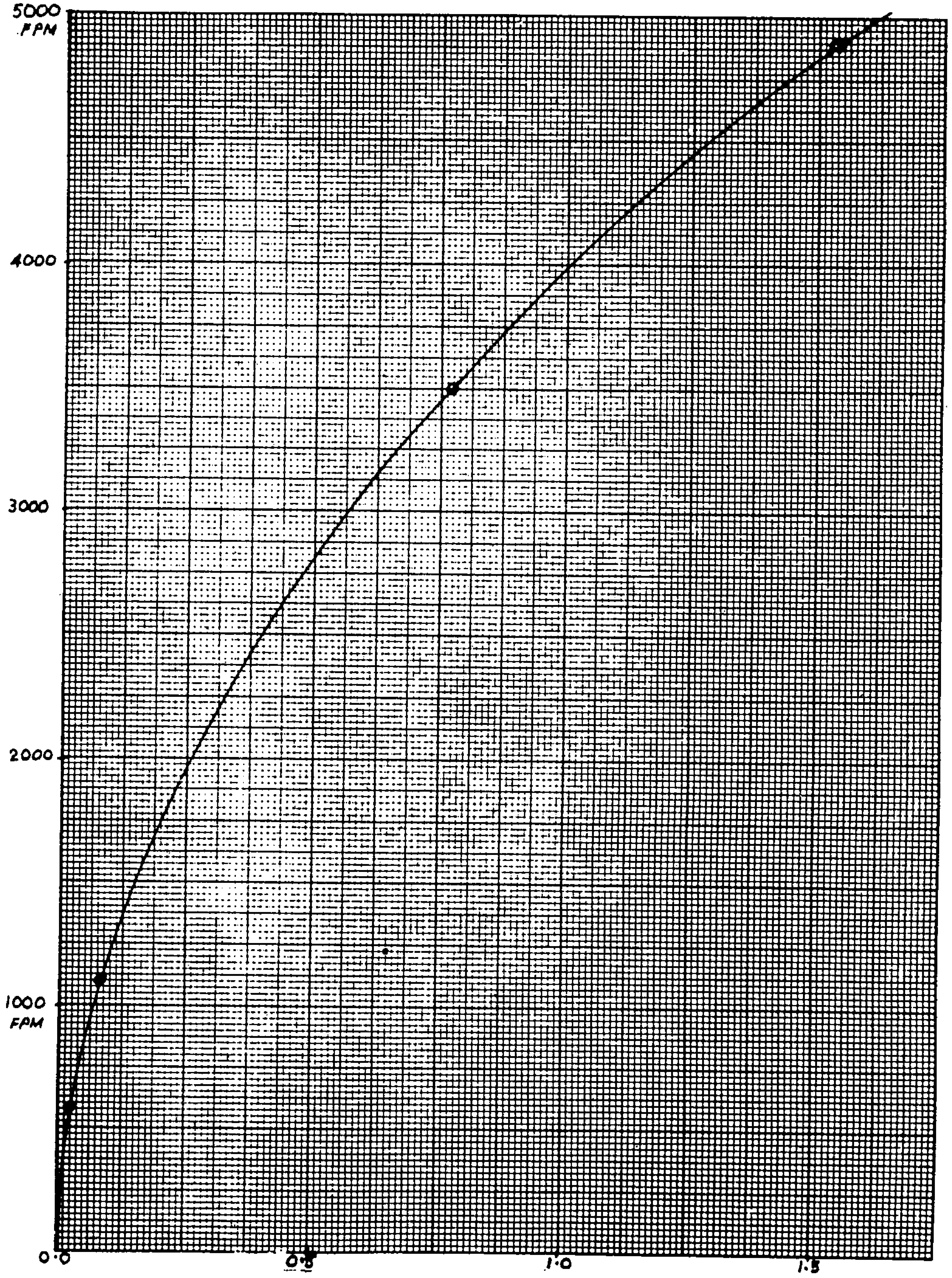


Fig. 8 Δp across 6' x 6' to 2' x 2'

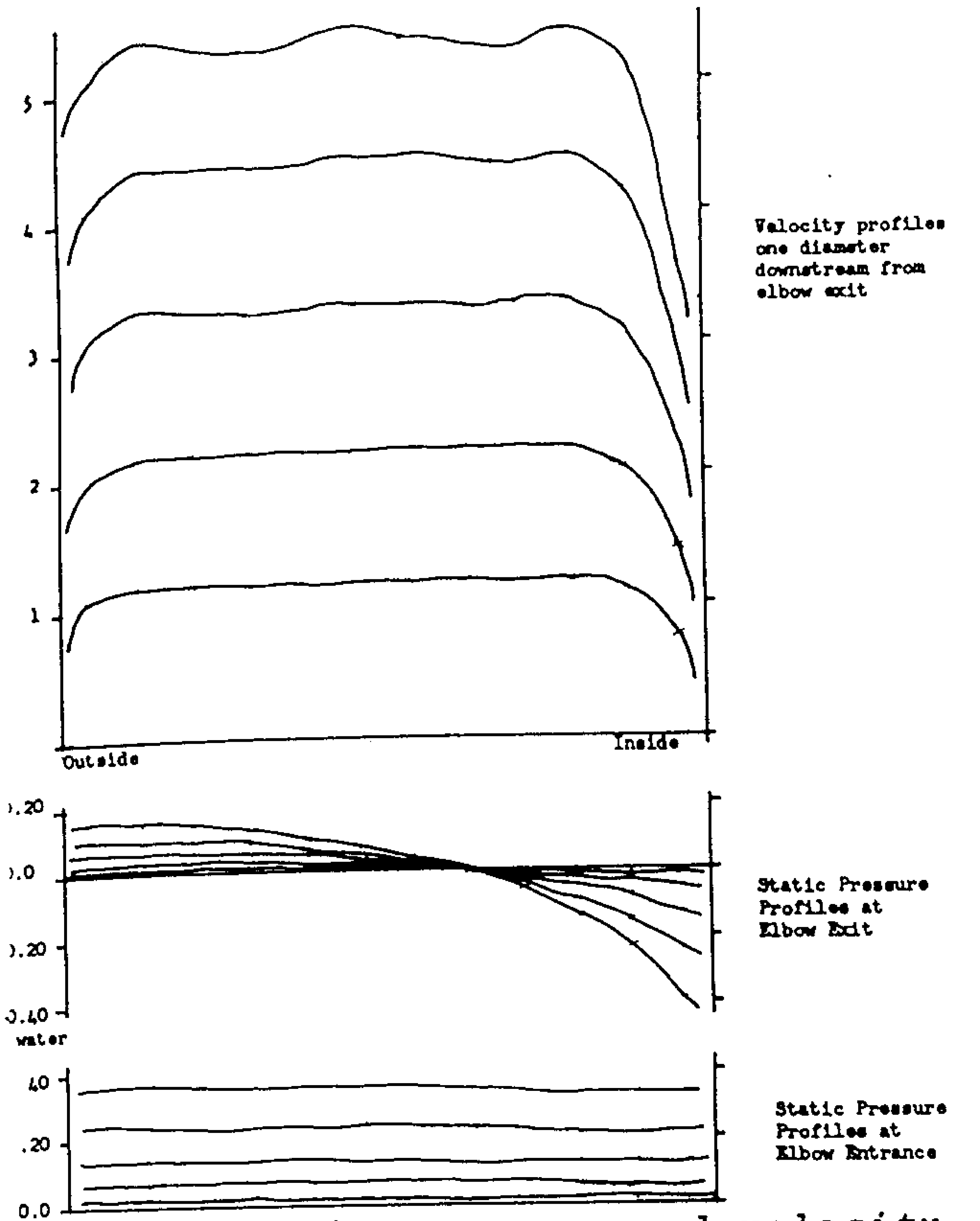


Fig. 9 Static pressure and velocity profiles for 2" radius single thickness vanes spaced 1"

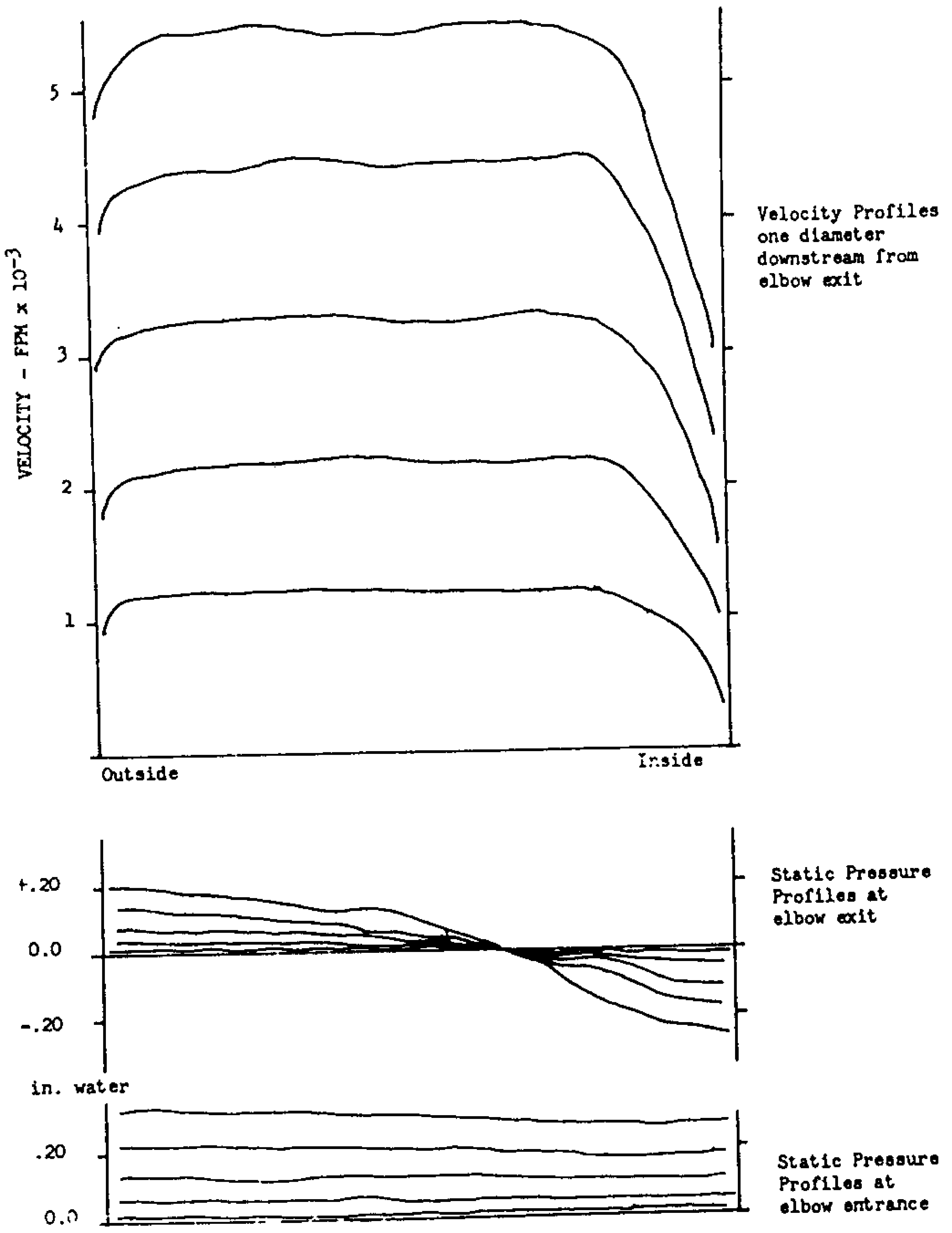


Fig. 10 Static pressure and velocity profiles for 2" radius single thickness vanes spaced 1.5"

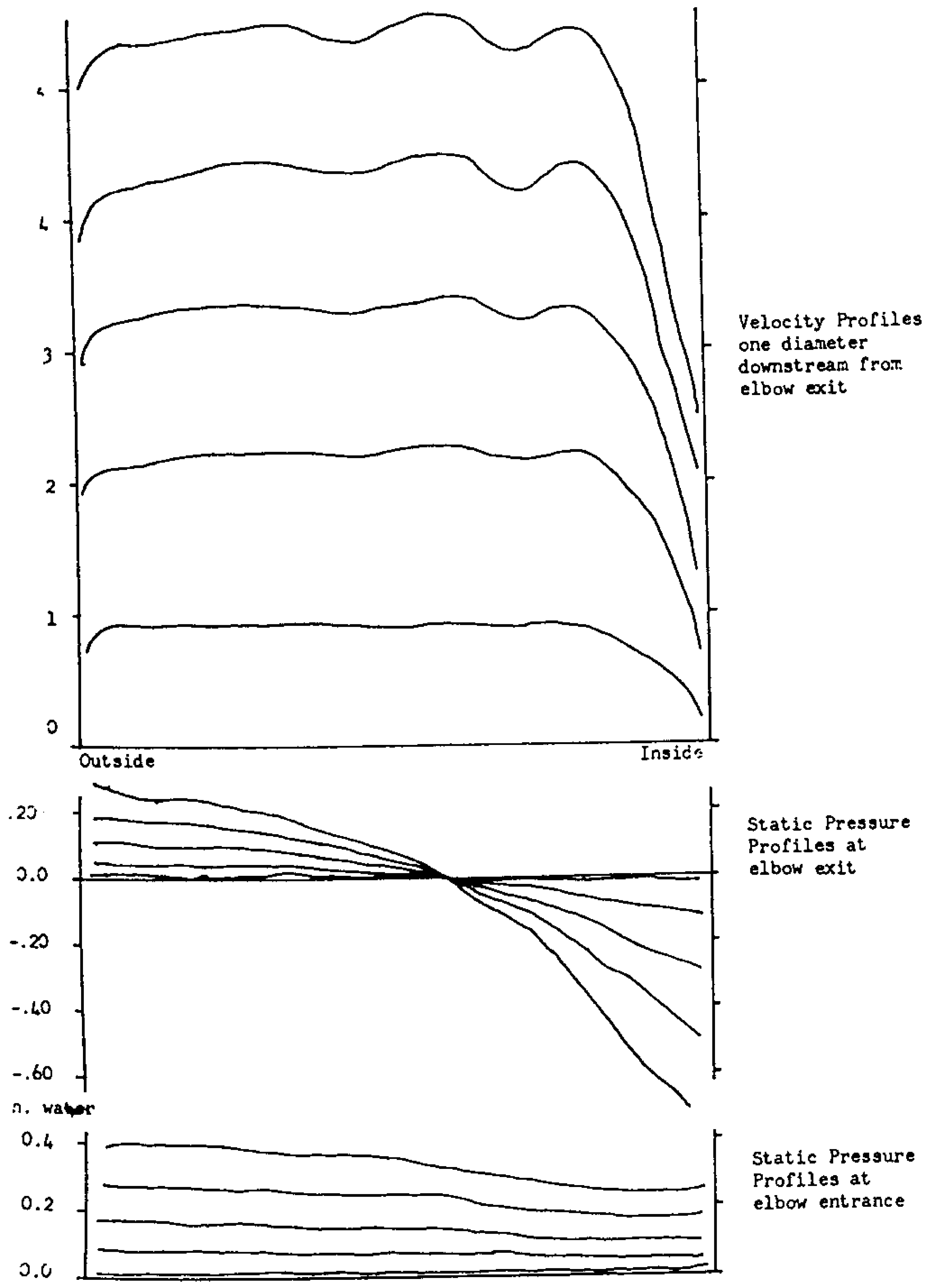


Fig. 11 Static pressure and velocity profiles for 2" radius single thickness vanes spaced 2"

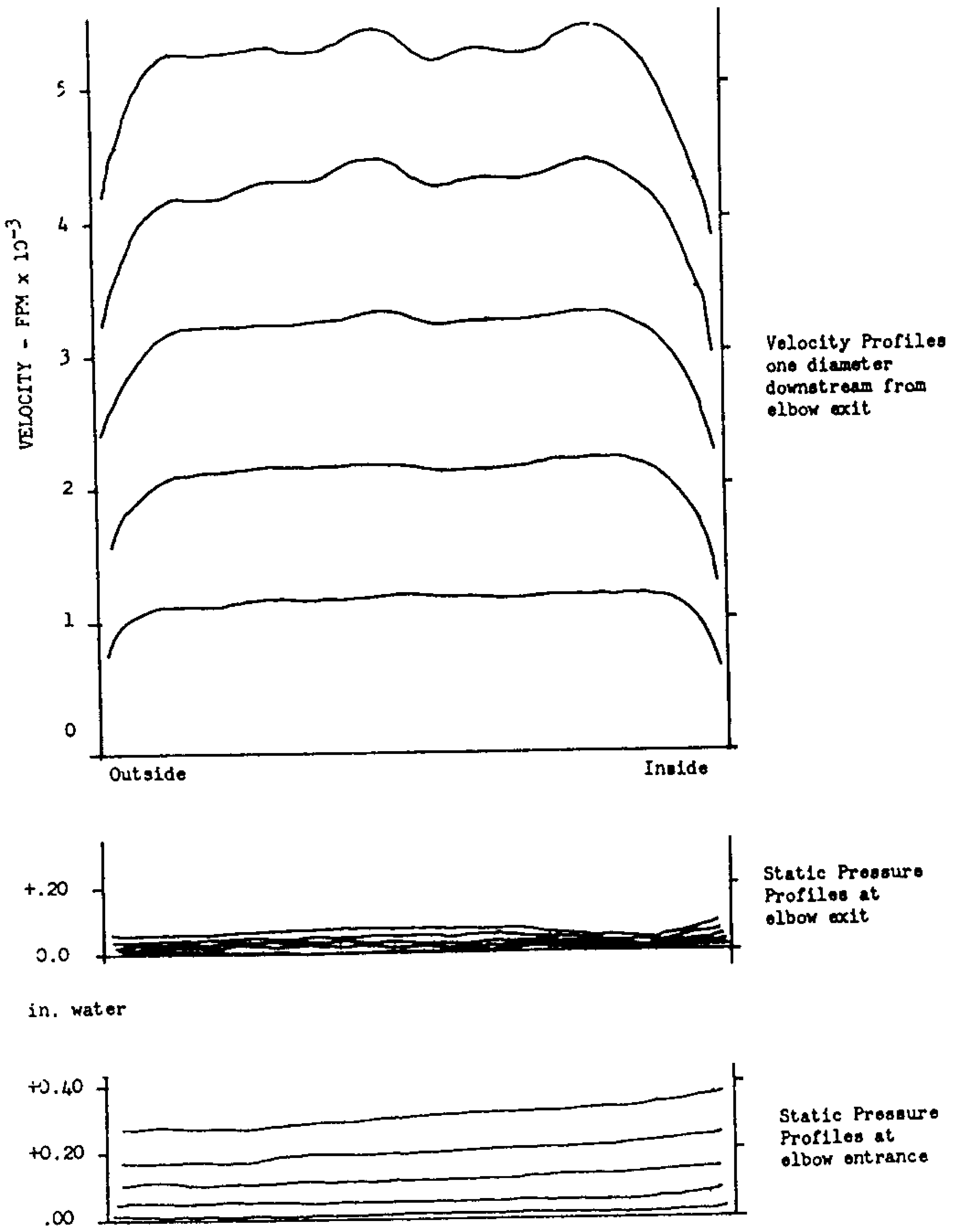


Fig. 12 Static pressure and velocity profiles for 2" radius single thickness vanes spaced 1.5" trailing edge extended

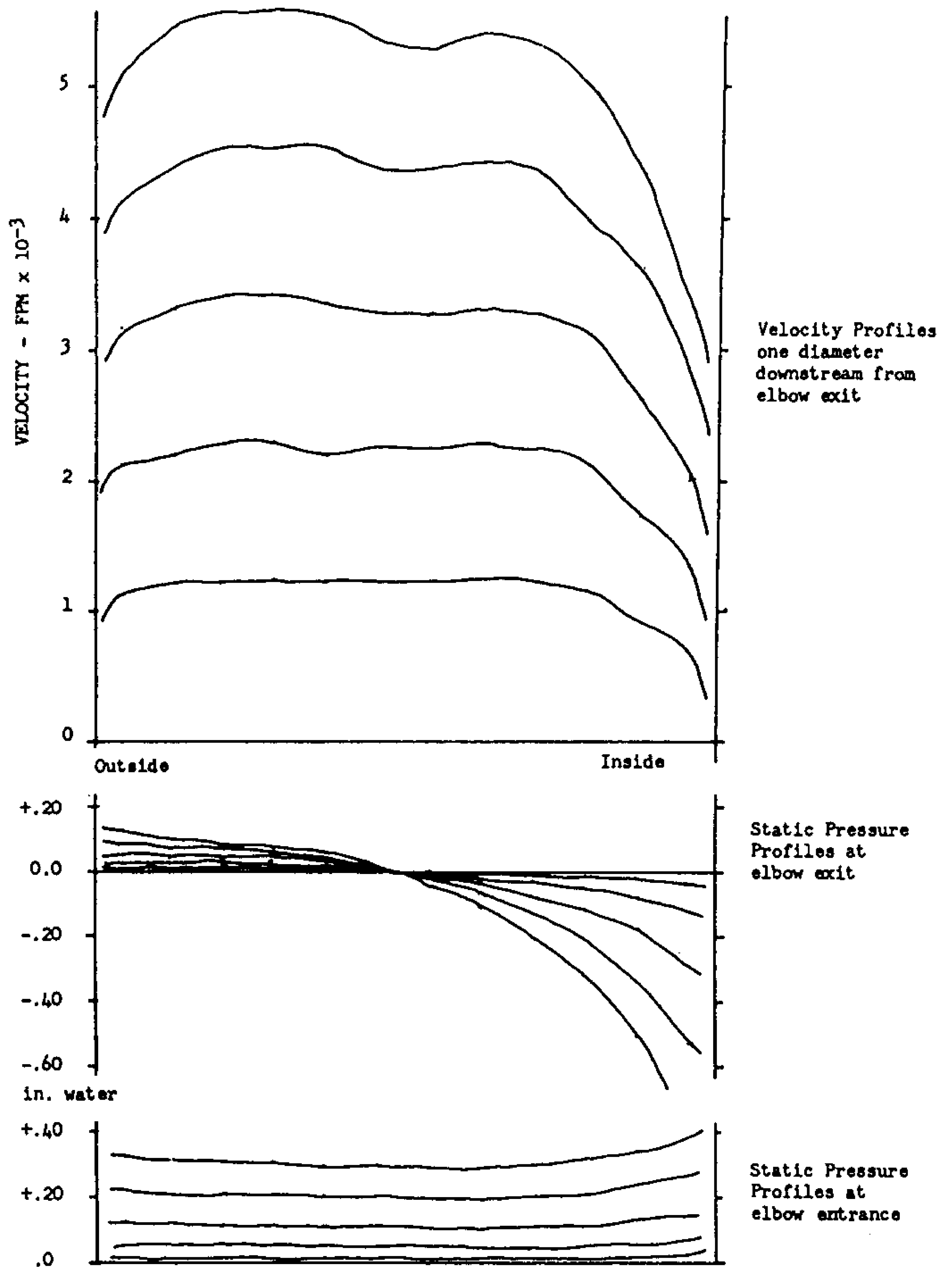


Fig. 13 Static pressure and velocity profiles for 4.5" radius single thickness vanes spaced 2.25"

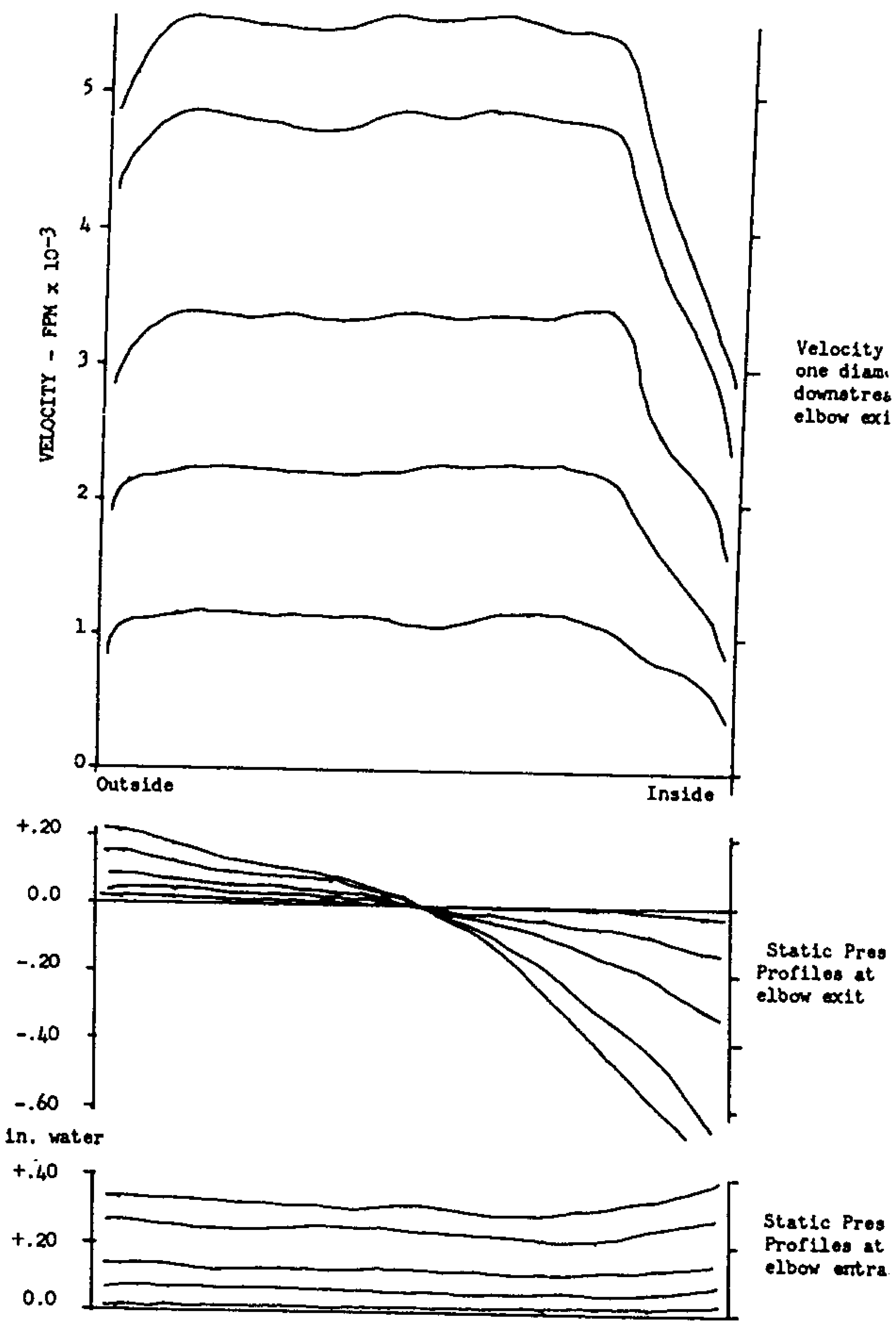


Fig. 14 Static pressure and velocity profiles for 4.5" radius sir thickness vanes spaced 3.25"

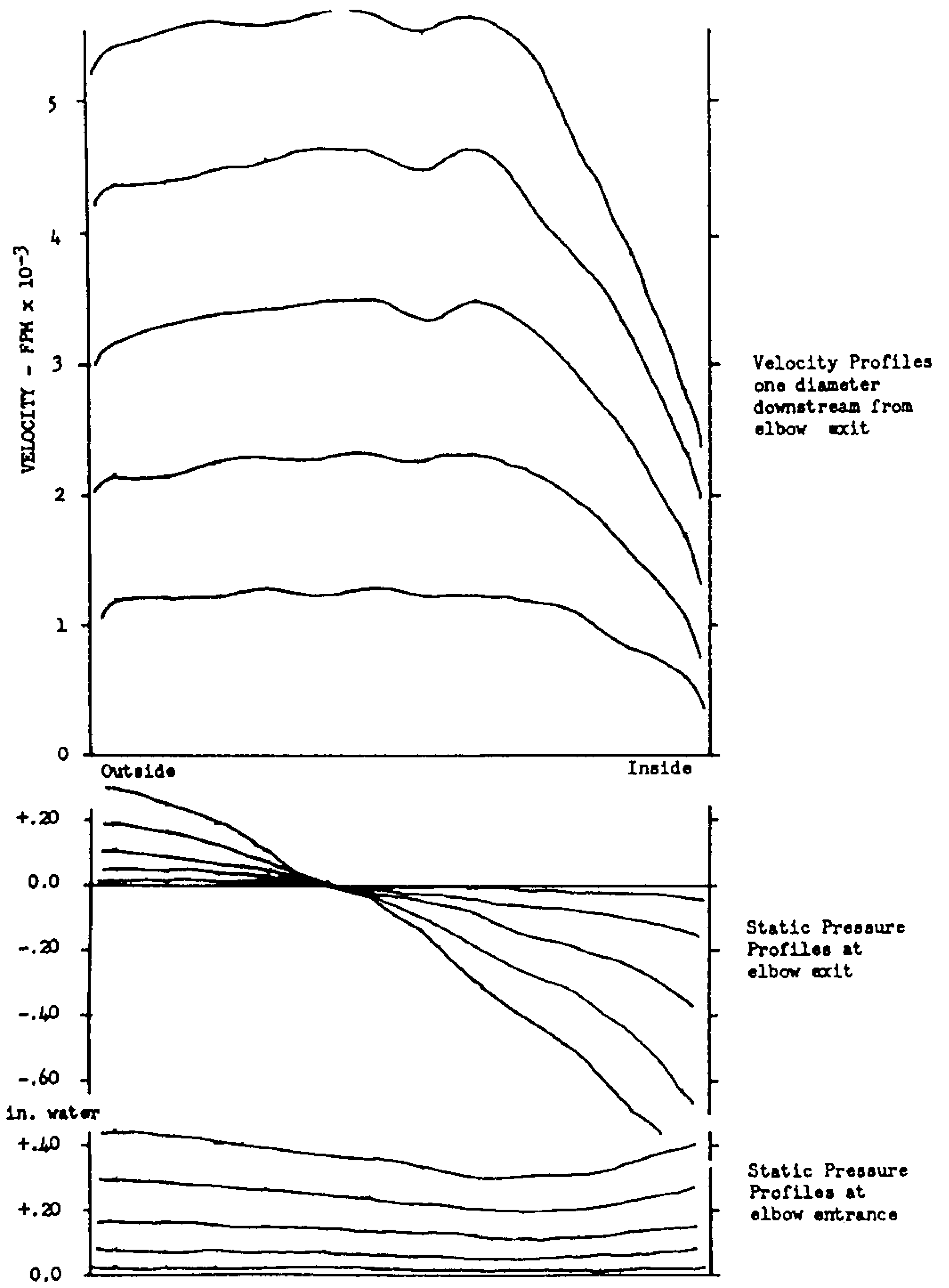
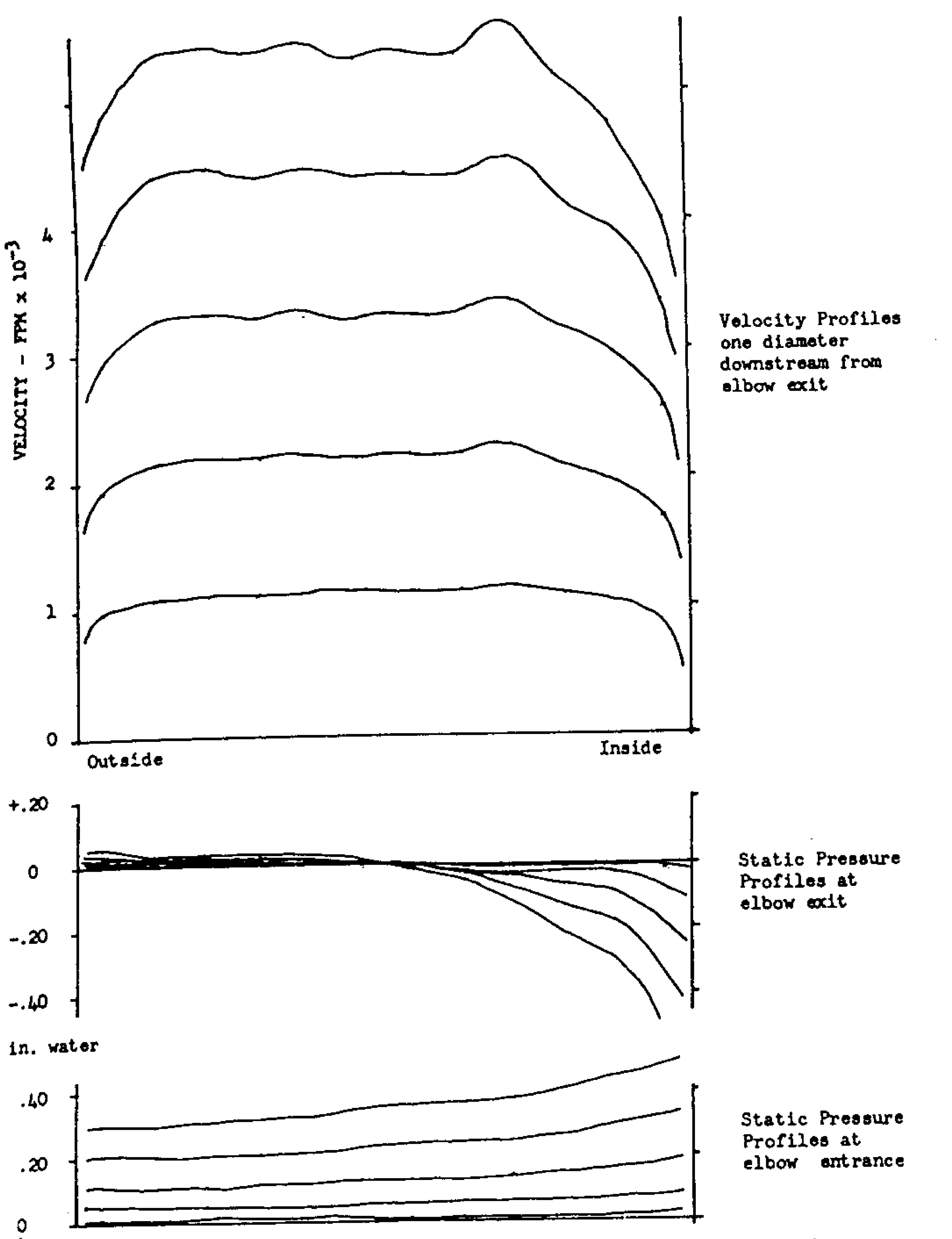


Fig. 15 Static pressure and velocity profiles for 4.5" radius singl thickness vanes spaced 4.5"



16 *Static pressure and velocity profiles for 4.5" radius single thickness vanes spaced 3.25", trailing edge extended*

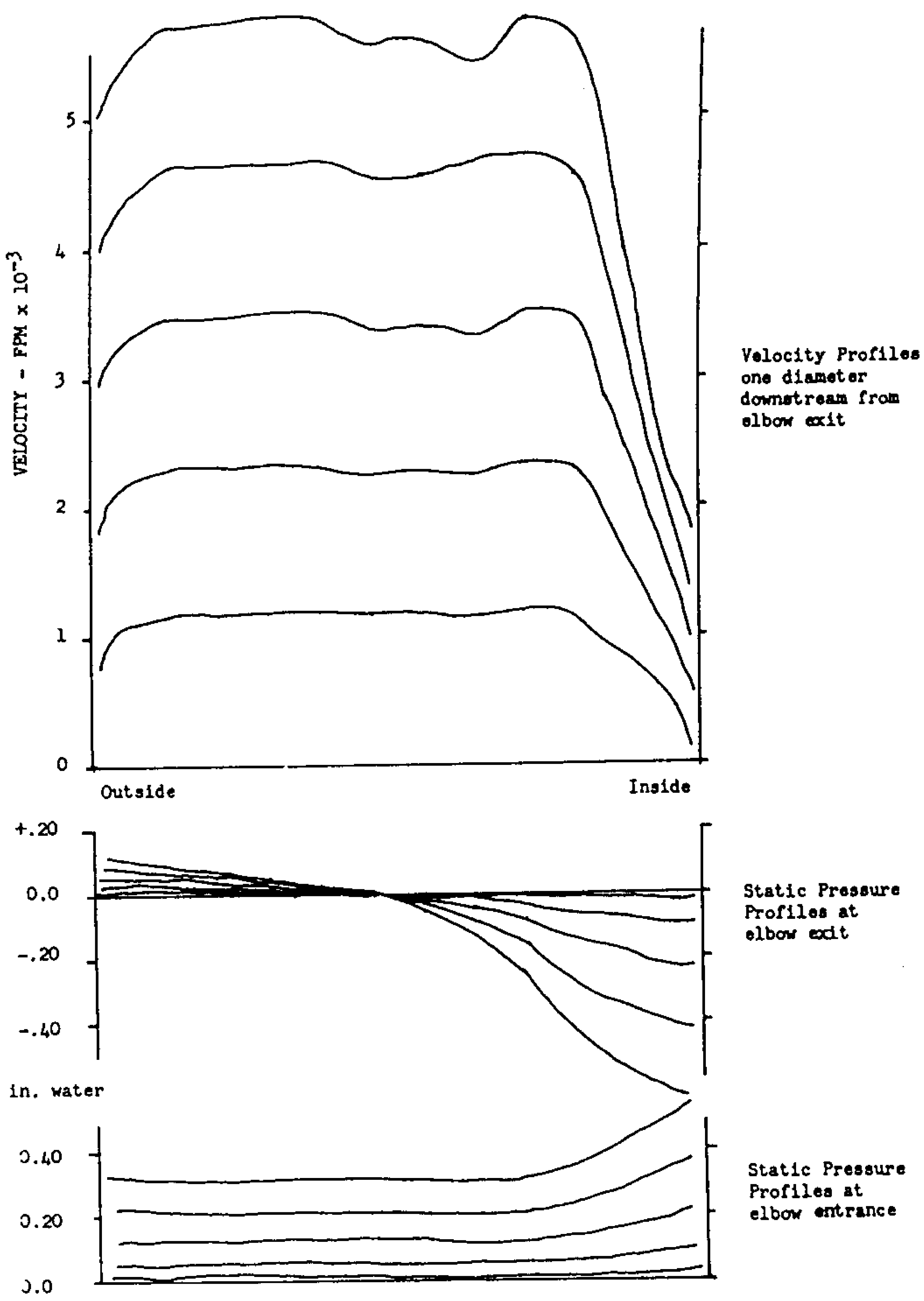


Fig. 17 Static pressure and velocity profiles for 6" radius single thickness vanes

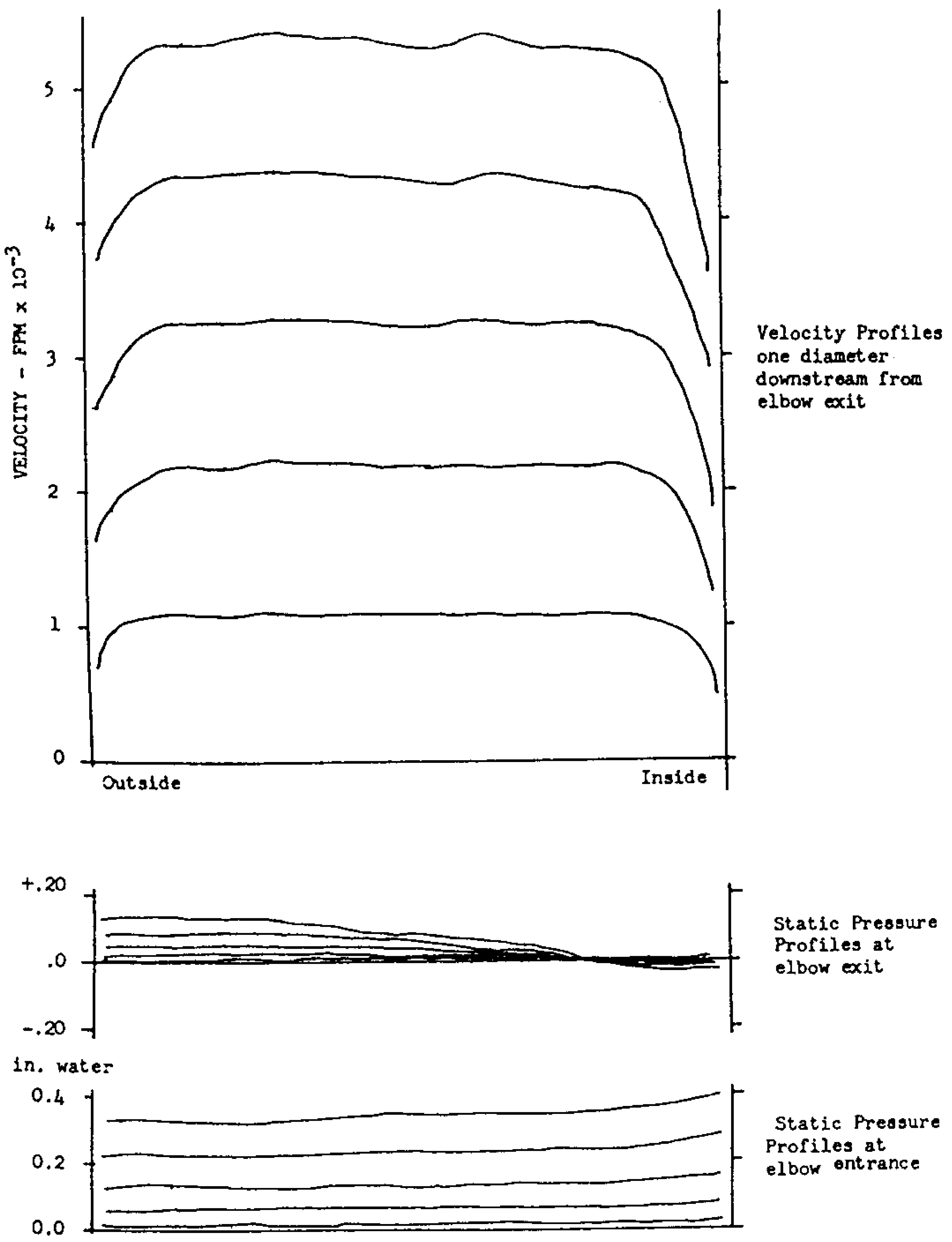


Fig. 18 Static pressure and velocity profiles for single thickness proprietary vane No. 1

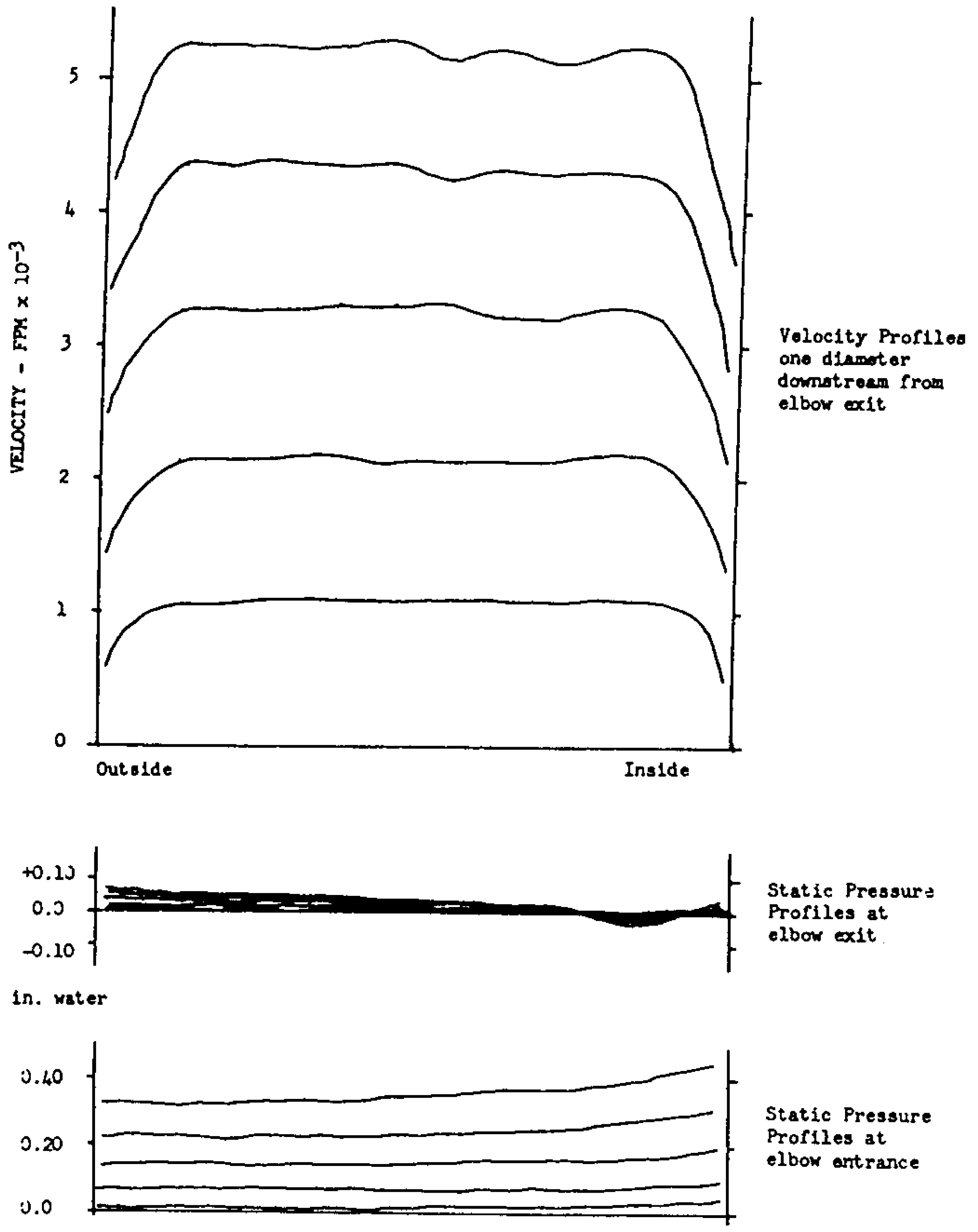


Fig. 19 *Static pressure and velocity profiles for 2" radius double thickness vanes spaced 1.5"*

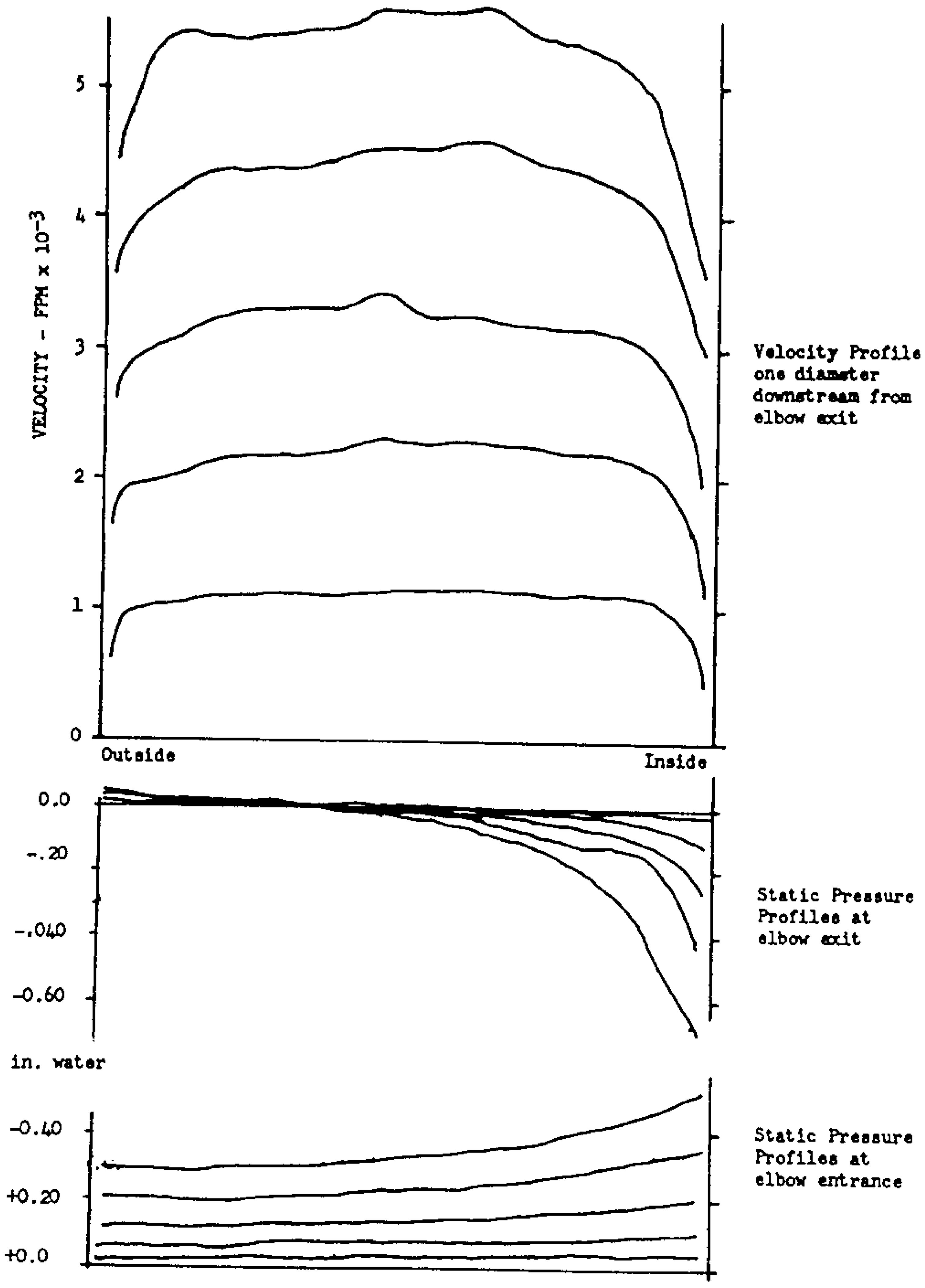


Fig. 20 *Static pressure and velocity profiles for 4.5" radius doul thickness vanes spaced 3.25"*

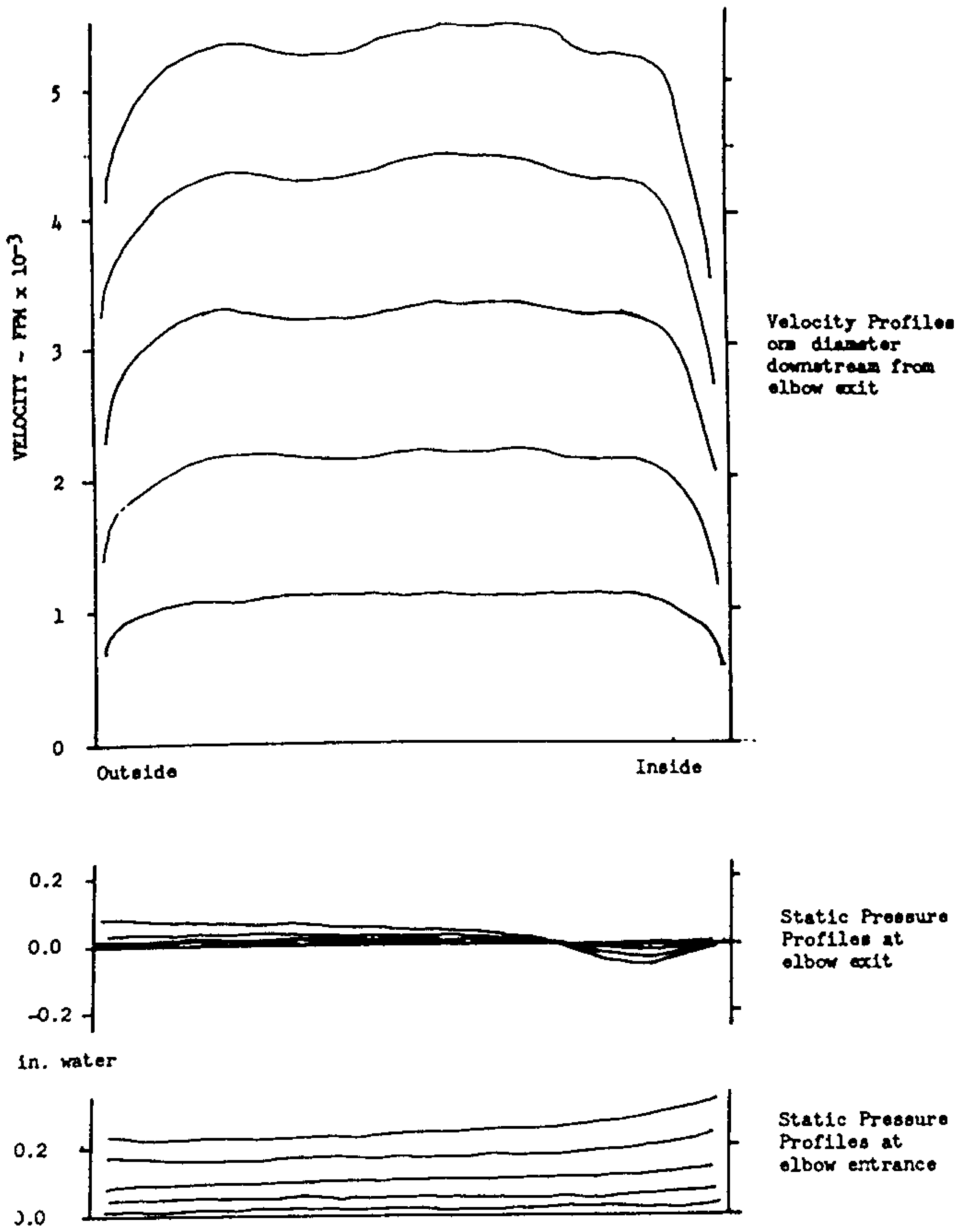


Fig. 21 Static pressure and velocity profiles for 2.63" radius double thickness proprietary vane No. 2

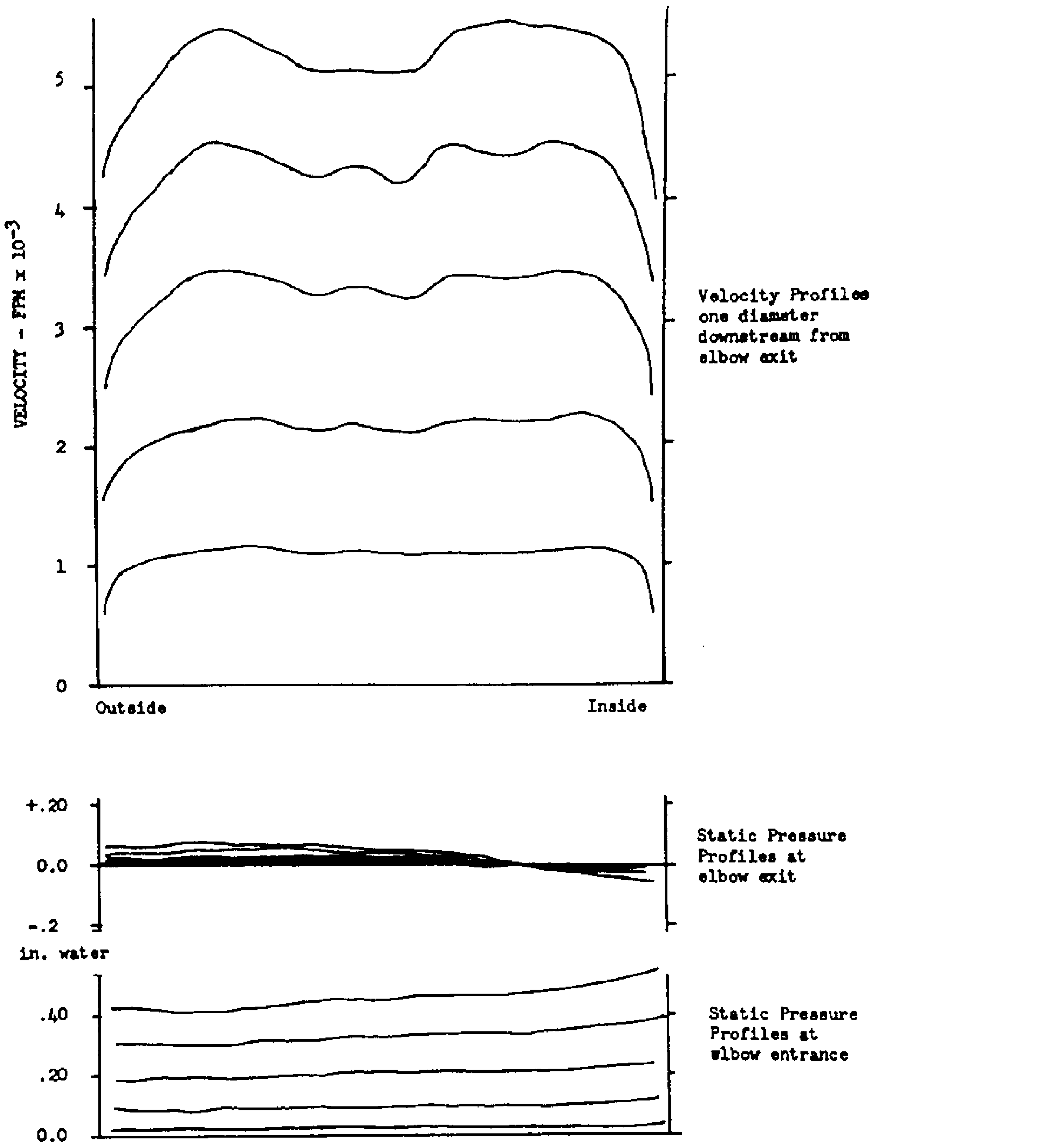


Fig. 22 Static pressure and velocity profiles for 2" radius double thickness proprietary vane No. 3

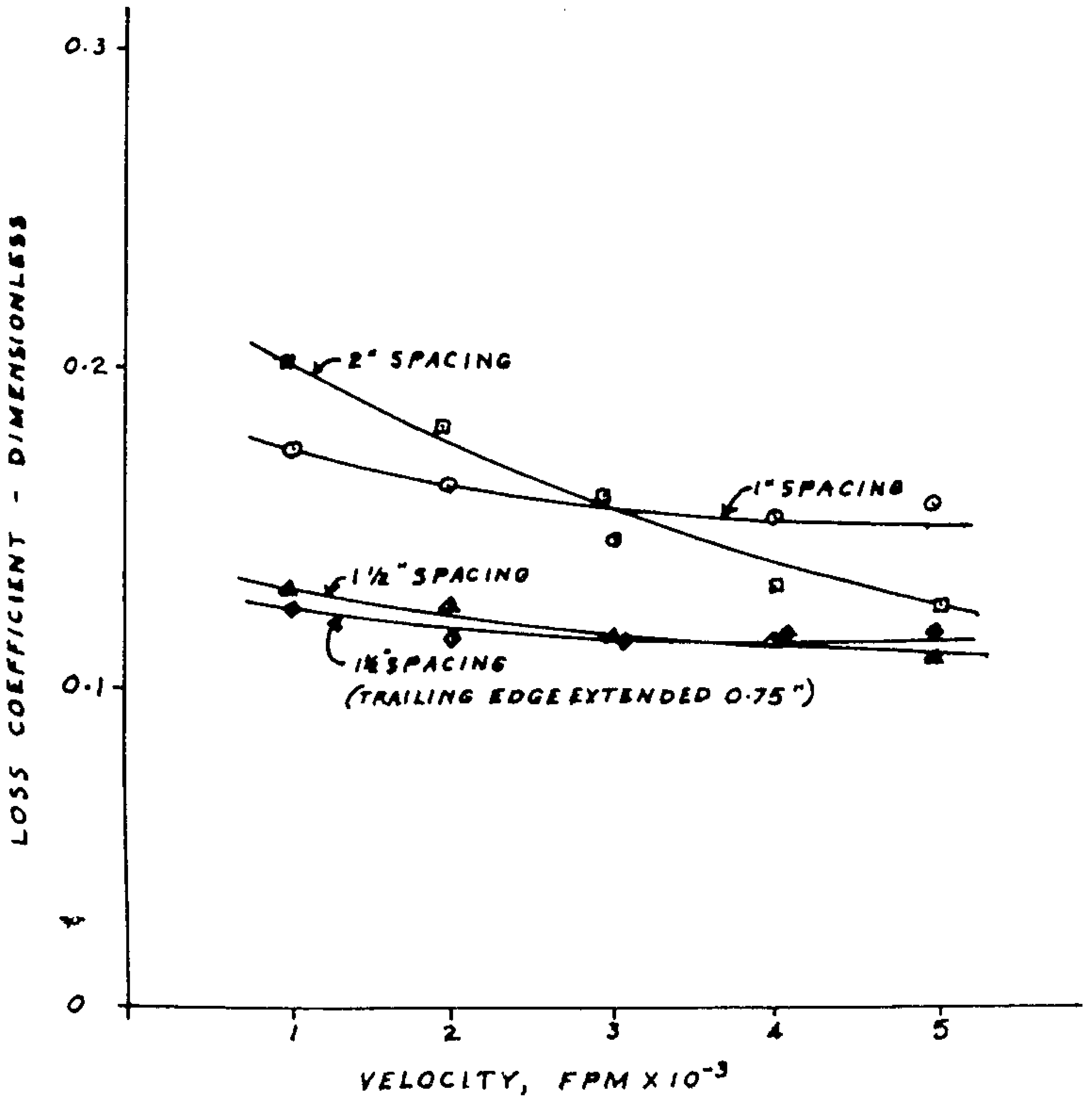


Fig. 23 Loss coefficients for single thickness vanes of 2" radius in 24" x 24" 90° elbow

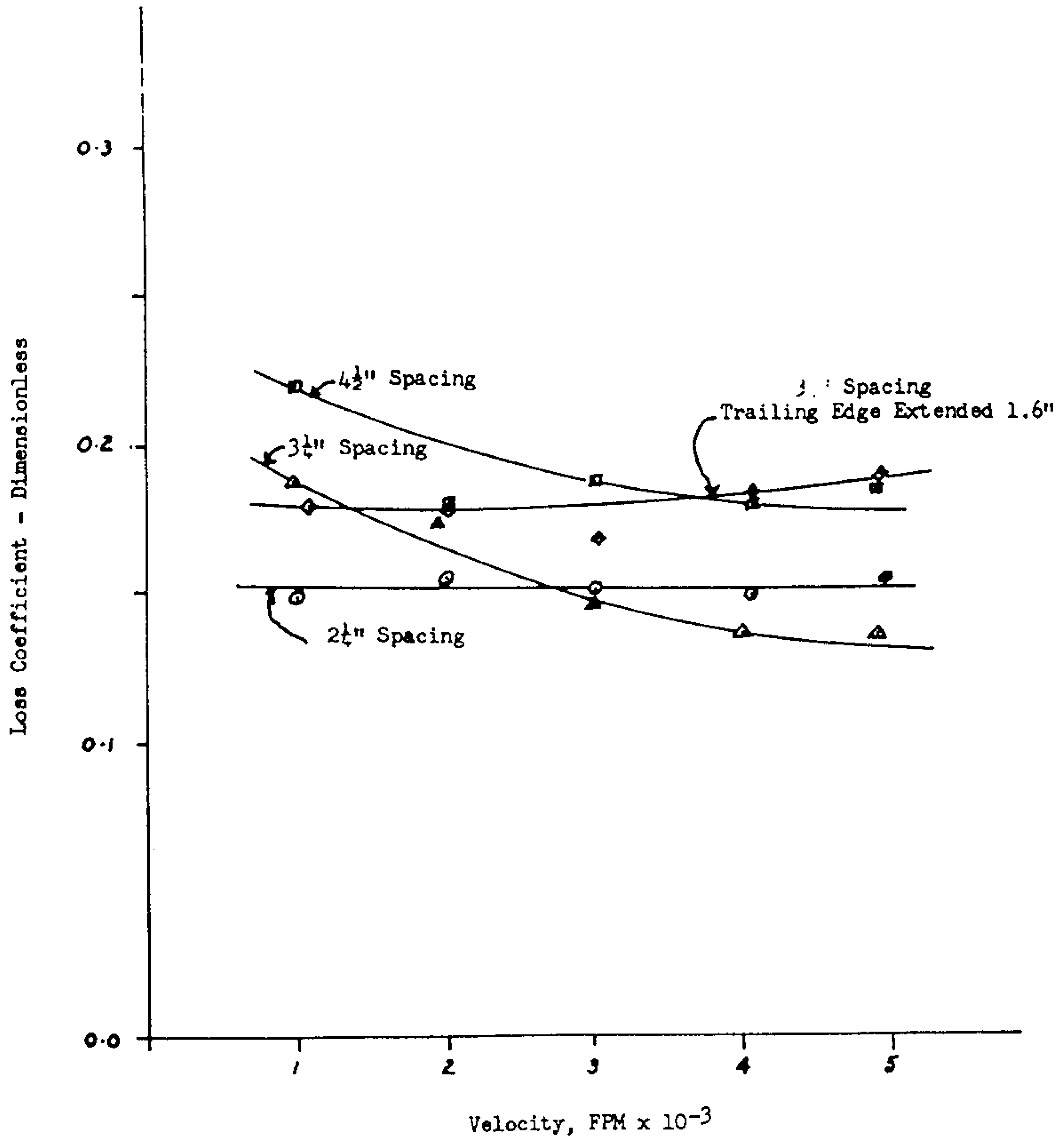


Fig. 24 Loss coefficients for single thickness vanes of 4.5" radius in 24" x 24" 90° elbow

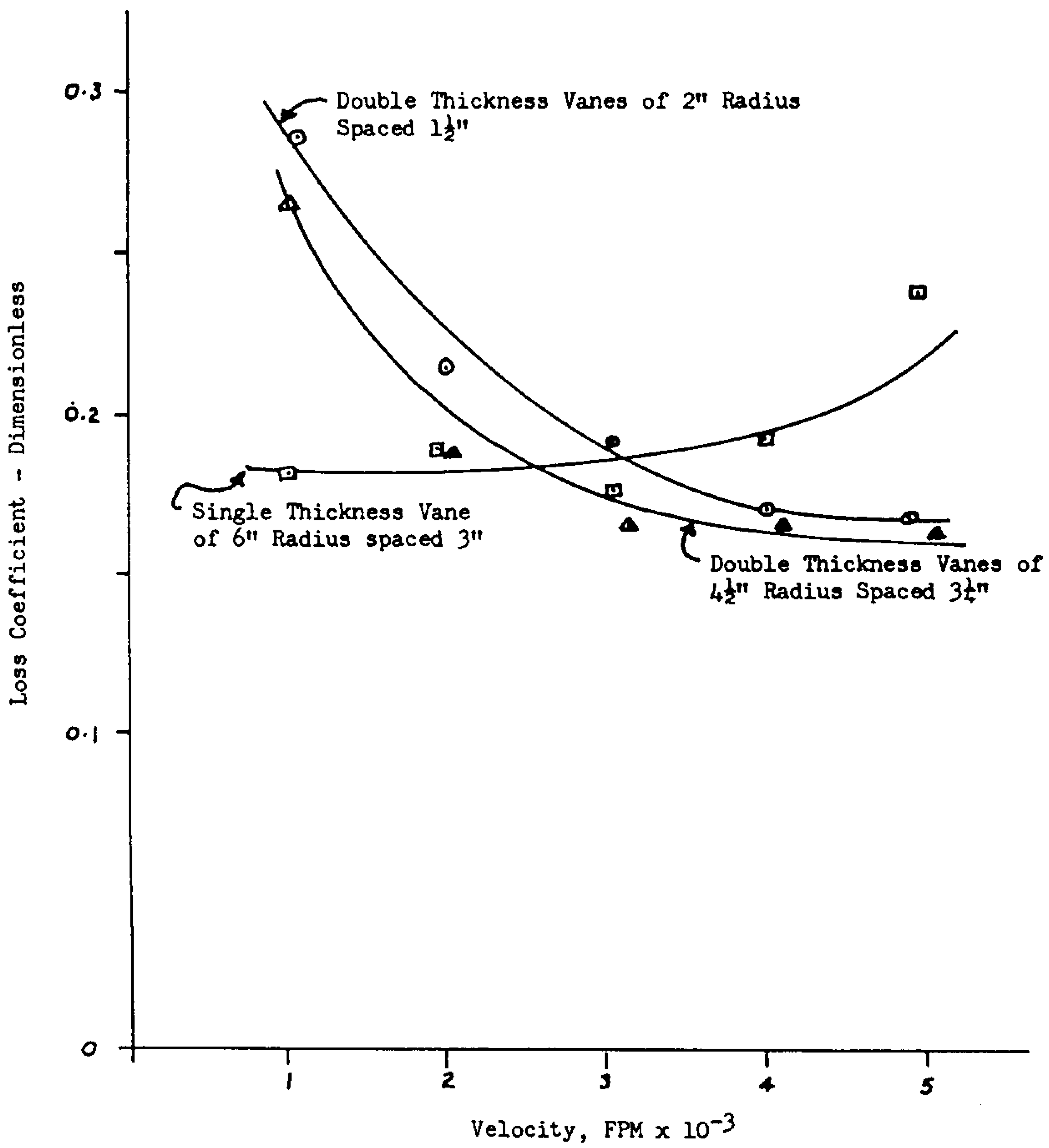


Fig. 25 Loss coefficients for two double thickness vanes and 6" radius single thickness vane in 24" x 24" 90° elbow

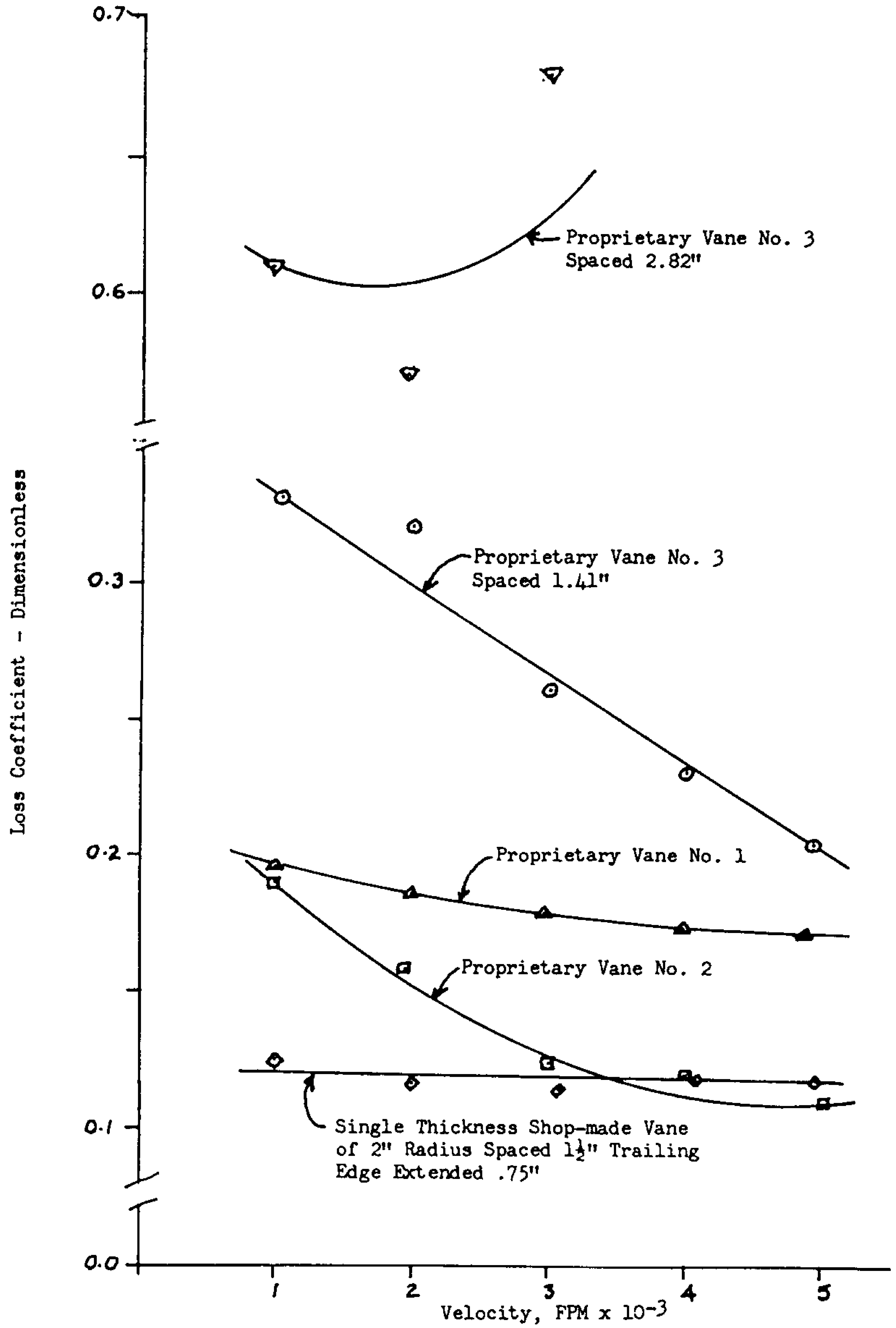


Fig. 26 Loss coefficients for three proprietary vanes and one shop-made vane in 24" x 24" 90° elbow

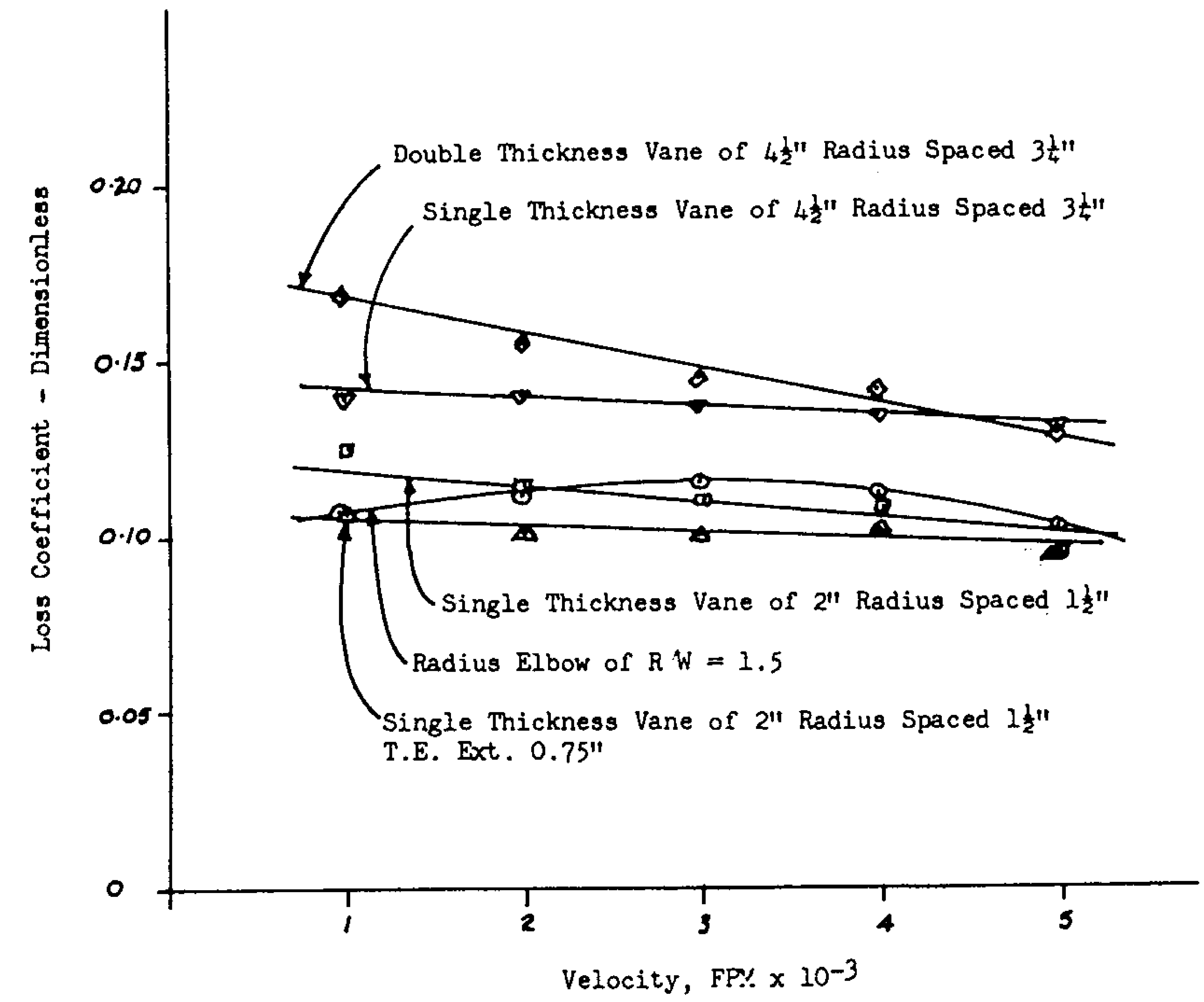


Fig. 27 Loss coeffients for four selected vanes in 24" x 24" 90° elbow and one 24" x 24" radius elbow as obtained from system for combined elbows

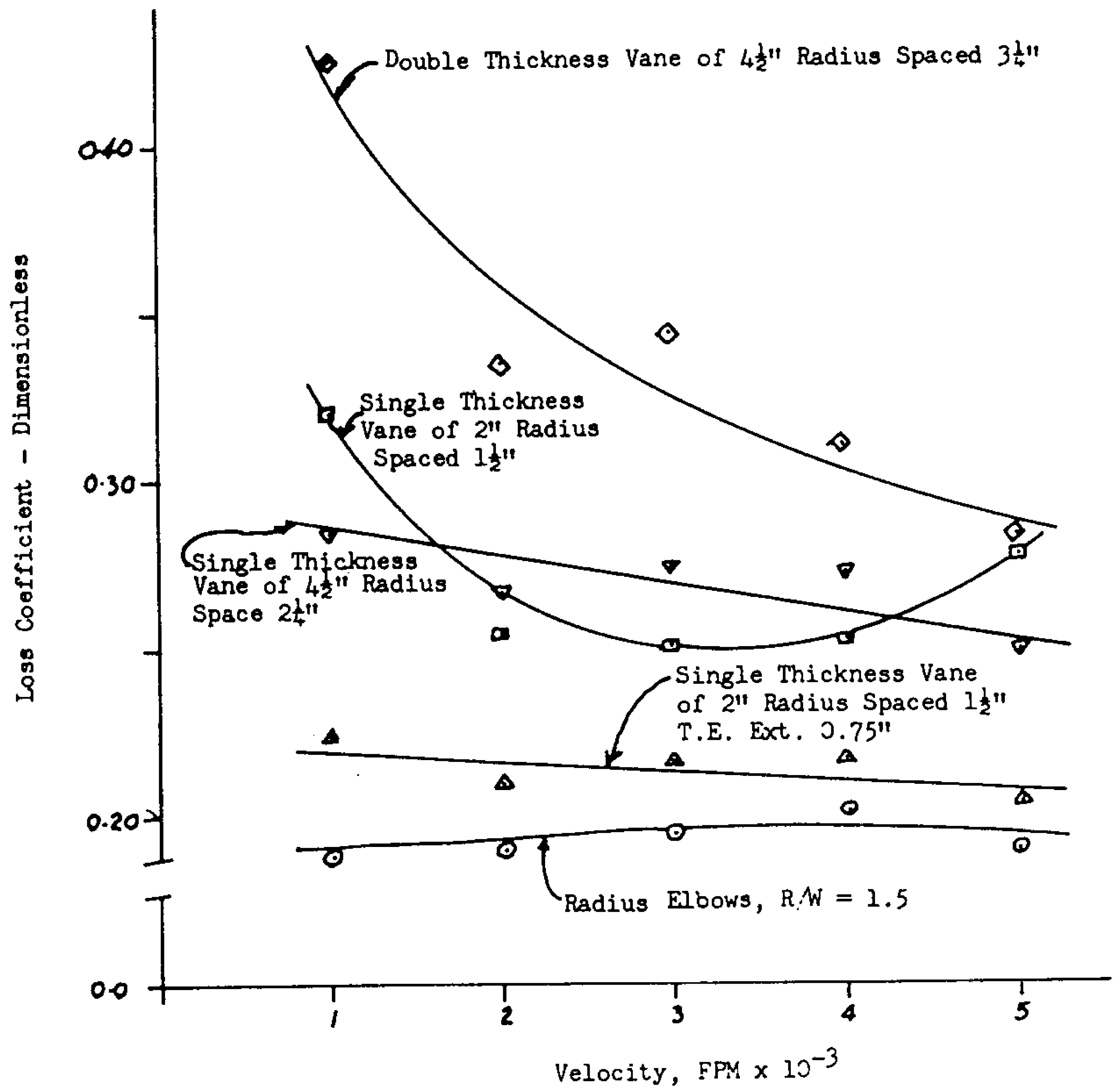


Fig. 28 Loss coefficients for four selected vanes and radius elbows in close coupled offset

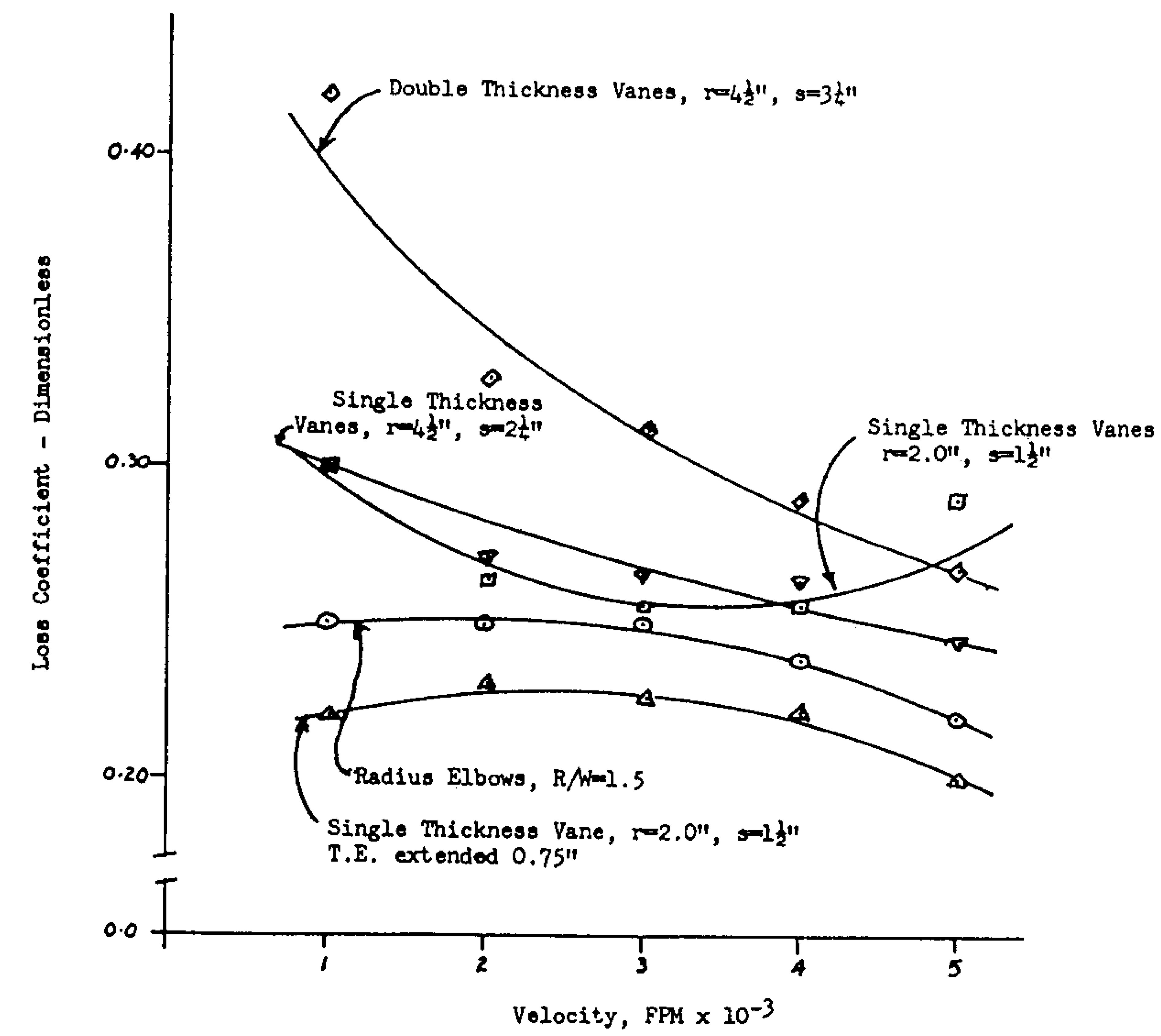


Fig. 29 Loss coefficients for four selected vanes and radius elbow in offset separated two diameters

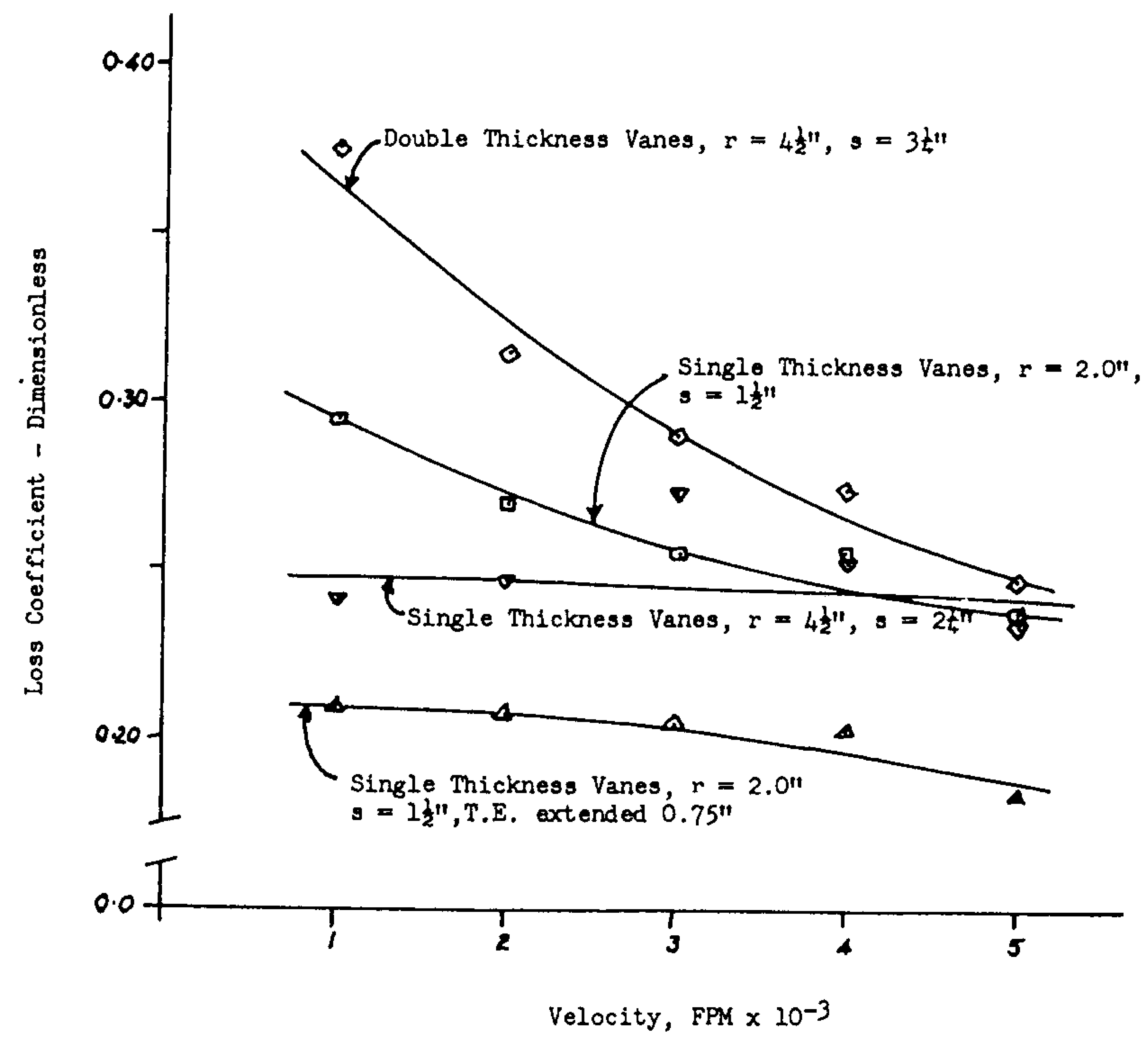


Fig. 30 Loss coefficients for four selected vanes in close-coupled "U" turn

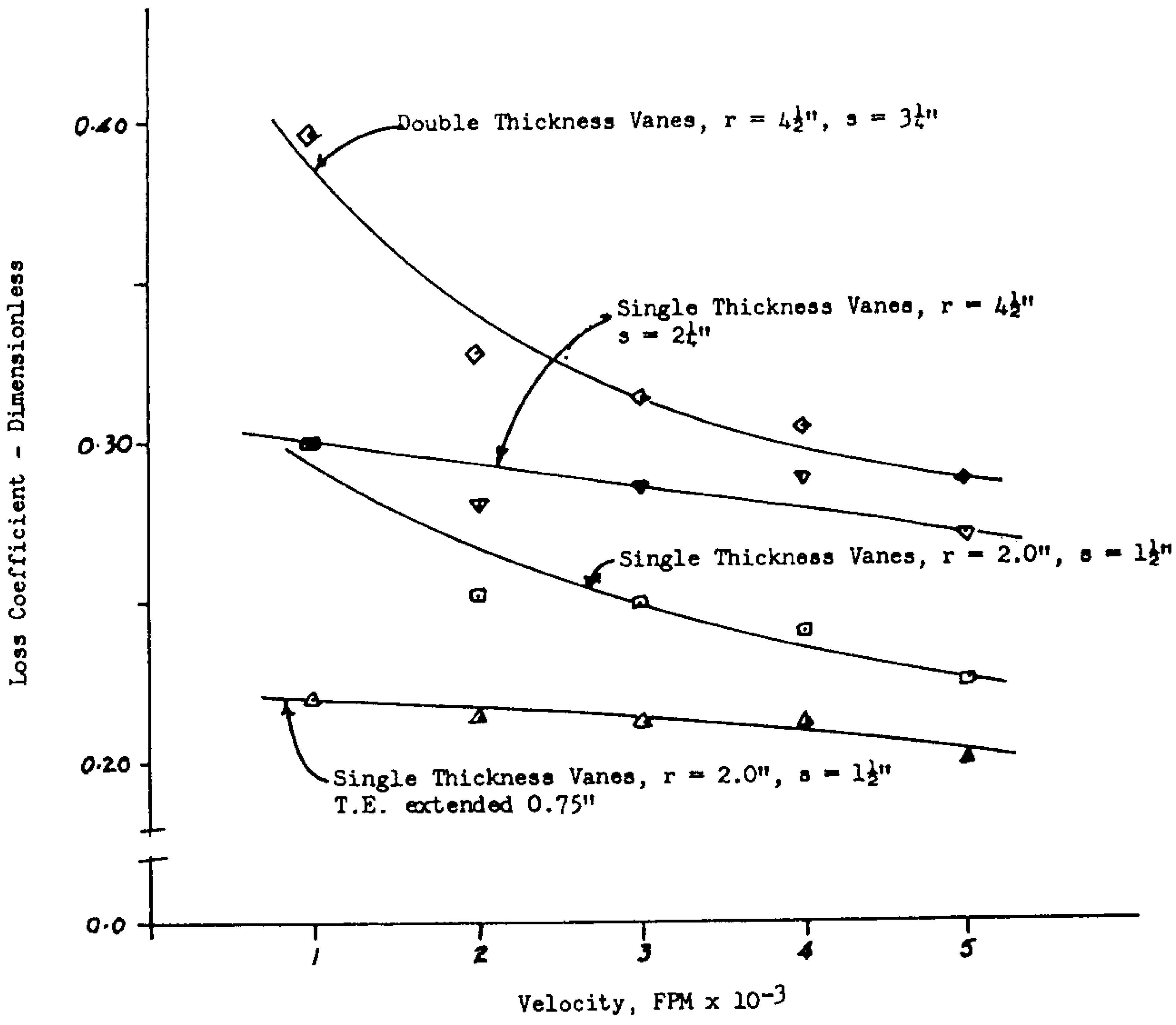


Fig. 31 Loss coefficients for four selected vanes in close-coupled change of plane

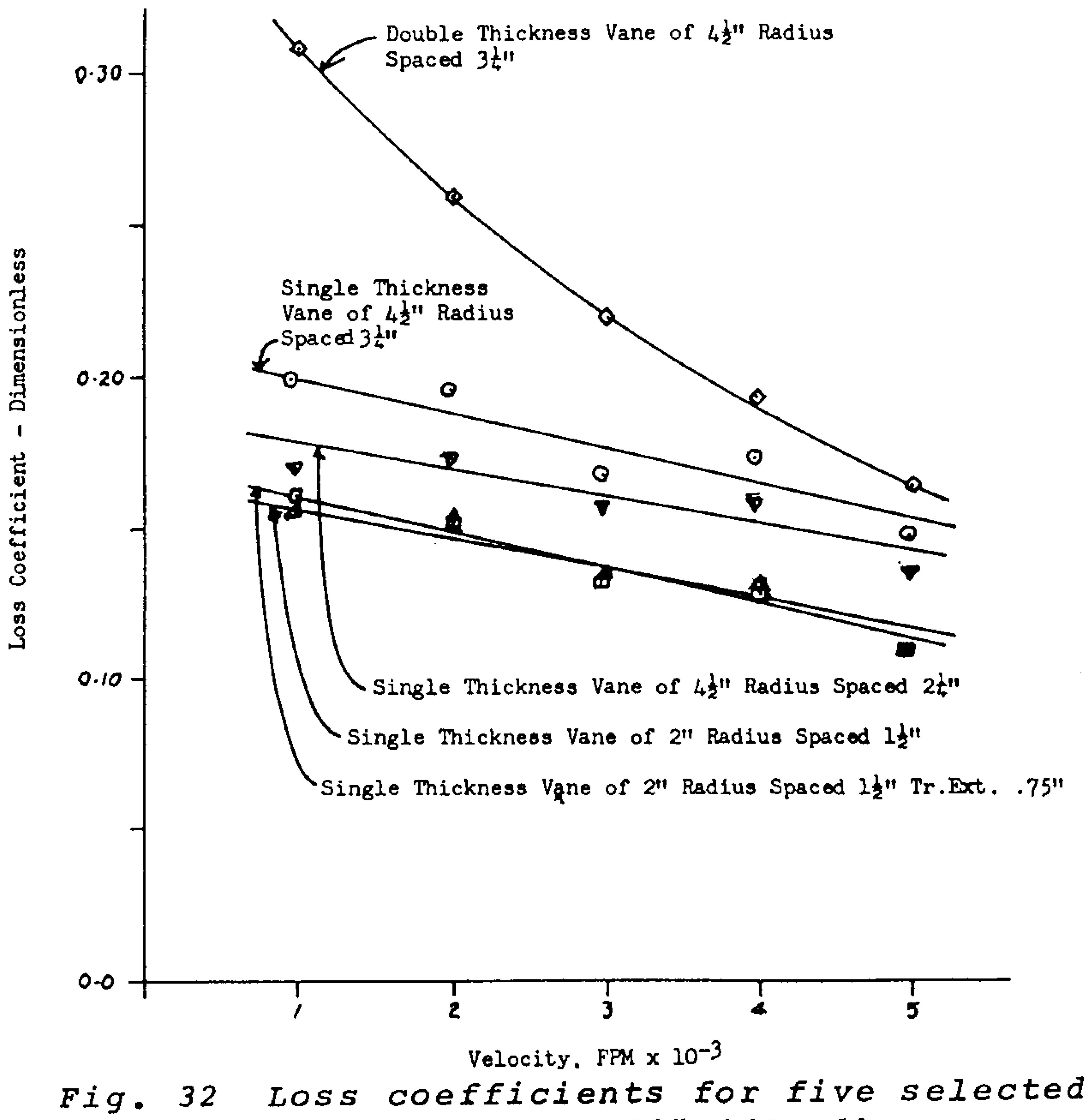


Fig. 32 Loss coefficients for five selected vanes in 48" x 12" 90° elbow

DISCUSSION

A. ELMAHDY (Carleton University, Ottawa, Ontario, Canada): Can you describe the velocity profile in the duct after having contraction and then diffusion in your apparatus? It seems to me that the turbulance level will be so high that it may affect the C_L coefficient.

J. MORRIS ROZELL: The system for testing 48 in. x 12 in. elbows in Figure 7, was the same as for testing 24 in. square elbows in combination (Figure 6). In Figure 6 the pitot probe rake was located at the upstream reading station. The straightener installed in this system at the upstream offset is described in the paper but not shown in the drawing. Since excellent velocity profiles were obtained after adjustment of the straightener, good profiles were expected in the Figure 7 system.

There were three test points in the system for testing flow conditions upstream from the test elbow. The first was the pitot probe rake section which was placed just upstream from the transition from 2 ft x 2 ft to 4 ft x 1 ft. Being so close to the 2 ft x 2 ft elbow, readings taken with it were not considered very reliable. However, a number of complete traverses were taken and the straightener was re-adjusted until good profiles were obtained.

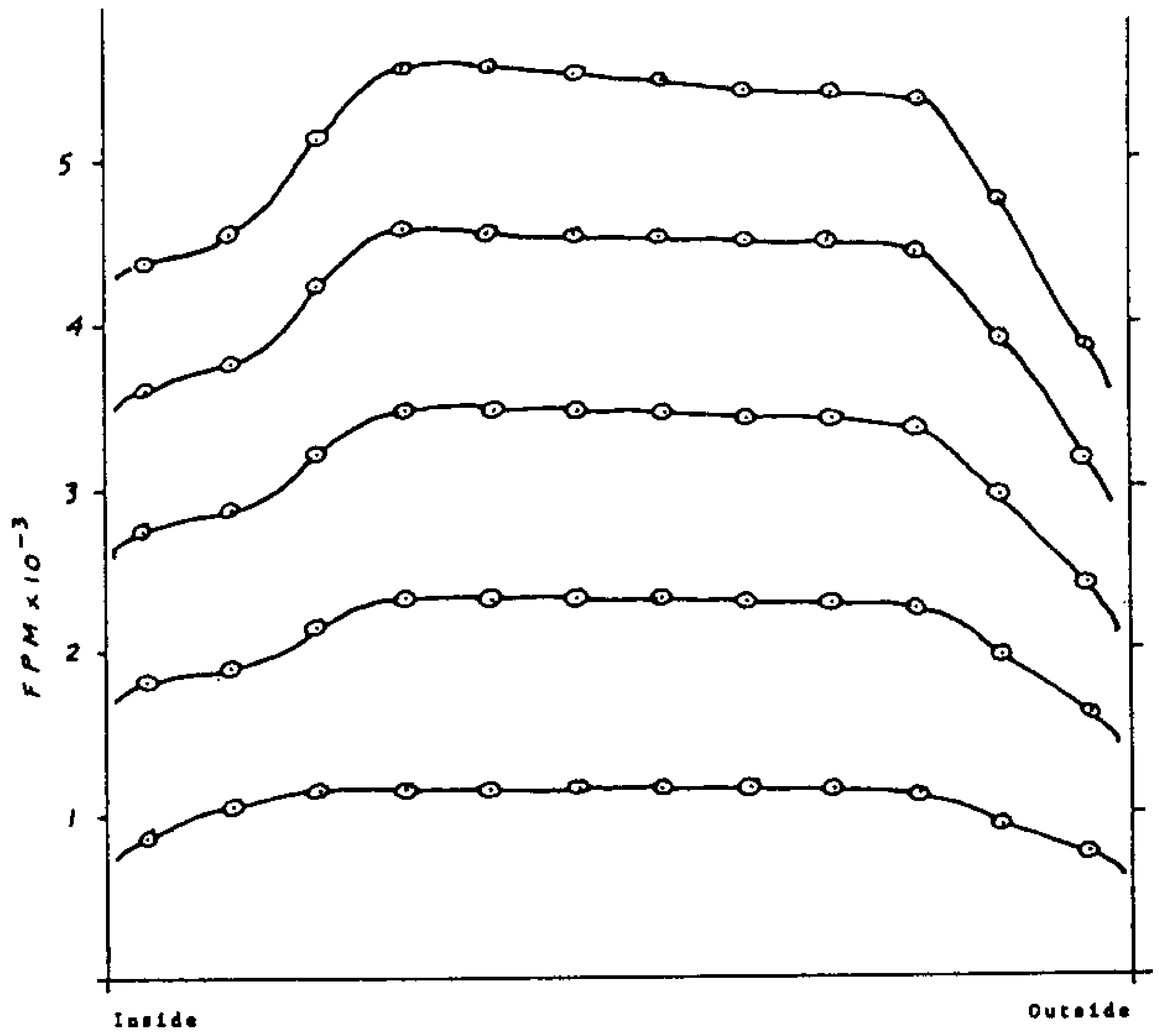
The second test point was the upstream reading station 2.6 diameters from the test elbow. As throughout the work, static pressure readings were taken at the centre of each side wall individually and their mean was used to compute Δp. These pressures were consistently higher on the inside (with respect to the bend) for the straight duct test than on the outside by approximately 10% in this system only. The difference was from 5 to 7% with the elbow inserted. Tabulated below are the readings obtained for the straight duct and the elbow with various vanes.

Comparison of static pressures in a straight 4 ft x 1 ft duct and with an elbow inserted for vanes as shown at the upstream reading station:

U	Str. duct		r=4.5 s=3.25D		r=4.5 s=3.25S		r=4.5 s=2.25S		r=2.0 s=1.5S		r=2.0 s=1.5S ext.	
	Inside	Outside	In.	Out.	In.	Out.	In.	Out.	In.	Out.	In.	Out.
1000	.067	.063	1.41	1.36	1.20	1.14	1.12	1.04	1.10	1.06	1.12	1.08
2000	2.35	2.10	4.72	4.50	4.2	4.0	4.05	3.80	3.85	3.65	3.85	3.6
3000	5.04	4.56	9.7	9.2	8.8	8.2	8.7	8.1	8.2	7.7	8.2	7.6
4000	8.60	7.71	16.1	15.1	15.5	14.5	15.0	14.0	14.0	13.0	14.0	13.1
5000	13.0	11.7	23.0	22.0	22.3	21.0	21.8	20.3	20.5	19.3	20.8	18.8

Pressures are in mm of alcohol. "S and D indicate single and double thickness."

The third test point was 12 in. upstream from the test elbow where an aerofoil pitot probe holder was located with the intention of showing the increase in velocity from inside to outside of the duct because of the long path on the outside. The figure below shows profiles for one set of vanes. Those for the other vanes tested in this elbow were very similar. These profiles show flow conditions opposite to those expected, and the velocities are lowest on the outside with respect to the bend. Reading points were at 4 in. intervals, starting 2 in. from the duct side wall. The prevailing pattern in these profiles was considered to result from the combined effect of the upstream elbows; the floor, ceiling and side wall at each side of the duct which continued for a distance of 10 to 12 in. inwards; also to increase flow pattern irregularities in the transition. There seemed to be no indication that there was separation of the air stream from the transition walls. The transition itself was an excellent fitting 4 ft long smoothly curved on all four sides. Data in this figure was not included in the paper because it did not show the expected pattern of flow.



Velocity profiles on centerline of 4 ft x 1 ft duct, 12 in. upstream from test elbow fitted with single thickness 2 in. radius extended vanes spaced 1.5 in.

E. C. LITSINGER (University of California, Lawrence Berkeley Laboratory, Berkeley, CA): Were acoustical, double wall turning vanes tested (e.g. "Acoustiturn") and what was the effect on C_L?

J. MORRIS ROZELL: No perforated or acoustically treated vanes were tested. Table 1 in the paper gives the approximate vane geometry of the proprietary vanes tested.