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DEVELOPMENT OF A FLOW MODIFIER FOR REDUCING THE REACTION FORCE OF FIREFIGHTING NOZZLES

C.K. Krishnakumar, S. Atallah and K.A. Borows

Risk & Industrial Safety Consultants, Inc.
268 Howard Street
Des Plaines, IL 60018-1906

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SUMMARY

The purpose of this study was to develop a device that can be inserted upstream of conventional water firefighting nozzles in order to reduce their reaction force. The device was intended to create a resistance to flow that may be controlled by the firefighter holding the nozzle. Earlier laboratory tests carried out under Phase I of this study had shown that a flow modifier with two drag generators spaced a short distance apart could produce a significant reduction in the nozzle reaction force, provided that adequate water pressure was available at the pump. The reduction in reaction force could be achieved without destroying the stream solidity or throw range.

A laboratory test program was first carried out in order to examine the effects of modifier design variables on the reaction force. By using the results of these tests, a Model I prototype of the flow modifier was then constructed out of brass and tested successfully at Underwriters Laboratories hydraulic laboratory. A Model II prototype was then constructed out of a much lighter polymeric material. Only the drag generators were made of brass. This model was demonstrated successfully at Tyndall AFB using Tyndall's fire pumper and fire department personnel. Difficulty in adjusting the spacing between the drag generators at water pressures above 90 psi was encountered. A Model III prototype was then fabricated and delivered to Tyndall AFB with minor changes in the design in order to alleviate this problem.

PREFACE

This report is an account of a study carried out by Risk & Industrial Safety Consultants, Inc. (RISC) under a SBIR Phase II USAF contract No. F08635-90-C-0065. The study was sponsored by Headquarters Air Force Engineering and Services Center, Directorate of Engineering and Services Laboratory (HQ AFESC/RD), Tyndall Air Force Base, Florida 32403-6001. Mr. Douglas Schwartz, HQ/AFESC/RDCF, was project officer. The period of performance was March 21, 1990 - May 7, 1991.

RISC wishes to acknowledge the assistance provided by two subcontractors: Underwriters Laboratories for carrying out the large scale tests, and W-D Tool Engineering Inc. for constructing the flow modifiers. Mr. R. Vaitys assisted RISC in carrying out a stress analysis of the proposed device and provided his consulting services as a design mechanical engineer. Mr. C. Spina, a machinist and model builder assisted in the construction of the laboratory test apparatus.

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This technical report has been reviewed and is approved for publication.

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Douglas Schwartz
Project Engineer

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INTRODUCTION

A. BACKGROUND

Hand-held nozzles which are used by the USAF to discharge water for firefighting purposes can be classified as solid stream and spray nozzles. Solid stream nozzles are used to deliver extreme range and penetration, provide large volumes of water and cool sensitive objects such as ordnance and fuel tanks. On the other hand, spray nozzles are used to attenuate thermal radiation from a fire and thus protect firefighters during their approach and attack. Spray nozzles are also utilized to disperse hazardous vapor clouds. Most commercial nozzles are capable of discharging both types of spray.

Solid stream nozzles are available in diameters ranging from 0.25 inch to 2.5 inches. Nozzles of up to about 1.25 inch in diameter are considered adequate for hand-line operation by individual firefighters. The nozzle reaction force or "kickback" from larger sized nozzles operating at typical nozzle pressures of 45 to 60 psig is too great for a single firefighter. Not only is the firefighter's personal safety compromised when handling a high-reaction nozzle, but fire suppression efficiency is diminished as well. Firefighting becomes more difficult and complex when two or more firefighters are committed to stabilizing a high-reaction nozzle while directing the water stream at a fire. Thus, it would be highly desirable if the nozzle reaction force at the hose outlet could be reduced by a design modification without sacrificing nozzle performance.

USAF fire trucks have fire pumps capable of boosting water pressures to about 300 psig. However, a large fraction of this pressure is currently being wasted and reduced to about 90 psig as friction loss in a narrow-diameter hose, typically 1.75 inches in diameter. If the kickback force could be disproportionately reduced through a design modification, the USAF would be able to utilize higher nozzle pressures, with correspondingly greater water flow rates and throw ranges. Such a design modification would allow a more efficient deployment of firefighting personnel, greatly strengthen their attack capabilities and enhance their safety.

Department of Defense (DOD) fire services and governmental agencies, such as NASA and the Forestry Service, and fire departments serving urban areas, industrial facilities and airports, will also benefit from the development of a device that could reduce nozzle kickback force. Acknowledgement of this problem by major urban fire departments (such as those in Los Angeles, San Francisco and Chicago) is evident from an article which describes several projects undertaken by the fire departments of these cities (Reference 1). The objective of these studies was to achieve larger nozzle discharge rates without increasing the fire flow stress on firefighters through modifications of nozzle design. Although the results were not conclusive, and no theoretical analyses were given, the article concluded that low-pressure fog nozzles had considerably less reaction force and are less stressful to firefighters than solid stream nozzles.

Under Phase I of an SBIR research contract, Risk & Industrial Safety Consultants, Inc., (RISC) examined the feasibility of reducing the reaction force of a solid stream fire nozzle. This was successfully accomplished without altering the stream solidity or the water throw range by inserting a flow modifier upstream of the nozzle. Basically, these modifiers were drag generators, which utilized a fraction of the large pressure drop (currently being wasted as hose friction) to generate a force opposing the nozzle reaction force.

A laboratory apparatus for measuring the nozzle reaction force was designed and constructed. Flow modifiers of three different designs were tested upstream of an experimental Plexiglas nozzle and a commercial nozzle. The tests were designed to compare the kickback for each of the nozzles when used with and without the flow modifiers at different pump pressures and water flow rates.

Since the primary purpose of Phase I was to prove the concept, the modifiers developed during Phase I were not necessarily of optimum design. They had a fixed structure so that a firefighter would have no control on water flow rate should the pressure of the water supply change. It was thus desirable to optimize the flow modifiers in Phase II of the research program and to test and demonstrate the feasibility of an adjustable prototype in full scale tests.

B. OBJECTIVES

The objectives of Phase II were the following:

1. Determine by laboratory tests the optimal design of a prototype adjustable flow modifier which would be inserted between a commercial fire nozzle and a fire hose.

2. Establish the performance characteristics of the flow modifier by conducting full-scale tests.

3. Demonstrate the performance of the final prototype flow modifier by conducting field tests at Tyndall Air Force Base.

C. APPROACH

To achieve the objectives of Phase II, the following program was undertaken:

1. Several flow modifiers were designed and fabricated.

2. Laboratory tests were conducted to evaluate the effect of design variables on flow modifier performance.

3. Based on the results of the laboratory tests, an optimized prototype variable resistance flow modifier was designed and fabricated.

4. Full-scale tests were performed using the optimized flow modifier together with selected commercial fire nozzles. The results were analyzed and the design of the modifier was refined further.

5. Using the results of the full-scale tests, a second, lighter prototype flow modifier was fabricated and tested in the laboratory .

6. Demonstration field tests were conducted at Tyndall Air Force Base. The demonstrations involved USAF fire trucks and personnel. Additional changes in the material of the modifier were found necessary.

7. Two additional flow modifiers of the final design were fabricated and delivered to USAF.

D. REPORT ORGANIZATION

This report summarizes the results of Phase II. In Section II, the theoretical basis for the design of the flow modifier is discussed. The preliminary design concept, the laboratory apparatus and the initial laboratory tests of the various flow modifier configurations are described in Section III. In Section IV, the first prototype design and the results of full-scale tests at Underwriters Laboratories are presented. A description of the final design and the results of the demonstration tests at Tyndall Air Force Base are discussed in Section V. RISC's conclusions and recommendations are summarized in Section VI. References are listed in Section VII. Appendices include the experimental data collected at RISC's laboratory and at Underwriters Laboratories as well as a stress analysis of the flow modifier.

SECTION II

THEORETICAL CONSIDERATIONS

A. THEORETICAL BACKGROUND

The following equations are available in the literature for the calculation of the nozzle reaction force (References 2, 3).

The reaction force of a solid stream nozzle is given by:

$$F_R \quad = \quad 1.5 \times d^2 \times p \qquad (1)$$

where

F_R = Nozzle reaction force (lb_f)

d = Nozzle tip diameter (inch)

p = Nozzle pressure (psig)

Equation (2) gives the momentum flux of the exiting water:

$$M_e \quad = \quad \dot{m} \times V_e/g_c = \rho \times A_e \times V_e/g_c \qquad (2)$$

where

M_e = Momentum flux of exiting water (lb_f)

$\dot{m}$ = Mass flow rate of water (lb_m/s)

V_e = Exit velocity (ft/s)

ρ = Density of water (lb_m/cu ft)

A_e = Cross-sectional area of opening (sq ft)

= $(\pi/4) \times (d^2/144)$

g_c = Conversion factor (32.2 poundal/lb_f)

The exit velocity through the nozzle, V_e, may be related to the pressure drop across the nozzle by the equation:

$$V_e \quad = \quad C_n \sqrt{2g_c\ (-\Delta P/\rho)} \qquad (3)$$

where

C_n = Discharge coefficient of the nozzle ≈ 0.98

$-\Delta P$ = Pressure drop across the nozzle (lb_f/sq ft) = 144p

Substitution of Equation (3) into Equation (2) yields:

$$M_e \quad = \quad 2C_n^2\ (-\Delta P)\ A_e \tag{4}$$

Substituting a value of 0.98 for C_n, and appropriate expressions for $-\Delta P$ and A_e gives:

$$M_e \quad = \quad 2\ (0.98)^2 (144p)\ (\pi/4)\ x\ (d^2/144)$$

$$= \quad 1.5\ x\ d^2\ x\ p$$

which is identical to Equation (1).

According to References 2 and 3, the reaction force of a spray nozzle is given by:

$$F_R \quad = \quad 0.0505\ x\ Q\ x\ \sqrt{p} \tag{5}$$

where

F_R = Reaction force (lb_f)

Q = Water flow rate (gpm)

p = Nozzle pressure (psig)

Equations (1) and (5) are identical if the flow areas of the solid stream and the spray nozzle are equivalent.

Substituting for Q into Equation (5) gives:

$$Q \quad = \quad V_e\ x\ A_e\ x\ 7.48\ (gal/cu\ ft)\ x\ 60\ (s/min)$$

From Equation (3):

$$V_e \quad = \quad 0.98\ \sqrt{2\ x\ 32.2\ (144p/62.4)} = 11.95\ \sqrt{p}$$

$$A_e \quad = \quad (\pi/4)\ x\ (d^2/144) = 0.00545\ d^2$$

$$Q \quad = \quad (11.95\ \sqrt{p})(0.00545\ d^2)\ (7.48\ x\ 60)$$

$$= \quad 29.23\ x\ d^2\ x\ \sqrt{p}$$

Substituting into Equation (5) yields:

$$F_R \quad = \quad 0.0505\ x\ (29.23\ x\ d^2\ x\ \sqrt{p})\ x\ \sqrt{p}$$

$$= \quad 1.5\ x\ d^2\ x\ p$$

which, again, is identical to Equation (1).

Therefore, for both solid stream and spray nozzles:

$$F_R \quad = \quad 1.5\ x\ d^2\ x\ p = \dot{m}\ V_e/g_c \tag{6}$$

This equation shows that, for a given flow rate and exit velocity (which implies a given nozzle pressure and flow area), the nozzle reaction force has a constant value. This is true only for a nozzle attached to a rigid pipe. The reaction force of a nozzle attached to a flexible hose depends on the hose tension, the nozzle pressure, the exit velocity and the water flow rate. The hose tension, in turn, may depend on the hose diameter, the unsupported length of the hose, the approach configuration and the water pressure. This dependence is discussed in the following section.

B. REACTION FORCE OF A NOZZLE ATTACHED TO A FLEXIBLE HOSE

Figure 1 shows the forces acting on a control volume which encompasses the firefighting nozzle held in a realistic position by the firefighter. By applying Newton's Second Law of Motion to this control volume, one gets Equation (7):

F_x = Momentum flux out - momentum flux in

$$(P_i\ A_i)_x + R_x - (T_x + W_x) = (\dot{m}\ V_e)_x - (\dot{m}\ V_i)_x \qquad (7)$$

where

F_x = Sum of x-components of all external forces acting on the control volume

P_i = Gauge water pressure at inlet of control volume

A_i = Cross-sectional area at inlet

A_e = Cross-sectional area at outlet

T_x = x-component of pull exerted by the hose on the control volume

R_x = x-component of force exerted by the firefighter (The opposite of the nozzle reaction force)

W_x = x-component of the weight of the nozzle

$\dot{m}$ = Mass flow rate of water

Noting that W_x is relatively small and that V_i is much less than V_e, Equation (7) may be approximated as follows:

$$R_x = (\dot{m}\ V_e)_x - (P_i\ A_i - T)_x \qquad (8)$$

Equation (8) clearly shows that increasing the value of the quantity $(P_i\ A_i - T)_x$ would reduce the nozzle reaction force.

When the nozzle is attached to a rigid pipe, the hose tension, T, will be equal to the pressure force, P_iA_i. In this case:

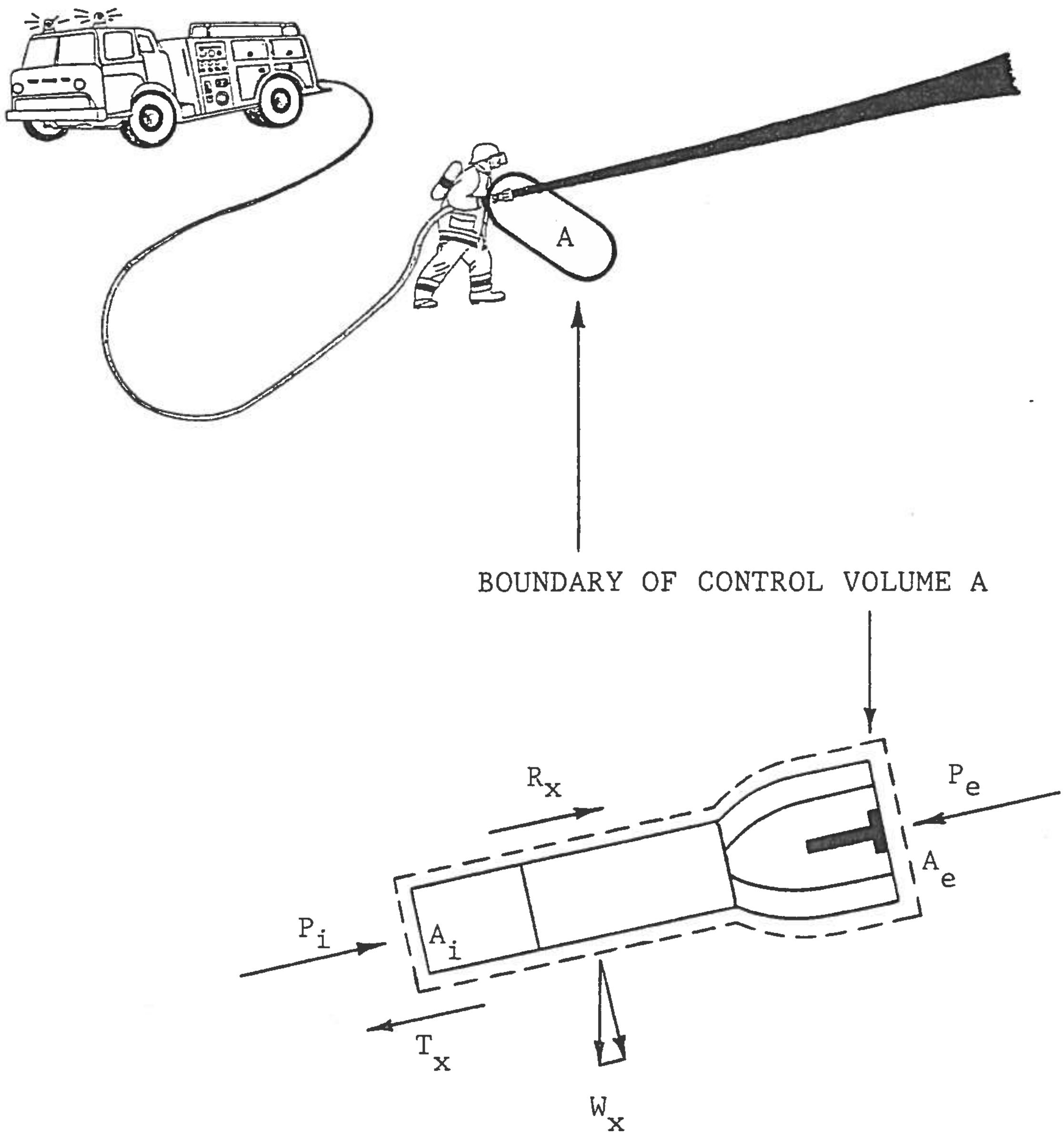


FIGURE 1: ANALYSIS OF NOZZLE REACTION FORCE

$$R_x = F_R = (\dot{m} V_e)_x \tag{9}$$

which is identical to Equation (6).

C. THE CONCEPT OF THE FLOW MODIFIER

The flow modifier is designed to increase the value of the quantity $(P_1 A_1 - T)_x$ for any given value of water flow rate. A pressure drop is created across the flow modifier, which is situated between the discharge end of the hose and the nozzle inlet. The value of the quantity $(P_1 A_1)_x$, for a control volume taken around the flow modifier and the nozzle assembly, will be larger than a control volume around the nozzle alone. The hose tension, T_x, will also increase with an increase in pressure P_1. However, an increase in the hose tension, T_x, will be smaller than an increase in the pressure force, $P_1 A_1$, provided there is some slack in the hose. Therefore, the quantity $(P_1 A_1)_x$ would increase as the value of P_1 increases, again, provided there is some slack in the hose.

Any flow obstacle inserted between the hose and the nozzle, such as a valve, would tend to increase the value of the quantity $(P_1 A_1 - T)_x$. However, to be advantageous, the drag generator should achieve this goal without affecting the stream solidity or the throw range of the discharge stream. It should also allow suspended matter to flow through without obstructing the flow.

SECTION III

LABORATORY PROGRAM

A. DESIGN CONCEPT OF THE FLOW MODIFIER

The firefighter must be able to vary the flow resistance of the flow modifier, so as to adapt to the available water pressure. A flow modifier with a constant value of flow resistance, such as the flow modifiers used in Phase I, may perform satisfactorily at the design value of the available water pressure. However, if the water pressure were to drop, the water flow rate would also drop. A nonadjustable flow modifier would not allow the operator to reduce the flow resistance in order to restore the water flow rate to its previous value. If the available water pressure should increase, it would not be possible to control the nozzle reaction force, since an increase in the flow resistance of the flow modifier would then be required. It was therefore important to redesign the flow modifier to allow variations in its flow resistance during actual use.

The following criteria governed the design of the flow modifiers used in the laboratory tests carried out during Phase II:

- o Optimal performance: to give the largest water flow rate for a prescribed value of nozzle reaction force.
- o Ease of attachment: to allow simple insertion between a commercial fire nozzle and a fire hose.
- o External adjustment: to allow for control and variation of the nozzle reaction force.
- o Clog-free operation: to permit the passage of debris.
- o Compactness: the weight and the overall length of the modifier was to be as small as practicable.
- o Optimal design: the design of the modifier was to be simple and its materials of construction light, durable and inexpensive.

The basic design principle of the flow modifier is shown in Figure 2. The design consists of two drag generators housed within a cylinder. The upstream drag generator has several circular openings, while the downstream drag generator has one centrally-located circular opening. The distance between the drag generators can be adjusted externally. As the drag generators approach each other, the total drag force increases. When the separation distance between them increases, the drag force decreases.

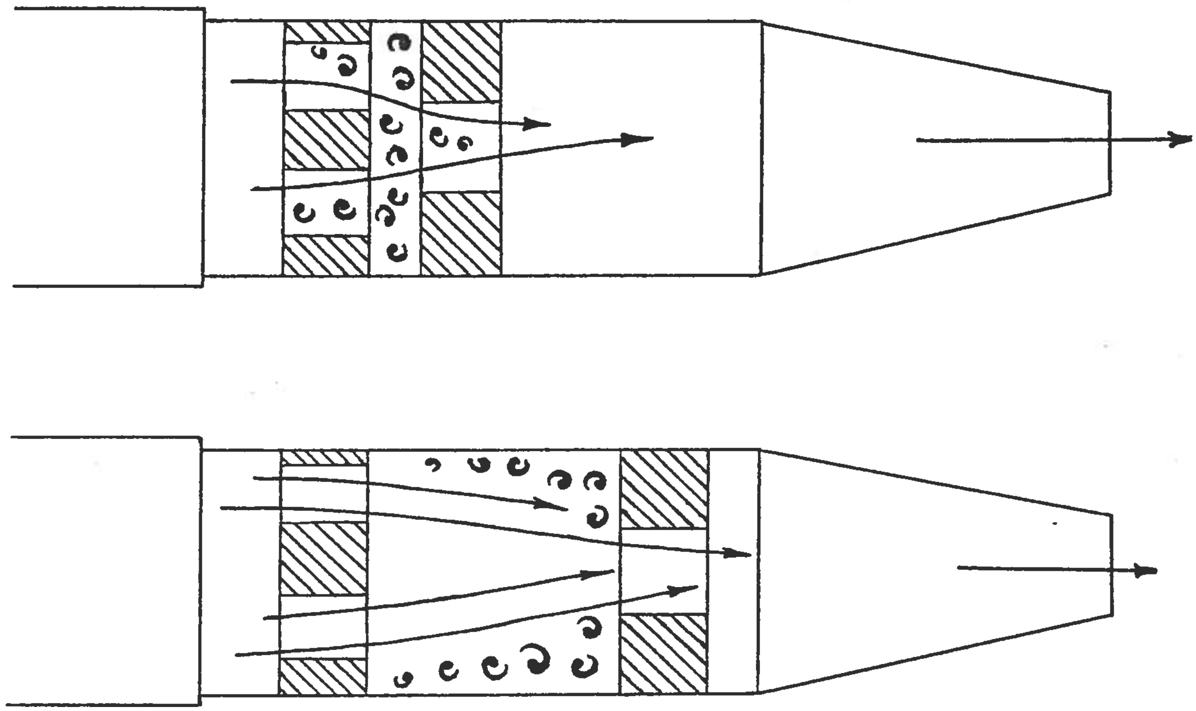


FIGURE 2: DESIGN PRINCIPLE OF THE FLOW MODIFIER

B. DESIGN AND FABRICATION OF THE DRAG GENERATORS

Four configurations of drag generators were designed and fabricated for use in the laboratory tests. The experimental flow modifiers consisted of two drag generators enclosed by a cylindrical housing shown in Figure 3.

The upstream drag generator had three, 0.375-inch diameter holes, which were displaced from the center. The downstream drag generator had one axial hole, 0.75-inch in diameter. Both drag generators would create pressure forces that oppose the nozzle reaction force. The goal was to regulate the nozzle reaction force by varying the distance between the drag generators. For the laboratory device, spacer rings of various thicknesses were placed between the drag generators in order to vary the separating distance between them and thus, the resistance to flow.

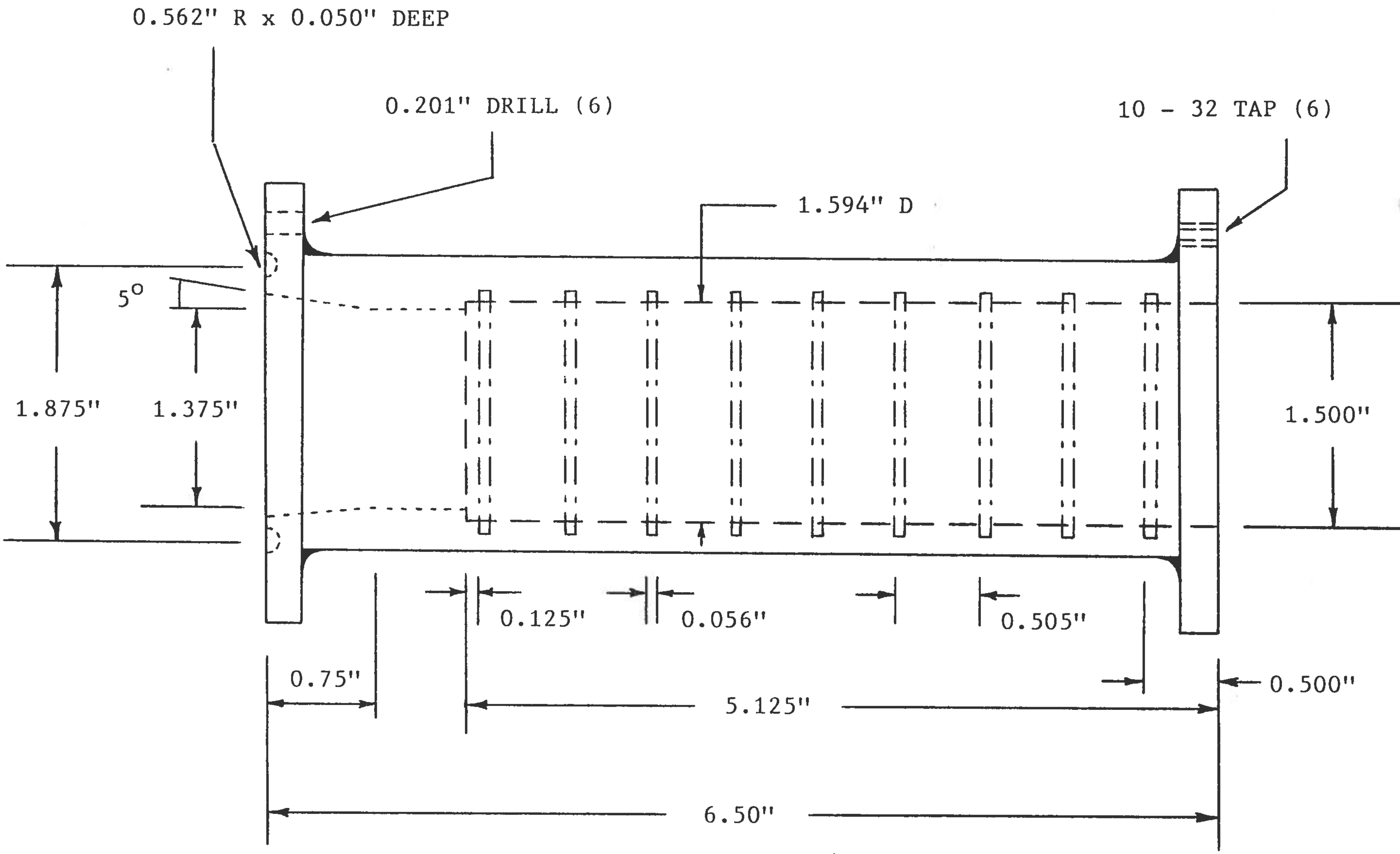


FIGURE 3: LABORATORY FLOW MODIFIER HOUSING

Figure 4 shows the profile of the downstream drag generator, which was used with all four of the upstream drag generators in the initial laboratory tests. Figure 5 shows the profiles of the four upstream drag generators used in these laboratory tests. Figure 5A shows Profile A, with a conical front surface and a flat back surface. Figure 5B shows Profile B, with a spherical front surface and a flat back surface. Figure 5C shows Profile C, with a flat front surface and a flat back surface. Figure 5D shows Profile D, with a flat front surface and a concave back surface. All the drag generators were made out of brass, while the cylindrical housing was constructed of stainless steel.

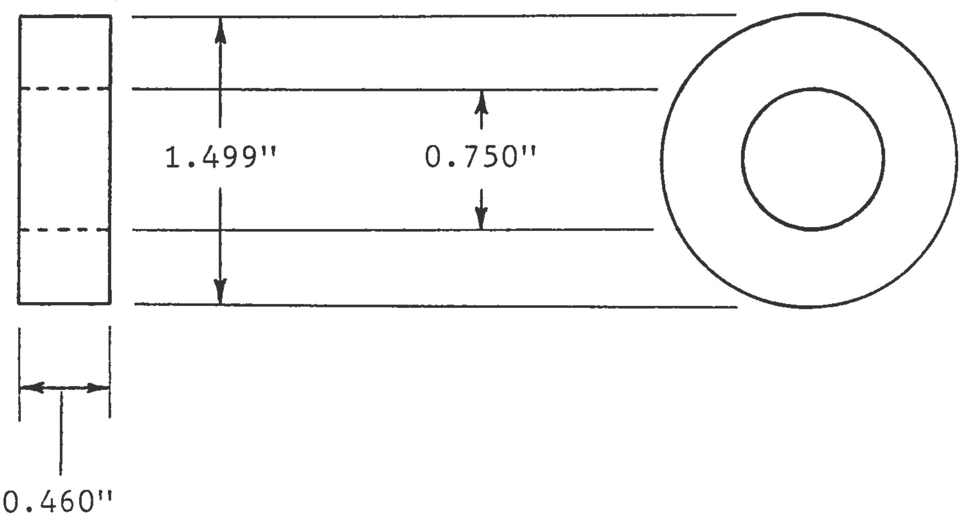


FIGURE 4: DOWNSTREAM DRAG GENERATOR

C. LABORATORY APPARATUS

The laboratory apparatus used during Phase II is shown in Figure 6. In many respects, it was similar to that used during Phase I. While the objective of the Phase I tests was to verify concept feasibility, the objective of the Phase II tests was to optimize the design of the flow modifiers. A higher degree of accuracy of test results was also required. Further, it was considered necessary to obtain the data with a more realistic fire hose and nozzle approach configuration.

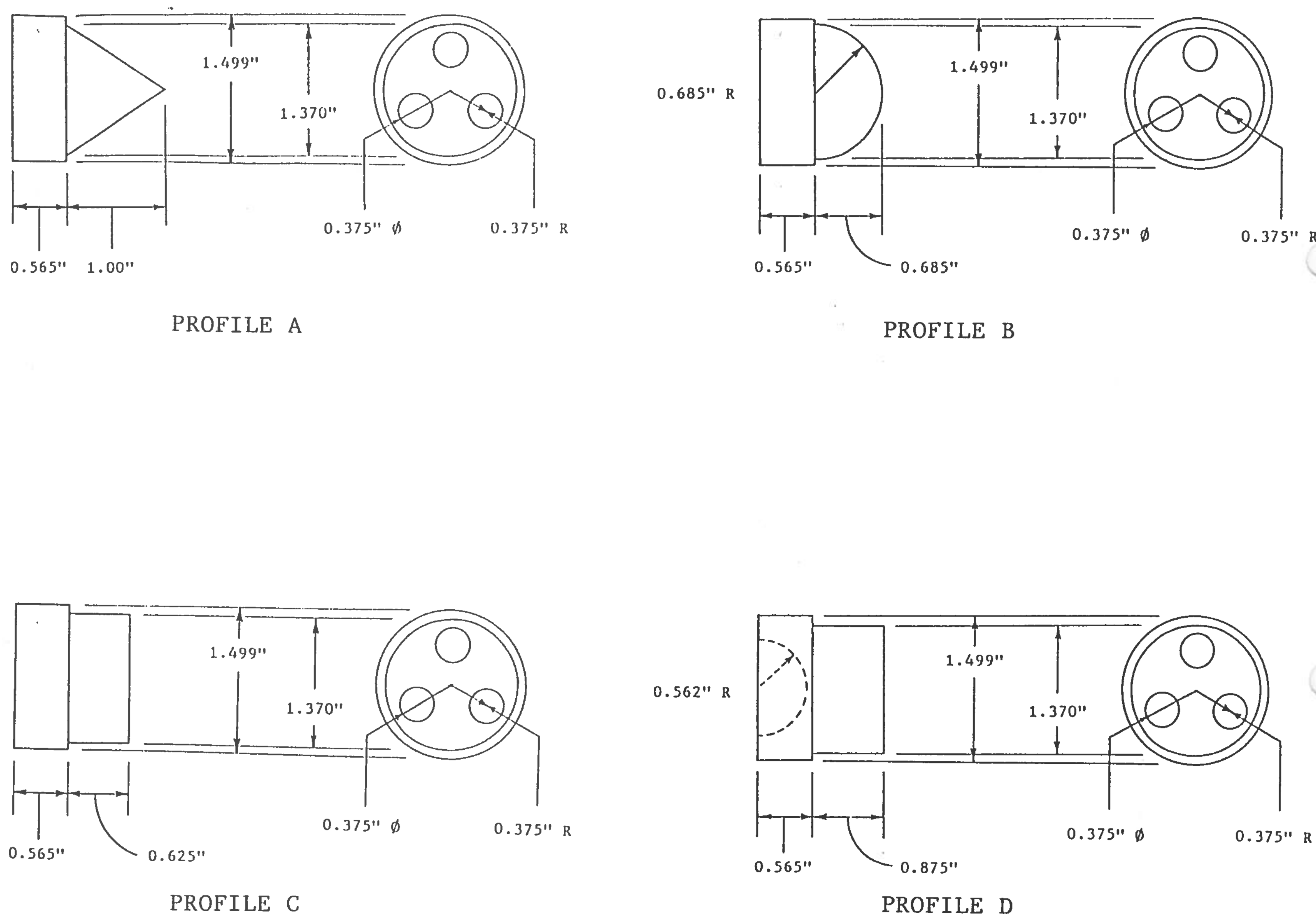


FIGURE 5: LABORATORY UPSTREAM DRAG GENERATOR PROFILES

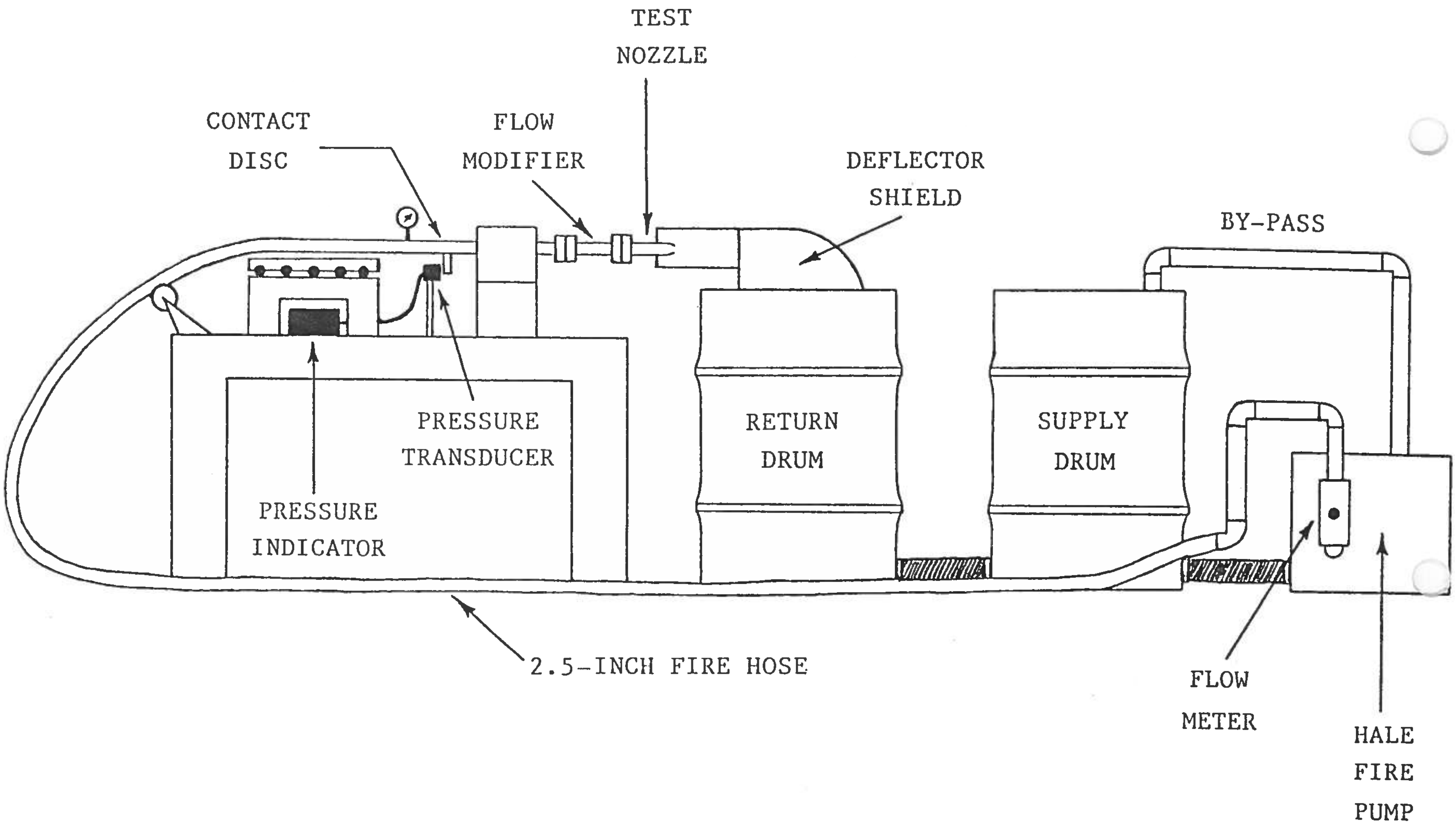


FIGURE 6: LABORATORY TEST APPARATUS

A Hale fire pump (Model 25FB-B25), driven by an 11-horsepower Briggs and Stratton gasoline engine, supplied water to the system. Water from the pump was discharged through an Omega rotameter (Model FL-75M) mounted near the pump outlet. From the rotameter, water flowed through a 2.5-inch fire hose, through the flow modifier-nozzle assembly and was discharged against a deflector shield, falling into an 85-gallon return drum. The flow modifier and nozzle assembly were positioned so that the free end projected slightly into the hood of the deflector shield to prevent splashing. A short pipe segment at the outlet end of the fire hose passed through a horizontal linear bearing, which permitted unobstructed longitudinal movement of the entire nozzle assembly. The fire pump recycled the water from another 85-gallon supply drum which was connected in series to the return drum. The water flow rate was regulated by adjusting the engine speed and/or the pump discharge valve. Finer control of the water flow rate was obtained by adjusting the valve on the bypass line.

The entire nozzle assembly was supported on rollers which moved on a polished, horizontal steel plate. The nozzle reaction force was transmitted to a piston, which then applied a uniform fluid pressure to the surface of an Omega pressure transducer (Model PX102-100SV), as shown in Figure 7. The transducer had a diameter of 0.75-inch which corresponds to an area of .442-square inch. The force was read on an Omega digital pressure indicator (Model DP350). A pressure gauge, which was attached to the pipe leading to the nozzle, indicated the nozzle pressure. A second pressure gauge registered the pressure downstream of the flow modifier. This was used as a backup to the transducer during tests with the prototype modifier.

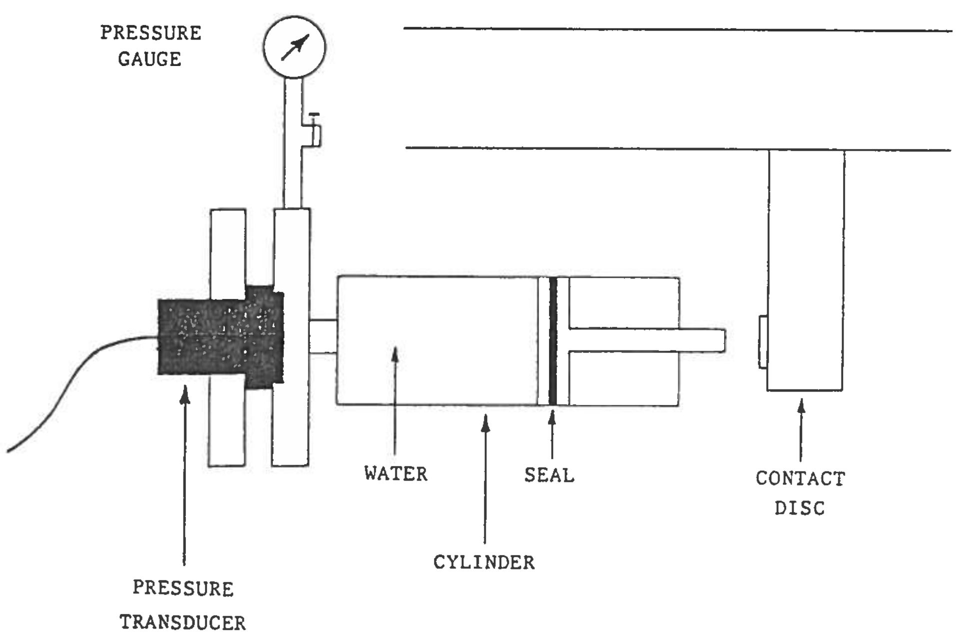


FIGURE 7: REACTION FORCE MEASUREMENT TRANSDUCER

D. LABORATORY TEST PROCEDURE

Numerous qualitative and quantitative laboratory tests were performed with different combinations of upstream and downstream drag generators. The objective of these tests was to determine the effect of the flow areas, the axial spacing and the upstream modifier profile on the nozzle reaction force.

The procedure for a typical nozzle performance test was as follows:

The desired pair of drag generators was positioned inside the flow modifier assembly. The pump discharge valve was set at a nearly closed position. The nozzle assembly was checked for unconstrained axial movement. The fire pump was then started and the discharge valve was gradually opened. The by-pass valve was left fully open and the discharge valve was adjusted to set the desired nozzle discharge rate. The nozzle pressure, the rotameter reading and the nozzle reaction force were then recorded. The test was then repeated for several water flow rates. Test data were checked for repeatability. After the completion of each test, the pump was turned off. The distance between the drag generators was varied and the procedure described above repeated. After completing the tests with one pair of drag generators, either one or both of the drag generators were changed for the next series of tests.

The axial spacing between the drag generators was varied in discrete increments using the spacer rings. The first few tests revealed that the openings in both the upstream and downstream drag generators were too small resulting in very high pressure drops across the flow modifier even when the drag generators were wide apart. It was also noted that the profile of the upstream drag generator did not affect the results significantly. This was based on the observation that the pressure drop across the flow modifiers was practically unchanged for any given water flow rate and spacing between the drag generators.

A second set of drag generators with larger openings was fabricated and tested. Four combinations of the two upstream and two downstream drag generators, illustrated in Figure 8, were tested. Upon completion of these tests, it was decided to use a simple flat profile for the upstream drag generator (see Profile C in Figure 5C) in all subsequent tests. The final laboratory test model consisted of an upstream drag generator with four, 0.375-inch holes, and a downstream drag generator with a 0.75-inch axial hole. This pair of upstream and downstream drag generators is shown in Figure 9.

Difficulties were encountered in replicating the readings of the reaction force transducer between runs carried out on different days. This was attributed to variations in the frictional forces between the hose and the laboratory floor and the pushing of the hose against the laboratory walls. However, the readings were consistent within each run.

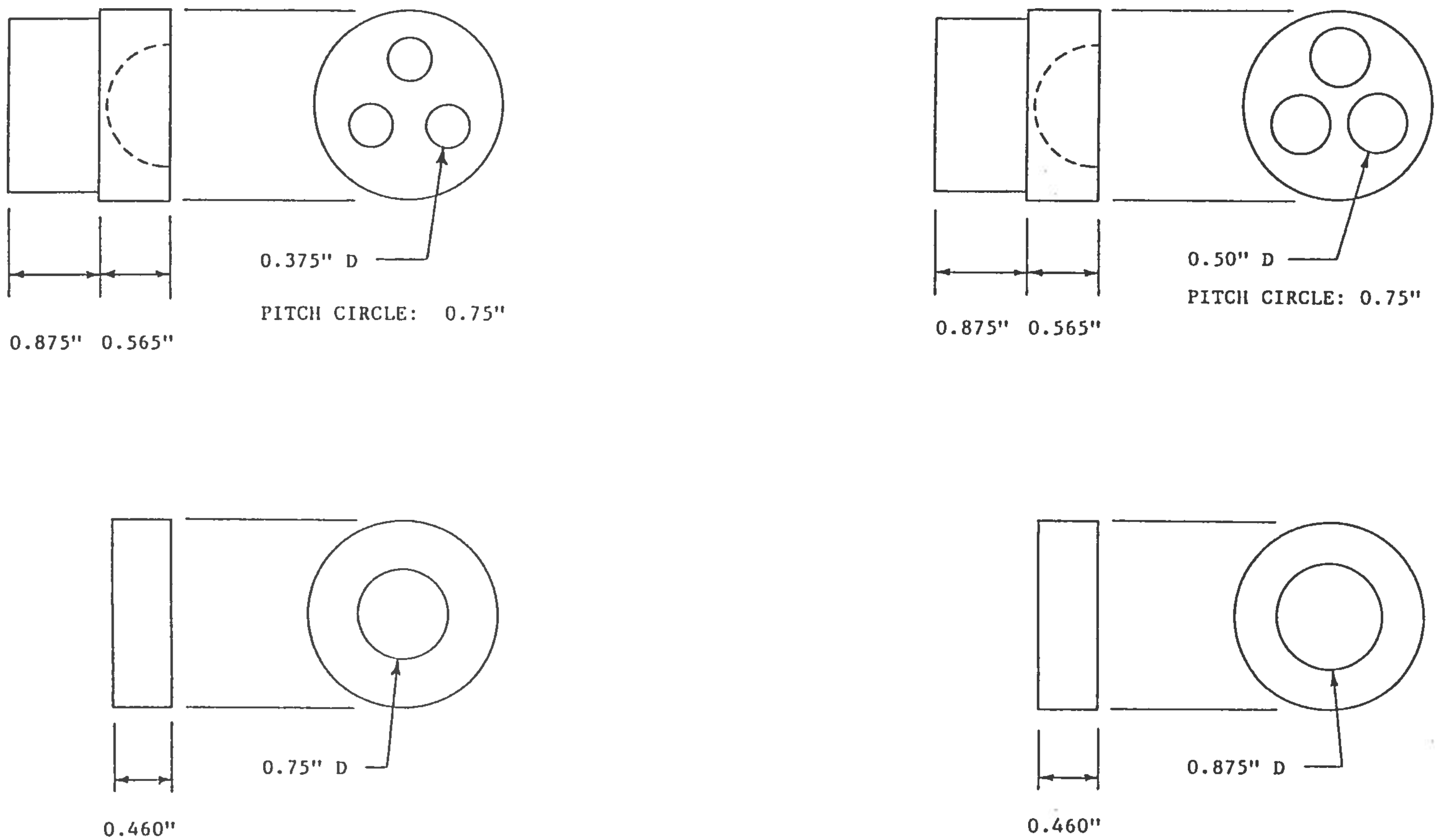


FIGURE 8: SECOND SET OF LABORATORY DRAG GENERATORS

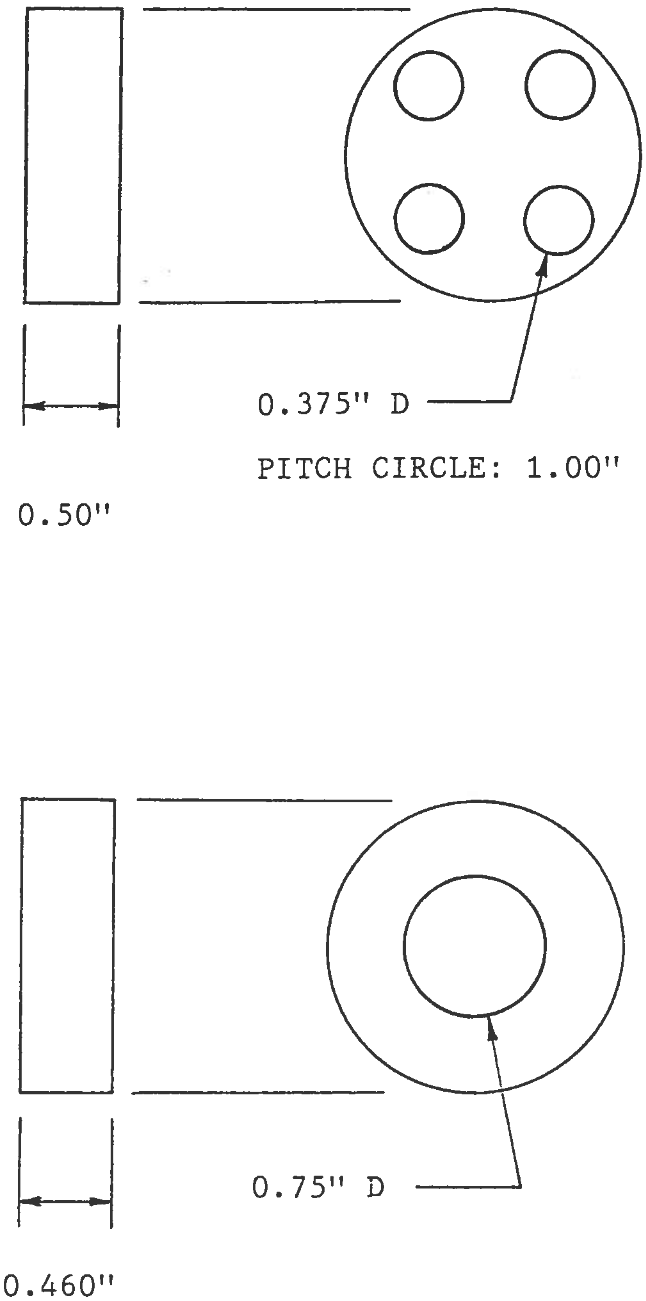


FIGURE 9: FINAL LABORATORY FLOW MODIFIER DESIGN

E. LABORATORY TEST RESULTS

The raw experimental data from the laboratory tests are tabulated in Appendix A. Figure 10 is a plot of the transducer readings, which are linearly related to the nozzle reaction force, versus the quantity $Q\sqrt{p}$, with and without the modifier depicted in Figure 9.

A review of Equations (5) and (8) shows that the reaction force for a nonrigid pipe may be written as follows:

$$F_R = f(Q\sqrt{p}) \qquad (10)$$

According to Bernoulli's equation, Q is a function of $\sqrt{p}$. For a smooth circular orifice, assumed in the derivation of Equation (5), the nozzle coefficient C_n was assumed to be a constant and equal to .98. This is not necessarily true for commercial firefighting nozzles. Furthermore, the dependence of the second term in Equation (8) on the nozzle pressure is not known. Thus, the quantity $Q\sqrt{p}$ was a convenient, experimentally-determined parameter that could be used to observe the changes in the reaction force, F_R when p and Q were changed.

Examination of Figure (9) shows that there is a significant change in the dependence of the reaction force, F_R on $Q\sqrt{p}$. With no modifier upstream of the nozzle, the reaction force increased monotonically with $Q\sqrt{p}$. When the modifier was inserted upstream of the nozzle, the reaction force actually decreased with an increase in $Q\sqrt{p}$. The effect was more pronounced as the resistance to flow was increased by narrowing the axial space between the upstream and downstream generators.

As a result of the laboratory experiments, the following important observations were made:

- The profile of the upstream modifier did not have a significant effect on the nozzle reaction force. This is probably because the turbulence generated by the drag generators is overshadowed by the turbulence generated downstream of the flow modifier.

- The drag generator flow areas and the distance between the drag generators within the flow modifier had a major effect on the nozzle reaction force.

- The nozzle reaction force responded very strongly and was significantly reduced as the space between the upstream and downstream drag generators was reduced and thus the resistance to flow increased.

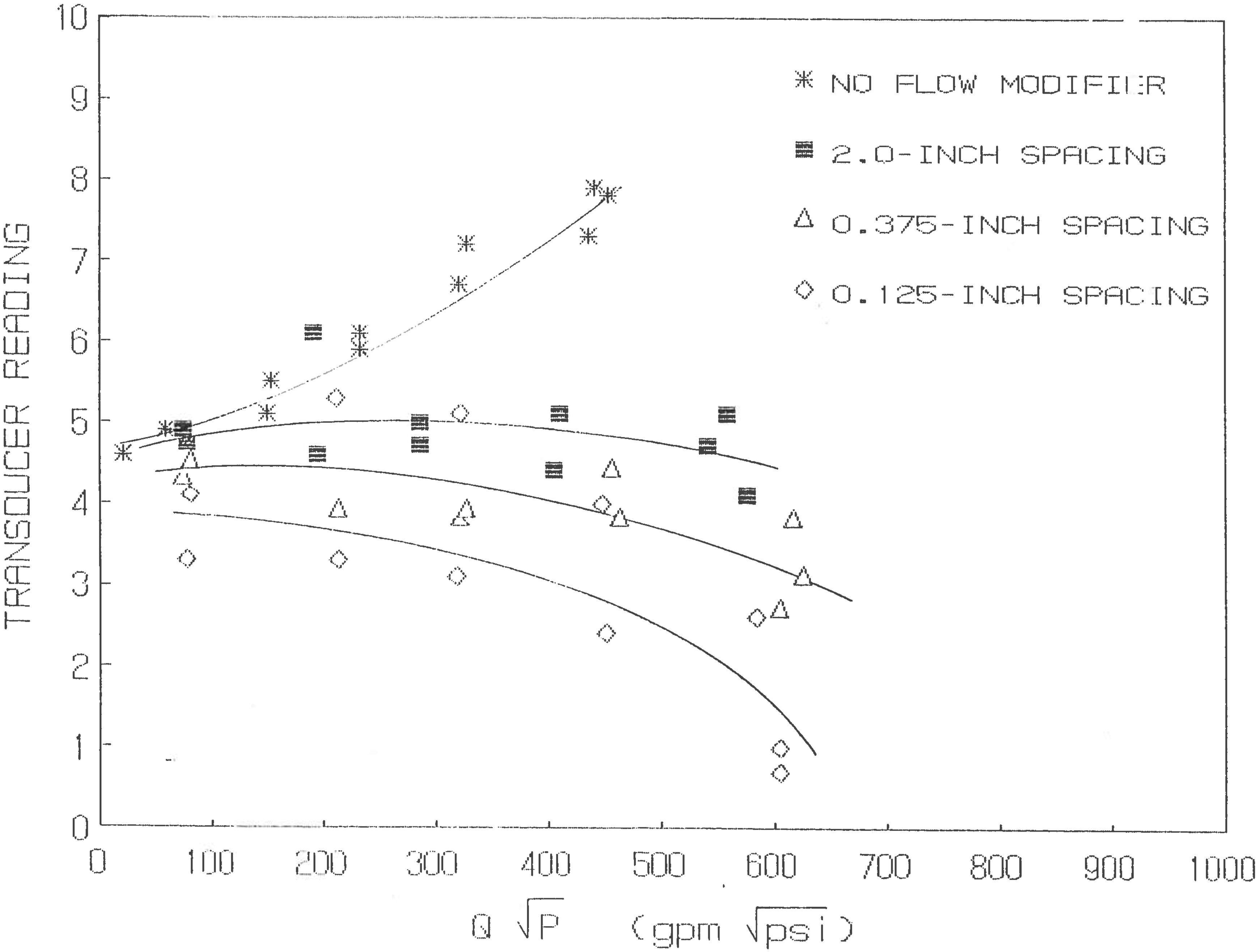


FIGURE 10: REACTION FORCE VS Q $\sqrt{P}$ FOR FINAL LABORATORY DRAG GENERATORS

SECTION IV

PROTOTYPE DEVELOPMENT AND TEST PROGRAM

This section describes the design and construction of the Model I prototype of the adjustable flow modifier and the subsequent tests at the hydraulics laboratory of Underwriters Laboratories, Inc. (UL), Northbrook, IL.

A. DESIGN OF MODEL I PROTOTYPE

Figure 11 shows the details of the Model I prototype of the flow modifier. This prototype was made of brass in order to permit alterations if necessary. It weighed about 16 pounds.

The upstream drag generator was force-fit into the end of the upstream tube. The downstream drag generator was also force-fit inside the downstream tube. The distance between the two drag generators could be varied by rotating one tube relative to the other. Both ends of the tubes had male national hose (NH) threads for direct connection to the fire hose and the commercial nozzle assembly. A handle was originally included in the design to facilitate the turning of the outer downstream tube. However, the handle added to the weight of the modifier, and was found to be bulky and unnecessary and was removed. Grooves were machined into the outer body to facilitate manual rotation.

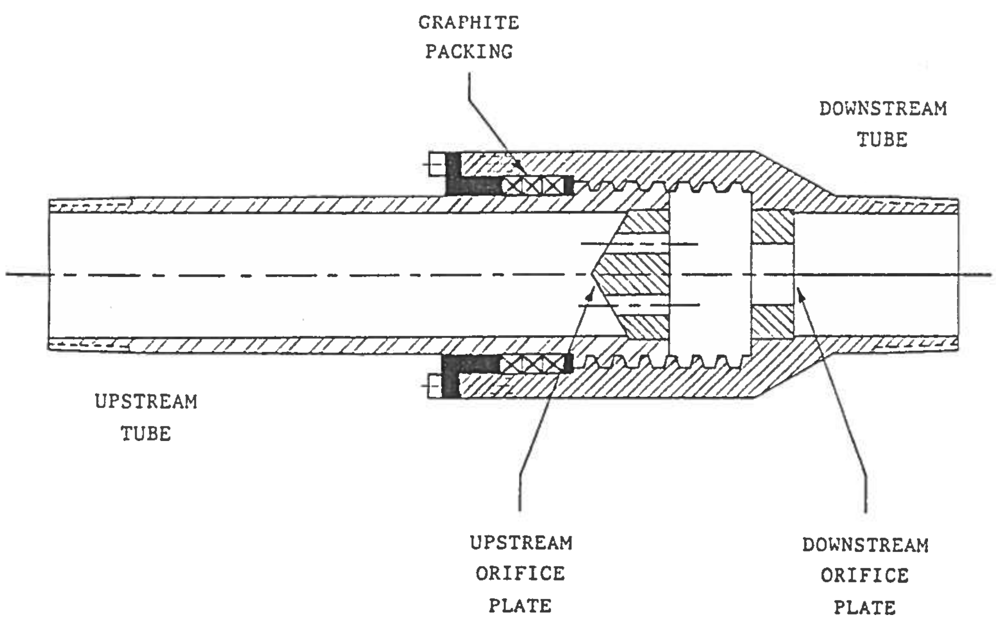


FIGURE 11: MODEL I PROTOTYPE FLOW MODIFIER

B. DESCRIPTION OF TEST NOZZLES

Measurements were made using three different commercial nozzles. Two nozzles were of the adjustable stream type: an Akromatic nozzle, with a 1.5-inch base, suitable for 2.5-inch hose (W.S. Darley & Co., Catalog No. T 319) and an Elkhart "Chief" constant flow nozzle, also with a 1.5-inch base, suitable for 1.75- and 2-inch hose (W.S. Darley & Co., Catalog No. W 139). The third nozzle was a 0.75-inch solid stream brass nozzle, with a 1.5-inch hose connection (W.S. Darley & Co, Catalog No. E 137). The adjustable nozzles were tested in the solid stream mode.

C. PROTOTYPE TEST APPARATUS

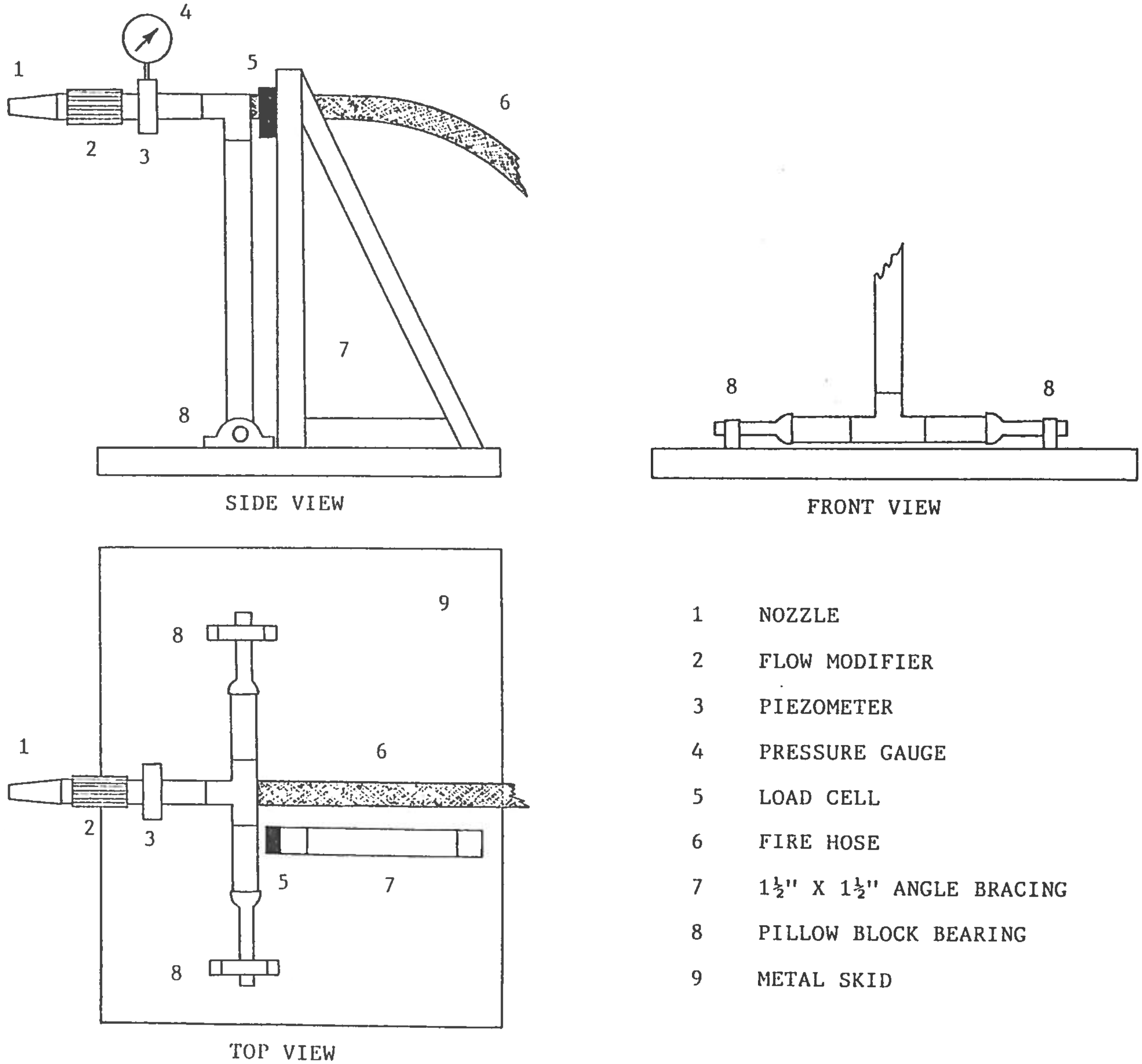
The test apparatus consisted of a high-capacity, high-pressure water supply system, a fire hose and a test stand. The stand supported the fire nozzle, the flow modifier (when used), a piezometer and a pressure gauge. The test stand consisted of an inverted "T" support stand mounted on a metal pallet. The stand was held by two pillow block bearings as shown in Figure 12.

A Perma-Cal pressure gauge (Model 85FA5PG) was used to record the inlet nozzle pressure, while a calibrated Sensotec load cell (Model 41-572-01) was used to measure the reaction force. A calibrated Daytronic digital indicator (Model 9515A) was used to register the nozzle reaction forces in pounds. Both the water flow rate and the supply pressure could be varied independently over a wide range. The flow rates were measured by a Newport flow meter (Model 29FA5M), with a range of 0 to 3,000 gpm.

D. TEST PROCEDURE

In a typical test, the flow modifier was set at its wide open position (Setting 1). The pump discharge regulator was adjusted to bring the water flow rate to a specified value. Values of the reaction force (the force transducer reading), the nozzle pressure and the throw distance were noted. The pump discharge was then increased to set the water flow rate to a higher value and a similar set of readings was taken. In these tests, water flow rates were varied from about 60 gpm to 120 gpm. Readings were also taken while flow rates were decreased.

The pump discharge was again adjusted to set the water flow rate to a specified value. The flow modifier was then partially closed to Setting 2. This caused the water flow rate to decrease. The pump discharge rate was brought back to the initial value while keeping the modifier setting unaltered. Nozzle pressure, water flow rate and throw distance were recorded again. Next, the flow modifier was adjusted to its most closed position (Setting 3), and the procedure was repeated.



1 NOZZLE
2 FLOW MODIFIER
3 PIEZOMETER
4 PRESSURE GAUGE
5 LOAD CELL
6 FIRE HOSE
7 $1\frac{1}{2}$" X $1\frac{1}{2}$" ANGLE BRACING
8 PILLOW BLOCK BEARING
9 METAL SKID

FIGURE 12: MODEL I PROTOTYPE TEST APPARATUS

The flow modifier was then opened to Setting 1 and the pump discharge adjusted to bring the water flow rate to the next higher specified value. As before, values of the nozzle pressure, water flow rate and throw distance were noted for each setting of the flow modifier. Care was exercised to limit the maximum hose pressure to about 275 psi for safety reasons.

Another series of tests was conducted with the Akromatic adjustable nozzle and the Elkhart constant flow nozzle to evaluate the effectiveness of the flow modifier qualitatively. The objective of this series of tests was to simulate real life conditions, and to note if an operator could feel a reduction in the magnitude of the reaction force. The test stand was removed. A short segment of 2.5-inch fire hose was used immediately upstream of the nozzle. The hose and nozzle assembly were hand-held by the operator. After setting the water flow rate with the flow modifier wide open, the modifier was partially closed, causing a reduction in the flow rate. The water flow rate was then restored to its previous value by adjusting the pump discharge pressure.

During all tests, the throw distance was visually observed using a series of evenly-spaced calibration marks on the pavement directly in front of the nozzle outlet. The solidity of the discharge stream was also observed and photographed.

E. UL TEST RESULTS

The experimental data from the full-scale tests at Underwriters Laboratories are tabulated in Appendix B. In these tables, "wide open" refers to Setting 1, "partially closed" refers to modifier Setting 2, and "nearly closed" refers to modifier Setting 3. The settings were marks on the modifier body that indicated the number of turns or fractions of turns used to adjust the spacing between the two drag generators.

Throw range measurements are approximate because of the windy condition that prevailed during the test period. The wind tended to disintegrate the stream and made accurate measurements difficult.

The reaction force, F_R, the nozzle pressure, p, and the water flow rate, Q, data in Appendix B were converted to plots of the reaction force F_R versus the quantity $Q\sqrt{p}$. These plots are shown in Figures 13, 14 and 15. The theoretical line given by Equation (5) for a <u>rigid</u> pipe is also shown on the same diagram. By plotting the results in this manner, the performance of the tested nozzles with and without the modifier can be compared with each other and with the theoretical predictions of Equation (5) for a rigid pipe.

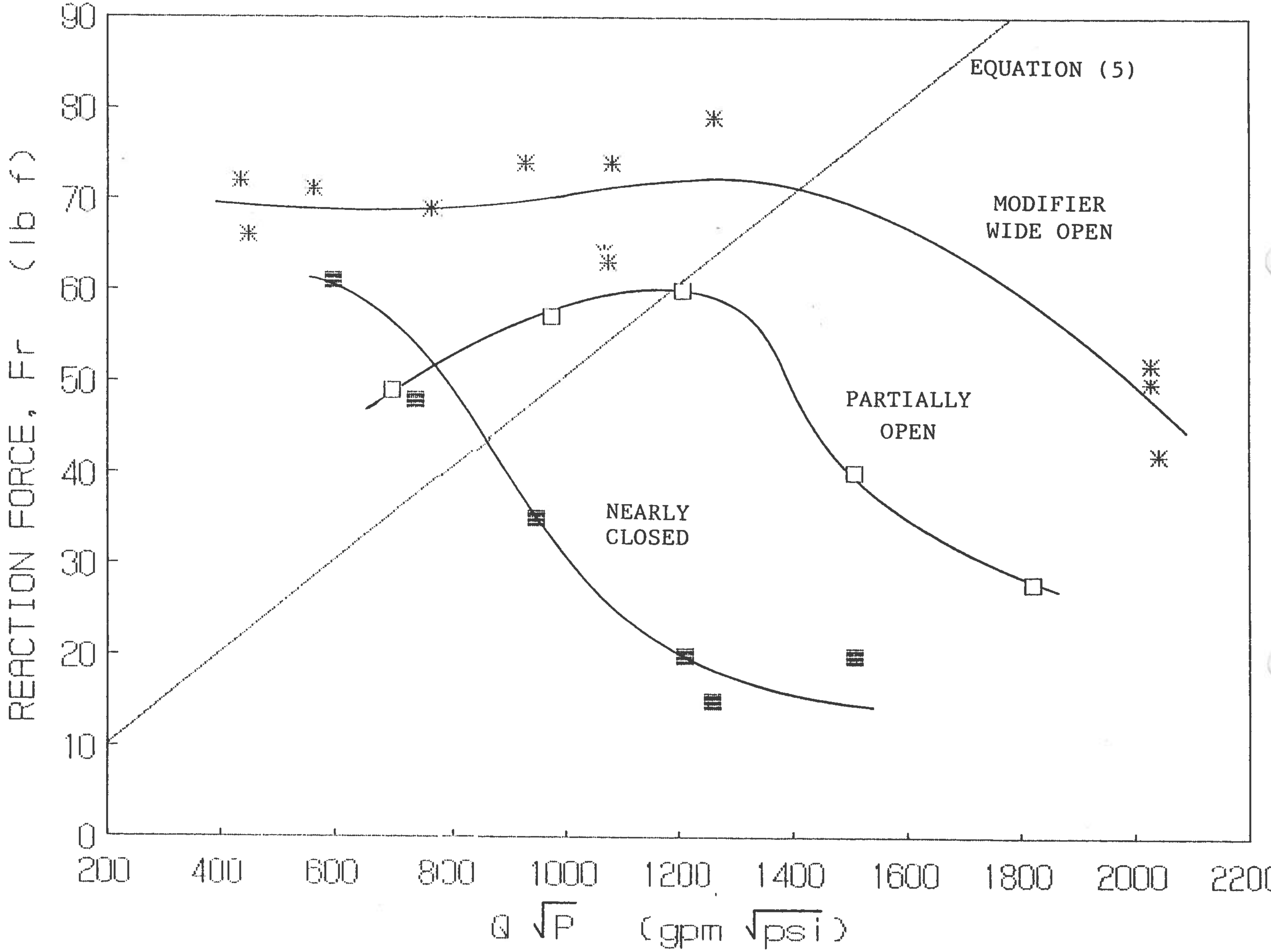


FIGURE 13: PLOT OF F_R VS $Q\sqrt{P}$ FOR THE AKROMATIC NOZZLE WITH AND WITHOUT THE MODEL I FLOW MODIFIER

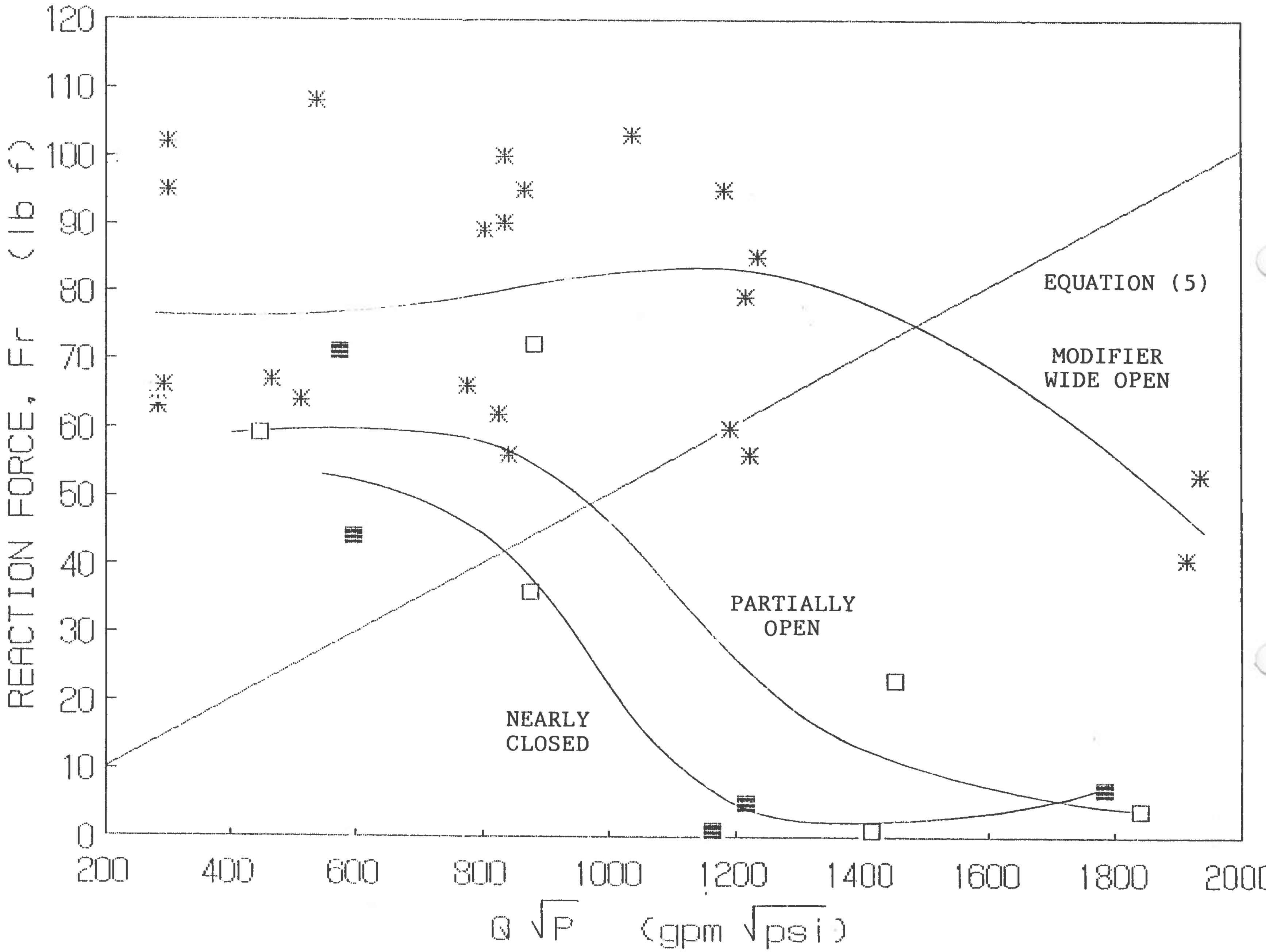


FIGURE 14: PLOT OF F_R VS $Q\sqrt{P}$ FOR THE ELKHART NOZZLE WITH AND WITHOUT THE MODEL I FLOW MODIFIER

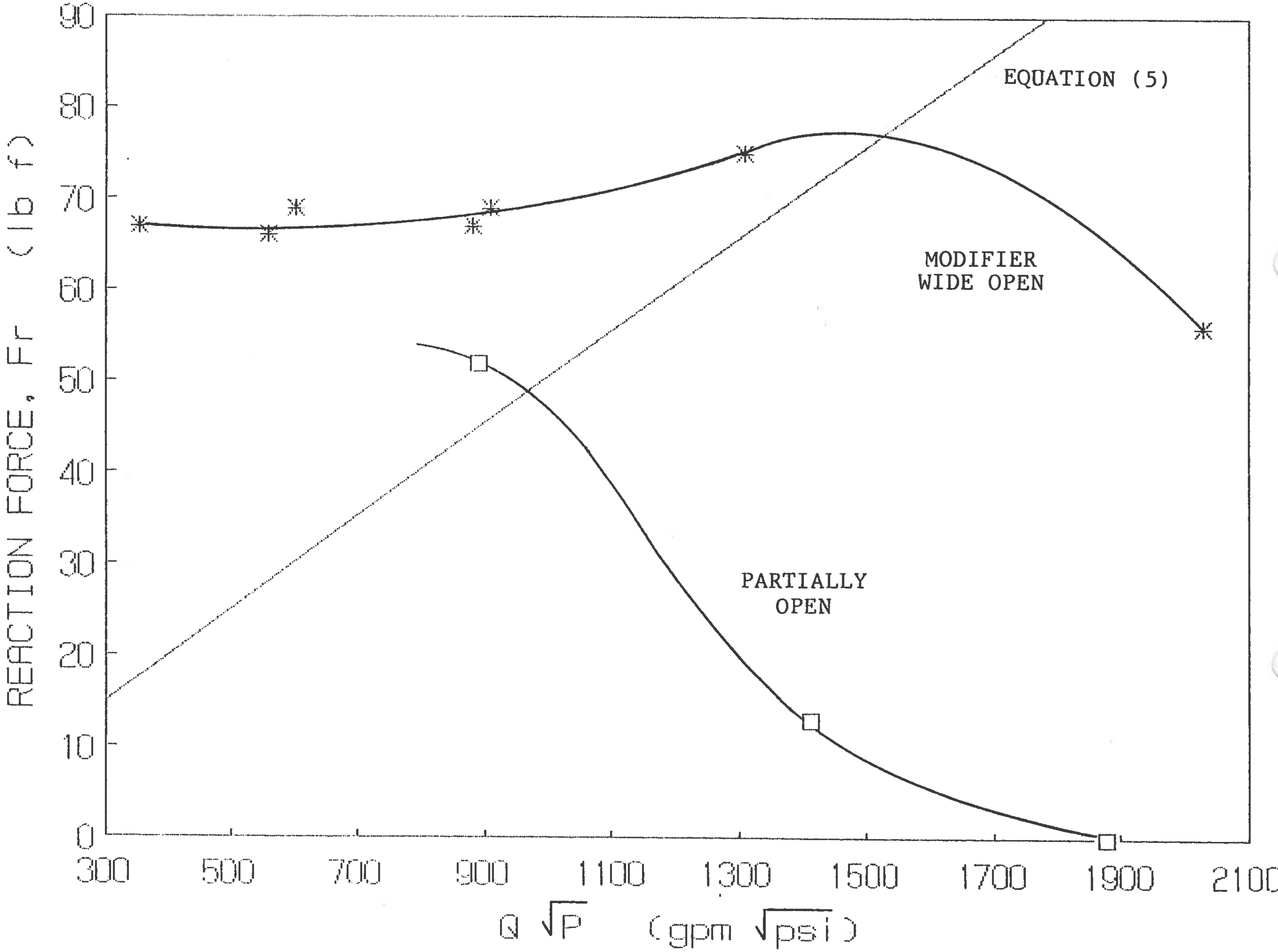


FIGURE 15: PLOT OF F_R VS $Q\sqrt{P}$ FOR THE SOLID STREAM NOZZLE WITH AND WITHOUT THE MODEL I FLOW MODIFIER

F. DISCUSSION OF RESULTS

Inspection of the results for the case when the modifier was wide open shows that the theoretical plot of F_R vs Q/p, often cited in the fire protection literature and used by the fire industry to predict reaction force, does not represent the reaction force from commercial nozzles.

As shown in Section II, Equation (5) is only applicable to rigid pipe systems and to orifices with an assumed <u>constant</u> orifice or nozzle discharge coefficient, C_n of 0.98. Neither one of these assumptions is necessarily applicable to a commercial firefighting nozzle connected to a hose. As can be seen, each commercial nozzle has a characteristic curve which predicts a somewhat constant reaction force of between 65 lb_f and 75 lb_f for values of Q/p of up to 1,400 (gpm/psi). The measured values of F_R were considerably higher than those predicted by Equation (5), especially at values of Q/p << 1,400. For Q/p > 1,400, the reaction force fell below that predicted by Equation (5).

The effect of the flow modifiers on the reaction force is clearly evident in Figures 13, 14 and 15. As can be seen, for the same values of Q/p, there was a dramatic reduction in the values of F_R when the modifiers were introduced upstream of the nozzles and set to generate a higher resistance to flow. For example, at Q/p of 1,400 gpm/psi, the Akromatic nozzle with the modifier wide open (Setting 1) gave a reaction force of about 70 lb_f. When the modifier was partially closed (Setting 2), the reaction force dropped to about 50 lb_f, a 29% reduction, and to about 14 lb_f when it was nearly closed (Setting 3), an 80% reduction. At the same Q/p of 1,400, the reaction force of the Elkhart nozzle dropped by about 90% at Setting 2 and was almost completely eliminated when the modifier was in the nearly closed position (Setting 3). Similarly, the reaction force of the brass solid stream nozzle dropped by about 80% when the modifier was partially closed (Setting 2).

No noticeable changes were observed in the throw range or the solidity of the discharge stream when the modifier was closed and the water flow rate was restored to its previous value. The solidity of the stream was found to be independent of the modifier setting or its presence. The reason for this is that the solidity of the water stream is highly dependent upon the turbulence generated within the commercial nozzle. The turbulence generated within the nozzle is so high that it overshadows any disturbances introduced in the flow by the flow modifier.

The Model I prototype tests indicated a strong relationship between the spacing and opening areas of the drag generators within the flow modifier and the nozzle reaction force. This dependence had also been observed during the laboratory tests. This implied a strong response of the reaction force and the nozzle pressure to the modifier adjustments.

According to the two individuals who carried out the qualitative demonstration tests, considerable reduction in nozzle kickback was experienced when the modifier was nearly closed and the pressure increased to give the original water flow rate.

As a result of these tests, it was decided to design and construct a second prototype out of lighter materials such as polymers. The new prototype design was also to be such that it would be easier to adjust and control the spacing between the drag generators than Model I.

SECTION V

MODEL II PROTOTYPE DESIGN AND DEMONSTRATION TESTS

A. DESIGN OF MODEL II PROTOTYPE FLOW MODIFIER

Figure 16 is a sketch which shows the design details of the Model II prototype of the flow modifier. This modifier was constructed out of a high strength plastic, commercially known as Turcite-A. The drag generators were constructed of brass. The overall dimensions of the modifier were reduced considerably for compactness and ease of handling. In contrast with the brass Model I prototype which weighed about 16 pounds, this unit weighed about two pounds.

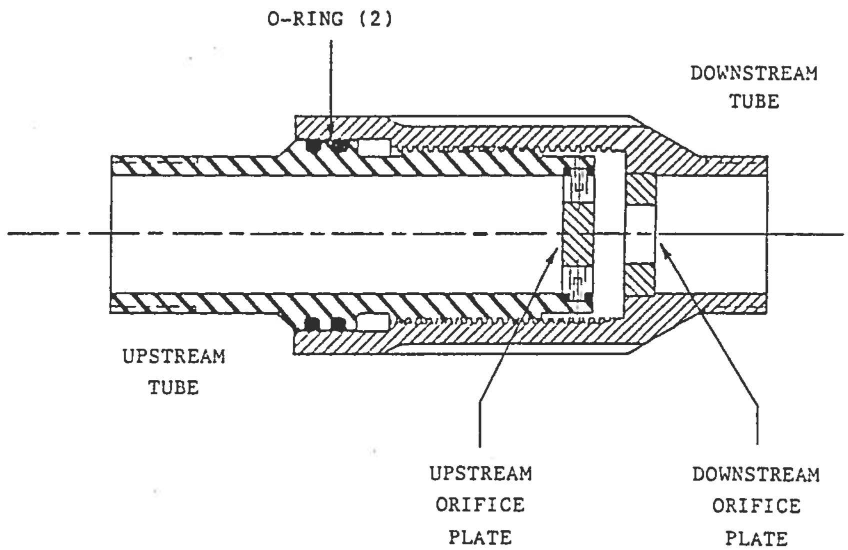


FIGURE 16: SKETCH OF MODEL II PROTOTYPE (not to scale)

The design of the upstream and the downstream drag generators was changed to minimize the pressure drop across the plates when the modifier was in its most open position. This was necessary to insure the best stream discharge characteristics when the available pump pressure was low.

The design of the mating threads on the upstream (outer) and the downstream (inner) tubes was also modified to effect a smooth and gradual response of the nozzle pressure and reaction force to flow modifier adjustments. The design of the seal between the upstream and the downstream tubes was also changed to make it easier to rotate one tube relative to the other. The graphite packing used in the Model I unit was replaced by a pair of O-rings.

A complete stress analysis was carried out for the Model II modifier prototype. The analysis is given in Appendix C.

B. LABORATORY TESTS WITH MODEL II PROTOTYPE FLOW MODIFIER

Diagnostic laboratory tests were conducted using the Model II prototype flow modifier. Although the purpose of these tests was primarily to insure that the modifier was easy to operate and that there were no leaks, some experimental data were recorded. In these tests, the water flow rates were varied from 40 gpm to 70 gpm. The nozzle pressures ranged from about 20 psig to 90 psig. The reaction pressure, as measured by the pressure gauge located at the transducer connection was also recorded. All of the tests were made with the Akromatic adjustable nozzle downstream of the flow modifier. Readings were taken at six different settings of the modifier. These settings ranged from Setting 1, which corresponded to the widest open position, to Setting 6, which corresponded to an almost closed position.

The results of these tests are given in Appendix D and are plotted in Figure 17. In this figure, the reaction pressure, P_R (psi) is plotted against the quantity Q√p (gpm√psi) for the six different modifier settings. Also plotted on the same diagram is the theoretical plot of Equation (5) corrected for the fact that the reaction force was acting on a hydraulic piston which was one-inch in diameter (or 0.785 square inch in area).

This plot is interesting in many respects. At the relatively low values of Q√p that could be achieved in the laboratory, and when the modifier offered little resistance to flow (i.e., curves 1, 2, 3 and 4), the reaction pressure (or the corresponding nozzle reaction force) started out greater than that predicted by the theoretical Equation (5) for rigid pipe. As Q√p increased, the reaction force dropped to a minimum then began to increase at a slope very roughly equal to the slope of the theoretical line. There is some evidence that the same behavior occurs when the resistance to flow is increased further by reducing the spacing between the drag generators within the modifier and increasing the value of Q√p. This behavior appears to begin to occur for curve 6. A similar behavior was also noted during the tests at Underwriters Laboratories (see the nearly closed curves in Figures 13 and 14).

This behavior should be expected at high pressures, when the hose walls are tightly stretched and the hose/nozzle assembly begins to behave like a rigid pipe.

C. DEMONSTRATION TESTS AT TYNDALL AIR FORCE BASE

Demonstration tests in which the Model II prototype modifier was attached to the commercial nozzles normally used by USAF were carried out at Tyndall Air Force Base on April 4, 1991. The pressure of hydrant water was boosted by a USAF fire truck to a maximum of about 300 psig. A 200-foot length of hose normally attached to the pump was removed and replaced by a 50-foot, 1.5-inch diameter section.

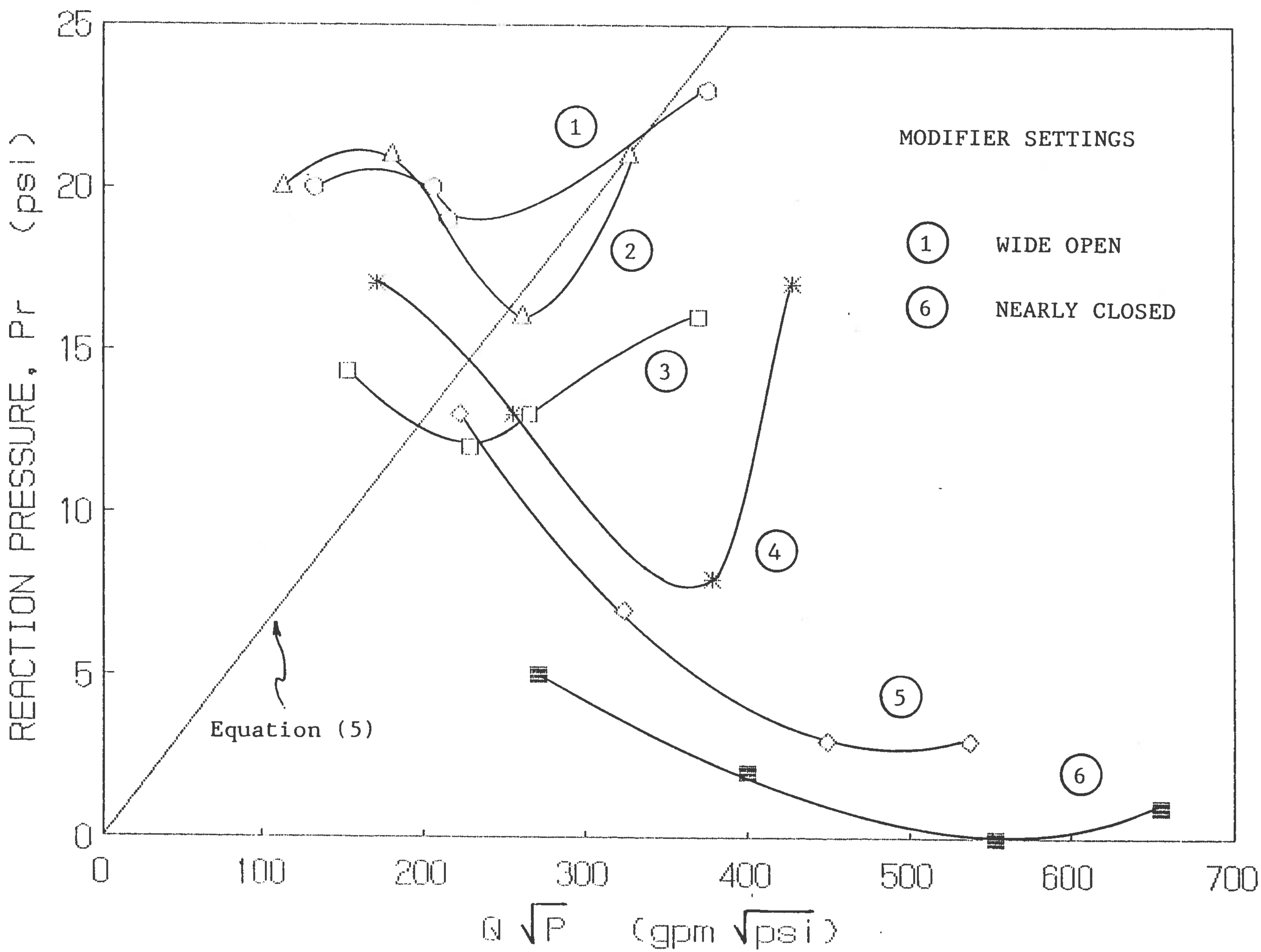


FIGURE 17: REACTION PRESSURE VS $Q\sqrt{P}$ AT DIFFERENT SETTINGS OF THE MODEL II PROTOTYPE FLOW MODIFIER

A 1.5-inch Task Force Tips adjustable fire nozzle and a 1.5-inch discharge playpipe nozzle were provided by the Tyndall AFB Fire Department. The Akromatic fire nozzle used by RISC at Underwriters Laboratories was also tested.

In each demonstration, the Model II prototype modifier was inserted between the fire hose and the commercial nozzle and was set at the wide open position (Setting 1). The hose and nozzle assembly was held by a firefighter from Tyndall Air Force Base. The fire pump was turned on and the nozzle was adjusted to the solid stream position. The pump discharge pressure was set at 150 psig, which is considered to be about the normal operating pressure. Considerable "kickback" was experienced by the firefighter.

The pump was shut off and the modifier adjusted to a partially closed position (Setting 2). The pump was then turned on. The pump discharge pressure and corresponding water flowrate were increased until the throw distance approximately equaled that obtained earlier with the modifier wide open. In the absence of a water flow rate measurement system on the fire truck necessitated the use of throw distance as a measure of the water flow rates.

Although the pump discharge pressure was much larger, the firefighter experienced a much smaller reaction force when the modifier was partially closed. The pump was then turned off and the modifier adjusted to its nearly closed position (Setting 3), and the test was repeated. The maximum pump discharge pressure used in these tests was about 290 psig.

The tests were repeated with all three commercial nozzles and by other personnel from the fire department. Everyone agreed that there was a considerable reduction in the reaction force for modifier Settings 2 and 3, as compared to that experienced at modifier Setting 1, the wide open position.

D. THE MODEL III PROTOTYPE FLOW MODIFIER

As a result of the Tyndall AFB demonstration tests, two major drawbacks of the Model II prototype modifier were noted. One was that the modifier could not be manually adjusted at the high water pressures developed by the Tyndall fire pump. This problem was not evident during the laboratory diagnostic tests with the same modifier because the laboratory test pressures never exceeded 90 psig. It was obvious that the problem was due to the radial pressure and expansion of the internal "upstream" plastic tube against the O-rings which slide against the inside wall of the external "downstream" tube. This problem was not observed during the full-scale tests at the Underwriters Laboratories because the Model I prototype modifier was made of brass.

To alleviate this problem, a Model III prototype was fabricated with the following changes:

- The upstream tube of the Model III prototype was made out of aluminum,

- The number of O-rings was reduced to one, and

- The depth of the O-ring groove was decreased so as to reduce friction with the outer tube.

The slight increase in weight of the unit due to the change to aluminum will be of the order of a few ounces.

The second problem noted during the demonstration tests was that the modifier added about 6-inches to the length of the hand-held nozzles, particularly to the comparatively long Task Force Tips nozzle. The modifier ended up at a location close to the firefighter's elbow. This made it difficult for the firefighter to adjust the modifier setting while fighting a fire. This problem was not unexpected. Ideally, future modifiers should be designed to fit within the commercial nozzle body. However, this requires that manufacturers adapt the design philosophy developed and proven in this study to their specific nozzle designs.

SECTION VI

CONCLUSIONS AND RECOMMENDATIONS

A. CONCLUSIONS

1. The literature suggests that for a nozzle discharging water at a given volumetric flow rate and nozzle pressure, the reaction force can be predicted by the equation:

   $$F_R = .0505 \times Q \times \sqrt{p} = 1.5 \times d^2 \times p$$

   This equation is only applicable to a nozzle with a fairly smooth circular outlet, attached to a rigid pipe. In this case, the reaction force would be equal to the momentum flux of the water discharged from the nozzle.

2. For a nozzle attached to a flexible fire hose, the nozzle reaction force depends on the water pressure and the tensile pull in the material of the hose near its discharge end. In this case, the pressure force and the tension in the hose, which oppose each other, need not be equal in magnitude. The tension in the hose depends on such factors as the unsupported length of the hose, the hose diameter, the material of the hose, its approach configuration, and the water pressure. At high water flowrates and nozzle pressures, the reaction force begins to follow the equation cited above.

3. Full-scale experiments carried out as part of this project have shown that the actual reaction force developed by commercial nozzles were roughly constant and of the order of 70 lb_f up to a certain point where $Q\sqrt{p}$ was about 1,400 (gpm$\sqrt{\text{psi}}$). In this region, the reaction force was greater than that predicted by the rigid pipe model. For values of $Q\sqrt{p} > 1,400$, the measured reaction force was below the theoretical predictions. Also, for the same values of $Q\sqrt{p}$, the reaction force differed from one commercial nozzle to another.

4. The results of this study have shown that significant reductions in the reaction force of a firefighting nozzle can be realized by the use of a properly designed flow modifier consisting of two drag generators with an adjustable spacing, placed upstream of a commercial firefighting nozzle.

Given the availability of a reasonably high pressure at the pump, the flow modifier permits an increase in the water flow rate and throw range, without exceeding humanly manageable levels of the reaction force or affecting the solidity of the nozzle discharge stream.

B. RECOMMENDATIONS

1. This study addressed the development of flow modifiers that can be attached upstream of any commercial water firefighting nozzle. Because of the wide variety of commercial nozzle designs, no attempt was made at developing a modifier that can be incorporated in any one specific commercial nozzle. It is highly recommended that manufacturers incorporate the proven concept developed in this study into the design of their specific commercial nozzles. This would make the combined nozzle and flow modifier assembly more compact and lightweight.

2. The design of the flow modifier can be further refined so that it can function as a shut-off valve for the firefighting nozzle. This would eliminate the need for the conventional ball valve found in most existing high flow nozzles.

SECTION VII

REFERENCES

1. Fornell, David P., "Taking the Pressure Off: Low-Pressure Fog Nozzles Hit the Streets," Fire Engineering, 143 (12), pp 58-68, 1990.

2. NFPA, Fire Protection Handbook, 16th Edition, published by the National Fire Protection Association, Quincy, MA, 1986.

3. Task Force Tips, Inc., "A Guide to Automatic Nozzles," published by Task Force Tips, Inc., Valparaiso, IN, 1984.

APPENDIX A:
LABORATORY TEST RESULTS
WITH FINAL LABORATORY FLOW MODIFIER
(See Figure 9)

TABLE A-1

LABORATORY TEST DATA:
NO FLOW MODIFIER

FLOWRATE (gpm)	NOZZLE PRESSURE (psi)	TRANSDUCER READING	Q/P gpm/psi
70	42	7.8	454
60	30	7.2	329
50	22	6.1	235
40	15	5.5	155
25	5.5	4.9	59
70	39	7.3	437
60	29	6.7	323
50	22	5.9	235
40	14.5	5.1	152
25	5.5	4.6	22
70	40	7.9	443

TABLE A-2

LABORATORY TEST DATA:
FLOW MODIFIER SPACING SET AT 2-INCHES

FLOWRATE (gpm)	NOZZLE PRESSURE (psi)	TRANSDUCER READING	Q/P gpm/psi
70	68	4.1	577
60	46	4.4	407
50	33	5.0	287
40	24	4.6	196
25	9	4.9	75
70	64	5.1	560
60	47	5.1	411
50	33	4.7	287
40	23	6.1	192
25	10	4.6	79
70	60	4.7	542

TABLE A-3

LABORATORY TEST DATA:
FLOW MODIFIER SPACING SET AT 0.375-INCH

FLOWRATE (gpm)	NOZZLE PRESSURE (psi)	TRANSDUCER READING	Q√P gpm√psi
70	75	2.7	606
60	58	4.4	457
50	43	3.9	328
40	29	3.9	215
25	9	4.3	75
70	78	3.8	618
60	60	3.8	465
50	42	3.8	324
40	29	3.9	215
25	11	4.5	83
70	80	3.1	626

TABLE A-4

LABORATORY TEST DATA:
FLOW MODIFIER SPACING SET AT 0.125-INCH

FLOWRATE (gpm)	NOZZLE PRESSURE (psi)	TRANSDUCER READING	Q√P gpm√psi
70	70	2.6	586
60	56	4.0	449
50	42	5.1	324
40	28	5.3	212
25	10	3.3	79
70	75	0.7	606
60	57	2.4	453
50	41	3.1	320
40	29	3.3	215
25	11	4.1	83
70	75	1.0	606

APPENDIX B: PROTOTYPE TEST DATA

TABLE B-1

FULL-SCALE TEST RESULTS WITH AKROMATIC ADJUSTABLE NOZZLE

FLOWRATE (gpm)	INLET PRESSURE (psi)	MODIFIER SETTING	REACTION FORCE (lb_f)	THROW RANGE (ft)
60	53	1	72	40
60	57	1	66	40
60	100	3	61	40
60	137	2	49	40
60	153	3	48	40
70	65	1	71	45
70	185	3	35	nr
80	93	1	69	50
80	149	2	57	50
80	230	3	20	50
90	107	1	74	50
90	180	2	60	nr
90	196	3	55	45
100	115	1	64	40
100	116	1	63	50
100	118	1	74	55
100	227	2	40	50
100	228	3	20	nr
110	132	1	79	55
110	274	2	28	55
150	183	1	50	55
150	183	1	52	nr
150	186	1	42	55

nr = not recorded

TABLE B-2

FULL-SCALE TEST RESULTS WITH ELKHART CHIEF NOZZLE

FLOWRATE (gpm)	INLET PRESSURE (psi)	MODIFIER SETTING	REACTION FORCE (lb_f)	THROW RANGE (ft)
60	23	1	63	nr
60	23	1	65	15
60	24	1	66	20
60	25	1	95	15
60	25	1	102	nr
60	55	2	59	15
60	92	3	71	nr
60	99	3	44	nr
80	34	1	67	20
80	40	1	64	20
80	41	1	64	20
80	45	1	108	nr
80	120	2	36	15
80	121	2	72	nr
80	212	3	1	nr
80	231	3	5	nr
100	60	1	66	20
100	65	1	89	20
100	68	1	62	20
100	70	1	90	nr
100	70	1	100	nr
100	71	1	56	20
100	75	1	95	nr
100	201	2	1	20
100	211	2	23	nr
110	89	1	103	nr
100	263	3	7	nr
112	270	2	4	nr
120	97	1	95	nr
120	99	1	60	25
120	103	1	79	20
120	104	1	56	25
120	106	1	85	25
150	163	1	41	30
150	167	1	53	35

nr = not recorded

TABLE B-3

FULL-SCALE TEST RESULTS WITH SOLID STREAM BRASS NOZZLE

FLOWRATE (gpm)	INLET PRESSURE (psi)	MODIFIER SETTING	REACTION FORCE (lb_f)	THROW RANGE (ft)
60	35	1	67	30
80	49	1	66	nr
80	57	1	69	40
80	124	2	52	40
100	78	1	67	nr
100	83	1	69	nr
100	200	2	13	nr
115	267	2	0	nr
120	119	1	75	nr
150	183	1	56	nr

nr = not recorded

APPENDIX C:
STRESS ANALYSIS FOR MODEL II PROTOTYPE MODIFIER

Presented below is the stress analysis of the Model II flow modifier shown in Figure C-1. The analysis covers the two critical regions of the two main components that are fabricated from plastic. The candidate material chosen for this application was Turcite A, a plastic manufactured by W. S. Shamban & Co., of California. This material has a relatively high strength and inherent lubricity (low coefficient of friction). It is commercially available in the form of extruded rod, and its principal physical properties are as follows:

σ_{tu} = 7,600 psi (tensile ultimate)

σ_{by} = 11,000 psi (flexural yield stress)

σ_{cu} = 11,500 psi (compressive ultimate, cubic specimen)

τ_u = 5,800 psi (shear ultimate)

ϵ_u = 15 percent (elongation at tensile rupture)

α = 5.2 x 10^{-5} in/in-F (coefficient of thermal expansion)

ρ = 1.49 g/cc (specific weight)

μ = 0.08 (coefficient of friction, Turcite vs. polished steel or Turcite vs. Turcite)

The only load that was considered in the subsequent analysis is the internal pressure of water which, for the sake of conservatism, was taken as 300 psi. This value represents the maximum available supply pressure, and it would be felt by all components of this device when in the zero-flow condition (i.e., when the nozzle is fully closed). When the nozzle is open and full-flow conditions prevail, the value of static pressure in the flow modifier is obviously lower than 300 psi, but the ensuing pressure drop has been neglected for the sake of conservatism in estimating the stresses.

A. DOWNSTREAM (OUTER) TUBE

Two modes of potential failure were investigated: tensile rupture due to hoop stress and tensile rupture in the longitudinal direction, at a section where the cross section changes abruptly.

1. Hoop Strength

Hoop stress due to an internal pressure of 300 psi (see Figure C-2) occurs at Section A-A, which is anywhere in Region B, identified in Figure C-1:

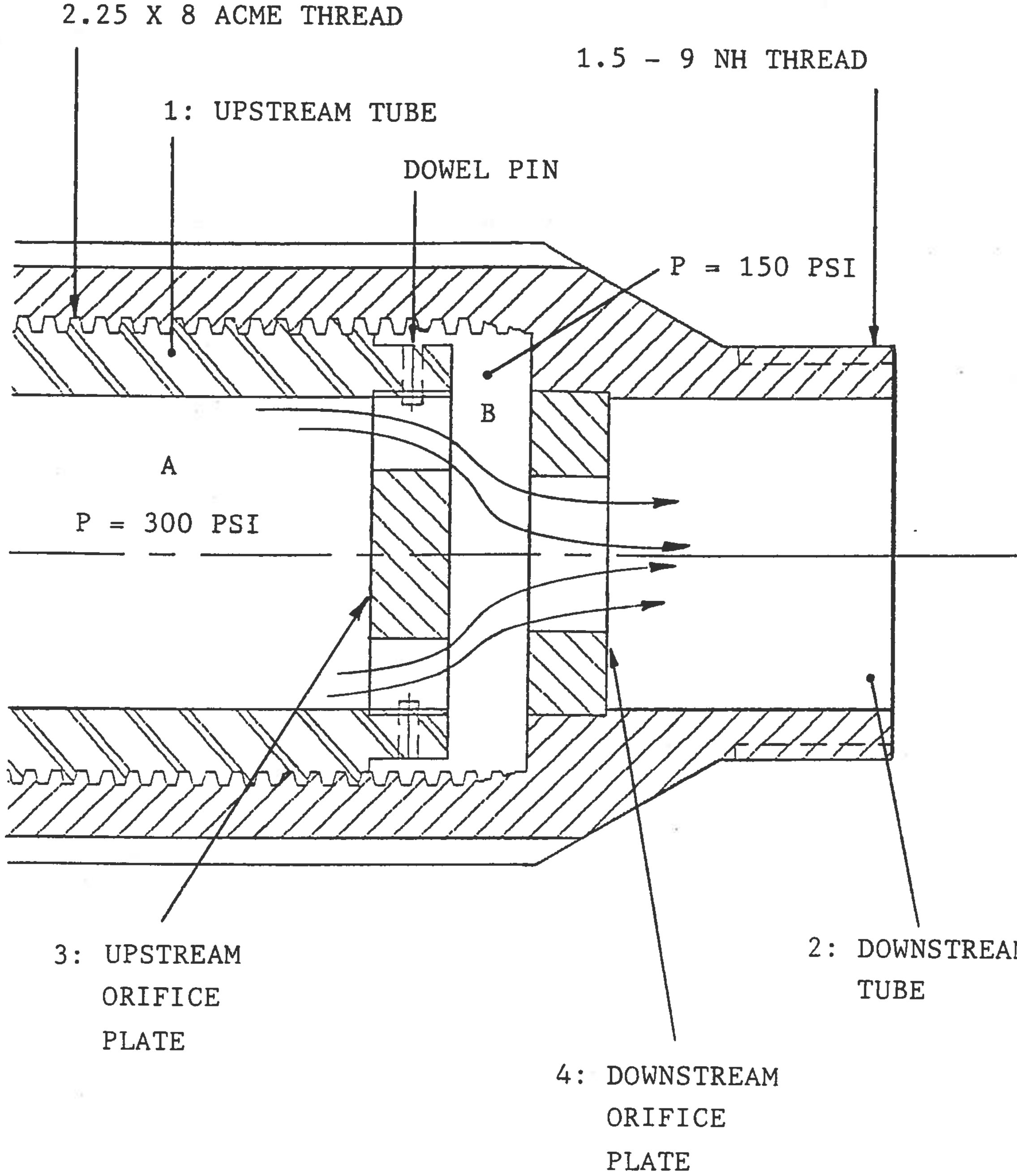


FIGURE C-1: PARTIAL LONGITUDINAL SECTION OF MODEL II

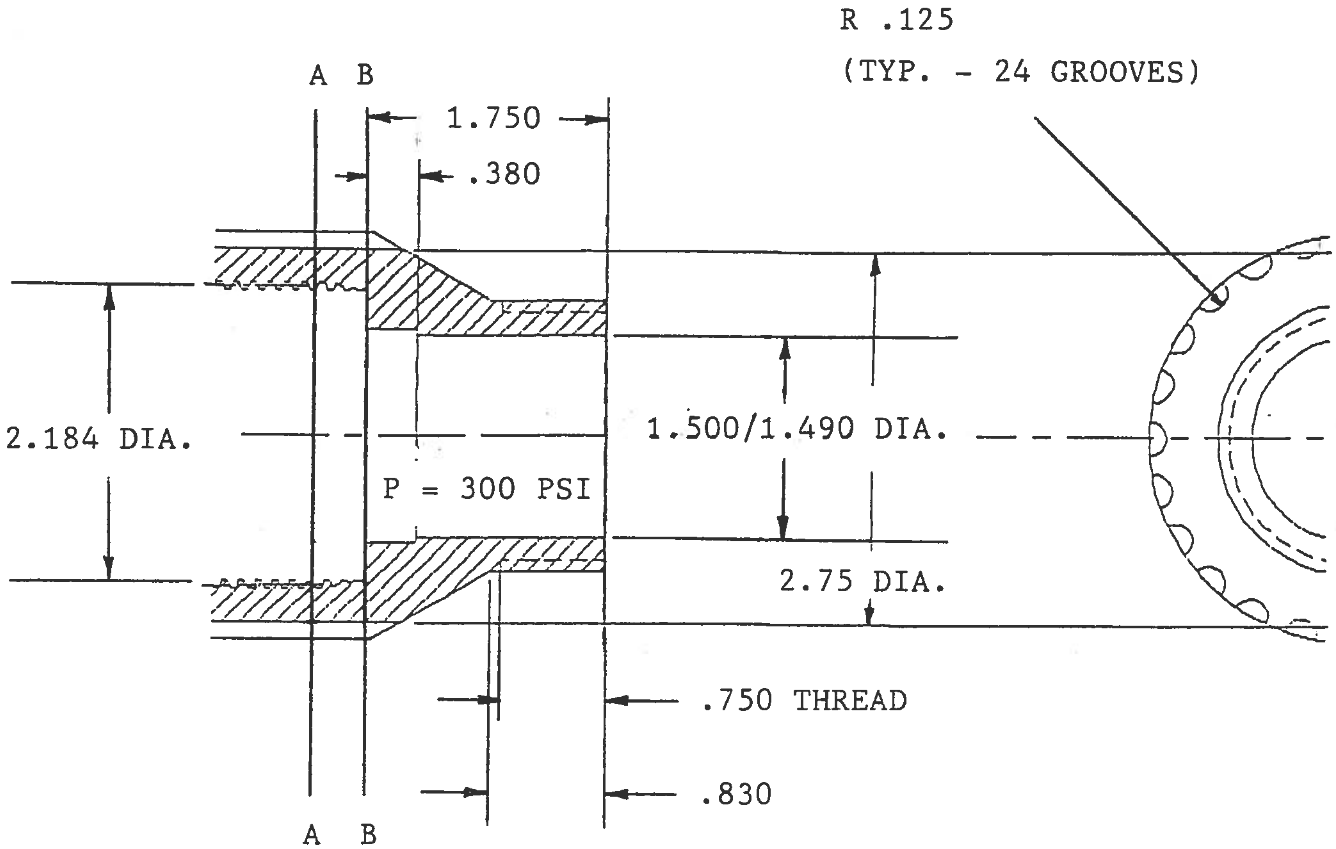


FIGURE C-2: LOCATION OF SECTIONS A-A & B-B IN DOWNSTREAM TUBE

$$\sigma_t = K_t(D_p\ p)/2t = 2.78\ (2.184 \times 300)/(2 \times .283)$$

$$= 3{,}218 \text{ psi} \qquad \text{(C1)}$$

where:

D_o = 2.75 inch = the tube O.D., measured to the bottom of longitudinal grooves

D_p = 2.184 inch = the pitch diameter of internal ACME thread

t = $(D_o - D_p)/2$ = the minimum material wall thickness

p = 300 psi

K_t = 2.78 = the stress concentration factor due to longitudinal grooves (Reference C-1)

The margin of safety, M.S., against hoop stress failure (based on tensile ultimate) is given by:

$$M.S. = (7,600/3,218) - 1 = 1.36 \quad (C2)$$

2. Axial Strength at Section B-B

Maximum possible axial load at Section B-B (Figure C-2) occurs when the nozzle is fully closed and the full supply pressure of 300 psi prevails in Region B. This axial load is:

$$F_a = (\pi/4)D_i^2 p = (\pi/4)(2.110)^2\ 300 = 1,049 \text{ lb} \quad (C3)$$

The resulting maximum tensile stress at the fillet indicated in Figure C-2 is given by:

$$\sigma_x = K_x F_a/\{\pi(D_o^2 - D_i^2)/4\} \quad (C4)$$

$$= 3.04 \times 1,049\{\pi(2.75^2 - 2.11^2)/4\} = 1,305 \text{ psi}$$

where:

D_o = 2.75 inch = the tube O.D.

D_i = 2.110 inch = the local I.D.

K_x = 3.04 = the stress concentration factor due to abrupt change in cross section with a fillet radius of 0.03 inch.

The margin of safety against axial rupture, based on tensile ultimate stress, is:

$$M.S. = (7,600/1,305) - 1 = 4.82 \quad (C5)$$

B. UPSTREAM (INNER) TUBE

The upstream tube is checked for strength against hoop rupture, shear failure of ACME threads, shear failure of 1.5-9 NH threads and shear-out failure at the four dowel pin holes.

1. Hoop Strength

Hoop stress in the smooth-O.D. region upstream of the ACME threads location is given by:

$$\sigma_t = D_i p/2t = (1.500 \times 300)/(2 \times .242) = 928 \text{ psi} \quad (C6)$$

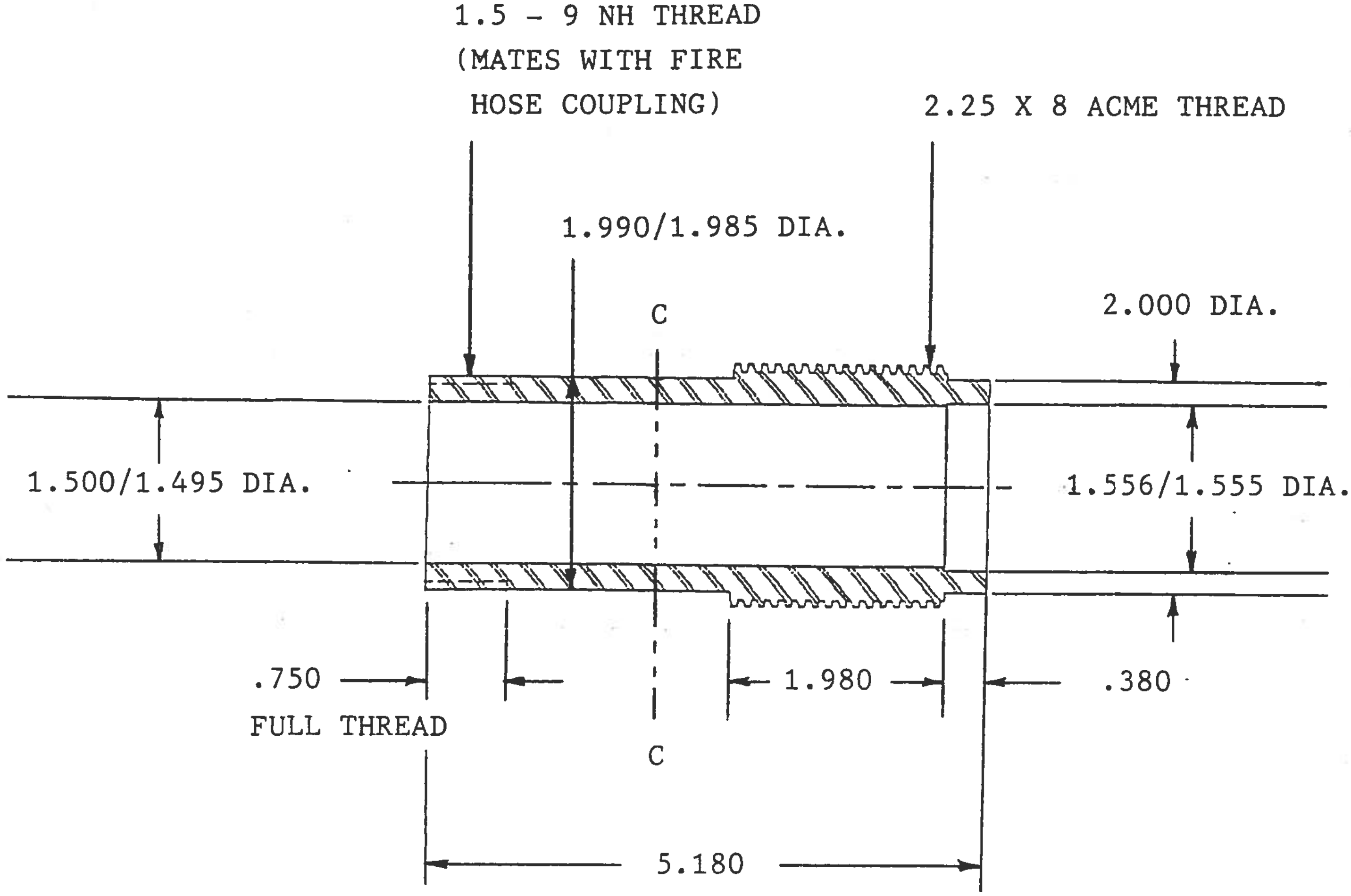


FIGURE C-3: LOCATION OF SECTION C-C IN UPSTREAM TUBE

where:

D_i = 1.500 inch = the tube I.D.

t = .242 inch = the local wall thickness

The margin of safety against tensile rupture in the tangential direction is:

$$M.S. = (7{,}600/928) - 1 = 7.18 \quad (C7)$$

2. Shear Strength of ACME Threads

Assume that shear failure would take place along a cylindrical surface whose diameter is the pitch diameter of the ACME thread. Thus, the effective shear area is:

$$A_s = \pi D_p L\beta = \pi \times 2.184 \times 1.980 \times 0.5 = 6.792 \text{ in}^2 \quad (C8)$$

The axial load applied to the threads (assuming a closed nozzle condition) has the same value as was determined in Section A-2:

$$F_a = 1{,}049 \text{ lb} \tag{C9}$$

The average shear stress in the threads is:

$$\tau = F_a/A_s = 1{,}049/6.792 = 155 \text{ psi} \tag{C10}$$

where:

D_p = 2.188 inch = the pitch diameter of 2.25 x 8 ACME threads

L = 1.98 inch = the thread engagement length

β = 0.5 = a factor accounting for the fact that one-half of this area belongs to the female thread

The margin of safety against shear failure is given by:

$$M.S. = (5{,}800/155) - 1 = 36.4 \tag{C11}$$

3. Shear Strength of 1.5-9 NH Thread

The effective shear area of threads, assuming the length of engagement is 0.5 inch (based on formula from Reference C-2):

$$A_s = 2.356\ L\ (D_p - 1.0825/N)$$

$$= 2.356 \times 0.5\ (1.918 - 1.0825/9) = 2.118 \text{ psi} \tag{C12}$$

The average shear stress in the threads is:

$$\tau = F_x/A_s = (\pi/4)D_i^2p/A_s \tag{C13}$$

$$= (\pi/4)(1.50)^2 \times 300/2.118 = 250 \text{ psi}$$

where:

D_p = 1.918 inch = the pitch diameter of the thread

N = 9 = the number of threads per inch

L = 0.5 inch = the length of thread engagement

The margin of safety against shear failure is:

$$M.S. = (5{,}800/250) - 1 = 22.2 \tag{C14}$$

4. Shear-out Strength of Dowel Pin Holes

Location of the four dowel pins is shown in Figures C-1 and C-4. These dowel pins are used to secure the upstream orifice plate (see Figure C-1) to the downstream tube. In the maximum-flow condition the pressure differential across the upstream orifice plate is approximately 150 psi (see Figure C-1). The downstream pressure of 105 psi, Region B, has been determined elsewhere from hydrodynamic considerations. The net axial force acting on the orifice plate is evaluated as follows:

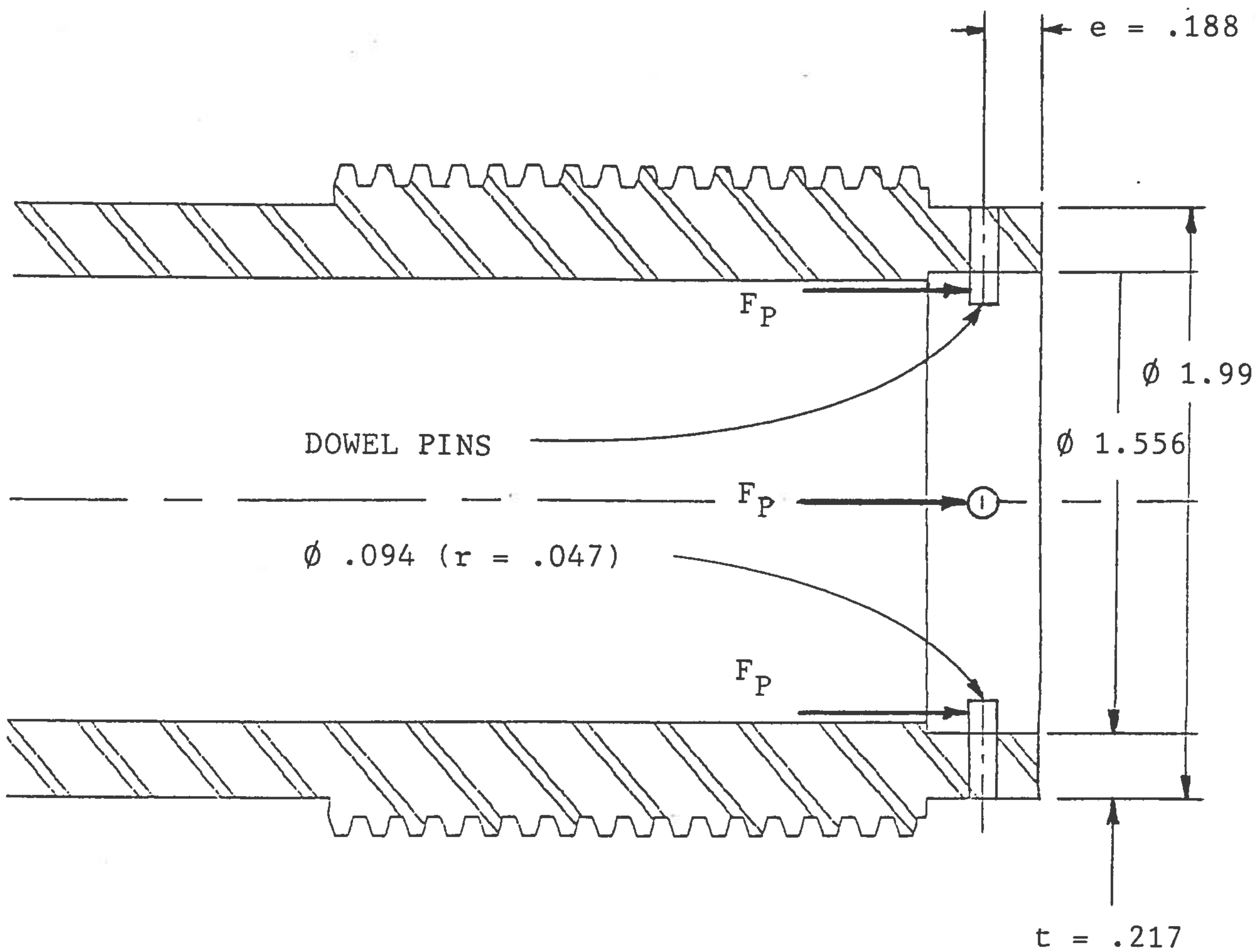


FIGURE C-4: DOWEL PIN LOCATIONS IN UPSTREAM TUBE

$$F_x = A_p \Delta p = 0.715 \times 150 = 107.2 \text{ lbs} \tag{C15}$$

The shear area that is effective in resisting the bearing load imposed by each dowel pin is given by:

$$A_s = 2(e - r \cos 45^\circ)t = 2(.188 - .047 \times .707)(.217)$$

$$= 0.067 \text{ inch}^2 \tag{C16}$$

where:

A_p = 0.715 inch^2 = the area of the orifice plate that is blocking the flow

Δp = 150 psi = the pressure differential across the orifice plate

e = 0.188 inch = edge distance of pins (Figure C-4)

r = 0.0468 inch = one-half the diameter of dowel pins

The average shear stress in the downstream tube material caused by the dowel pin loads is given by:

$$\tau = F_x/4A_s = 107.2/(4 \times .067) = 399 \text{ psi} \tag{C17}$$

The margin of safety against shear-out failure is:

$$\text{M.S.} = (5{,}800/399) - 1 = 13.5 \tag{C18}$$

C. OPERATING TORQUE

The torque required to rotate the downstream tube with respect to the upstream tube may be evaluated from the equation (Reference C-3):

$$T = F_a \left[\tfrac{1}{4}\mu D + \left(\frac{.866 \tan\lambda + \mu}{.866 - \mu \tan\lambda}\right)\left(\frac{D}{2} - \frac{.3248}{N}\right)\right] \qquad \text{(C19)}$$

$$= 1{,}049 \left[\frac{.08 \times 1.980}{4} + \left(\frac{.866 \times .0182 + .08}{.866 - .08 \times .0182}\right)\left(\frac{1.980}{2} - \frac{.3248}{8}\right)\right]$$

$$= 152 \text{ lb}$$

where:

F_a = 1,049 lb = the maximum axial laod acting on the ACME threads (see Section B-2)

D = 1.980 inch = the pitch diameter of ACME thread

μ = 0.08 = the coefficient of friction at the male/female thread interface

N = 8 = the number of ACME threads per inch

λ = 1.043° = the helix angle of threads

The torque value of 152 inch-lb corresponds to a blocked-flow condition (nozzle closed, fire hose pressurized with 300 psi). In the full-flow condition, when pressure in Region B is 150 psi, the value of the operating torque would be one-half of the above value or 76 inch-lb. Reference C-4 indicates that about 133 inch-lb may be applied by a twisting motion of the wrist or forearm of 90-percentile of adult human males. By comparing the calculated value to the torque-applying capacity of the 90-percentile human male, we see that an average-strength operator should be able to make dynamic adjustments to the flow modifier with no difficulty.

D. REFERENCES

C-1: Oberg, E., and F. D. Jones, Machinery's Handbook, The Industrial Press, New York, 18th Edition, 1979

C-2: Peterson, R. E., Stress Concentration Factors, J. Wiley & Sons, New York, 1974

C-3: Faupel, J. H., Engineering Design, J. Wiley & Sons, New York, 1964

C-4: HUMANSCALE: Human Strength Dial, designed by H. Dreyfuss Associates

APPENDIX D: LABORATORY TEST DATA FOR MODEL II PROTOTYPE

TABLE D-1

MODEL II PROTOTYPE LABORATORY TEST RESULTS

MODIFIER SETTING	FLOWRATE (gpm)	PRESSURE (psi)	P_R (psi)	Q/P gpm/psi
1(fully open)	40	11	20	133
	50	17	20	206
	60	13	19	216
	70	29	23	377
2	40	8	20	113
	50	13	21	180
	60	19	16	262
	70	22	21	328
3	40	$(14.7)_3$	$(14.3)_3$	153
	50	21	12	229
	60	$(19.5)_2$	$(13.)_2$	265
	70	28	16	370
4	40	18	17	170
	50	26	13	255
	60	40	8	379
	70	$(37.3)_3$	$(17.)_3$	428
5	40	31	13	223
	50	42	7	324
	60	56	3	449
	70	59	3	538
6(nearly closed)	40	46	5	271
	50	64	2.*	400
	60	85	0.*	553
	70	88	1.*	657

$(\ )_3$ indicates the average of three values.

$(\ )_2$ indicates the average of two values.

* indicates uncertain values near low end of the scale.