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REPORT STAT

INTERIM PROGRESS REPORT
ON THE
PROCESSOR DEVELOPMENT PROGRAM

STATINTL

February 1965

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

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FOREWORD

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[redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted]

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Approved: [redacted]

Research Manager

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ABSTRACT

This document presents the program undertaken and the results obtained in the period ending February 1965. Separate technical reports covering specific contract assignments started during this period are included.

SECTION 1
INTRODUCTION

1.1 GENERAL

The contract was signed July 1, 1964. For the purpose of this interim report, the program will be divided into the following major levels of effort: 1) Procurement and installation of a GFE cleanroom in which the GFE HTA-5 processor will be installed as a test vehicle. This installation will be used for an evaluation program to determine the cross effect of environment on film processing and vice versa. 2) A research program directed primarily, but not exclusively, toward improvement in the liquid/air bearing concept of film processing.

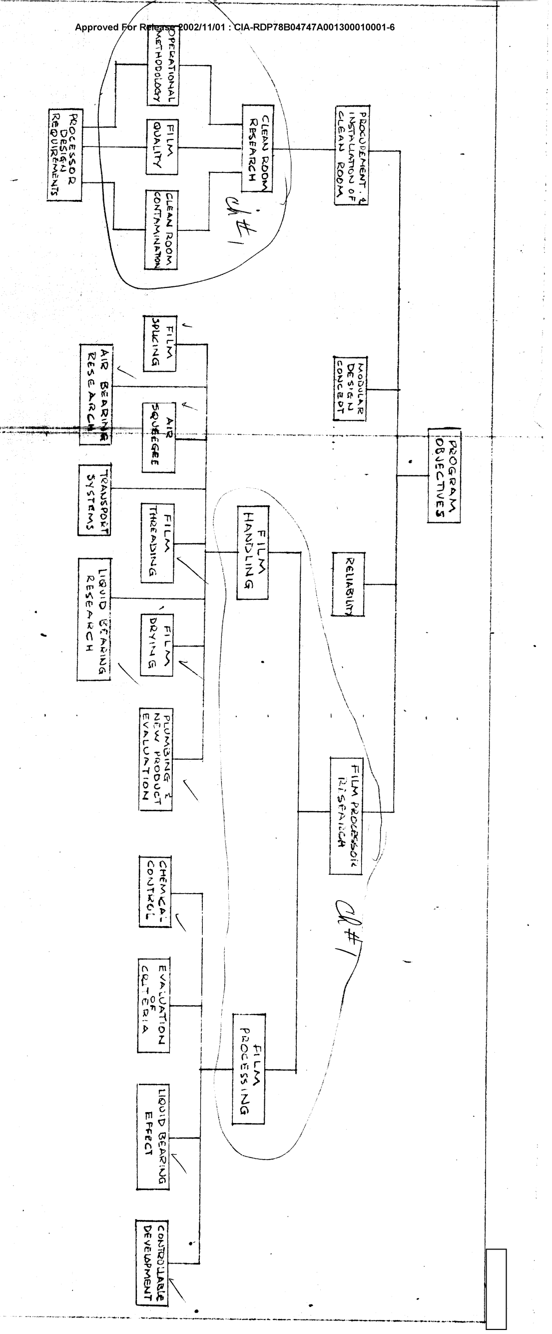
The broad objectives of the program are laid down in the Design Objectives of the contract. The portion of the total research program undertaken in this term of the contract is intended to provide essential design criteria as well as the development of specific items of equipment.

1.2 BACKGROUND

A typical immersion processor of conventional design usually consists of the following sections.

1) Load end, containing the film magazine or cabinet, and splicer.
2) Accumulator, to permit film splicing without interrupting the processing cycle.
3) Chemical and wash sections, which are basically tanks containing the various chemical solutions and wash water.
4) Drier section or cabinet.
5) Film takeup or spooling assembly.

Depending on the film type, width, and extent of additional facilities required, equipment such as a viewing panel, air squeegee, central



SECTION 1
FIG. I-1

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control system and many other features, may be provided. In the past, all processors of this general type have had one feature in common; film is carried through the entire processing cycle on a series of driven and idler rollers.

Much attention has been directed to detailed improvements in the film transport systems. In all machines, some form of ganged chain drive to some or all of the rollers is employed, in conjunction with the use of friction or magnetic particle clutches to equalize the tension of the film throughout the processor. Much attention has also been paid to the finish and coating of the rollers to minimize film slippage.

STATINTL In 1960, a concept was advanced by [redacted] STATINTL a subsidiary of [redacted] to minimize the mechanical handling of film during processing. This concept is based on the use of air and liquid bearings in place of the traditional rollers. The film is transported on cushions of liquid or air formed between the bearing and the film. The use of the air/liquid bearing-principle, even in the drying cabinet, virtually eliminates direct contact between the film and mechanical parts of the processor.

On the basis of this concept, and from demonstrations given on a working model, a contract was awarded for a prototype air/liquid bearing film processor, identified as the HTA-5. This processor was built and tested; much data about the functional requirements were obtained. However, some unexpected problems were encountered during the test program, indicating the need for careful test work and data analysis to provide advanced design criteria.

In addition to improvements resulting from the elimination of virtually all mechanical film handling, more efficient processing was obtained due to the impingement of the chemicals directly on the emulsion as it passed over the bearings. More efficient wash action was also obtained for the same reason. Effective film drying, using heated air in the drier air bearings, was obtained and a decrease in film tension during transport was noted.

On the problem side, difficulty was encountered in retaining stable liquid and air cushions capable of withstanding the loads generated at a high transport speed of 60 feet per minute. Adequate surplus water removal, film tracking, and large pressure losses in standard pipe fittings and associated plumbing were among the other problems encountered. An unfavorable total power consumption compared to that required for a conventional processor was also experienced.

1.3 OBJECTIVES

The objectives of this processor development program are clearly stated in the Development Objectives exhibit forming part of the contractual documents, and are summarized in Figure 1-1. The program in work in this phase of the contract is directed towards providing data on which the design of a future processor, modular in concept and designed for cleanroom operation, can be based. The work is authorized by the following assignments, summarized in Figure 1-2.

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SECTION 2

2.1 LABORATORY FACILITY AND CLEANROOM INSTALLATION

The configuration of the cleanroom was finalized in September, 1964, and the laboratory facilities were planned accordingly. Electrical and air conditioning requirements were engineered in October, 1964. A final conference was held at the [redacted], on October 29, 1964, at which time a detailed examination of all requirements was made. Agreement was reached with [redacted] on a statement of work forming Exhibit "A" of the subcontract. STATINTL STATINTL

The first delivery of cleanroom parts was scheduled for November 13, 1964. However, due to procurement difficulties, the delivery of the steel framing for the raised floor was delayed and erection of the cleanroom did not start until November 23, 1964. Prior to this date, architectural drawings of the extensive changes to the area housing the cleanrooms were prepared and the approval of local area authorities was obtained.

In order to isolate the cleanrooms and laboratory installation from the shop power system and to provide the power required, electrical engineering drawings were prepared for electrical services from the area and the provision of a new main power and distribution system.

A concrete pad designed to support the HTA-5 processor, was poured. Delivery of the floor framing was made within a few days of the revised date, and assembly of the framing started the first week in December, 1964.

The erection schedule is shown in Figure 2-1 and has been generally adhered to in spite of detailed changes found necessary as erection proceeded. These changes were made for safety, some to meet local area codes, and others to conform to the contract specification.

STATINTL A final progress meeting was held between personnel of [redacted] STATINTL [redacted] on January 20, 1965 to check general

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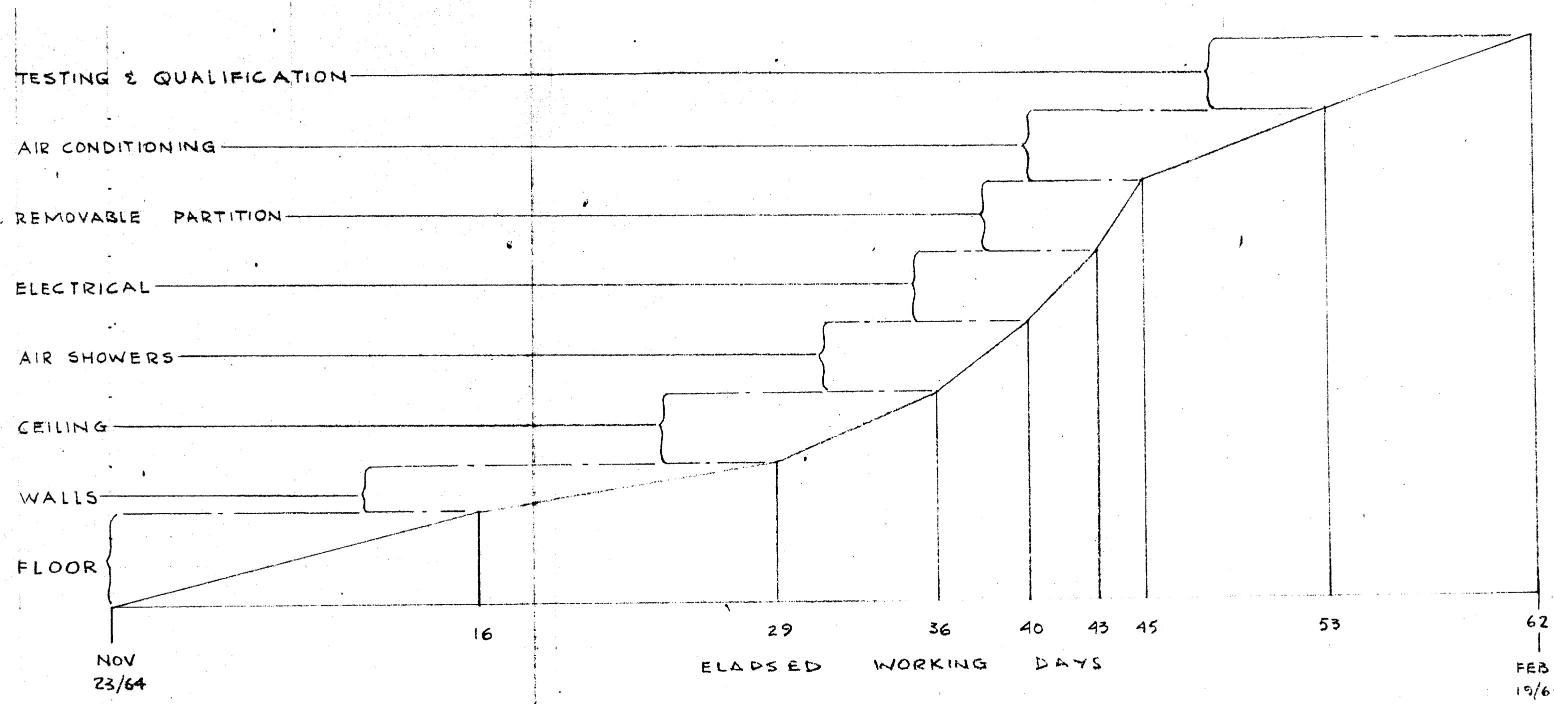
progress and to reach agreement on specific problems.

The layout of the complete facility is shown in Figure 2-2. Typical views of erection progress are shown in Figures 2-3 through 2-9.



SECTION 2
FIG. 2-1

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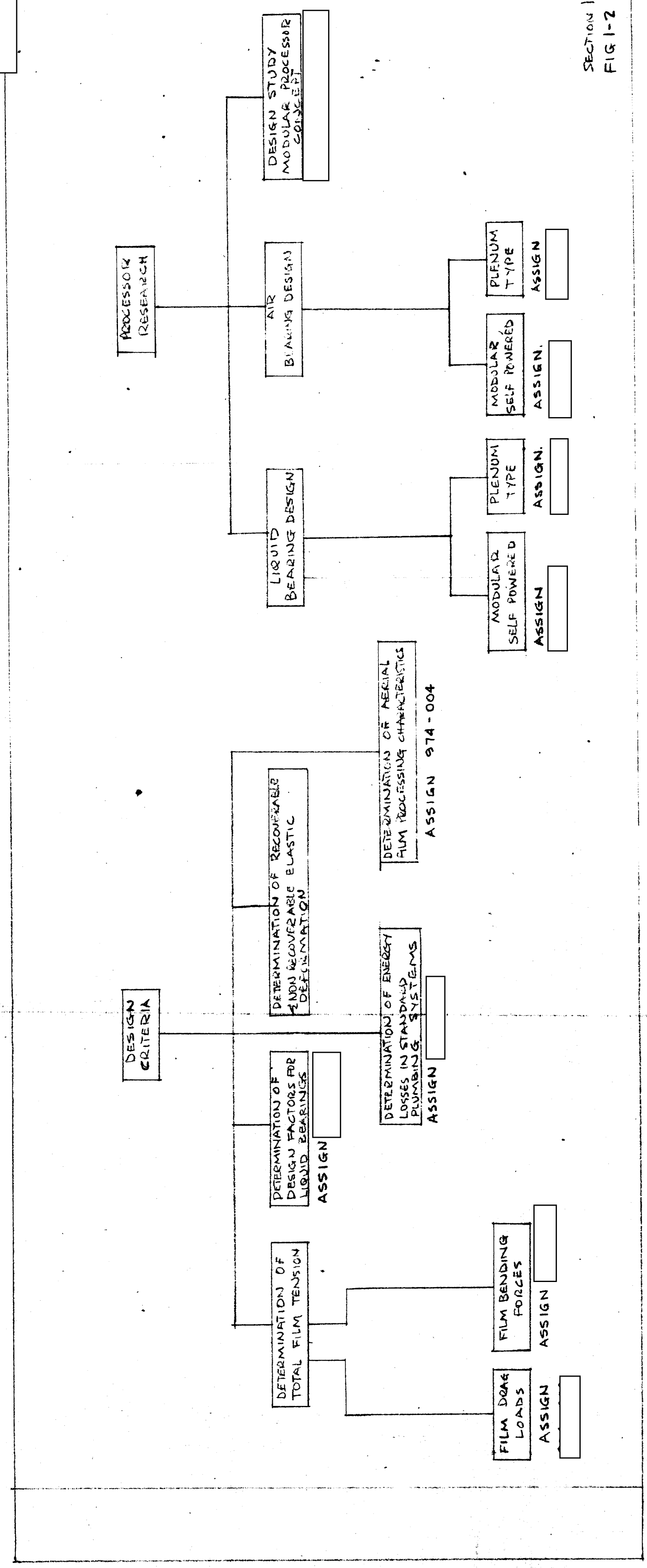

SECTION 1
FIG 1-2

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R AND D FACILITY AND CLEANROOM INSTALLATION

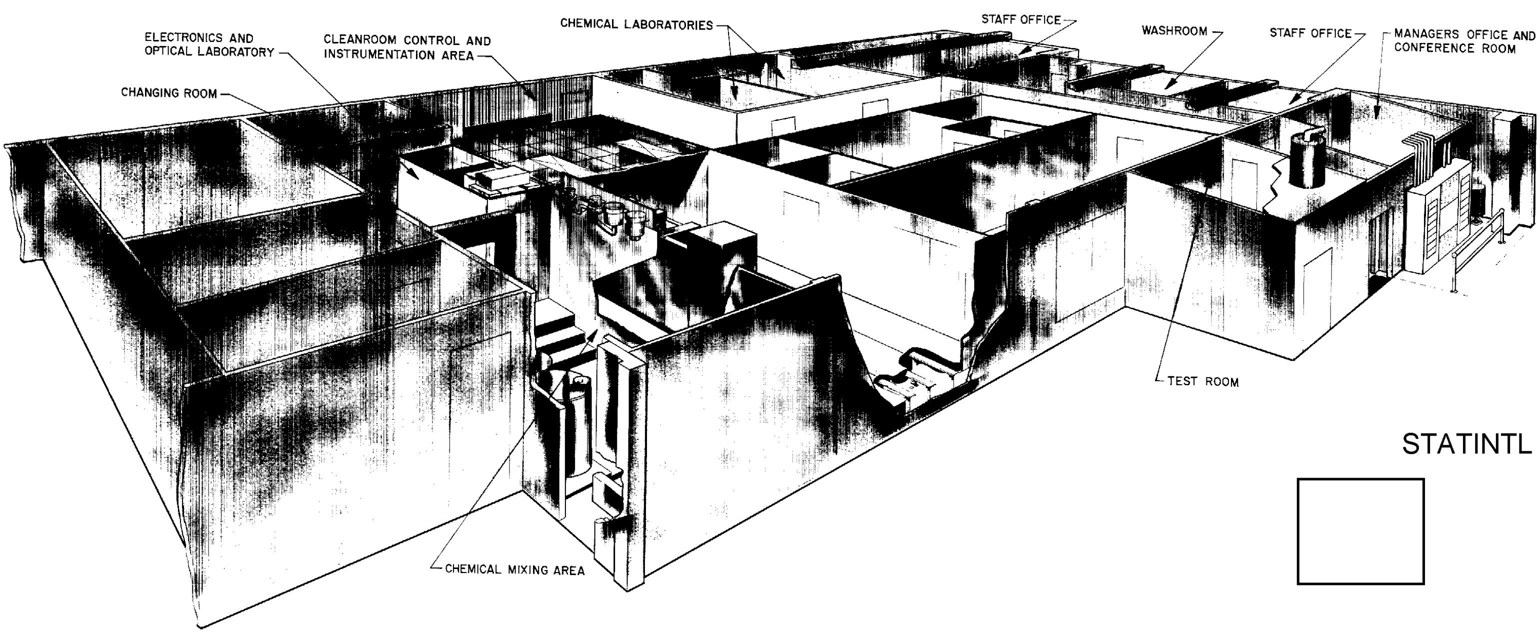

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FIG. 2-2

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FIG. 2-3

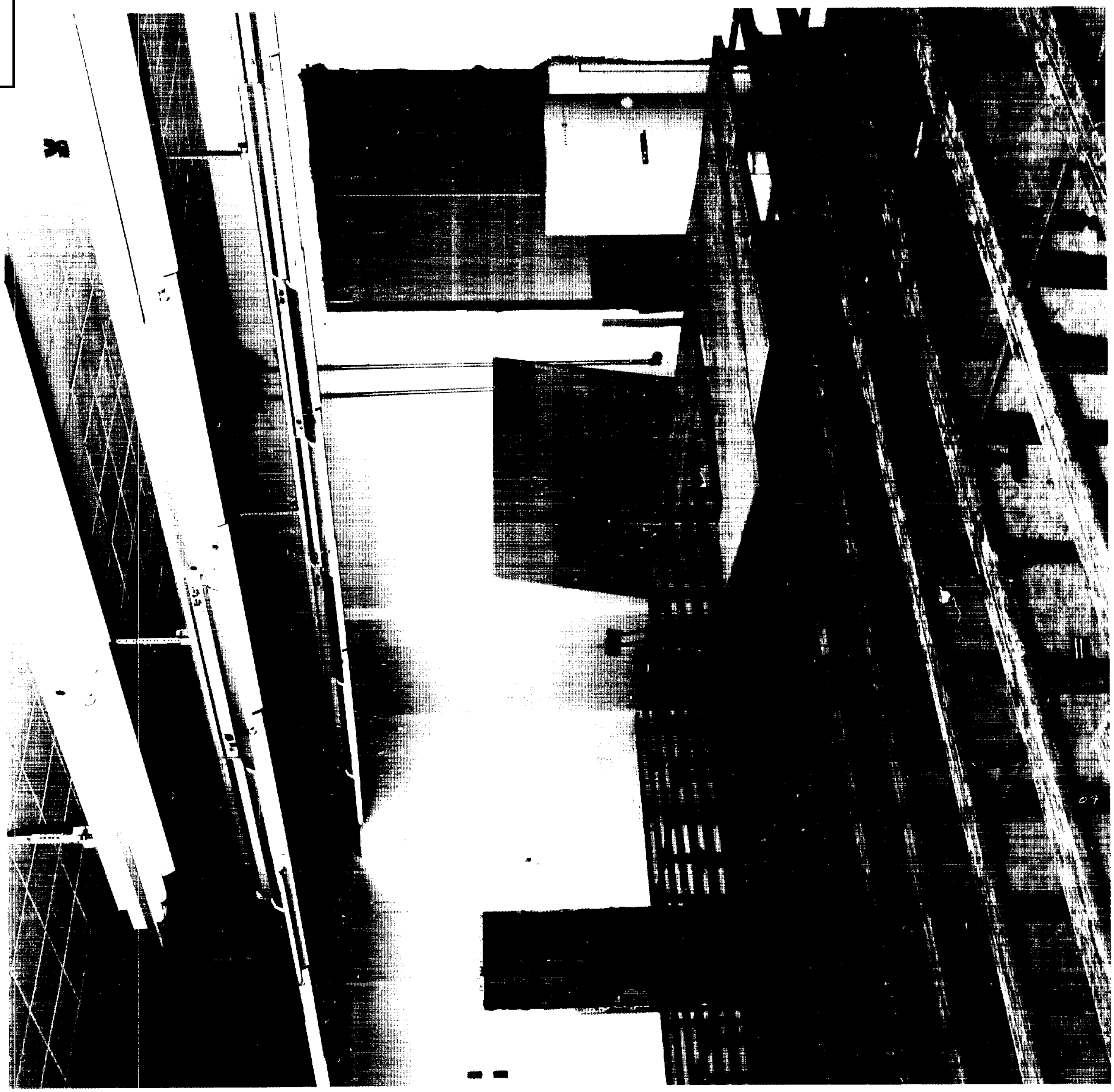

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FIG. 2-4

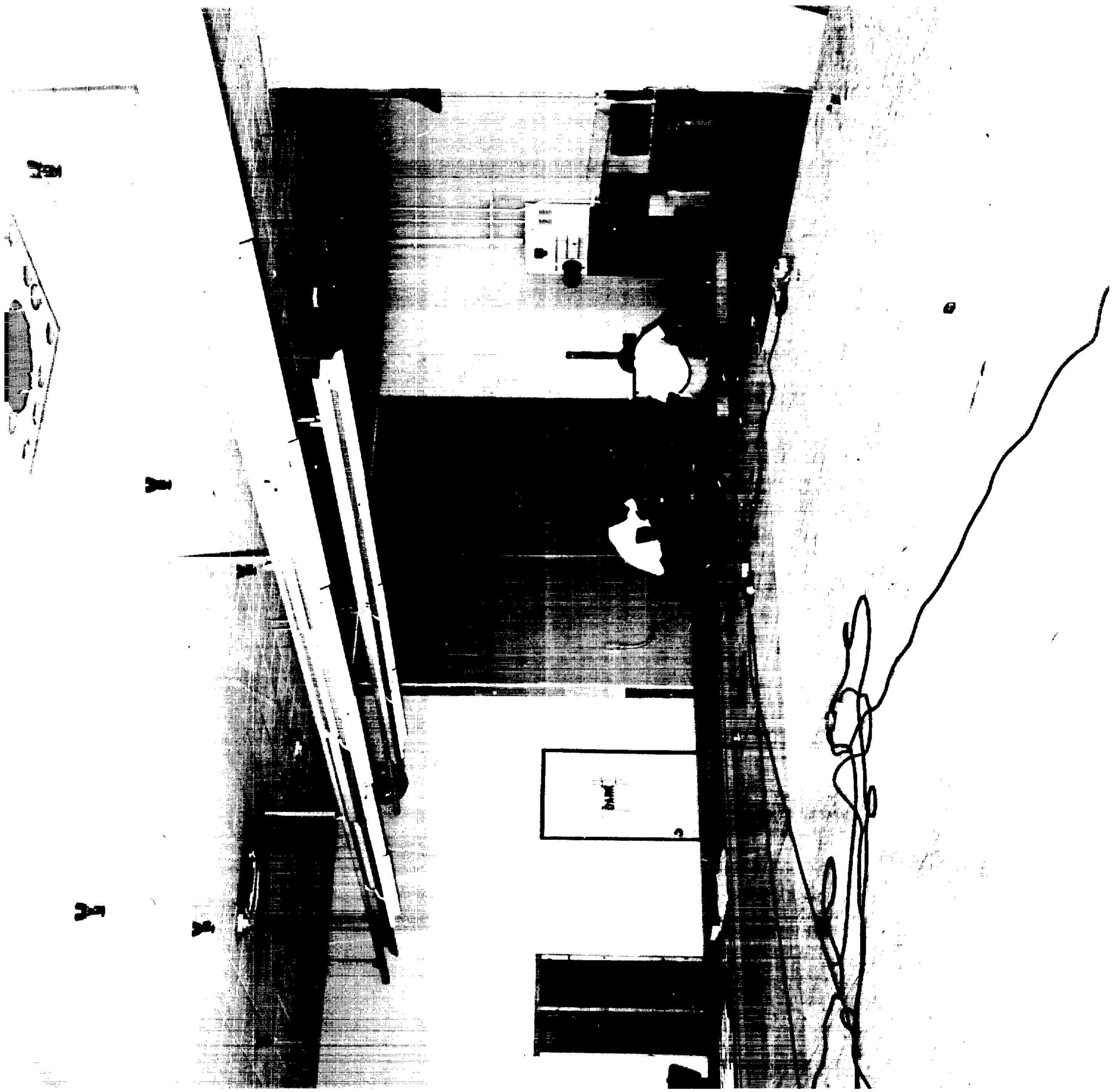

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FIG 2-5

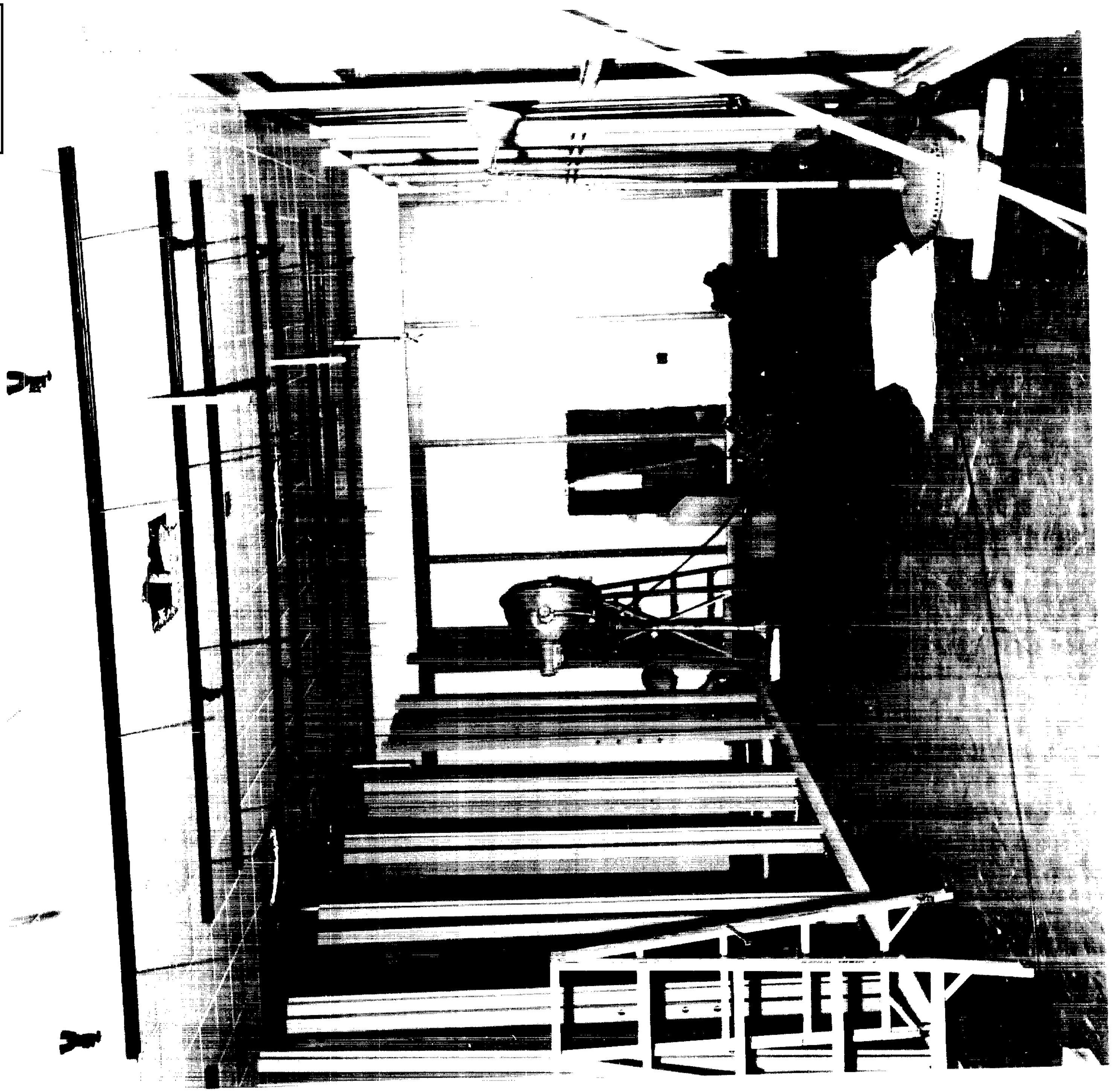

FIG. 2-6

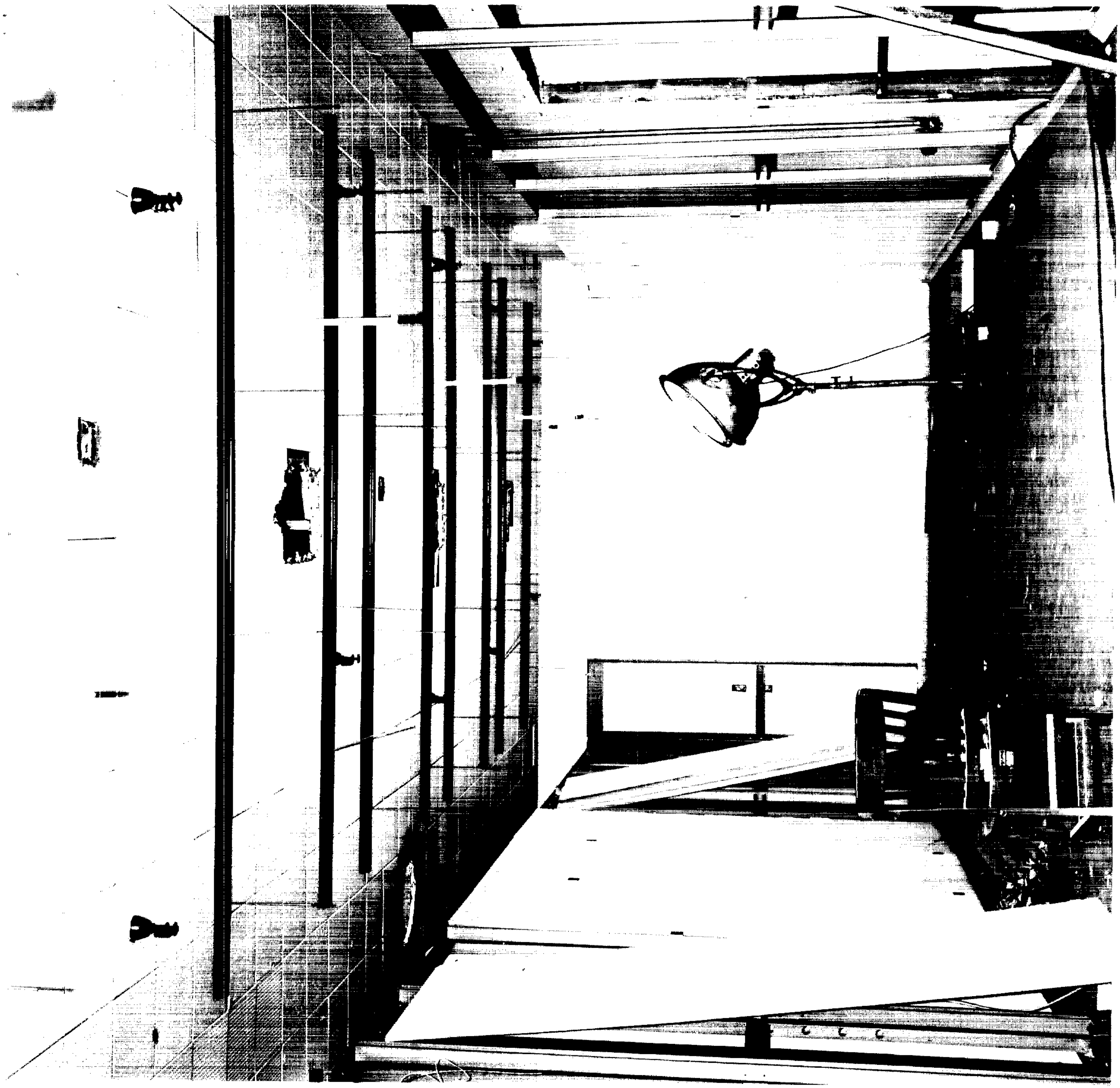

FIG 2-7

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FIG. 2-8

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FIG. 2-9

SECTION 3

3.1 ORGANIZATION

This contract is being handled in the Research and Development Department, the organization of which is shown in Figure 3-1. A research manager has been given responsibility for the program and work progress and a contract administrator has been assigned to the program to insure that contractual requirements are met.

3.2 PROGRAM CONTROL

Research projects are controlled by an assignment system. A project is selected from the main program (Figure 1-2) and an assignment form prepared. This form details the work required, the objective, and the proposed method of handling the project to obtain the desired data. On the back of the form is a summary of data from which the estimated engineering and manufacturing time and cost can be checked against actual time and cost incurred. The assignments are identified by the contract number prefixed to a three-digit numeral suffix [redacted] The report covering the investigation and the results, together with necessary charts, sketches, and photographs, is identified by the same number as the authorizing assignment. The assignment control flow chart is shown in Figure 3-2.

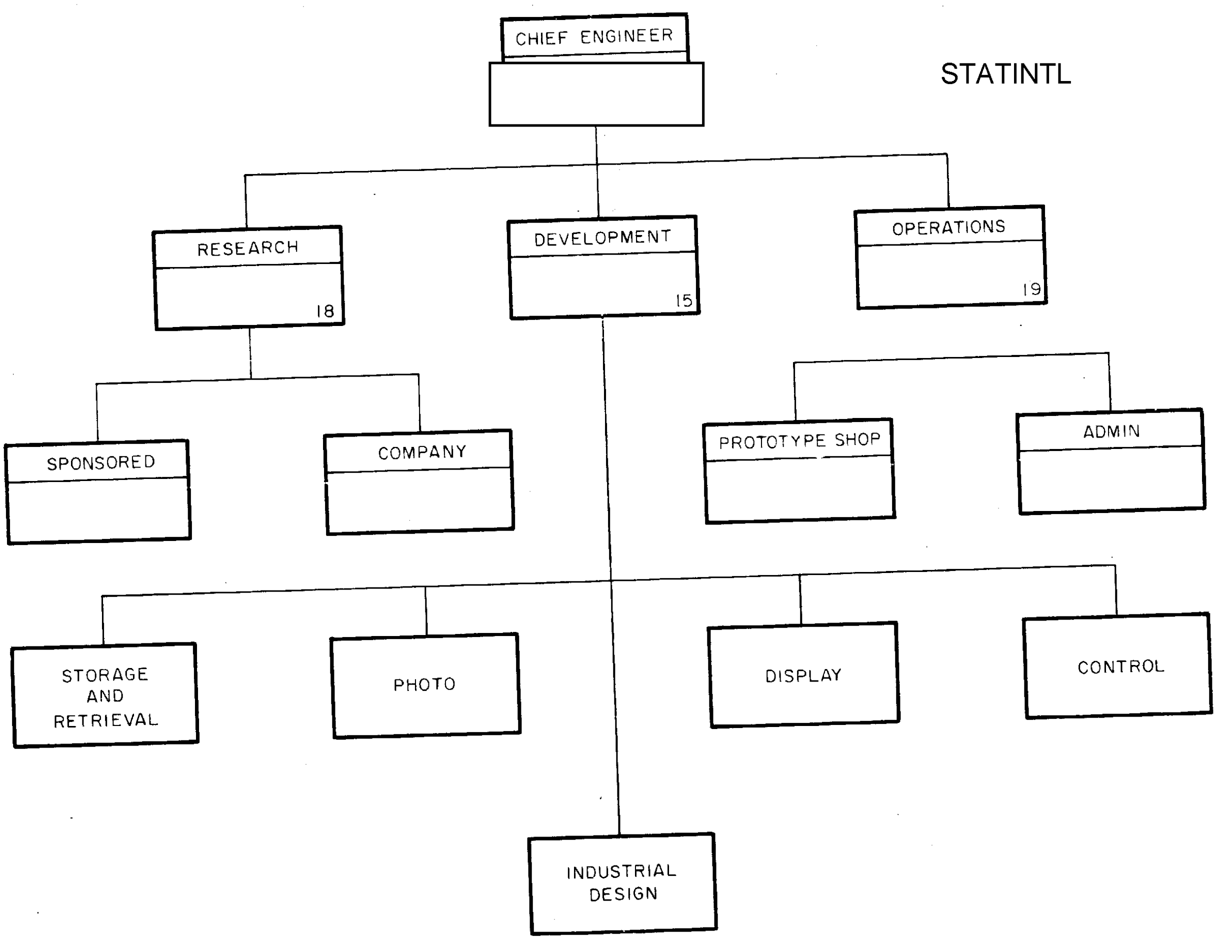

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Figure 3-1. R & D Organization

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R AND D CONTRACT ASSIGNMENT CONTROL

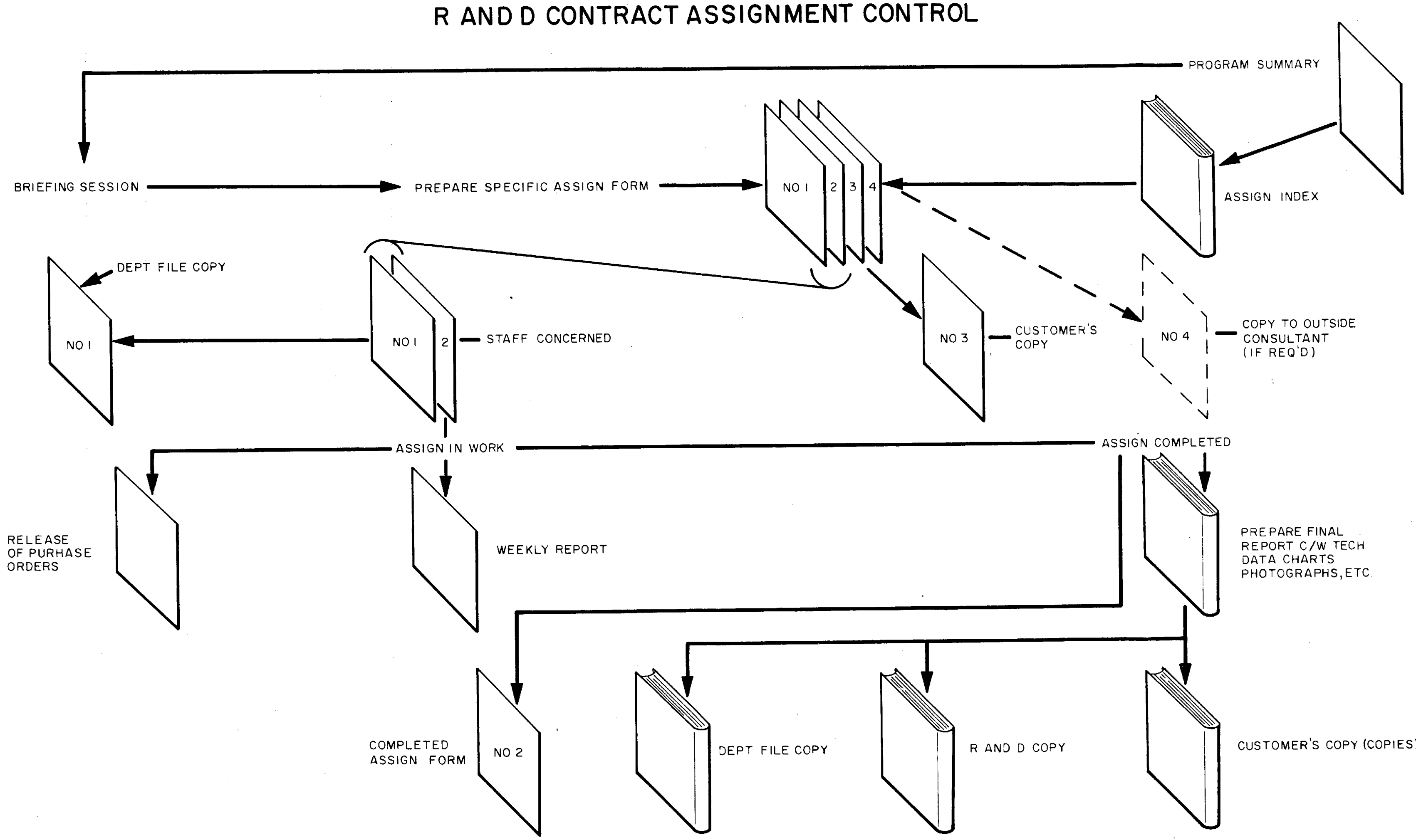

FIG. 3-2

SECTION 4
RESEARCH PROGRAM

4.1 GENERAL

The program objectives are to determine essential design criteria and to develop specific items of equipment that satisfy the conceptual and engineering advances called for in the contract.

The specific assignments detailed in Subsection 4.3 are directed toward this goal. These studies will concentrate on the development of a processor of compact modular design, lower power consumption, and the incorporation of improved liquid and air bearings.

All assignments listed in Subsection 4.2 are identified by their authorizing number and title. Each assignment is cross-referenced to the applicable section of the Development Objectives exhibit of the contract.

4.2 INDEX OF ASSIGNMENTS

Assignment	Description	Exhibit Ref.
	Measuring the pressure drop across standard PVC fittings	3-4
	Determining the force required to bend film	3-14
	Hydromatic liquid bearing assessment	3-1
	The effects of high processing temperatures on aerial films	3-8
	Comprehensive film processing data chart	3-8
	The performance evaluation of a positive-pressure transport capstan	3-3
	Calculating the efficiency of a tank with incorporated thermal control	3-8
	Evaluating the requirements of a cleanroom liquid/air bearing module	3-8

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Assignment	Description	Exhibit Ref.
[redacted]	Determining the coefficient of friction of film	3-14
	Determining the vertical spacing of liquid bearings	3-8
	Photographic record of cleanroom erection	1
	Designing and testing intertank air bearings	3-8 3-1
	Designing, constructing and testing a liquid bearing incorporating a built-in pump	3-8 3-1

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4.3 SPECIFIC ASSIGNMENTS

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4.3.1 Assignment [redacted]

In the HTA-5 prototype liquid/air bearings, pressure losses between the pumps and the bearings were experienced - these losses were in excess of the calculated allowance. In some cases, these losses necessitated the replacement of the pumps provided,for pumps of higher horsepower. This assignment was planned to accurately measure the pressure drop and flow across various standard PVC fittings.

It would be of undoubted value to extend this test series to include standard stainless-steel piping and fittings, and sanitary piping and fittings used in the food processing industry. This would then present a cross-reference of values against such parameters as cost, assembly time, reliability of joints, and difference in pump energy saved. With due regard to other investigations of higher priority, such an extension is beyond the present scope of funding.

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4.3.2 Assignment [redacted]

This assignment was issued to determine the forces required to bend film over rollers of varying diameters. These

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forces are one of the essential design parameters necessary in the calculation of the loads on bearings, required capstan torque, and film tension.

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4.3.3 Assignment

Various diameters of liquid bearings and slot configurations were tested under this assignment to provide a background for the design of an improved liquid bearing of the sleeve type. Much of the data obtained remains to be analyzed at this date in order to establish a mathematical relationship and to develop, if possible, formulae on which designs can be based. Based on the undigested data obtained, a bearing was designed with the following objectives in mind:

1) Eliminating edge guides
2) Reducing horsepower requirements
3) Attempting to achieve self-centering of film.

Testing of this bearing will continue during the remaining period of the contract.

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4.3.4 Assignment

To determine the best configuration for a processing module, it was necessary to obtain the range of processing speeds at given developing temperatures.

Since the concept of a modular processor provides three variable parameters in the form of transport speed (FPM), temperature, and the number of modules required for a given developing process, this assignment was planned to obtain data not only for these criteria, but also for the expected processed film quality in terms of fog level, contrast, resolution, speed and granularity at selected increments of processing temperature.

This assignment will continue until the end of the current contract.

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4.3.5 Assignment [redacted]

This assignment was planned to incorporate, in chart form, all the available data on original aerial negative and duplicating films, together with the results obtained from other assignments producing applicable and correlative data.

No report will be issued on this assignment at this date.

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4.3.6 Assignment [redacted]

This assignment was planned to investigate the performance of a vacuum capstan, the design of which was based on a new technique for generating and applying a vacuum to a rotating film drive capstan from positive air pressure. A plenum for applying positive pressure to the outside of film wrapped 180 degrees around the capstan was also manufactured. Due to other investigations of higher priority, however, further investigation of this technique is beyond the present scope of work.

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4.3.7 Assignment [redacted]

In the HTA-5 processor, the bearing pumps and the recirculation and temperature control system for each tank were mounted in separate assemblies some distance from the processor itself (Figure 4-1). This separation led to BTU gains in the tank cooling systems and losses in the heating system in addition to pressure losses through the pipes and fittings to and from the temperature control equipment.

This assignment authorized a study directed toward incorporating a method of thermal control in the tank itself, with the objective of reducing the overall machine size, complexity, and power consumption.

This study is incomplete at this time, since its progress is tied to developments resulting from Assignment [redacted] under which a design study of a modular tank assembly is being made. STAT

4.3.8 Assignment

This assignment initiated a design study to evaluate a processor designed on the modular principle specifically for cleanroom use. Ideally, each standard module would be self-contained and interchangeable with any other, thus making it possible to adapt to any specific processing requirement in black and white or color. Simply adding the proper number of self-aligning units, then, would permit maximum flexibility, enabling the processor to be "tailor-made" for each process.

In concept, the modules for the wet end would be constructed of double-wall stainless steel with internal insulation to minimize heat transfer. Integral plate-coils for heat exchange, plug-in bearings, self-contained filters, and temperature control would eliminate all external service-unit functions such as pumps, plumbing, and thermostatic exchangers. The optimal design would then be neat, compact, easy to clean and maintain, high in reliability, and extremely flexible in application.

Further assignments will be issued to study associated equipment requirements, including cleanroom requirements for the modular design of the accumulator, drier, and takeup assemblies. The same modular concept would be applied to the design of intertank plug-in air-transfer bearings.

4.3.9 Assignment

This assignment involves determination of film drag or, more correctly, skin friction of the film when pulled through water or near-water solutions. This determination is essential as a liquid/air bearing processor design parameter. The values obtained from the tests initiated by this assignment will provide the C_F constant in a formula developed for thin, flat plates. This is applicable when a correct value can be given to the coefficient of friction for film. Other factors such as bending forces and film tensioning devices must also be considered at this time. With a method of calculating the total tension in the film available, a logical sequence of processor design can be applied as follows:

1) Processor performance requirements are obtained initially from a customer's specification, which details the films to be processed, the processing rate in feet per minute, and the sensitometric characteristics required. Processing times and temperatures are obtained from this information.

2) With this criteria established, the physical sizes of the tanks, the immersed length of film, and the number of bearings required can be determined.

3) Using the formula previously described to obtain the total film tension load, the load on the individual liquid and air bearings and the transport capstan torque loads can then be calculated.

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4.3.10 Assignment [redacted]

This assignment was planned to determine the maximum vertical spacing of liquid bearings of 1-1/2, 1-3/4, and 2 inches nominal diameter plus a 1/8-inch built-up cushion.

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4.3.11 Assignment [redacted]

This assignment was issued for administrative purposes to provide a photographic record of the cleanroom erection. Some of these interim documentary prints of significant highlights of the schedule have already been appended to the Monthly Progress Reports submitted to the Contracting Officer. No separate report will be issued.

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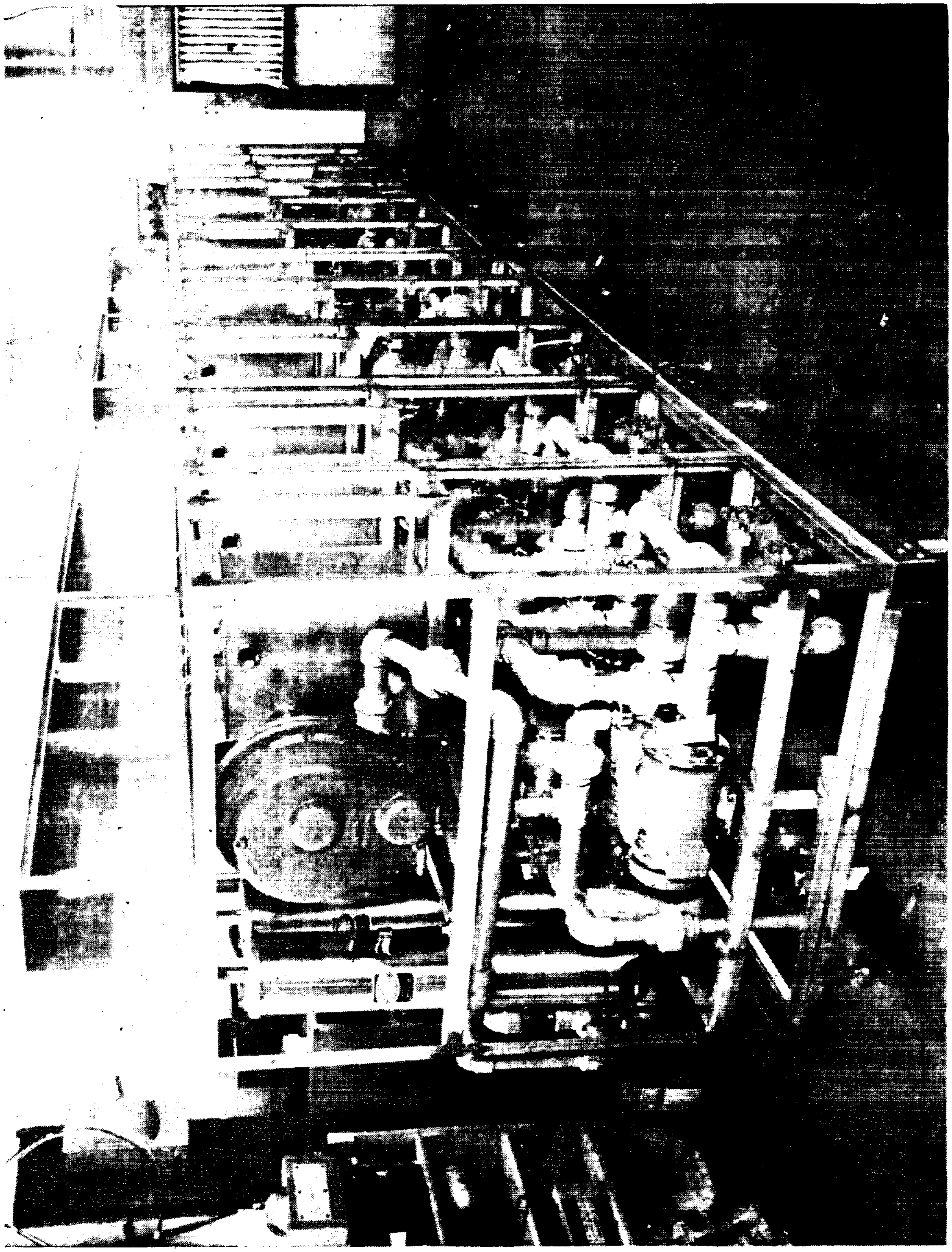

Figure 4-1. Service Unit Modules in Position

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REPORT

MEASUREMENT OF PRESSURE DROPS ACROSS STANDARD PIPE AND FITTINGS

Pressure Drop—
Pipe & Fittings

STATINTL

February 1965

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FOREWORD

[redacted] submits this report in compliance with Item 3.4 of the Development Objectives of [redacted] This report should be read in conjunction with Report [redacted] of which it forms part.

Approved: [redacted]

Research Manager

SECTION 1
INTRODUCTION

1.1 DATA LIMITATIONS

When the liquid bearing concept was first considered, the state-of-the-art in processor design required only the movement of film through the various steps of developing and fixing by means of rollers or sprockets. Hydrodynamic and fluid mechanic complexities introduced by the new bearing, in which the film was supported on a liquid cushion, required [redacted] engineers to depend heavily on available technical data - pump capacities, pipe and fitting losses, pressure drops through filters, and frictional coefficient buildup with photochemical deposits. Inadequacies in the published data parameters quickly became apparent when pump capacities had to be virtually doubled to compensate for line losses, even though supposedly ample design safety factors had been incorporated.

1.2 PURPOSE AND OBJECTIVES

One of the foremost objectives of the assignment was to satisfy the need for these missing parameters and provide, generally, a more complete technical documentation of fundamental engineering data germane to processor design. One important byproduct of the research program, then, was to eliminate rule-of-thumb calculations in which the pressure drop in a 45-degree elbow was assumed to be one-half of that for a 90-degree elbow, or that in a valve four times a 90-degree elbow, with a safety factor of 20 percent or better.

With the shortcomings of technical literature in mind, the objectives of the research project were formulated. The following list comprises the most important research objectives for this part of the program:

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1) Check as many different fittings (including straight pipe) as feasible in the light of time and budget.

2) Begin experimentation on 1-1/4-inch rigid polyvinyl chloride (PVC) pipe and threaded fittings. Measure Δp with unburred fittings and pipe. Repeat tests with burred fittings and internal taper.

3) Repeat tests outlined in objective (2) with socket-type fittings.

4) Repeat the series of tests with polished, sanitary stainless steel dairy pipe and fittings.

5) Determine the effect of pump inlet pipe size.

6) Determine the effect of restricted inlet pipe size.

7) Study input of pump, mechanical efficiency, losses, and the effect of a dropping head on pump output.

8) Make a long run breakdown test of pump, using actual photographic chemical solutions.

9) Check the interrelationship of pump outlet angle on delivered gpm.

10) Make effectivity comparisons among various types of flowmeters - rotameter, orifices, venturi, and newer types.

SECTION 2
TECHNICAL DISCUSSION

2.1 EQUIPMENT AND INSTRUMENTATION

The pressure drop test apparatus is illustrated in Figures 2-1, 2-2, 2-3, and 2-4. All instrumentation and fittings are described in detail in Appendix F. A stainless-steel hold tank, on loan from [redacted] formed the core of the circulatory setup. From its center bottom outlet, a 2-inch ID PVC pipe fed a 2-horsepower centrifugal pump.

On both the inlet and outlet sides of the pump, thermometer wells were provided for measuring T_1 and T_2 respectively. Unions were installed on both sides to enable easy removal of the unit without disturbing the rest of the apparatus. On the downstream side, a valved tee for drainage and a pressure gage to read P_1 were provided. The piping then led directly to a 1-1/4-inch ball throttling valve and the flowmeter, and from the latter to the remaining test apparatus. The test piping and fittings were all 1-1/4-inch PVC, with the exceptions noted (Appendix F). They were supported on two tiers by wooden racks.

The lower level was a straight run of pipe over 10 feet long; a riser led to the upper level and to a union leading to a tee. The left branch of the tee was arbitrarily designated Branch I and the right, Branch II. Each of these two branches returned to the hold tank. Branch I embodied three test fittings and Branch II, four. The wooden supporting racks were carefully leveled so that both the upper and lower stages were precisely horizontal. Each fitting was provided with an upstream and downstream pressure tap for Δp measurement. These consisted of holes drilled and tapped for 1/8-inch standard pipe thread. The tapping depth was controlled so that when the flanged brass tubing adapters were screwed in, their bottoms would be flush with the inside of the pipe in accordance with Hydraulic

Institute Standards (Ref: 11). The test apparatus was completely assembled from a scale drawing by two shop plumbers. The only specific instruction given them was to use standard shop practice in cutting, fitting, and threading pipe and to use "Proseal" (flexible two-component epoxy mixture) in making up the joints. The completed test rack closely approximated the assembly technique incorporated in any standard [redacted] gear.

All pressure drops were measured with either a vertical U-tube manometer or a sensitive inclined mercury manometer. The tank temperature, T_3, was measured with an accurate Centigrade thermometer.

2.2 PRESSURE DROP EXPERIMENTATION

The first step in the research project was the calibration of the flowmeter. This was done by accurately timing, with a stopwatch, the filling of a standard bucket whose exact capacity had been measured. Enough runs were made at each 2 gpm flow increment on the rotameter scale to assure an accurate mean average. The data are presented graphically in Figure 2-5 and tabularly in Table 2-1. Based on the same data, the Reynolds numbers were calculated and plotted against friction coefficients (both are dimensionless) for PVC pipe (Figure 2-6). The data for various commercial pipes and tubes were obtained from the literature (References 1 and 2). It is interesting to note how much less the coefficients of friction are for plastic than for glass, supposedly the epitome of smoothness.

The pressure drops and Reynolds numbers were measured on the horizontal 10.020-foot section of the 1-1/4-inch PVC pipe (lower level). So that the total pressure drop for the section could be measured simultaneously, a long 1/4-inch diameter copper tube was connected to the upstream pressure tap and brought to the downstream end. When all lines were bled free of air, the readings were taken on the inclined mercury manometer.

Time, rate of flow, temperature, inlet pressure, and pressure drop were recorded in a typical series of tests. The flow was changed from maximum to minimum rotameter readings in 5 gpm increments. Enough rechecks were made to assure reproducibility of readings. As the tests progressed, it was found better practice to proceed from the lowest to the highest flow reading. Use of this technique resulted in less overall temperature variation (since the tank was nonadiabatic) for a series which might take as long as 26 minutes. Corrections for density, viscosity, etc., with temperature were made in the observed results (Appendix A).

Since the accurate calorimetric thermometers used were of the total immersion type, stem temperatures were recorded during the early runs. A sample calculation (Appendix B) showed the stem correction to be negligible in the 69^{o} to 77^{o}F ambient operating temperature range used, so it was neglected.

Pressure drops on the 1-1/4-inch tee were recorded across each leg independently, with the opposite leg blocked off, and again with both legs open. Data obtained for pressure drops with both legs of the tee open were omitted because their intervariation was slight and in all cases, the readings were less than those with one leg blocked off. Since design would be based on maximums, these data lost their significance. Note that the pressure drops across the leg leading to Branch II were higher than those leading to Branch I. Two explanations are possible: 1) An internal aberration in the plastic die not removed by the burring operation was responsible, or 2) The increased pressure drop in Branch II (in all cases higher than Branch I) was reflected back to the leg of the tee.

In only three instances could comparable data be found in published charts, those for straight pipe, a 90-degree elbow, and a tee. These are presented, together with our data, in Figures 2-7, 2-8, and 2-9. Some of the proprietary data seems overly optimistic. Note that the Cherry-Burrell

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data is not exactly comparable, since the closest size to our 1-1/4-inch ID pipe is their 1.402-inch stainless.

The remaining data are presented graphically in Figures 2-10 through 2-19 and tabularly in Tables 2-2 through 2-20. Each set of data presents a comparison between pressure drops in the fittings with unreamed pipe and with reamed pipe. The latter data were obtained in the following manner. After all tests were made on the original setup, the components were carefully identified and the apparatus completely disassembled. All fittings were internally deburred and each end of the connecting pipes faired with a special tool (Figure 2-20). The apparatus was then reassembled with Proseal in exactly the original order and orientation. With no other change, the flow was increased 6.7 percent. This result points to possible economies in reduced pump sizing on large production machines.

When the test apparatus was first assembled, a source of sweep fittings to check against the common, standard pipe thread, short-turn types could not be located. Continued market research uncovered a line of specialized electrical conduit fittings manufactured by Kraloy. The tests were subsequently performed on two of these PVC Schedule 40 conduit turns (Figure 2-13) fitted with female adapters, slip to thread. The pressure drops in the 90-degree sweep elbow (Figure 2-12 and Table 2-14) were almost exactly equal to those of a straight pipe of equivalent length. In neither the case of the 45-degree sweep nor that of the short-turn elbow were the pressure drops half of those of the 90-degree elbow. They were more. This phenomenon cannot be explained by inaccuracies of mensuration (see discussion of errors, Appendix C). The appendix also includes calculations of pump heads and effect of discharge angle on delivery.

TABLE 2-1
FLOWMETER CALIBRATION DATA AND REYNOLDS NUMBER CAL CULATION

Flowmeter Reading gpm	Measured Flow gpm	T_2 °F	ρ gm/ml	ft^3/sec	V ft/sec	$1/V^2$ sec^2/ft^2	f	μ lb/ft sec	$1/\mu$ ft. sec./lb.	ρ $lb./ft^3$	R_ϵ
41.2	40.8	71.05	.99720	.0893	10.36	.00932	.00218	.000608	1645	62.25	1.11×10^5
39.6	40.8	72.55									
37.7	38.2	72.75	.99718	.0833	9.66	.01072	.00214	.000607	1649	62.25	1.04×10^5
35.8	38.2	72.85									
33.9	33.8	72.95	.99716	.0724	8.40	.01417	.00215	.000606	1651	62.25	9.05×10^4
32.1	31.2	72.45									
30.2	29.6	72.55									
28.3	26.9	72.65	.99716	.0618	7.17	.01945	.00228	.000605	1652	62.25	7.73×10^4
26.3	26.3	72.70									
24.4	23.7	72.85									
22.5	21.5	72.90	.99714	.0509	5.90	.02873	.00349	.000605	1654	62.25	6.37×10^4
20.6	19.7	73.00									
18.7	18.2	73.10	.99713	.0400	4.64	.04645	.00259	.000604	1655	62.25	5.01×10^4
16.7	15.8	73.20									
14.7	13.9	73.30	.99712	.0291	3.38	.08754	.00315	.000603	1657	62.25	3.65×10^4
12.8	11.1	73.45									
10.9	10.3	73.50									
9.0	7.9	73.55	.99710	.0182	2.11	.2246	.00332	.000603	1660	62.25	2.29×10^4
6.9	7.2	73.65									
4.9	4.5	73.75									
-	2.6	74.05	.99706	.0071	..823	1.476	.00417	.000600	1667	62.25	8.95×10^3

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Table 2-2

Head Loss In Feet Of Water/100 Feet Of 1-1/4-Inch PVC Pipe

(Unreamed)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	Δ ø Zero	High	Zero	Low	Total	Total/2	Ft. Water $\times \frac{13.56}{12} - 1$	Ft. Loss/100' $\times \frac{100}{10.02}$		
40.6	41.1	76.08	13.9	.03	2.95	.54	4.25	6.63	3.32	3.48	34.73		
37.7	37.7	76.21	14.4		2.46		3.78	5.67	2.84	2.97	29.64		
33.0	32.6	76.32	15.1		1.86		3.00	4.29	2.15	2.25	22.46		
28.3	27.8	76.39	15.7		1.45		2.44	3.32	1.66	1.74	17.37		
23.5	22.9	26.45	16.2		1.07		1.95	2.45	1.23	1.29	12.87		
18.7	18.0	76.56	16.7		.68		1.47	1.58	.79	.83	8.28		
13.8	13.0	76.62	17.3		.40		1.19	1.02	.51	.53	55.29		
9.0	8.2	76.76	17.8		.17		.81	.41	.21	.22	2.19		
4.1	4.1	77.06	18.3		.03		.61	.07	.04	.04	.40		
40.7	41.2	76.65	13.9		2.88		4.28	6.59	3.30	3.46	34.53		

Table 2-3

Head Loss In Feet Of Water/100 Feet Of 1-1/4-Inch PVC Pipe

(Reamed)

43.2	44.3	70.30	13.6	.53	4.63	.03	3.57	7.64	3.82	4.00	39.92		
40.0	40.4	70.25	14.1		4.07		3.12	6.63	3.32	3.48	34.73		
35.0	34.7	70.22	14.9		3.29		2.43	5.16	2.58	2.70	26.95		
30.0	29.5	70.16	15.6		2.57		1.86	3.87	1.94	2.03	20.26		
25.0	24.4	70.15	16.0		2.00		1.43	2.87	1.44	1.51	15.07		
20.0	19.3	70.13	16.5		1.48		.96	1.88	.94	.98	9.78		
15.0	14.2	70.15	16.9		1.04		.59	1.07	.54	.57	5.69		
10.0	9.2	70.15	17.8		.74		.32	.50	.25	.26	2.59		
5.0	4.8	70.27	18.2		.56		.13	.13	.07	.07	.70		

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Table 2-4

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Union

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	$\Delta\phi$ Zero	High	Zero	Low	Total	Total/2	Ft. Water x $\frac{13.56-1}{12}$	Correction Factor Ft. Loss/Inch	Tap Distance x 14.4 In.	Corrected Reading Hd Loss in Ft.
40.5	41.0	70.33	13.8	.25	1.14	.09	.57	1.37	.69	.72	.0297	-.43	.29
35.0	34.7	70.35	14.6		.90		.50	1.06	.53	.55	.0211	-.30	.25
30.0	29.5	70.42	15.4		.77		.43	.86	.43	.45	.0158	-.23	.22
25.0	24.4	70.47	15.9		.67		.37	.70	.35	.37	.0117	-.17	.20
20.0	19.3	70.55	16.6		.57		.32	.55	.28	.29	.0082	-.12	.17
15.0	14.2	70.62	17.1		.47		.27	.40	.20	.21	.0051	-.07	.14
10.0	9.2	70.67	17.6		.40		.23	.29	.15	.16	.0024	-.03	.13
5.0	4.8	70.69	18.1		.33		.23	.22	.11	.11	.0009	-.01	.10

Table 2-5

Head Loss In Feet Of Water For 1-1/2" Rigid PVC Union

(Reamed Pipe)

43.2	44.3	70.61	13.6	.50	1.48	.08	.52	1.42	.71	.74	.0333	-.48	.26
40.0	40.4	70.56	14.2		1.37		.43	1.22	.61	.64	.0289	-.42	.24
35.0	34.7	70.55	15.0		1.17		.36	.95	.48	.50	.0225	-.32	.18
30.0	29.5	70.50	15.6		.98		.28	.68	.34	.36	.0169	-.24	.12
25.0	24.4	70.42	16.0		.85		.25	.52	.26	.27	.0126	-.18	.09
20.0	19.3	70.40	16.7		.72		.19	.33	.17	.18	.00816	-.12	.06
15.0	14.2	70.40	17.1		.62		.16	.20	.10	.11	.00475	-.07	.04
10.0	9.2	70.40	17.7		.55		.12	.09	.05	.05	.00217	-.03	.02

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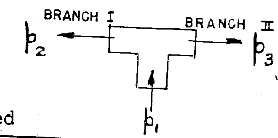
Table 2-6

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Tee

($\angle \phi_1 \phi_2$ For Unreamed Pipe) Branch I Open - Branch II Closed



Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	$\Delta \phi_1 \phi_2$ Zero	High	Zero	Low	Total	Total/2	Ft. Water $x \frac{13.56}{12} - 1$	Correction Factor Ft. Loss/In.	Tab Distance x14.45	Corrected Reading Hd Loss in Ft.
40.5	41.0	70.52	13.9	.25	2.90	.09	2.44	5.00	2.50	2.62	.0297	-.43	2.19
35.0	34.7	70.56	14.7		2.26		1.97	3.89	1.95	2.04	.0211	-.31	1.73
30.0	39.5	70.58	15.6		1.77		1.48	2.91	1.46	1.53	.0158	-.23	1.30
25.0	24.4	70.62	15.8		1.36		1.12	2.14	1.07	1.12	.0117	-.17	.95
20.0	19.3	70.66	16.6		1.00		.80	1.46	.73	.76	.0082	-.12	.64
15.0	14.2	70.73	17.1		.72		.54	.92	.46	.48	.0051	-.07	.41
10.0	9.2	70.82	17.5		.51		.36	.53	.27	.28	.0024	-.03	.25
5.0	4.8	70.94	18.2		.35		.21	.22	.11	.12	.0009	-.01	.11

Table 2-7

Head Loss In Feet Of Water For 1-1/4 Rigid PVC Tee

($\Delta \phi_1 \phi_3$ For Unreamed) Branch I Closed - Branch II Open

				$\Delta \phi_1 \phi_3$ Zero	High	Zero	Low					x14.55	
40.0	40.4	71.33	13.8	.26	3.30	.10	2.64	5.58	2.79	2.92	.0289	-.42	2.50
35.0	34.7	71.42	14.8		2.66		2.34	4.64	2.32	2.43	.0211	-.31	2.01
30.0	29.5	71.45	15.5		2.13		1.74	3.51	1.76	1.84	.0158	-.23	1.61
25.0	24.4	71.47	16.1		1.62		1.39	2.65	1.33	1.39	.0117	-.17	1.22
20.0	19.3	71.52	16.7		1.20		1.02	1.86	.93	.97	.0082	-.12	.85
15.0	14.2	71.58	17.2		.89		.69	1.22	.61	.64	.0051	-.07	.57
10.0	9.2	71.62	17.6		.60		.42	.66	.33	.35	.0024	-.03	.32
5.0	4.8	71.75	18.0		.42		.26	.32	.16	.17	.0009	-.01	.16

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Table 2-8

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Tee

(Δp_1 p_2 For Reamed) Branch I Open - Branch II Closed

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	Zero	Δp_1 High	p_2 Zero	Low	Total	Total/2	Ft. Water x $\frac{13.56}{12}$ -1	Correction Factor Ft. Loss/In.	Tab Distance x 14.45	Corrected Reading Hd Loss in Ft.
43.2	44.3	70.82	13.8	.51	3.28	.08	2.63	5.32	2.66	2.78	.0333	-.48	2.30
40.0	40.4	70.80	14.3		2.85		2.24	4.50	2.25	2.35	.0289	-.42	1.93
35.0	34.7	70.73	15.0		2.33		1.76	3.50	1.75	1.83	.0225	-.33	1.50
30.0	29.5	70.70	15.6		1.83		1.32	2.56	1.28	1.34	.0169	-.24	1.10'
25.0	24.4	70.67	16.0		1.47		1.00	1.88	.94	.98	.0126	-.18	.80
20.0	19.3	70.66	16.6		1.09		.68	1.18	.59	.62	.00816	-.12	.50
15.0	14.2	70.71	17.1		.87		.44	.72	.36	.38	.00475	-.07	.31
10.0	9.2	70.70	17.7		.66		.27	.34	.17	.18	.00217	-.03	.15
5.0	4.8	70.90	18.1		.53		.13	.07	.04	.04	.00058	-.01	.03

Table 2-9

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Tee

(Δp_1 p_2 For Reamed) Branch I Closed - Branch II Open

42.7	43.7	69.38	13.8	.51	3.38	.08	2.71	5.50	2.75	2.88	.0327	-.48	2.40
40.0	40.4	69.38	14.2		3.02		2.41	4.84	2.42	2.58	.0289	-.42	2.11
35.0	34.7	69.37	14.9		2.46		1.89	3.76	1.88	1.97	.0225	-.33	1.64
30.0	29.5	69.37	15.5		1.97		1.46	2.84	1.42	1.49	.0164	-.25	1.24
25.0	24.4	69.38	16.0		1.54		1.07	2.02	1.01	1.06	.0126	-.18	.88
20.00	19.3	69.40	16.6		1.20		.73	1.34	.67	.70	.00816	-.12	.58
15.0	14.2	69.40	17.1		.97		.51	.89	.45	.47	.00475	-.07	.40
10.0	9.2	69.55	17.6		.72		.30	.43	.22	.23	.00217	-.03	.20
5.0	4.8	69.60	18.2		.55		.13	.09	.05	.05	.00058	-.01	.04

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Table 2-10

Head Loss In Feet Of Water For 1-1/4" Rigid PVC 90° Elbow

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	$\Delta\phi$ Zero	High	Zero	Low	Total	Total/2	Ft. Water x $\frac{13.56}{12} - 1$	Correction Factor Ft. Loss/In.	Tab Distance x14.45	Corrected Reading Hd Loss in Ft.
40.5	41.0	69.97	13.8	.50	2.76	.05	2.12	4.33	2.17	2.27	.0297	-.46	1.81
35.0	34.7	70.06	14.7		2.19		1.62	3.26	1.63	1.71	.0211	-.33	1.38
30.0	39.5	70.12	15.3		1.73		1.21	2.39	1.20	1.26	.0158	-.24	1.02
25.0	24.4	70.17	15.9		1.38		.89	1.72	.86	.90	.0117	-.18	.72
20.0	19.3	70.22	16.6		1.05		.62	1.12	.56	.59	.0082	-.13	.46
15.0	14.2	70.31	16.9		.83		.40	.68	.34	.36	.0051	-.08	.28
10.0	9.2	70.36	17.6		.63		.23	.31	.16	.17	.0024	-.04	.13
5.0	4.8	70.52	18.1		.52		.13	.10	.05	.05	.0009	-.01	.04

Table 2-11

Head Loss In Feet of Water For 1-1/4" Rigid PVC 90° Elbow

(Reamed Pipe)

43.2	44.3	71.66	13.8	.37	2.38	.20	1.98	3.79	1.90	1.99	.0333	-.51	1.48
40.0	40.4	71.65	14.3		2.05		1.72	3.20	1.60	1.68	.0289	-.45	1.28
35.0	34.7	71.65	14.9		1.64		1.33	2.40	1.20	1.25	.0225	-.35	.90
30.0	29.5	71.63	15.6		1.35		1.09	1.87	.94	.99	.0169	-.26	.73
25.0	24.4	71.64	15.9		1.12		.77	1.32	.66	.69	.0126	-.19	.50
20.0	19.3	71.65	16.5		.81		.62	.86	.43	.45	.00816	-.13	.32
15.0	14.2	71.65	16.9		.65		.47	.55	.28	.29	.00475	-.07	.22
10.0	9.2	71.67	17.4		.47		.32	.22	.11	.12	.00217	-.03	.09
5.0	4.8	71.80	18.3		.40		.23	.06	.03	.03	.00058	-.01	.02

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Table 2-12

Head Loss In Feet Of Water For 1-1/4" Flanged 90° Elbow

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	$\Delta\phi$ Zero	High	Zero	Low	Total	Total/2	Ft. Water $\times \frac{13.65}{12} - 1$	Correction Factor Ft. Loss/In.	Tab Distance x14.45	Corrected Reading Hd Loss in Ft.
40.2	40.6	69.44	14.1	.52	2.38	.08	1.72	3.50	1.75	1.83	.0291	-.43	1.40
35.0	34.7	69.48	14.7		1.96		1.29	2.65	1.33	1.39	.0211	-.31	1.08
30.0	29.5	69.53	15.5		1.53		.97	1.90	.95	.99	.0158	-.23	.76
25.0	24.4	69.56	15.8		1.26		.71	1.37	.69	.72	.0117	-.17	.55
20.0	19.3	69.63	16.6		.96		.49	.85	.43	.45	.0082	-.12	.33
15.0	14.2	69.67	17.0		.79		.35	.54	.27	.28	.0051	-.07	.21
10.0	9.2	69.74	17.6		.63		.22	.25	.13	.14	.0024	-.04	.10
5.0	4.8	69.88	18.1		.52		.15	.07	.04	.04	.0009	-.01	.03

Table 2-13

Head Loss In Feet Of Water For 1-1/4" Flanged 90° Elbow

(Reamed Pipe)

42.7	43.7	68.45	13.8	.39	2.37	.22	2.00	3.76	1.88	1.97	.0326	-.48	1.49
40.0	40.4	68.42	14.2		2.12		1.73	3.24	1.62	1.70	.0289	-.42	1.28
35.0	34.7	68.40	14.8		1.71		1.38	2.48	1.24	1.30	.0225	-.33	.97
30.0	29.5	68.40	15.6		1.35		1.08	1.82	.91	.95	.0169	-.25	.70
25.0	24.4	68.36	16.1		1.07		.83	1.29	.65	.68	.9126	-.18	.50
20.0	19.3	68.36	16.6		.82		.62	.83	.42	.44	.00816	-.12	.32
15.0	14.2	68.40	17.1		.63		.44	.46	.23	.24	.00475	-.07	.17
10.0	9.2	68.43	17.7		.49		.32	.20	.10	.10	.00217	-.03	.07
5.0	4.8	68.50	18.2		.42		.25	.06	.03	.03	.00058	-.01	.02

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Table 2-14

Head Loss In Feet Of Water For 1-1/4" Rigid 90° Sweep Elbow

(Reamed Pipe)

Meter Flow gpm	Corrected gpm	T_3 °C	P_1 psi	Δø Zero	High	Zero	Low	Total	Total/2	Ft. Water x $\frac{13.56}{12}$ -1	Correction Factor Ft. Loss/In.	Tab Distance x14.45	Corrected Reading Hd Loss in Ft.
45.0	46.9	14.50	24.9	.38	1.32	.18	1.22	2.54	1.27	1.33	.0458	-.67	.66
40.0	40.4	14.42	25.6		1.18		1.10	1.72	.86	.90	.0225	-.43	.47
35.0	34.7	14.34	26.4		.92		.90	1.26	.63	.66	.0225	-.33	.33
30.0	29.5	14.31	27.1		.81		.70	.95	.48	.50	.0169	-.25	.25
25.0	24.4	14.17	27.5		.67		.57	.68	.34	.36	.0126	-.18	.18
20.0	19.3	14.10	28.0		.57		.45	.46	.23	.24	.00816	-.12	.12
15.0	14.2	14.02	28.5		.49		.34	.27	.14	.15	.00475	-.07	.08
10.0	9.2	13.93	29.1		.43		.26	.13	.07	.07	.00217	-.03	.04
5.0	4.8	13.85	30.0		.38		.21	.03	.02	.02	.00058	-.01	.01
5.0	4.8												

Table 2-15

Head Loss In Feet Of Water For 1-1/4" Rigid 45° Sweep Elbow

(Reamed Pipe)

												x 14.67	
45.0	46.9	15.46	24.8	.32	2.30	.18	2.10	3.90	1.95	2.04	.0458	-.67	1.37
40.0	40.4	15.42	25.5		1.73		1.57	2.80	1.40	1.47	.0289	-.42	1.05
35.0	34.7	15.34	26.4		1.43		1.22	2.15	1.08	1.13	.0225	-.33	.80
30.0	29.5	15.30	27.1		1.13		.96	1.59	.80	.84	.0169	-.25	.59
25.0	24.4	15.21	27.6		.93		.73	1.16	.58	.61	.0126	-.18	.43
20.0	19.3	15.18	28.1		.74		.55	.79	.40	.42	.00816	-.12	.30
15.0	14.2	15.15	28.5		.58		.38	.46	.23	.24	.00475	-.07	.17
10.0	9.2	15.08	29.0		.46		.26	.22	.11	.12	.00217	-.03	.09
5.0	4.8	15.00	30.1		.40		.18	.08	.04	.04	.00058	-.01	.03

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Table 2-16

Head Loss In Feet Of Water For 1-1/4" Rigid 45° Elbow

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	Zero	$\Delta\phi$ High	Zero	Low	Total	Total/2	Ft. Water $x \frac{13.56}{12} - 1$	Correction Factor Ft. Loss/In.	Tab Distance x14.45	Corrected Reading Hd Loss in Ft.
40.2	40.6	70.27	13.9	.52	2.74	.08	2.30	4.46	2.23	2.33	.0291	-.43	1.90
35.0	34.7	70.37	14.7		2.22		1.76	3.40	1.70	1.78	.0211	-.31	1.47
30.0	39.5	70.41	15.3		1.78		1.32	2.52	1.26	1.32	.9158	-.23	1.09
25.0	24.4	70.42	15.8		1.38		.92	1.72	.86	.90	.0117	-.17	.73
20.0	19.3	70.51	16.6		1.06		.61	1.09	.55	.58	.0082	-.12	.46
15.0	14.2	70.66	17.1		.82		.39	.63	.32	.34	.0051	-.07	.27
10.0	9.2	70.66	17.5		.64		.22	.28	.14	.15	.0024	-.04	.11
5.0	4.8	70.77	18.1		.53		.10	.05	.03	.03	.0009	-.01	.02

Table 2-17

Head Loss In Feet Of Water For 1-1/4 Rigid 45° Elbow

(Reamed Pipe)

42.5	43.4	69.17	13.6	.37	2.34	.20	2.18	3.95	1.98	2.08	.0324	-.48	1.60
40.0	40.4	69.15	14.2		2.08		1.93	3.44	1.72	1.80	.0289	-.42	1.38
35.0	34.7	69.13	14.9		1.76		1.58	2.77	1.39	1.45	.0225	-.33	1.12
30.0	29.5	69.08	15.6		1.38		1.22	2.03	1.02	1.07	.0169	-.25	.82
25.0	24.4	69.07	16.2		1.05		.91	1.39	.70	.73	.0126	-.18	.55
20.0	19.3	69.07	16.6		.81		.67	.91	.46	.48	.00816	-.12	.36
15.0	14.2	69.10	17.1		.62		.48	.53	.27	.28	.00475	-.07	.21
10.0	9.2	69.15	17.6		.47		.32	.22	.11	.12	.00217	-.03	.09
5.0	4.8	69.20	18.2		.38		.23	.04	.02	.02	.00058	-.01	.01

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Table 2-18

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Ball Valve

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	Δφ Zero	High	Zero	Low	Total	Total/2	Ft. Water x $\frac{13.56}{12}$ -1	Correction Factor Ft. Loss/In.	Tab Distance x 16.50	Corrected Reading Hd Loss in Ft.
40.5	41.0	69.67	13.8	.50	1.89	.05	1.26	2.60	1.30	1.36	.0297	-.49	.87
35.0	34.7	69.75	14.7		1.54		.98	1.97	.99	1.04	.0211	-.35	.69
30.0	29.5	69.81	15.4		1.25		.73	1.43	..72	.75	.0158	-.26	.49
25.0	24.4	69.87	15.9		1.02		.65	1.12	.56	.59	.0117	-.19	.40
20.0	19.3	69.93	16.6		.83		.38	.66	.33	.35	.0082	-.14	.21
15.0	14.2	69.96	17.1		.68		.28	.41	.21	.22	.0051	-.08	.14
10.0	9.2	70.03	17.4		.57		.18	.20	.10	.10	.0024	-.04	.06
5.0	4.8	70.21	17.9		.50		.13	.08	.04	.04	.0009	-.01	.03

Table 2-19

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Ball Valve

(Reamed Pipe)

43.2	44.3	67.86	13.6	.37	1.27	.19	.83	1.59	.80	.84	.0333	-.55	.29
40.0	40.4	67.82	14.2		1.12		.78	1.34	.67	.70	.0289	-.48	.22
35.0	34.7	67.80	14.9		.94		.65	1.03	.52	.54	.0225	-.37	.17
30.0	29.5	67.80	15.5		.77		.52	.73	.37	.39	.0169	-.28	.11
25.0	24.4	67.76	16.1		.65		.42	.51	.26	.27	.0126	-.21	.06
20.0	19.3	67.75	16.6		.54		.36	.34	.17	.18	.00816	-.13	.05
15.0	14.2	67.77	17.1		.47		.28	.19	.10	.10	.00475	-.08	.02
10.0	9.2	67.75	17.6		.42		.23	.09	.05	.05	.00217	-.04	.01
5.0	4.8	67.90	18.3		.37		.20	.01	.01	..01	.00058	-.01	-

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Table 2-20

Head Loss In Feet Of Water For 1-1/4" Rigid PVC "Y" Valve

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	Zero	Δφ High	Zero	High	Total	Total/2	Ft. Water x $\frac{13.56}{12}$ -1	Correction Factor Ft. Loss/In.	Tab Distance x 16.50	Corrected Reading Hd Loss in Ft.
40.5	41.0	69.31	13.9	.45	3.24	.06	2.92	5.65	2.83	2.97	.0297	-.49	2.48
35.0	34.7	69.39	14.7		2.57		2.16	4.22	2.11	2.21	.0211	-.35	1.86
30.0	29.5	69.45	15.4		1.98		1.59	3.06	1.53	1.60	.0158	-.26	1.34
25.0	24.4	69.51	15.9		1.50		1.12	2.11	1.06	1.11	.0117	-.19	.92
20.0	19.3	69.56	16.6		1.13		.77	1.39	.70	.73	.0082	-.14	.59
15.0	14.2	69.62	17.1		.82		.43	.74	.37	.39	.0051	-.08	.31
10.0	9.2	69.72	17.6		.60		.25	.34	.17	.18	.0024	-.04	.14
5.0	4.8	69.85	18.1		.48		.11	.08	.04	.04	.0009	-.01	.03

Table 2-21

Head Loss In Feet Of Water For 1-1/4" Rigid PVC "Y" Valve

(Reamed Pipe)

43.2	44.3	68.20	13.6	.37	2.19	.19	2.02	3.65	1.83	1.92	.0333	-.55	1.37
40.0	40.4	68.16	14.1		1.97		1.74	3.15	1.58	1.66	.0289	-.48	1.18
35.0	34.7	68.12	14.8		1.59		1.47	2.50	1.25	1.31	.0225	-.37	.94
30.0	29.5	68.10	15.6		1.25		1.08	1.77	.89	.93	.0169	-.28	.65
25.0	24.4	68.10	16.2		.98		.82	1.24	.62	.65	.0126	-.21	.44
20.0	19.3	68.07	16.6		.75		.62	.81	.41	.43	.00816	-.13	.30
15.0	14.2	68.11	17.1		.58		.43	.45	.28	.29	.00475	-.08	.21
10.0	9.2	68.15	17.8		.43		.28	.15	.08	.08	.00217	-.04	.04
5.0	4.8	68.25	18.2		.37		.23	.04	.02	.02	.00058	-.01	.01

STAT

Table 2-22

Head Loss In Feet Of Water For 1-1/2" Rigid PVC Plug Valve

(Unreemed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	$\Delta\phi$ Zero	High	Zero	Low	Total	Total/2	Ft. Water x $\frac{13.65}{12} - 1$	Correction Factor Ft. Loss/In.	Tab Distance x 14.50	Corrected Reading Hd Loss in Ft.
40.0	40.4	69.90	13.8	.52	1.77	.08	1.13	2.30	1.15	1.20	.0289	-.42	.78
35.0	34.7	70.03	14.6		1.47		.98	1.85	.93	.97	.0211	-.31	.66
30.0	29.5	70.10	15.4		1.19		.67	1.28	.64	.67	.0158	-.23	.44
25.0	24.4	70.16	15.9		.98		.49	.89	.45	.47	.0117	-.17	.30
20.0	19.3	70.21	16.6		.81		.33	.56	.28	.29	.0082	-.12	.17
15.0	14.2	70.26	16.9		.68		.23	.33	.17	.18	.0051	-.07	.11
10.0	9.2	70.32	17.4		.57		.15	.14	.07	.07	.0024	-.03	.04
5.0	4.8	70.47	18.0		.51		.10	.03	.02	.02	.0009	-.01	.01

Table 2-23

Head Loss In Feet Of Water For 1-1/2" Rigid PVC Plug Valve

(Reamed Pipe)

42.7	43.7	68.95	13.8	.38	1.48	.19	1.17	2.08	1.04	1.09	.0326	-.47	.62
40.0	40.4	68.90	14.3		1.33		1.06	1.82	.91	.95	.0289	-.42	.53
35.0	34.7	68.90	14.9		1.13		.87	1.43	.72	.75	.0225	-.33	.42
30.0	29.5	68.90	15.7		.91		.68	1.02	.51	.53	.0169	-.25	.28
25.0	24.4	68.87	16.1		.74		.53	.70	.35	.37	.9126	-.18	.19
20.0	19.3	68.87	16.6		.59		.43	.45	.23	.24	.00816	-.12	.12
15.0	14.2	68.90	17.1		.49		.32	.24	.12	.13	.00475	-.07	.06
10.0	9.2	68.95	17.7		.40		.27	.10	.05	.05	.00217	-.03	.02
5.0	4.8	69.00	18.1		.39		.23	.05	.03	.03	.00058	-.01	.01

STAT

Table 2-24

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Coupling

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	Δφ Zero	High	Zero	Low	Total	Total/2	Ft. Water x $\frac{13.56}{12}$ -1	Correction Factor Ft. Loss/In.	Tab Distance x 14.50	Corrected Reading Hd Loss in Ft.
40.2	40.6	69.68	13.8	.52	1.48	.08	.82	1.70	.85	.89	.0291	-.42	.47
35.0	34.7	69.73	14.6		1.25		.63	1.28	.64	.67	.0211	-.31	.36
30.0	29.5	69.81	15.3		1.08		.51	.99	.50	.52	.0158	-.23	.29
25.0	24.4	69.85	16.0		.92		.38	.70	.35	.37	.0117	-.17	.20
20.0	19.3	69.85	16.5		.77		.29	.46	.23	.24	.0082	-.12	.12
15.0	14.2	69.91	69.9		.67		.22	.29	.15	.16	.0051	-.07	.09
10.0	9.2	69.95	17.4		.58		.17	.15	.08	.08	.0024	-.03	.05
5.0	4.8	70.14	18.1		.52		.12	.04	.02	.02	.0009	-.01	.01

Table 2-25

Head Loss In Feet Of Water For 1-1/4" Rigid PVC Coupling

(Reamed Pipe)

42.7	43.7	68.80	13.8	.39	1.22	.22	.79	1.40	.70	.73	.0326	-.47	.26
40.0	40.4	68.80	14.3		1.12		.73	1.24	.62	.65	.0289	-.42	.23
35.0	34.7	68.77	14.9		.96		.65	1.00	.50	.52	.0225	-.33	.19
30.0	29.5	68.75	15.6		.82		.53	.74	.37	.39	.0169	-.25	.14
25.0	24.4	68.73	16.1		.68		.45	.52	.26	.27	.0126	-.18	.09
20.0	19.3	68.75	16.6		.57		.39	.35	.18	.19	.00816	-.12	.07
15.0	14.2	68.75	17.1		.50		.32	.21	.11	.12	.00475	-.07	.05
10.0	9.2	68.80	17.7		.44		.27	.10	.05	.05	.00217	-.03	.02
5.0	4.8	68.88	18.2		.41		.23	.03	.02	.02	.00058	-.01	.01

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Table 2-26

Head Loss In Feet Of Water For 1-1/4" PVC Pipe And Fittings - Branch I

(Unreamed Pipe)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	High Max.	High Min.	Low Max.	Low Min.	Δ ø av.	Corrected Δ ø Head Loss in Ft.		
40.8	41.3	70.84	13.8	10.65	10.45	10.65	10.62	21.19	22.19		
35.0	34.7	70.91	14.7	8.01	7.85	8.03	7.80	15.85	16.59		
30.0	29.5	70.99	15.5	6.04	5.84	6.15	5.87	11.95	12.51		
25.0	24.4	71.20	15.9	4.19	3.95	4.20	4.04	8.19	8.57		
20.0	19.3	71.17	16.3	2.72	2.63	2.85	2.67	5.44	5.70		
15.0	14.2	71.21	17.0	1.69	1.54	1.78	1.70	3.36	3.52		
10.0	9.2	71.18	17.3	.83	.75	.96	.88	1.71	1.79		
5.0	4.8	71.37	18.1	.54	.50	.61	.57	1.11	1.16		

Table 2-27

Head Loss In Feet Of Water For 1-1/4" PVC Pipe and Fittings - Branch II

(Unreamed Pipe)

40.0	40.4	70.77	13.9	11.02	10.80	11.05	17.75	21.81	22.84		
35.0	34.7	70.81	14.6	8.47	8.19	8.47	8.25	16.69	17.47		
30.0	29.5	70.86	15.4	6.36	6.20	6.47	6.17	12.74	13.34		
25.0	24.4	70.95	15.9	4.60	4.50	4.65	4.58	9.17	9.60		
20.0	19.3	70.97	16.5	3.30	3.10	3.35	3.15	6.45	6.75		
15.0	14.2	71.02	16.9	2.20	2.10	2.25	2.10	4.33	4.53		
10.0	9.2	71.10	17.6	1.28	1.24	1.35	1.30	2.59	2.71		
5.0	4.8	71.27	18.2	.77	.70	.82	.78	1.54	1.61		

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Table 2-28

Head Loss In Feet Of Water For 1-1/4" PVC Pipe And Fittings - Branch I

(Reamed)

Meter Flow gpm	Corrected gpm	T_2 °F	P_1 psi	High		Low		Δ ø av.	Corrected Δ ø Head Loss in Ft.		
				Max.	Min.	Max.	Min.				
43.2	44.3	69.86	13.5	10.50	10.18	10.50	10.18	20.68	21.65		
40.0	40.4	69.86	14.0	9.08	8.78	9.03	8.80	17.85	18.69		
35.0	34.7	69.80	14.9	7.29	6.97	7.29	7.01	14.28	14.95		
30.0	29.5	69.80	15.6	5.29	5.18	5.37	5.18	10.51	11.00		
25.0	24.4	69.75	15.9	3.92	3.77	3.98	3.72	7.70	8.06		
20.0	19.3	69.75	16.6	2.65	2.43	2.67	2.53	5.14	5.38		
15.0	14.2	69.76	17.0	1.63	1.48	1.66	1.49	3.13	3.28		
10.0	9.2	69.80	17.6	.85	.77	.92	.80	1.67	1.75		
5.0	4.8	69.86	18.1	.35	.30	.35	.28	.64	.67		

Table 2-29

Head Loss In Feet Of Water For 1-1/4" PVC Pipe And Fittings - Branch II

(Reamed)

42.5	43.4	69.40	13.8	10.32		10.27		20.59	21.56		
40.0	40.4	69.35	14.2	8.95		9.00		17.95	18.79		
35.0	34.7	69.35	14.8	7.22		7.30		14.52	15.20		
30.0	29.5	69.35	15.6	5.33		5.40		10.73	11.23		
25.0	24.4	69.35	16.0	3.78		3.90		7.68	8.04		
20.0	19.3	69.35	16.6	2.52		2.64		5.16	5.40		
15.0	14.2	69.35	17.2	1.64	1.50	1.65	1.48	3.14	3.29		
10.0	9.2	69.35	17.6	.85	.79	.92	.87	1.77	1.85		
5.0	4.8	69.55	18.2	.52	.48	.61	.56	1.09	1.14		

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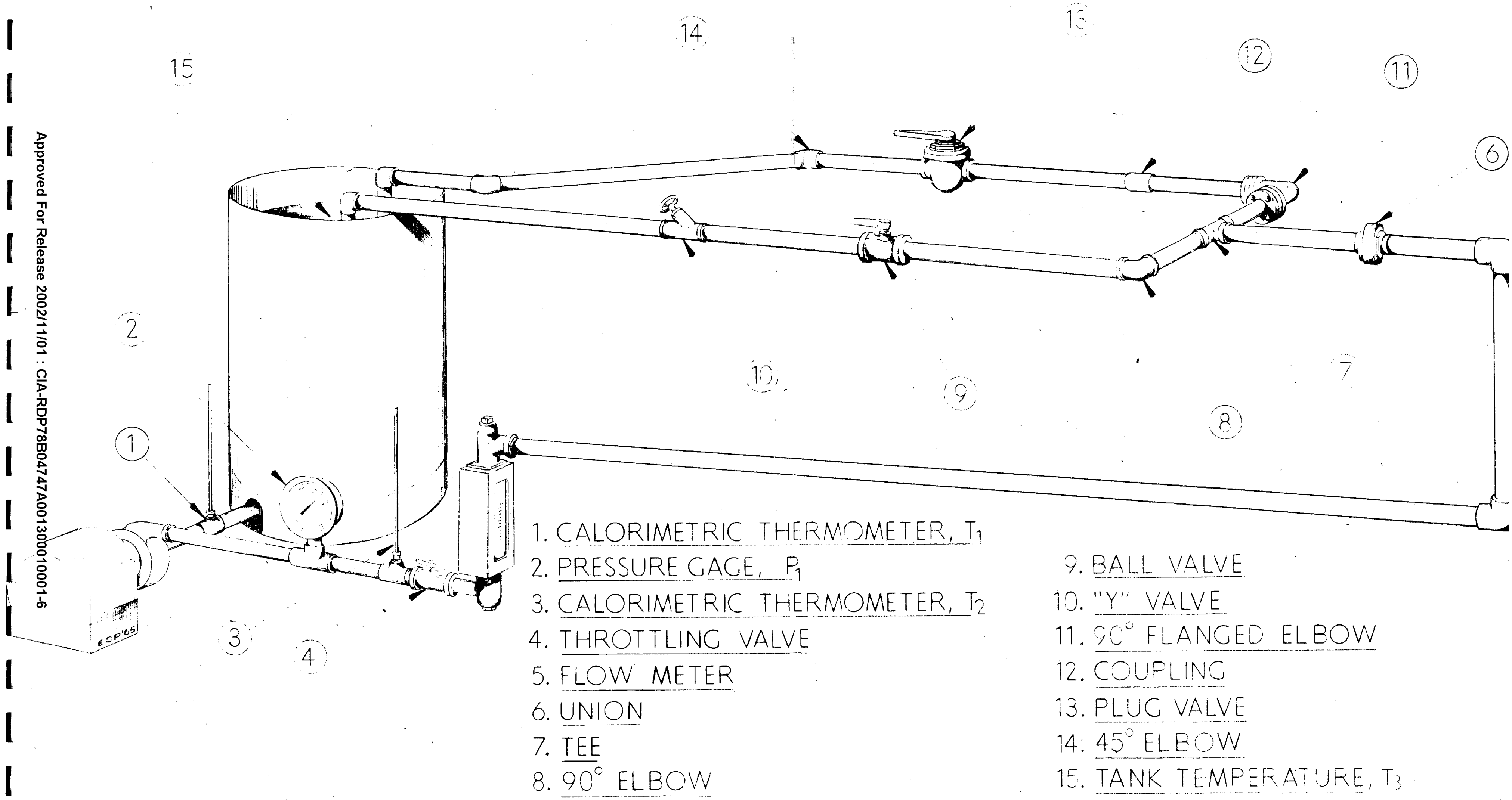

Figure 2-1. Pressure I Apparatus

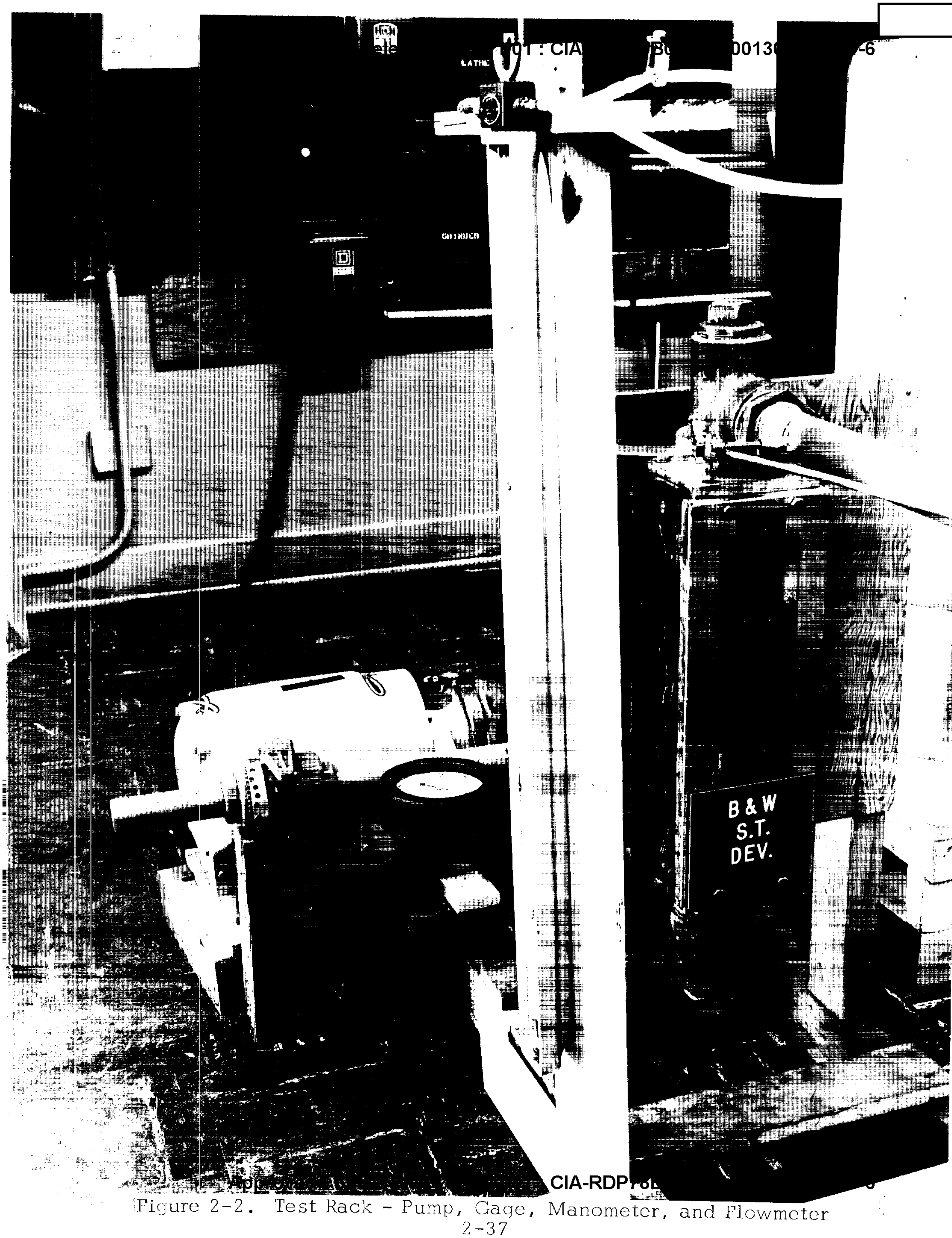

Figure 2-2. Test Rack - Pump, Gage, Manometer, and Flowmeter

Figure 2-3. Test Rack - Pump, Gage, Manometer, and Thermometers

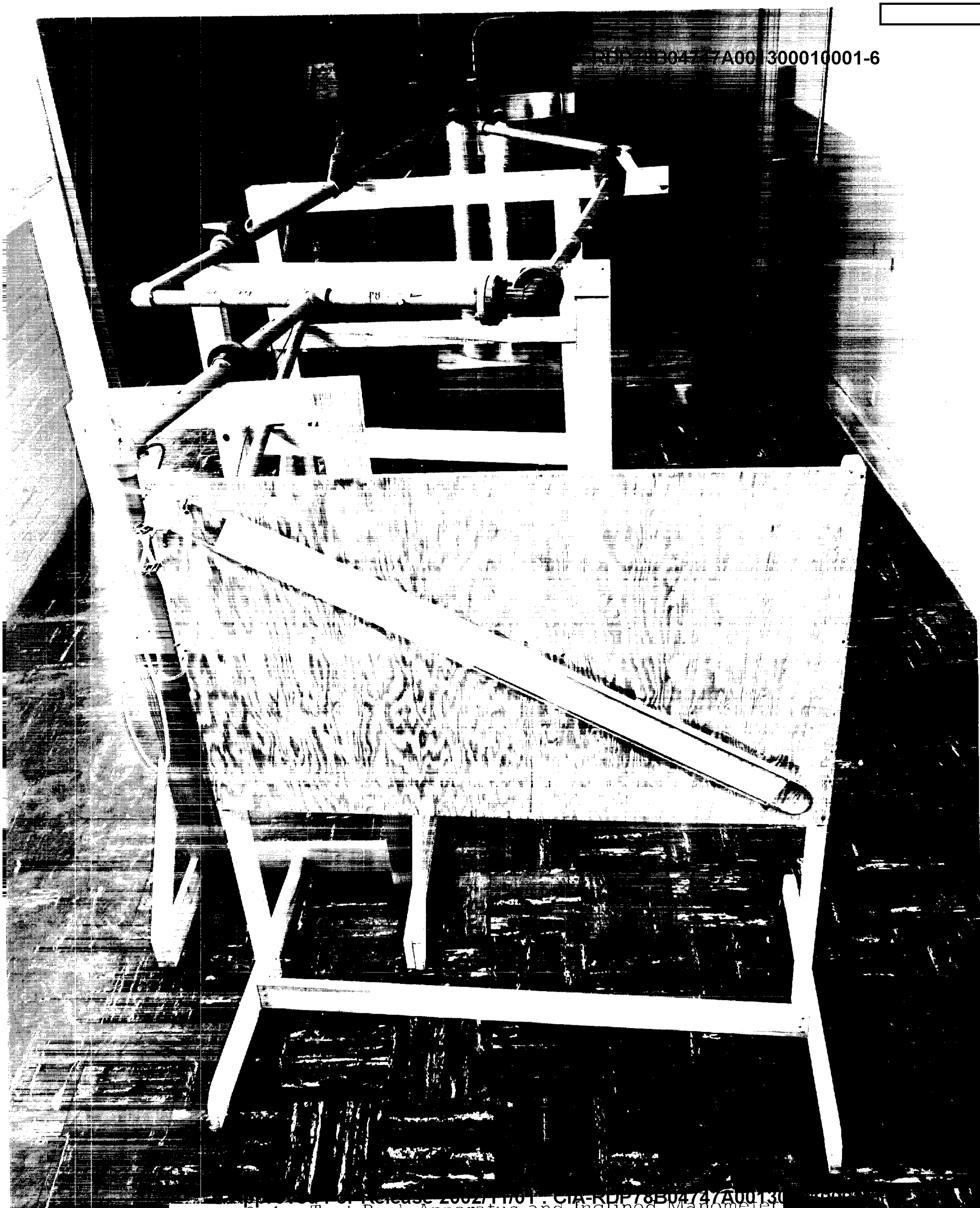

Figure 2-4. Test Rack Apparatus and Inclined Manometer

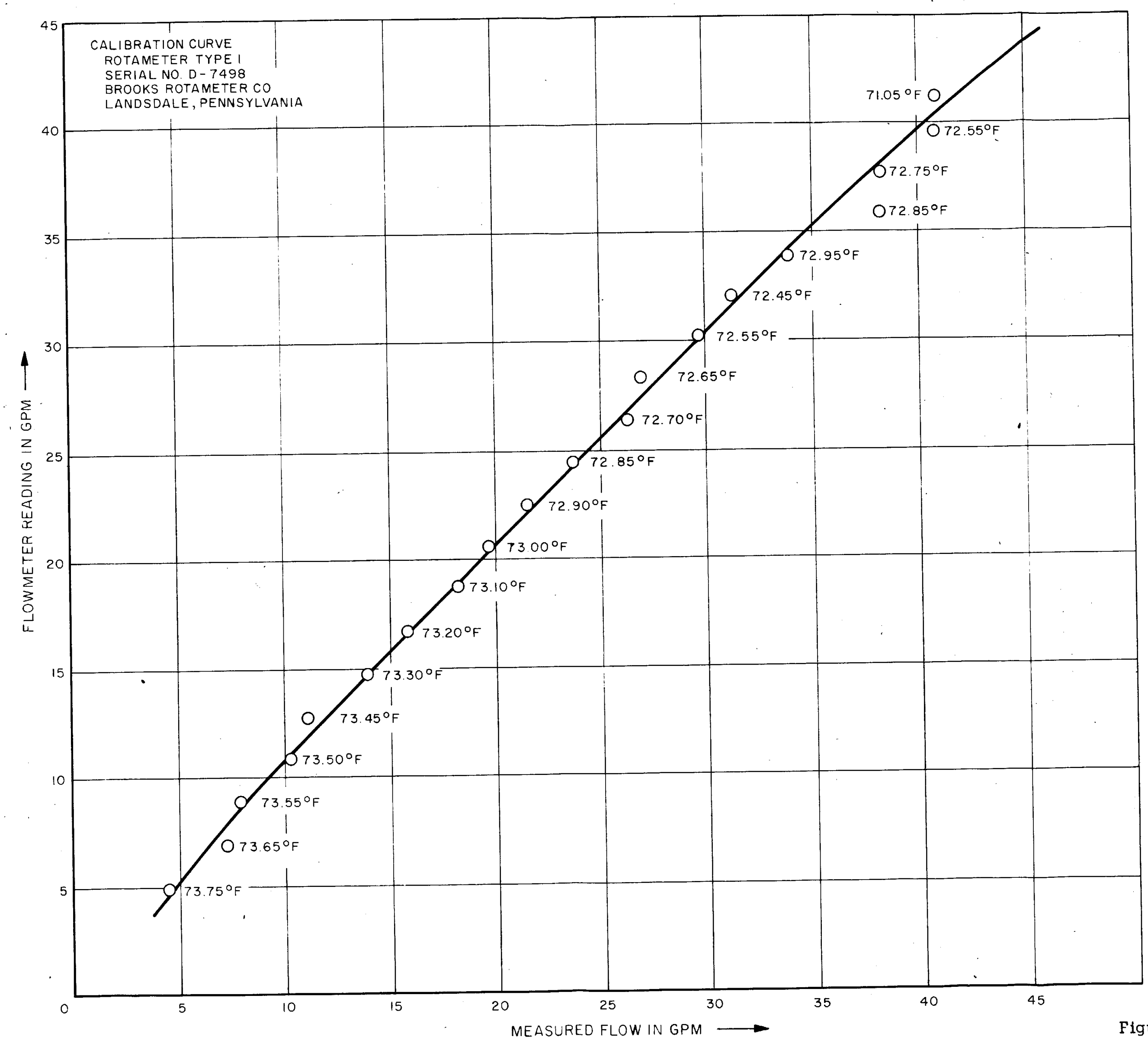

Figure 2-5. Flowmeter Calibra
Chart

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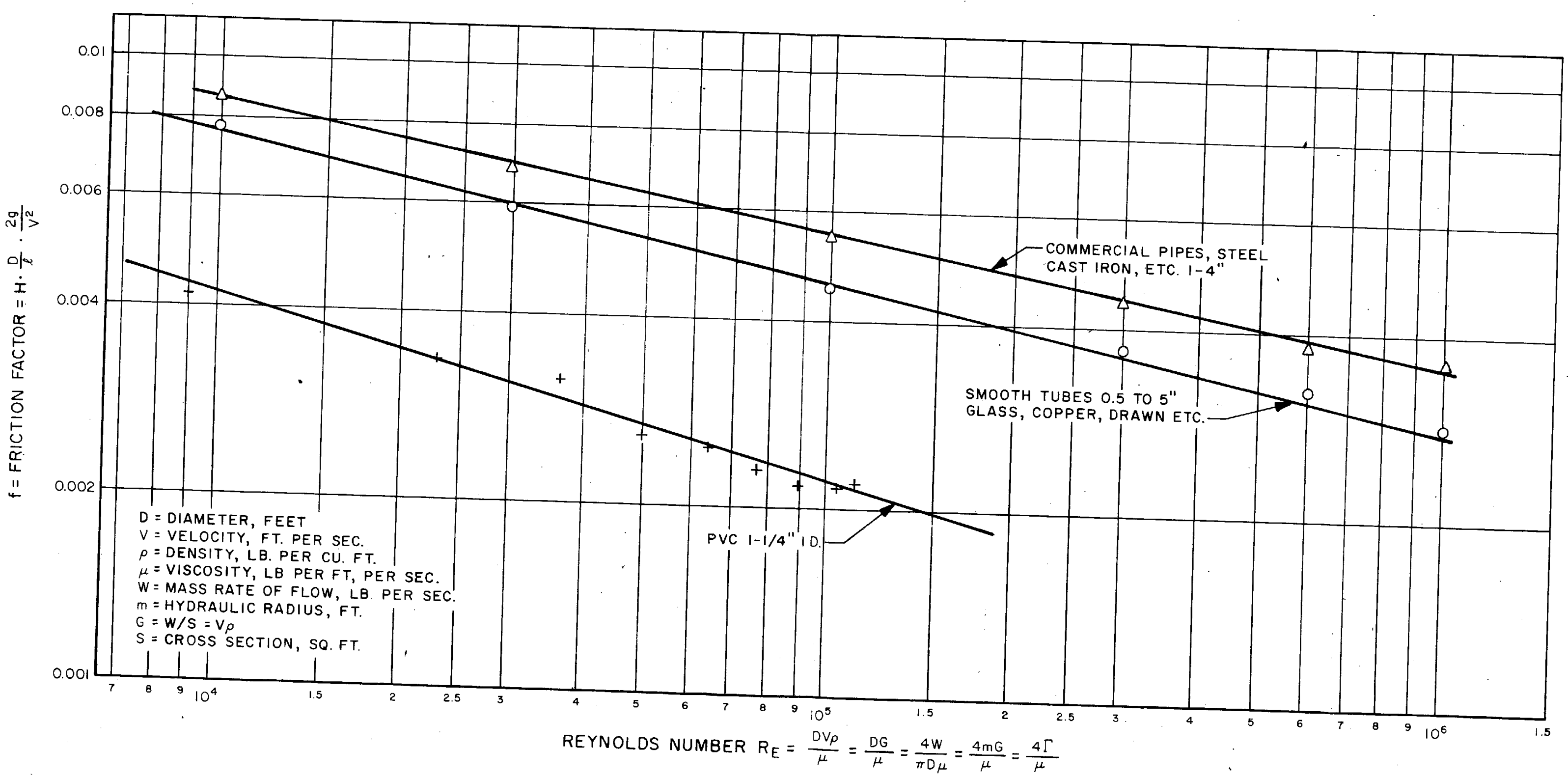

Figure 2-6. Reynolds Numbers Friction Coefficien

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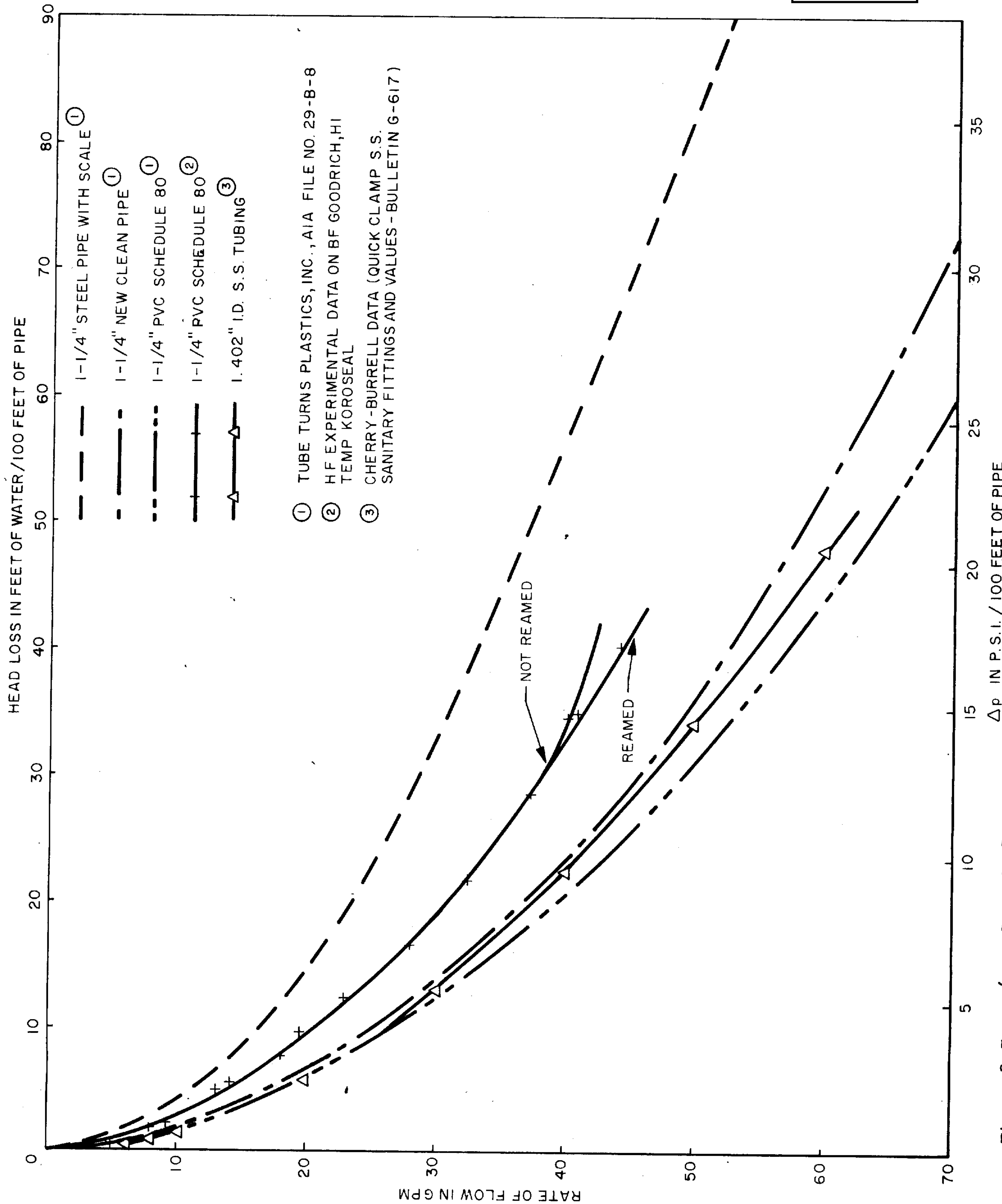

Figure 2-7. Δp For Straight Pipes

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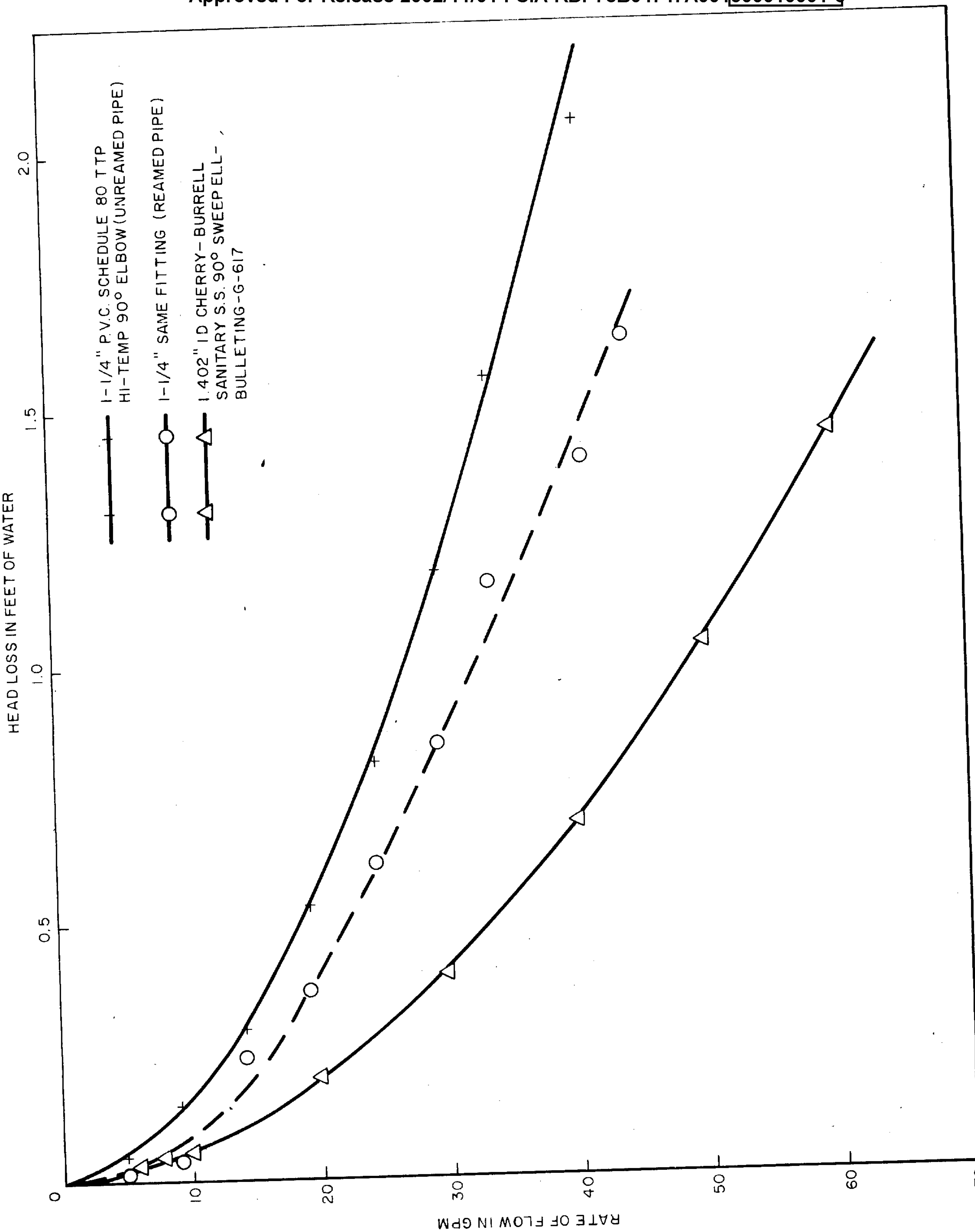

Figure 2-8. $\Delta\phi$'s for 90° Elbows

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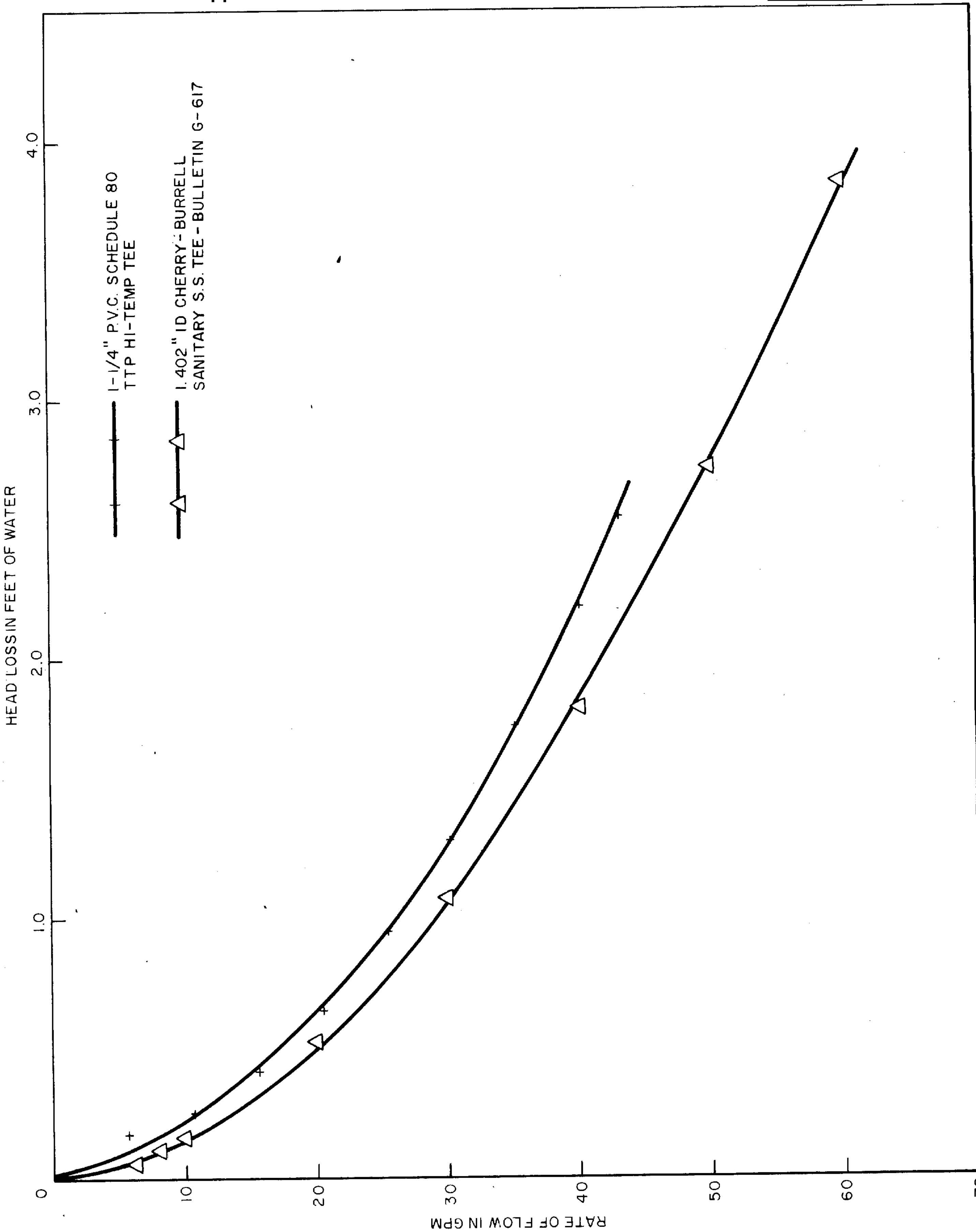

Figure 2-9. Δ p's for Tees

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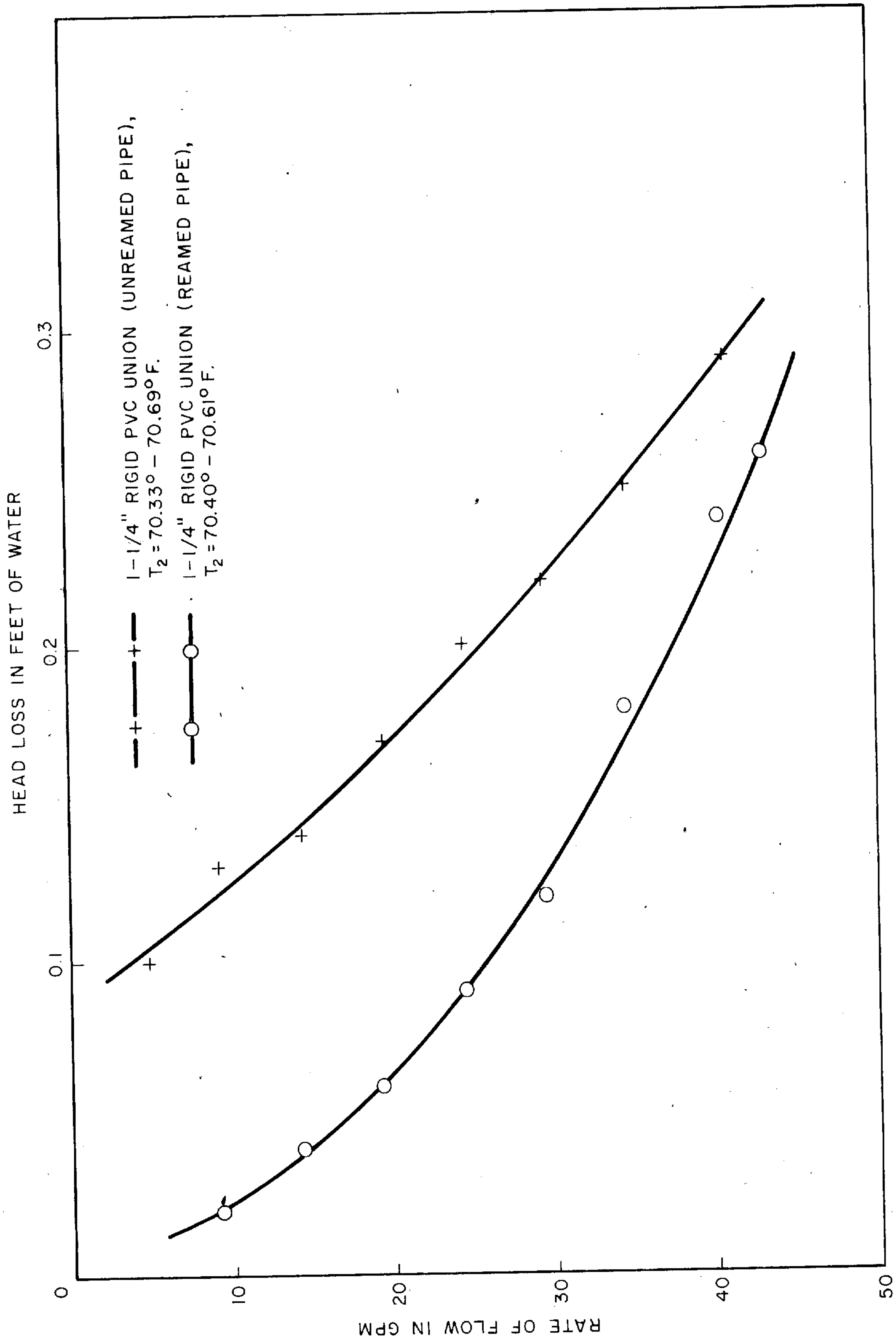


Figure 2-10. Δ p for PVC Union

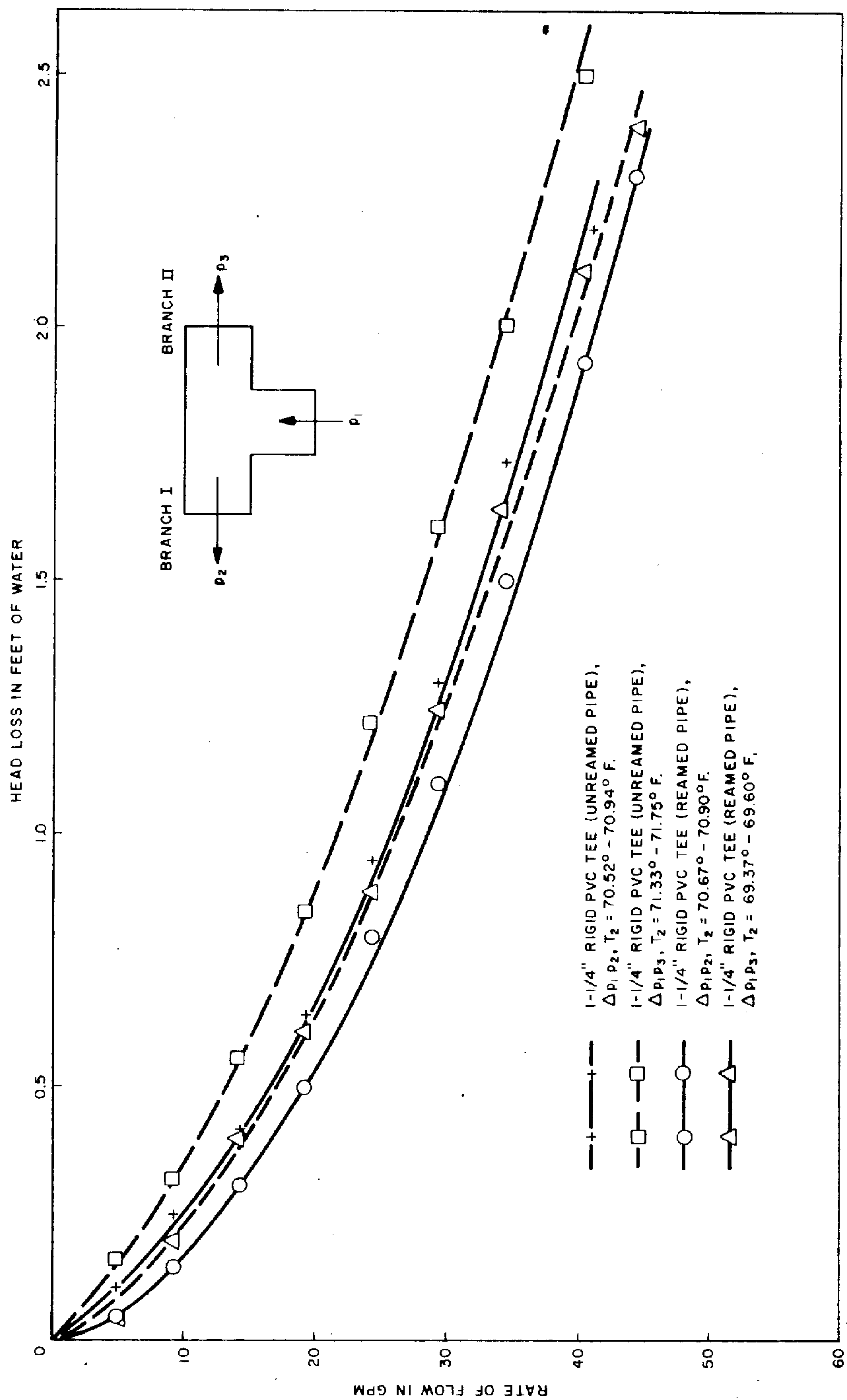


Figure 2-11. Δp for PVC Tee

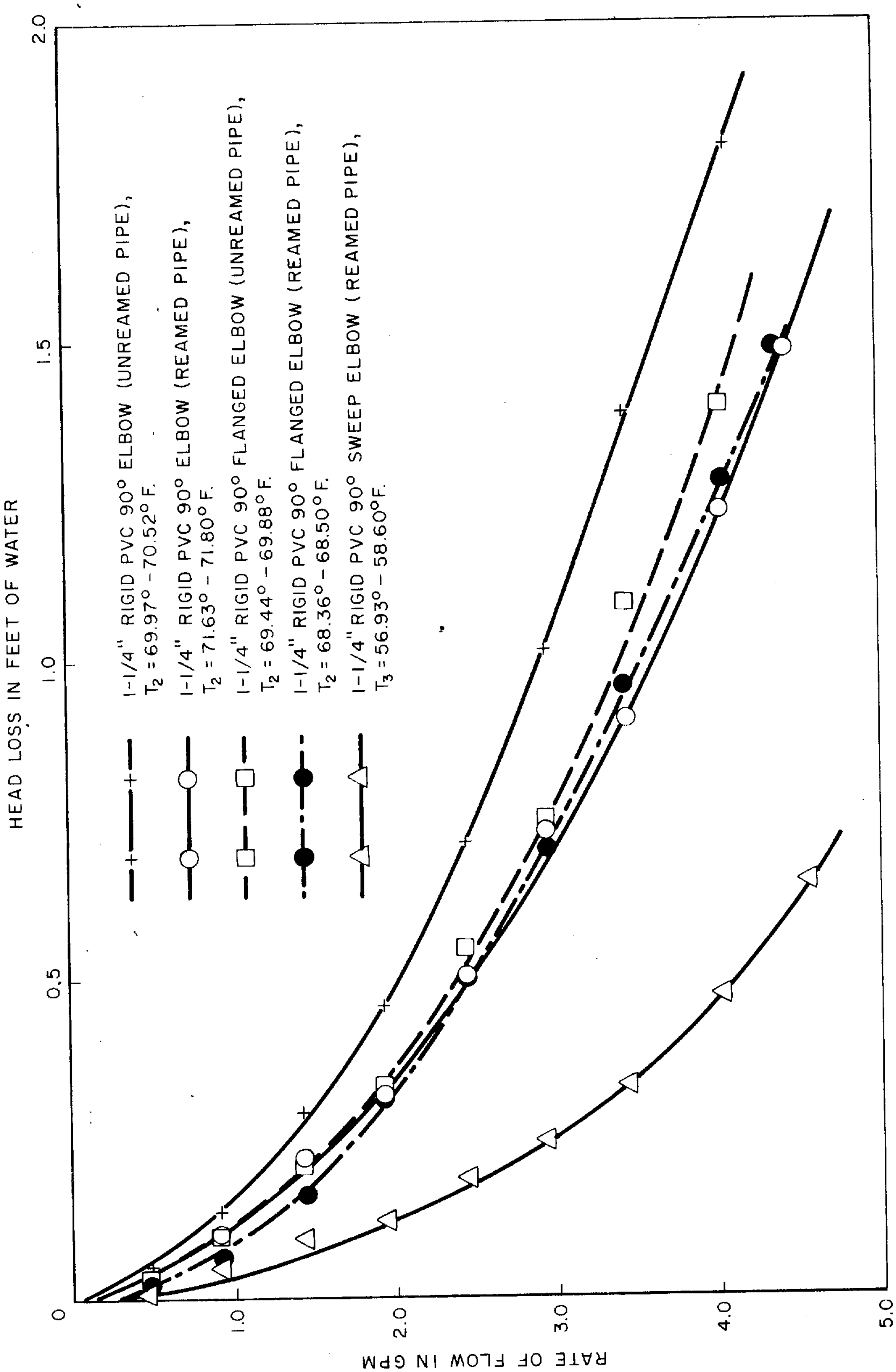
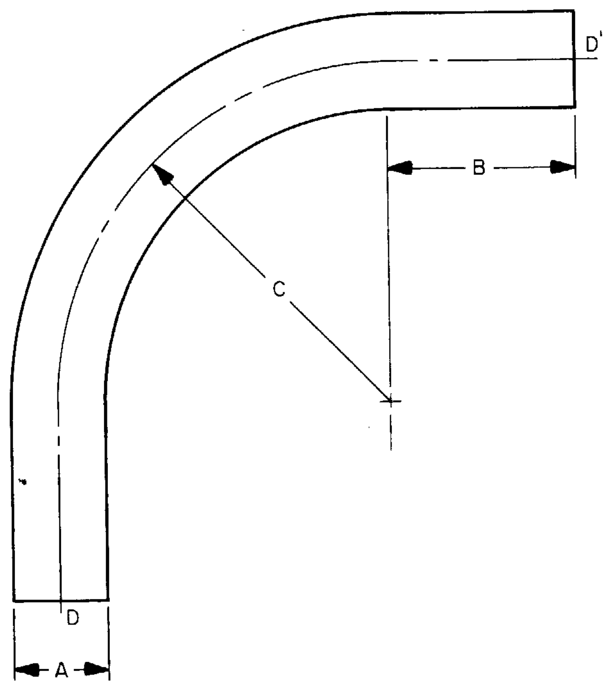
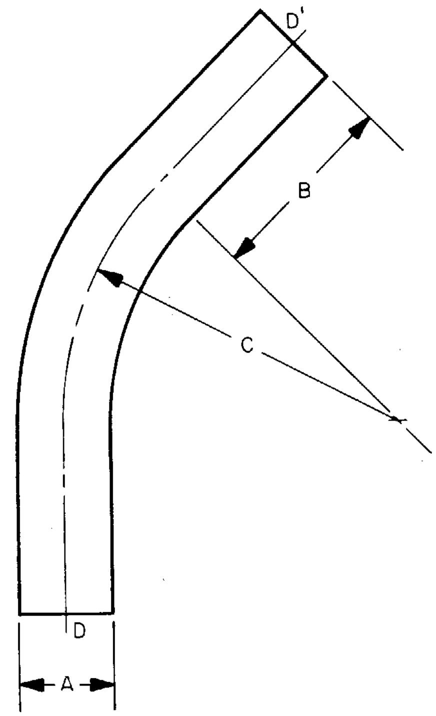


Figure 2-12. Δp's for 90-Degree PVC Elbows



90° PVC SWEEP ELBOW-SCHEDULE 40

SIZE	A	B MIN.	C	D-D'MIN
1-1/4"	1.660	2"	7-1/4"	15.40



45° PVC SWEEP ELBOW-SCHEDULE 40

SIZE	A	B MIN.	C	D-D'MIN.
1-1/4"	1.660	2"	7-1/4"	9.70

Figure 2-13. Dimensions of Sweep Elbows

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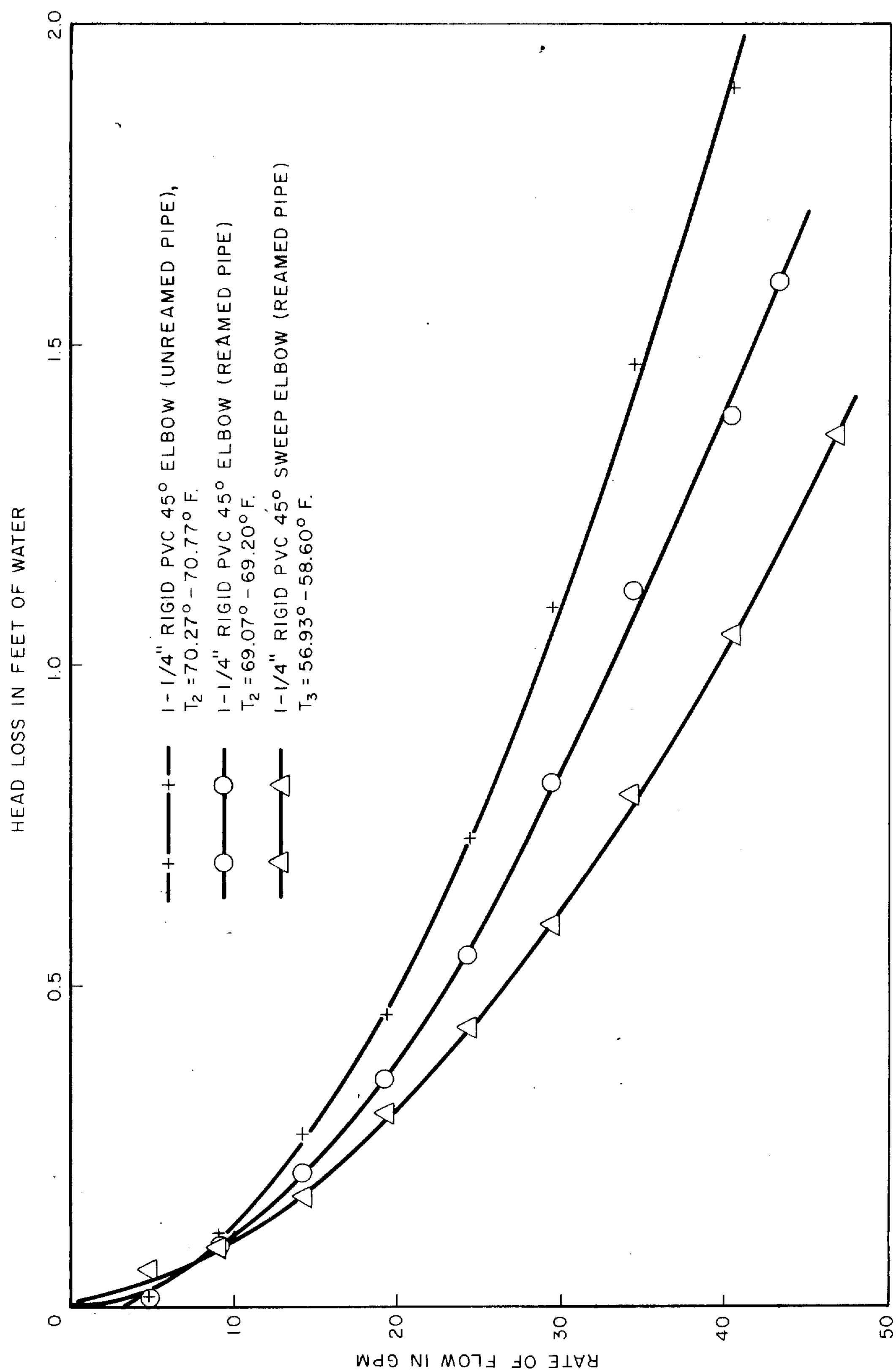

Figure 2-14. Δp's for 45-Degree PVC Elbows

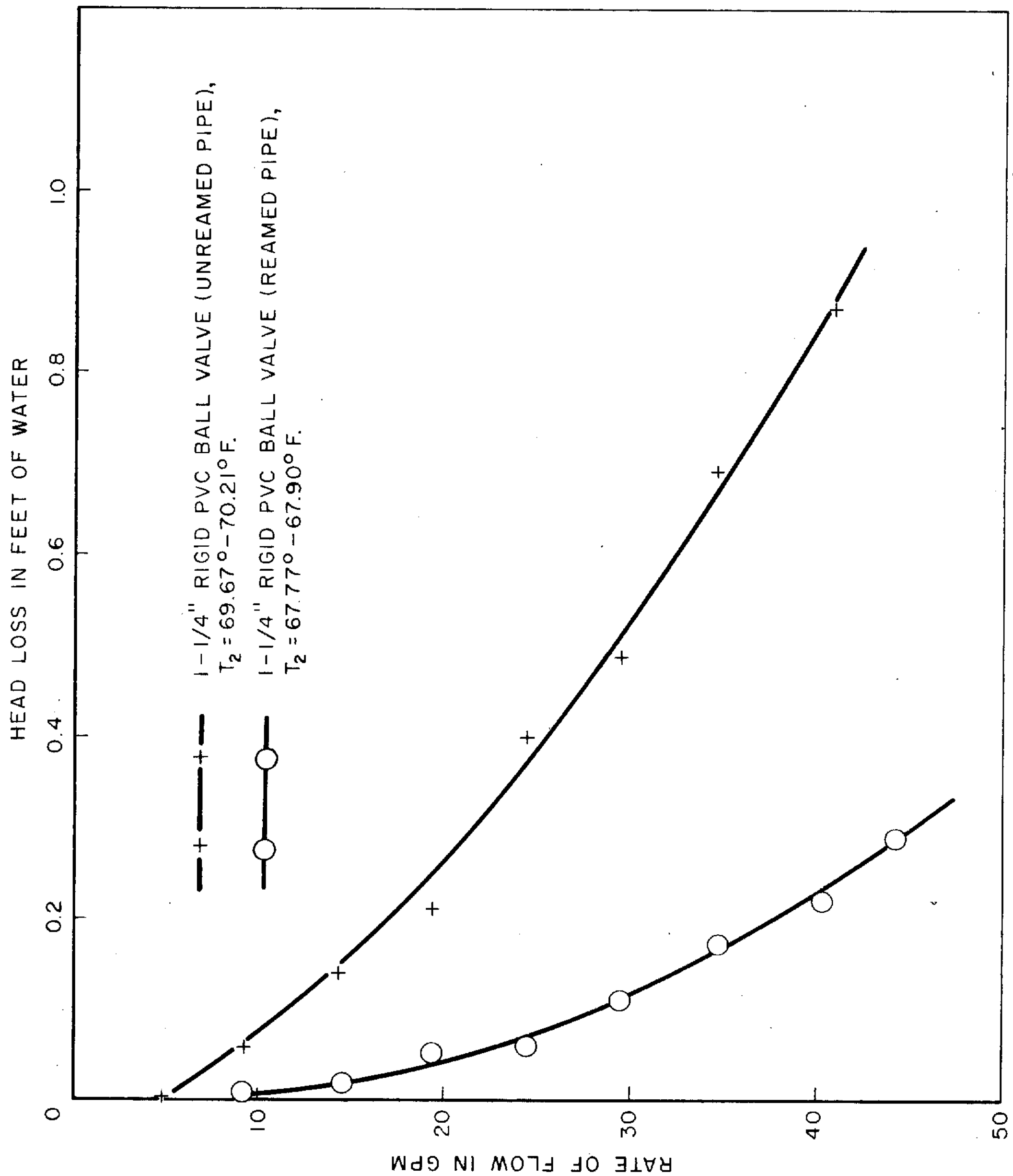

Figure 2-15. $\Delta\phi$ For PVC Ball Valve

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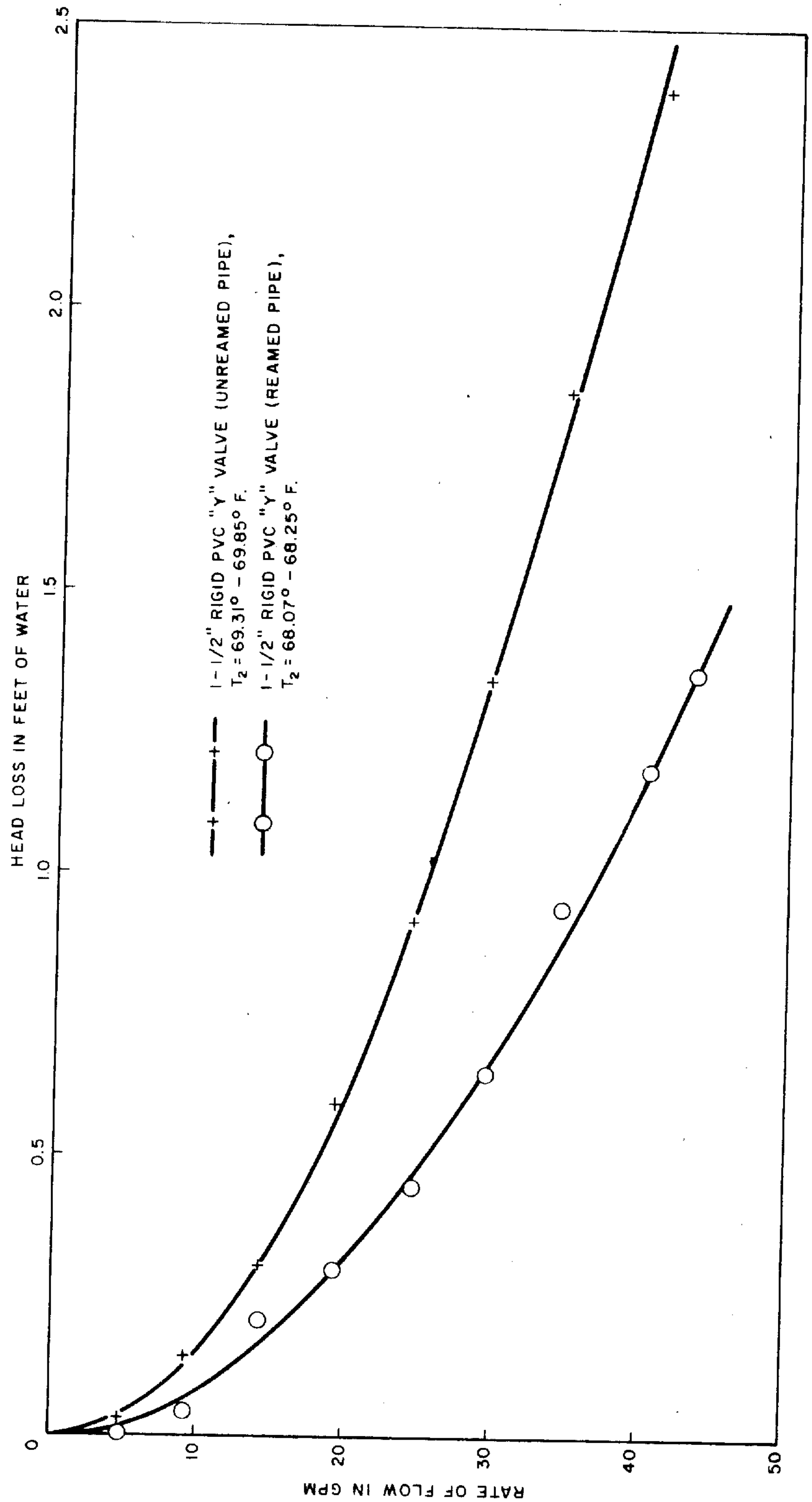


Figure 2-16. Δp For PVC "Y" Valve

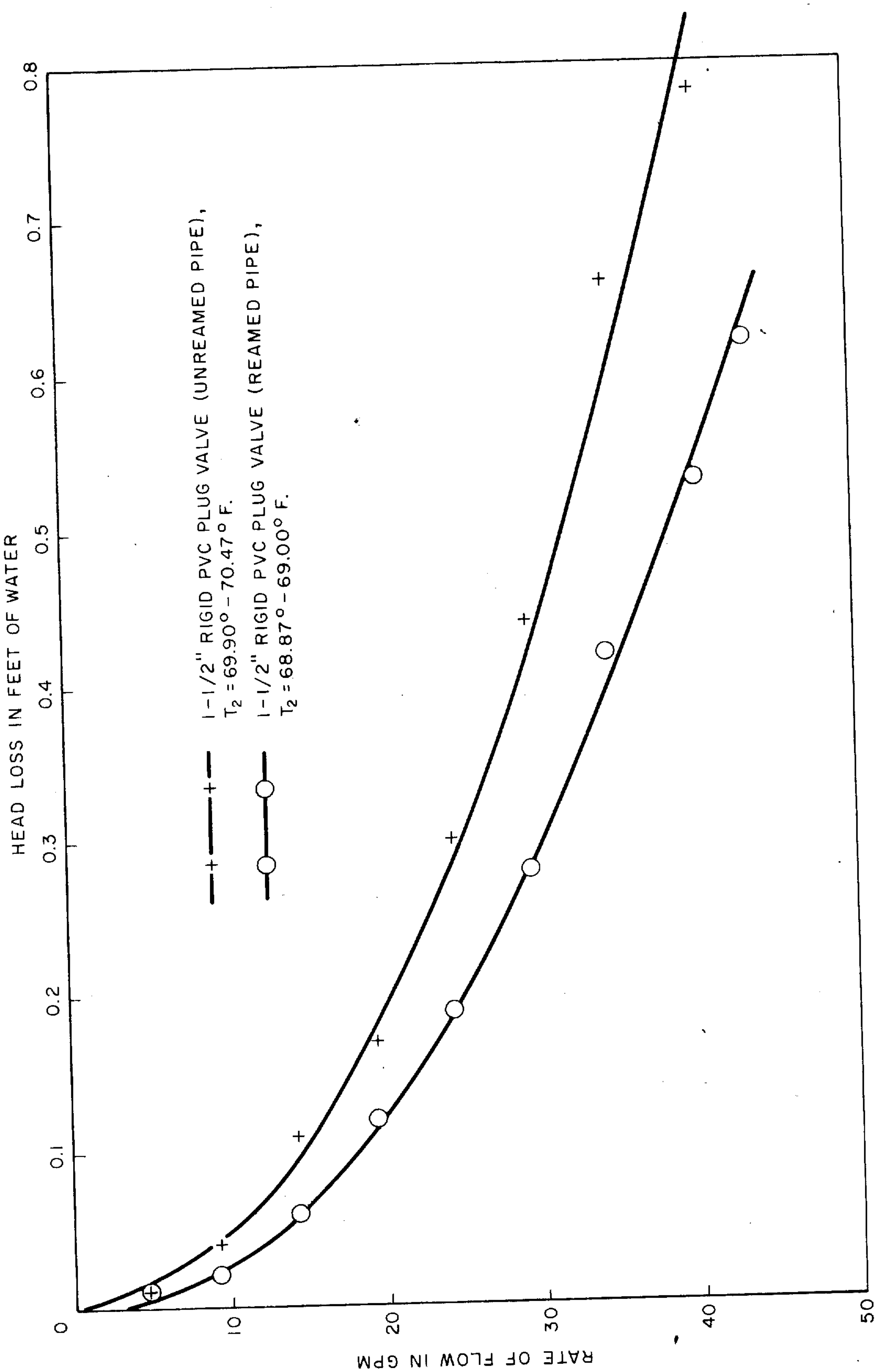


Figure 2-17. $\Delta\phi$ For PVC Plug Valve

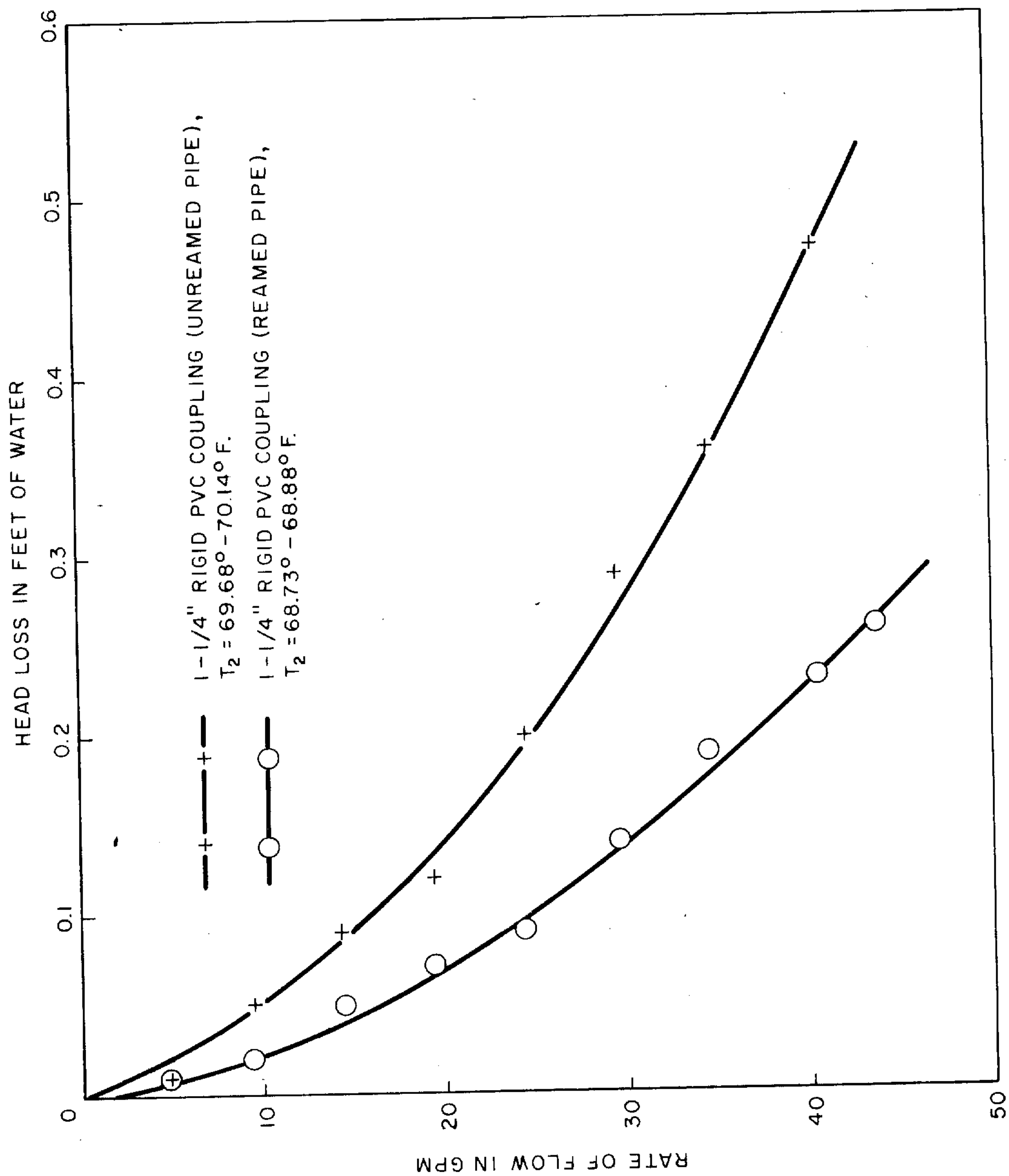


Figure 2-18. $\Delta\phi$ For PVC Coupling

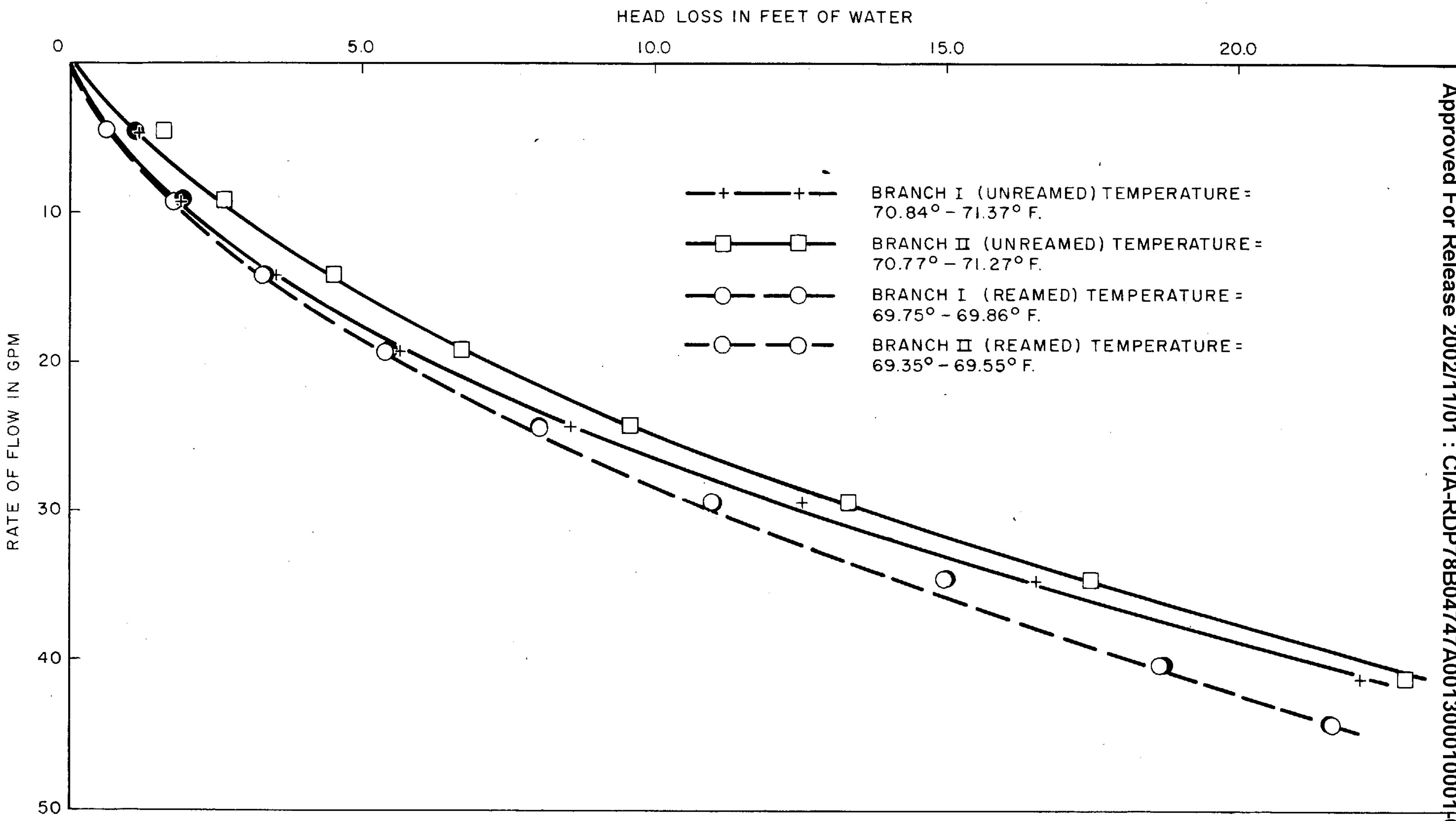
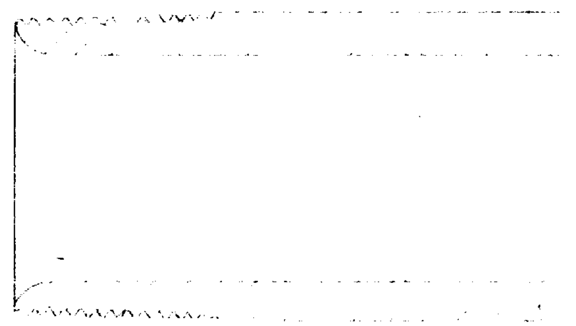
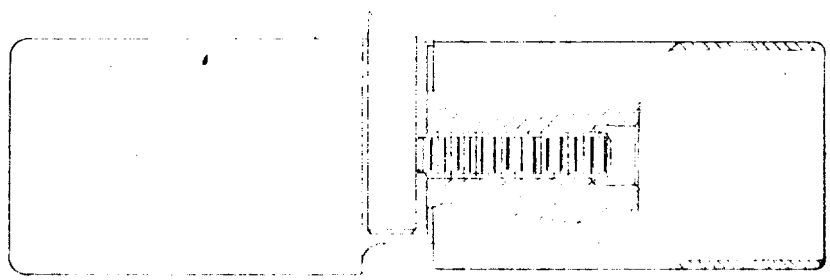


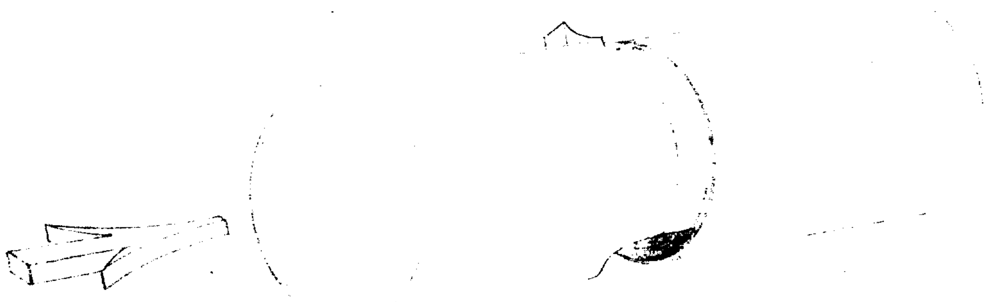
Figure 2-19. $\Delta\phi$'s for PVC Pipe and Fittings



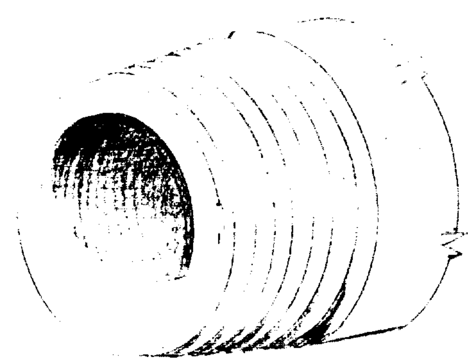
CROSS-SECTION OF
FAIRED PIPE



MOUNTING OF FAIRING TOOL BIT



FAIRING TOOL



UNREAMED PIPE

Figure 2-20. Fairing Tool and Pipe

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SECTION 3
CONCLUSIONS

3.1 PUMP TESTS

In order to apportion time and budget most efficiently among the high priority facets of the research contract, a rigid schedule for each was set up and followed. Thus, a few of the proposed tests itemized in the list of objectives were necessarily postponed until a later date. This was true of some of the pump tests outlined, but a number of significant conclusions can be drawn, nevertheless.

The pump had an excellent rated capacity for a given horsepower input. The drive motor used showed only a fraction of its rated rise (55°C.) after over three hours of continuous service. The flow was not sensitive to angle of discharge from the centrifugal plenum. It could be completely disassembled in less than a minute. It proved to be a simple shop procedure to modify the impeller to exactly match the load. The unit produced almost no entrainment of air, even without a bleed cock on the plenum. The plenum pressure plate, impeller, shaft, and gland fittings are all stainless steel, designed so that they can easily be cleaned or replaced. Optionally, the unit can be supplied with a splash-proof housing of polished stainless steel. Appendix D gives some typical pump calculations.

The design has not yet been tested with curved (an optional feature) blades or with ordinary corrosive photographic chemicals. The latter test will require some carefully planned safety precautions (automatic leak-sensing unit and circuit breaker) to avoid the hazards of continuous unattended operation. Three of the other tests outlined in the list of objectives were not performed, i.e., falling head, constricted inlet, and breakdown. Since there was some risk of permanent damage to either the pump or the motor and since the unit under test was on loan, this latter series should be run on a purchased unit.

3.2 PRESSURE DROP TESTS

As mentioned previously, some of the published proprietary data on pressure drops is suspect. This conclusion is based on the fact that the data do not agree with the standard charts of Reynolds numbers which are supported by more experimental data than even the International Critical Steam Tables.

Of the three types of valve tested, the ball valve proved to have the lowest pressure drop. This is the type used almost exclusively on [redacted] [redacted] designed equipment. Technically speaking, they are not "ball" valves, but plug valves in which the rotating unit is spherical rather than a truncated cone. They are well designed, have no sharp internal protrusions, and, because of their modular assembly, can be used in place of a standard union. They are not wholly satisfactory, however, when used as throttling valves to regulate flow.

The standard plug valve had the next lowest pressure drop. While the readings were approximately 2-1/2 times those of the ball valve, they cannot be accepted as fully definitive. The smallest valve of this type obtainable was 1-1/2 inches. It was designed for slip fittings and was modified by inserting slip-to-thread collars. This left an irregular plenum at each side which would have been largely eliminated in a 1-1/4-inch fitting.

The "Y" valve design was excellent for throttling, but offered the largest pressure drop of the three. This was to be expected because the flow pattern is subjected to two sharp changes in direction as it passes through the valve. All of the tests on the three valves were performed in the full-open position; any intermediate readings would have been meaningless.

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The measured pressure drops for the union and the coupling, with reamed pipe, were almost identical, besides being the lowest for the series. This was not surprising in the case of the coupling because of its short overall length, the only irregularity being a few exposed threads when the joint was made up. Note, however, that the internal section of the union was full diameter and quite smooth so as to be comparable to straight pipe, whose loss in feet of water per inch is only 0.0289 at 40 gpm.

Several pressure drop runs (Appendix E) were made across the rotameter alone. Psig readings were converted to inches of mercury and then reduced to an equivalent length of 1-1/4-inch PVC pipe. Thus, for a flow of 40.8 gpm, the pressure loss was equivalent to 30.2 feet of standard pipe. Lack of time and money prevented testing other types of flowmeters, particularly venturis, whose losses are relatively low. The other pressure-drop readings were not recalculated as equivalent lengths of pipe; this is rarely required in design and can be readily obtained from the data presented.

The data presented in this report, while somewhat limited in scope, have been carefully prepared and are unusually accurate. They should adequately fill a gap in present technical design reference literature.

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4.1 CONTINUED EXPERIMENTATION

The type of experimental measurement described in preceding sections is extremely painstaking and time consuming. Since its worth is invaluable to the design engineer, its further expansion should be funded by a group such as the Bureau of Standards or the National Science Foundation. It is difficult to justify fundamental research on a short-term development or state-of-the-art improvement contract, notwithstanding the immediate benefits to a design program such as could evolve. In this case, these benefits include smaller pump size, increased flow, increased pressure, less pulsation and air entrainment, and finally, no "cut-and-fit" trials for power sizing or line losses.

There was no opportunity, for instance, to run comparative tests on sanitary stainless-steel dairy pipe and fittings. These promise to have singular advantages. Despite the fact that a running foot of stainless costs more than five times that of the plastic in a comparable size, it is virtually indestructible. It is designed for minimum interstices (to be microorganism-free) which simplifies cleaning. Each joint can be broken open so that unions are unnecessary. Its pressure drops should be in the same range as plastic, plus the advantage of not being sensitive to thermal shock. With the exception of long runs, no supporting structure is necessary to prevent sag and fracture. No joint could be made up too tightly as each is joined with a single-lever quick clamp. The use of stainless steel should be thoroughly explored.

Some of the plastic fittings, particularly in plastic-to-metal joints, split after sitting for periods of a few days to a few weeks. This strain aging might be avoided by making up the joints with an adaptation of a torsion wrench. None is available for this purpose and not all joints can be assembled with the standard strap wrenches recommended for plastic. Profitable research results could be anticipated by the design and testing of such a specialized tool.

 STAT

There was little question that the dairy pump used for the tests was superior, in a number of parameters which could be compared, to one model of presently used equipment. The most salient of these features were outlined in Section 3 of this report. These included trouble-free performance, simplified maintenance, high efficiency, low line pulsation, freedom in choice of plenum orientation, low air entrainment, and ease of matching capacity to load. Additionally, the operation of the pump with the manufacturer's recommended inlet head was extremely quiet. This would tend to lessen greatly the ambient sound level when a number of pumps and blowers must be combined for an operating processor. The importance of supplementing the present data with life tests and the relative imperviousness of the gaskets to photographic solutions cannot be emphasized too strongly. Other manufacturers' sanitary pumps should be tested also.

It is further recommended that a series of comparative tests be performed to determine the optimum type of flowmeter for this application, i.e., most accurate, least expensive, easiest to maintain, and lowest pressure drop.

Because of the singular performance of the two sweep fittings tested, continued experimentation should be directed toward enlarging the variety of low-loss fittings such as these. For example, several types of sweep tees could be fabricated from sections of the sweep elbows and straight pipe. Predictably, the low pressure drops obtainable could usher in a whole new design concept for the plastic fabricators and a distinct advance in the state-of-the-art.

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 STAT

APPENDIX A

A1.1 Sample flowmeter calibration calculation.

A spin-flow stainless steel bucket held 33 lbs. of water at 75.00°F.

At start of run, 11:04 A.M.:

T_1= Temperature at pump inlet	= 71.15°F.
T_2= Temperature at pump outlet	= 71.05°F.
P_1= Pressure at pump outlet	= 14.0 ±0.5 psi
Flowmeter reading	= 41.2 gpm
Time to fill calibrated bucket (aver.)	= 5.8 seconds
T_3= Temperature in tank	= 22.45°C.

Density of water (Reference 3):

	°C	p gm/ml
	21	0.99802
71.05°F =	21.70	0.99787
	22	0.99780
	23	0.99756
75.00°F =	23.90	0.99734
	24	0.99732

W=Mass Rate of Flow #/sec

$$= \frac{33}{5.8} \times \frac{.99787}{.99734} = 5.693 \ \#/\text{sec}$$

$$\frac{\#}{\text{Sec.}} \Big| \times 60 \ \#/\text{min} \Big| \times .1198 \ \frac{\text{gal}}{\text{min.}} @39.2^{\circ}\text{F} \Big| \times .99787 \ \text{gal/min.} @71.05^{\circ}\text{F} \Big|$$

5.693 x 60 x .1198 x .99787 = 40.83 gpm measured vs
41.2 Flow Meter

A 1.2 SAMPLE REYNOLDS NUMBER CALCULATION

$$\text{Reynolds Number} = Re = \frac{DV\rho}{\mu} = \frac{DG}{\mu} = \frac{4W}{\pi D\mu} = \frac{4mG}{\mu} = \frac{4\gamma}{\mu}$$

Where:

D = Diameter, feet

V = Velocity, feet per second

ρ = Density, pounds per cubic foot

μ = Viscosity, pounds per cubic foot per second

W = Mass rate of flow, pounds per second

m = Hydraulic radius, feet

$G = W/S = V\rho$

S = Cross section, square feet

$$Re = \frac{4W}{\pi D\mu} = \frac{4 \times 5.693}{\pi \times .1048 \times .000648}$$

$$= 1.067 \times 10^5$$

Darcy-Weisbach Expression =

$$h = f \left(\frac{l}{D}\right) \frac{V^2}{2g}$$

where:

h = head loss in feet

l = length of pipe, feet

g = acceleration of gravity, feet per second^2

f = friction factor, or coefficient, dimensionless solving:

$$f = h\left(\frac{D}{l}\right) \quad \frac{29}{V^2}$$

$$= \frac{3.48 \times .1048 \times 2 \times 32.17}{10.02 \times (10.36)^2}$$

$$= .00218$$

APPENDIX B

B1. 1 SAMPLE THERMOMETER STEM CORRECTION.

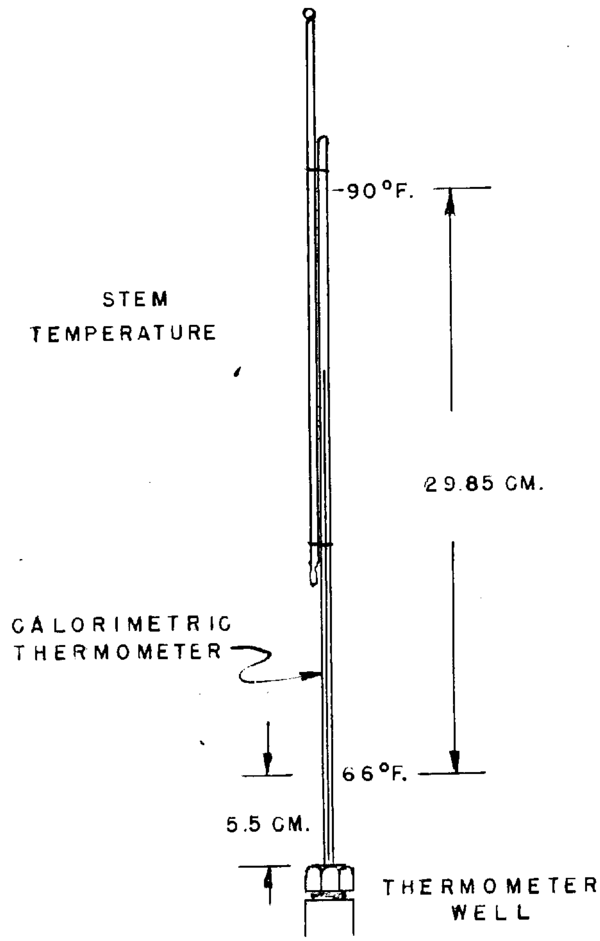

FIGURE B1-1

Tc= To+ fxlx(To-Tm)

Where:

Tc = corrected temperature

To = observed temperature

l = column length in degrees above liquid surface

Tm = mean temperature of mercury

f = correction factor = .000157 (Corning 0041)

To calculate l:

$$\frac{24^{\circ}F}{29.85\ cm} \times 5.5\ cm = 4.33^{\circ}F$$

$$l = 71.05 - 66.00 + 4.33 = 9.38^{\circ}F$$

$$Tc = 71.05 + .000157 \times 9.38\ (71.05 - 74.3)$$

$$= 71.05 - .006$$ Correction factor less than error of reading

APPENDIX C

C1. 1 CALCULATION OF PROBABLE ERROR OF MEASUREMENT

The most direct way to calculate the validity of the pressure drop data is to compare the actual total pressure loss in each Branch of the Test Rack with the value obtained when individual losses for each fitting and the connecting pipes are added together. This was done as follows:

I. $\sum \Delta p$ pipe and fittings. Branch I

-Where 24. 178 feet = total length of connecting pipe and . 3473 = pressure loss per foot in feet of water

24. 178 x . 3473 = 8. 397

90° elbow	4 x 1. 81	= 7. 24
Union	1 x . 29	= . 29
Tee	1 x 2. 19	= 2. 19
Ball Valve	1 x . 87	= . 87
"Y" Valve	1 x 2. 48	= 2. 48
		21. 467

Actual Reading = 22. 19

Calculated Reading= 21. 47

Percentage error = 3. 24%

II. $\sum \Delta p$ pipe and fittings. Branch II

	24. 465 x . 3473	= 8. 497
90° Elbow	3 x 1. 81	= 5. 43
Union	1 x . 29	= . 29
Tee	1 x 2. 50	= 2. 50
Flanged Elbow	1 x 1. 40	= 1. 40
Coupling	1 x . 47	= . 47
Plug Valve	1 x . 78	= . 78
45° Elbow	2 x 1. 90	= 3. 80
		23. 167

Calculated Reading	=	23.167
Actual Reading	=	22.84
Percentage Error	=	1.44%

The foregoing calculations were made at the maximum flow rates obtained in the two Branches. When it is considered that the calculated error in each case would be cumulative and that Branch I had a total of eight fittings, while Branch II had ten, the average error of measurement per fitting is less than 0.5 percent (0.41 percent for I and 0.14 percent for II).

In the instances of low flows for the small-loss fittings (ball valve, union, coupling, etc.), reference to the data tables shows that the correction for line loss between manometer taps and the fitting being measured was, in many cases, 1/3 to 1/2 of the total pressure drop. Since the manometer readings themselves were rounded off to the nearest hundredth, the percentage of error could be quite high. The significance of such error, however, is minimal as it occurs at flow rates much below operating levels of any stage in a large production processor.

As stated in Subsection 2.1, the manometer could be read to an accuracy of 0.2 percent full scale when temperature-compensated. The pressure fluctuations introduced by the pump were of greater magnitude, in all cases, than the error of reading. The two calorimetric thermometers could be read to within $\pm 0.01^{\circ}$F. and the Centigrade thermometer to $\pm 0.01^{\circ}$C.

The pipe lengths were measured with a 100-foot Lufkin Chrome-Clad surveyor's tape, accurate to 0.001 foot. The manometer tap distances were measured with an L. S. Starrett 12-inch steel rule, No. C305R, accurate to 0.01 inch. Since maximum pressure drops were on the order of 0.03 foot per inch of pipe, any conceivable error of measurement in pipe length would be much below manometer error.

APPENDIX D

D1. 1 SAMPLE PUMP CALCULATIONS

Two typical calculations follow in which total head, suction head, and liquid horsepower were determined for the pump and for two different bearings, a 1-1/2-inch ID copper bearing and an identical stainless steel one. In both cases, barometric pressure was 29.99 inches of mercury at the time of recording the data and thus, a correction was unnecessary.

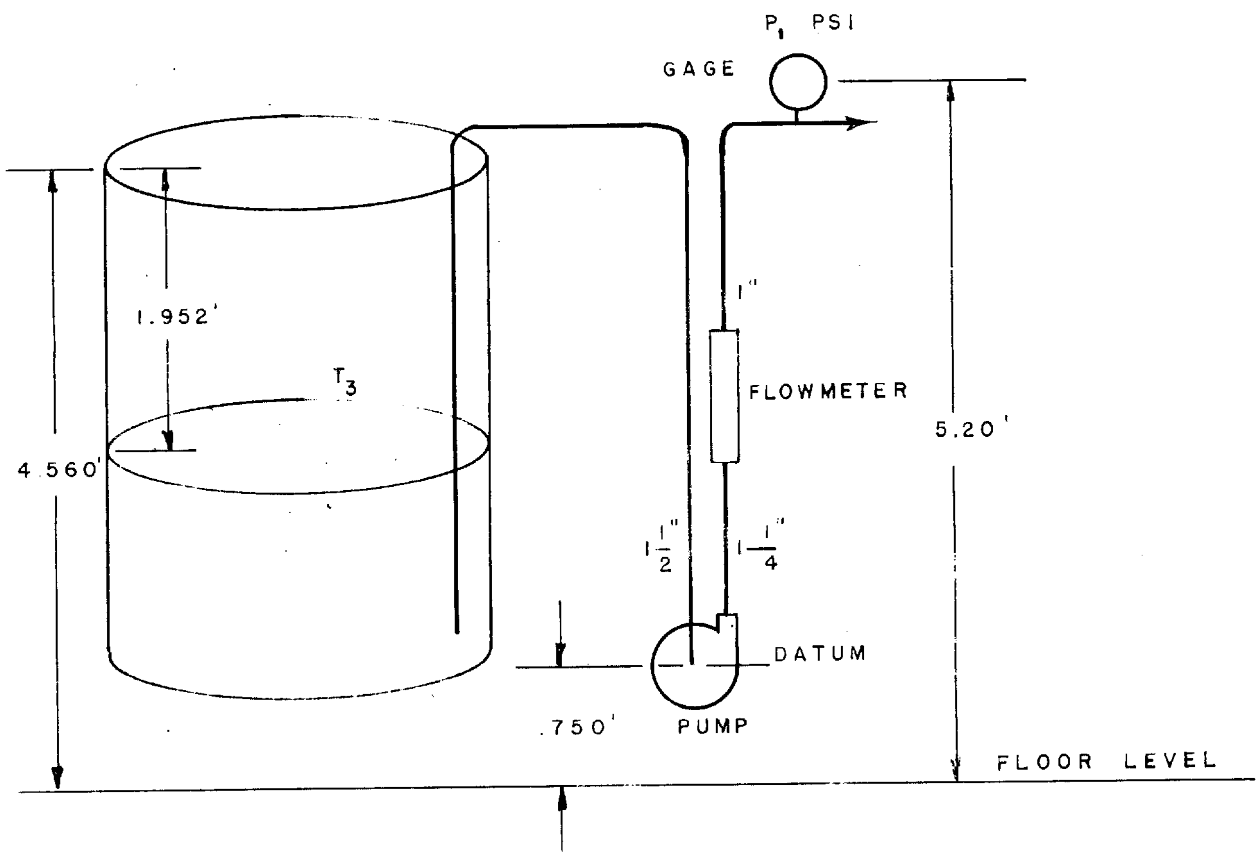

FIGURE D1-1

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Pump Suction Head = $h_s = h_{sg} + Z_s + \frac{V_s^2}{2g}$

Pump Discharge Head = $h_d = h_{dg} + Z_d + \frac{V_d^2}{2g}$

Where h_{dg} = discharge gage reading in feet of water

h_{sg} = suction gage reading in feet of water

Z_d = elevation of discharge gage zero above datum elev., feet

Z_s = elevation of suction gage zero above datum elev., feet

V_d = aver. water veloc. in discharge pipe @ discharge gage, ft/sec.

V_s = aver. water veloc. in suction pipe @ suction gage, ft/sec.

For 1.5" ID Cu Tube

$P_1 = 5.0$ psi

Flow = 34.0 gpm

$T_3 = 22.4^\circ C.$

$$h_{sg} = 4.560 - 1.952 - .750$$
$$= 1.858'$$

$$Z_s = 0$$

$$V_s = \frac{gal}{min} \times \frac{fT^3}{gal} \times \frac{1}{fT^2} = ft/min$$

$$V_s = 34.0 \times \frac{\overset{.1340}{0.1337}}{0.99782} \times \frac{1}{.0124}$$
$$= 367.4 \text{ ft/min}$$

Where $A = \left(\frac{1.507}{12}\right)^2 \frac{\pi}{4} = .0124$

gal to ft^3 @ $4^\circ C$ = x (0.1337)

$d_{21.9^\circ C} = .99782$

$$\frac{V_s^2}{2g} = \frac{\left(\frac{367.4}{60}\right)^2}{2 \times 32.17} = \frac{37.49}{64.34}$$
$$= .583'$$

$$\therefore \quad h_s = h_{sg} + Z_s + \frac{V_s^2}{2g}$$

$$= 1.858 + 0 + 0.583$$

$$= 2.441$$

$$h_{dg} = 5.0 \times \frac{27.673}{12} \times \frac{1}{.99782}$$

$$= 11.56'$$

Where:

psi to in. H_2O @ $4^{\circ}C = x(27.673)$

$$V_d = 34.0 \times \frac{0.1337}{0.99782} \times \frac{1}{.00617}$$

$$= 738.4 \text{ ft/min.}$$

$$\text{Where } A = \left(\frac{.942}{12}\right)^2 - \frac{\pi}{4} = .00785 \times .7854$$

$$= .00617$$

$$\frac{V_d^2}{2g} = \frac{\left(\frac{738.4}{60}\right)^2}{2 \times 32.17} = \frac{151.5}{64.34} = 2.355$$

$$h_c = h_{dg} + Z_d + \frac{V_d^2}{2g}$$

$$= 11.56 + 4.45 + 2.355$$

$$= 18.365$$

$$\text{Liquid Horsepower} = \text{whp} = \frac{(\#/\text{min}) \times (\text{total head})}{33,000}$$

$$= \frac{\left(34.0 \times 8.337 \times \frac{.99782}{.99905}\right) \times 15.924}{33,000} = \frac{283.1 \times 15.924}{33,000}$$

$$= 0.137 \text{ hp}$$

Where 8.337 # = 1 gal. water @ $60^{\circ}F$

$60^{\circ}F = 15.56^{\circ}C$

$d_{15.56^{\circ}C} = .99905$

Total Head = H = $h_d - h_s$

= 18.365 = 2.441 = 15.924

For 1.5" ID S.S. Tube

P_1 = 5.0 psi

Flow = 39.3 gpm

$T_3 = 22^{o}$ C

$$V_s = 39.3 \times \frac{.1337}{.99780} \times \frac{1}{.0124}$$

= 424.7 ft/min

$$\frac{V_s^{\ 2}}{2g} = \frac{\left(\frac{424.7}{60}\right)^2}{2 \times 32.17} = \frac{50.1}{64.34} = 0.779$$

h_s = 1.858 + 0 + .779

= 2.637

$$V_d = 39.3 \times .1340 \times \frac{1}{.00617} = 853.5$$

$$\frac{V_d^{\ 2}}{2g} = \frac{\left(\frac{853.5}{60}\right)^2}{2 \times 32.17} = \frac{202.4}{64.34} = 3.146$$

h_d = 11.56 + 4.45 - 3.146 = 19.156

$\therefore$ H = 19.156 - 2.637 = 16.519

$$whp = \frac{(39.3 \times 8.337 \times \frac{.99780}{.99905}}{33,000} = 16.519 = \frac{54052}{33000}$$

= 0.164 hp

D1.2 CONCLUSIONS

When this figure is compared with that for copper (0.137), the penalty in power requirements we are paying for the pressure drop in the welded and drawn stainless tubing can be seen. Since this parameter is so critical, we should select seamless tubing, drawn and polished exclusively for construction of liquid bearings. The higher cost of the seamless tubing will be more than offset by reduced pump horsepower and improved performance.

D1.3 EFFECT OF PUMP DISCHARGE ANGLE

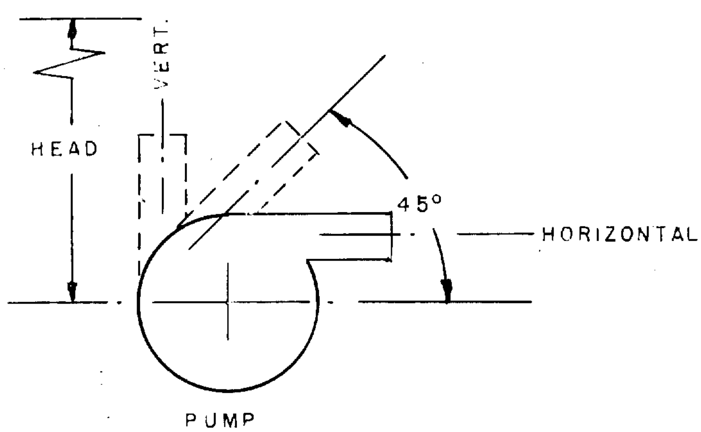

FIGURE D1-3

Eight feet of 1-5/8-inch ID flexible hose was added to circuit so plenum could be revolved easily. For each series, Branch II was open, Branch I closed.

APPENDIX E

E1.1 ROTAMETER PRESSURE LOSSES

In order to calculate the approximate pressure drop across the flowmeter, the inlet pressure P_1 was measured with the pressure gage and the outlet pressure P_2 with a mercury manometer. While converting pressure gage readings to inches of mercury introduced some error, it avoided extensive changes in the test apparatus.

Flow	T_2 °F	P_1 psi	P_2 in. Hg.	P_1 (feet water) x2.3066	@39.2°F	P_2 Corr.	Δp
40.8	72.90	14.0	20.76	32.29	32.21*	21.73	10.48
39.5	74.05	14.3	21.74	32.98	32.90*	22.76	10.14
37.5	74.35	14.6	22.62	33.68	33.59*	23.68	9.91

* Corrected for density at measured temperature.

°F	°C	gm/ml
72.90	22.73	.99762
74.05	23.35	.99748
74.35	23.53	.99743

In the case of the 40.8 gpm flow, the pressure drop is equivalent to 30.2 feet of 1-1/4-inch pipe.

$\frac{10.48}{.3473}$ = 30.2 ft. where .3473 is the pressure drop in $\frac{\text{feet}}{\text{foot}}$

 STAT

Plenum Position	Time	Flow gpm	T_3 ^oF	P_1 psi	Head feet	Impeller
Vertical	1:53 PM	42.8	69.69	13.9	6.500	SPL.
45^o	1:59 PM	42.8	69.69	13.9	6,500	SPL.
Horizontal	2:03 PM	42.5	69.69	13.9	6.500	SPL.
Vertical	2:42 PM	55.0	69.60	22.8	6.120	S
45^o	2:46 PM	55.5	69.70	23.0	6.120	S
Horizontal	2:48 PM	55.5	69.75	22.9	6.120	S

As can be readily seen from the above compilation, in no case does the position of the plenum discharge outlet make even 1 percent difference in flow. Note that two different impellers were used in the test.

APPENDIX F

1. 1 DETAILED DESCRIPTION OF APPARATUS AND INSTRUMENTATION

1. 1. 1 Hold Tank

The hold tank was [redacted] unit fabricated from stainless steel, welded and ground, with a rounded bottom and support skirt. Its capacity was 147.2 gallons. The inside diameter was 33-3/4-inches; the overall depth at the center was 39 inches and, at the edge, 37 inches. Its 2-inch bottom drain had a vortex-breaking cover and the tank had a single one-inch side drain.

1. 1. 2 Pump

The liquid flow was provided by a 2-horsepower Cherry-Burrell "Flexflow" pump, Model VAH, Model No. 12566-0, Serial No. 25720. The motor was manufactured by the Louis Allis Co., Milwaukee, Wisconsin. It is a three-phase, 220/440 volt, 5.2/2.6 ampere, Type CDIX, Frame 204C, Class AO9, rated at 3495 rpm and 55^{o}C. full load temperature rise. The pump used had three optional impeller lengths available. The shortest, "Spe," (special) was 3-inches long, "S" (short) was 3-3/4-inches long and "MM" (medium) was 4-7/16-inches long. All pressure drop tests except those for the sweep elbows were performed with the "Spe." blade and some additional pump tests with the "S" blade. Maximum rating with the "Spe." impeller was 35,000 pounds per hour (4198 gallons) at a 17-foot head. With the "S" impeller, rating was 30,000 pounds per hour (3598) gallons) at a 50-foot head, manufacturer's rating. The "M" blade was not used because the "S" gave greater flow than our rotameter capacity.

1.1.3 Pipe

The pump inlet pipe was Kraloy, Chemtrol Co. 2-inch LPS, Schedule 80, PVC II, CS 207-60, rated at 222 psi working pressure at 77°F. Interconnecting 1-1/4-inch PVC pipe was all B. F. Goodrich, Schedule 80, high temperature "Koroseal," WP 415 psi at 75°F PE, 150 psi at 180°F PE, 065, R28GK, 0050.

1.1.4 List of Fittings

Branch I

1) 1-1/4-inch TTP (Tube Turn Plastics, Inc.) PVC union.
2) 1-1/4-inch TTP, PVC tee.
3) 1-1/4-inch TTP, PVC 90-degree elbow.
4) 1-1/4-inch Chemtrol PVC ball valve, with Teflon gaskets.
5) 1-1/2-inch Walworth PVC "Y" valve, reduced to 1-1/4-inch IPS.

Branch II

1) 1-1/4-inch PVC 90-degree elbow, with flanges.
2) 1-1/4-inch TTP, PVC coupling.
3) 1-1/2-inch Injection molded UPVC plug valve, 125 psi air at 75°F, limited to 150°F, TTP reduced to 1-1/4-inch IPS.
4) 1-1/4-inch TTP, PVC 45-degree elbow.

All remaining connecting fittings were manufactured by Tube Turn Plastics, Inc., Louisville, Kentucky.

1.1.5 Thermometers

Two of the thermometers used (T1 and T2) were BKH-Braun SS Co., ASTM-56F Bomb calorimeter types, total immersion, graduated from 66° to 95°F in 0.05°F divisions. Their numbers were 1362102 and 1362112, respectively.

STAT

All tank temperatures (designated T3) were measured on a Taylor Instrument Company "Permafused," etched-stem Centigrade thermometer. It was a gas-filled mercury type, 381mm long, ASTM 63C precision, No. 4173820, with a range of -8° to $+32^{\circ}C$. It reads to $0.1^{\circ}C$ with interpolation to $0.01^{\circ}C$.

1.1.6 Flowmeter

The flowmeter was a Brooks Rotameter Company, Lansdale, Pennsylvania, instrument. It was designed to measure gpm of liquids with a specific gravity of 1.0 (see Table 2-1 and Calibration Chart, Figure 2-5). Its serial number was D-7498 and its range was 5 to 55 gpm.

1.1.7 Pressure Gage

The inlet pressure measurements were made on a United States Gauge, Model No. 1811-T "Supergauge." Its range was zero to 60 psi pressure, its movement, connection, and bourdon tube were made from Type 316 stainless steel and its gearing was Nylon. It was calibrated and certified by the W. R. Ladewig Company, Los Angeles, California, on 16 October 1964. It reads to 0.5 psi with interpolation to 0.1 psi.

1.1.8 Manometers

All pressure drops were read on one of two manometers. The first was a Meriam U-tube vertical cleanout type whose range was 30 inches, graduated in increments of 0.10 inch. With mercury, 0.10 inch, was equivalent to 0.0491 psi.

Since many of the Δp's were very small, a specially sensitive mercury inclined manometer was constructed. This second instrument was made of accurate-bore Pyrex tubing, 122 centimeters long and 7mm in diameter. Its maximum reading was 40 inches scale, which reduced to 20 inches actual because of its 2:1 slope ratio. This was equivalent to

9.82 psi gage. It could be read to 0.01 inch and had an accuracy of 0.2 percent full scale when temperature-compensated. Each leg was provided with a glass tee and pinch clamps for bleeding the lines of entrapped air and filling the instrument with water.

STAT

REPORT

Determination of The Force Required to Bend Film 180 Degrees Over Different Radii of Curvature

STAT

February, 1965

STATINTL

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FOREWORD

STATINTL [redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted] This report should be read in conjunction with Report [redacted] of which it forms a part.

STAT

STAT

Approved: [redacted]

Research Manager

STATINTL

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

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ABSTRACT

This report presents the results of investigations conducted to determine the force required to bend film 180 degrees over different radii of curvature. The tests were carried out as part of a series necessary to establich design criteria essential to the development of the liquid/air bearing concept of film processing.

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SECTION 1 - INTRODUCTION

In a liquid/air bearing type of film processor, the objective is to transport the film from the load end to the takeup end without mechanical contact between the film and the machine. This objective is attained by providing stationary sleeves in place of rollers in and between the tanks. Air, or the fluid in the tanks, is pumped into the sleeves and ejected at sufficient velocity and pressure to form a continuous cushion between the film and the bearing. The film is transported over this continuous cushion free of mechanical handling and with low tension on the film. Two main criteria are necessary to determine the cushion loads and capstan torque requirements: (1), the coefficient of friction of the film (film drag load) and (2), the additional tension required to bend the film around the bearings. This report determines the values of the latter parameter.

SECTION 2 - TECHNICAL DISCUSSION

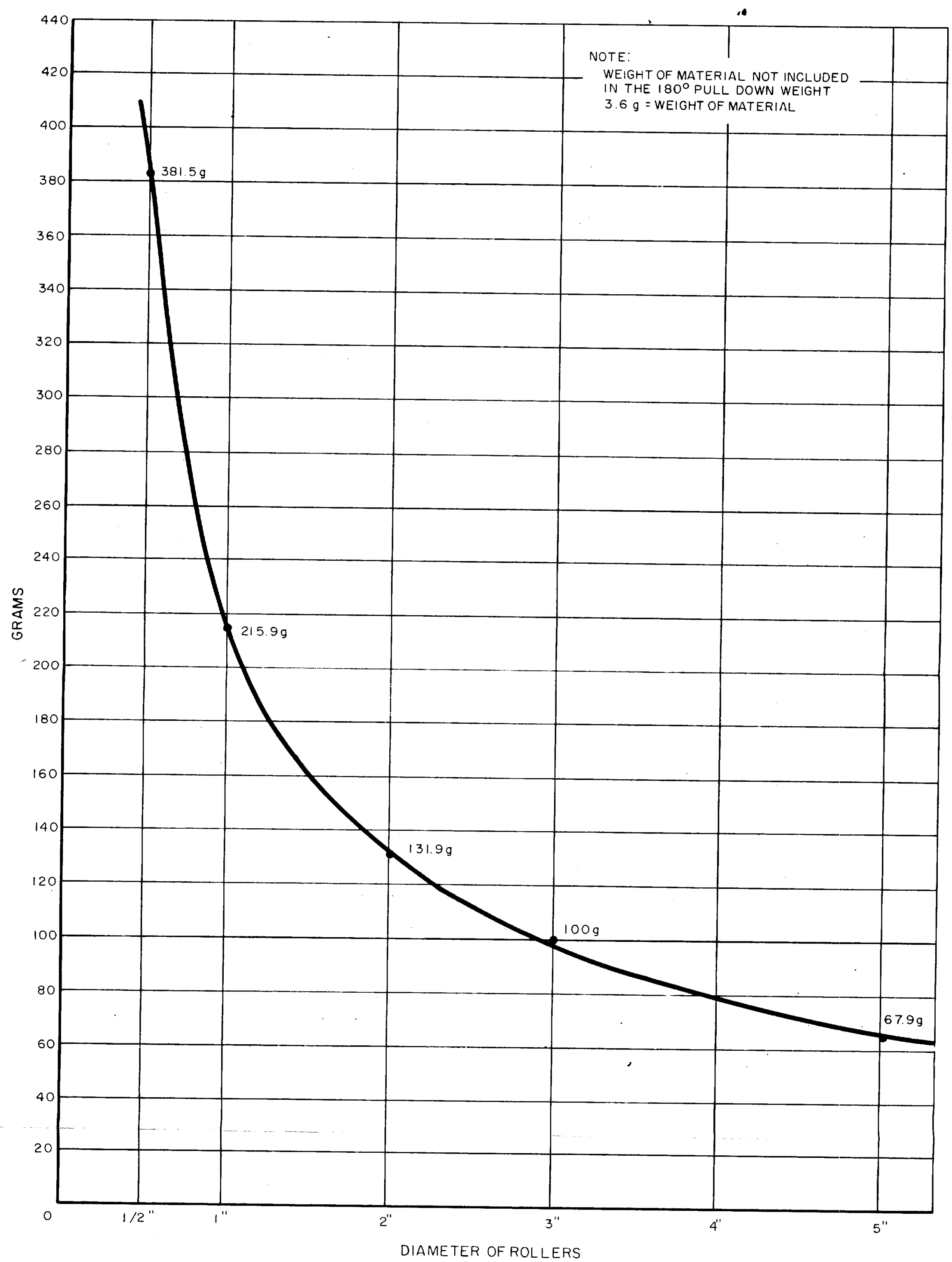
To obtain the values of the loads, a framework was constructed in which rollers were mounted as required. The diameters of the rollers used were 1/2, 1, 2, 3, and 5 inches. These diameters were selected as those giving a sufficient number of points for the plotting of a curve for each film thickness. A 15-inch length of each film sample was hung over each diameter of roller and weights were added until the film touched the roller over an angle of 180 degrees. New, dry, unprocessed film was used. The film tested was in 35mm, 70mm, 5-inch, and 9-1/2 inch widths, with base thicknesses of 0.00257, 0.0056, and 0.0085 inch.

The results obtained show that each test yielded a real half of a hyperbola, and that the break in the curves occurred for a diameter of 5 inches or greater.

SECTION 3 - CONCLUSIONS

Based on the dry film values obtained, it is concluded that capstans and bearings should be designed with a minimum diameter of 5 inches to minimize total film tension and maximum bearing and capstan loads. Where circumstances permit, bearings of even larger diameter should be used.

To give a design example, a processor using thirty-seven 1-inch diameter bearings (assuming use of 2.57 mil thin-base film, Figure 2-1) would require a total force of 153.6 grams x 37, or 12.5 pounds. Selection, in the same machine, of 5-inch diameter bearings would result in a total force of 17.5 grams x 37, or 1.43 pounds. This produces a significant reduction in film tension, bearing cushion loads, and drive capstan torque.



LEADER 35MM X 15" HEAVY-BASE FILM
08.5 MILS – THICKNESS OF MATERIAL
UNPROCESSED

Figure 12.

STAT

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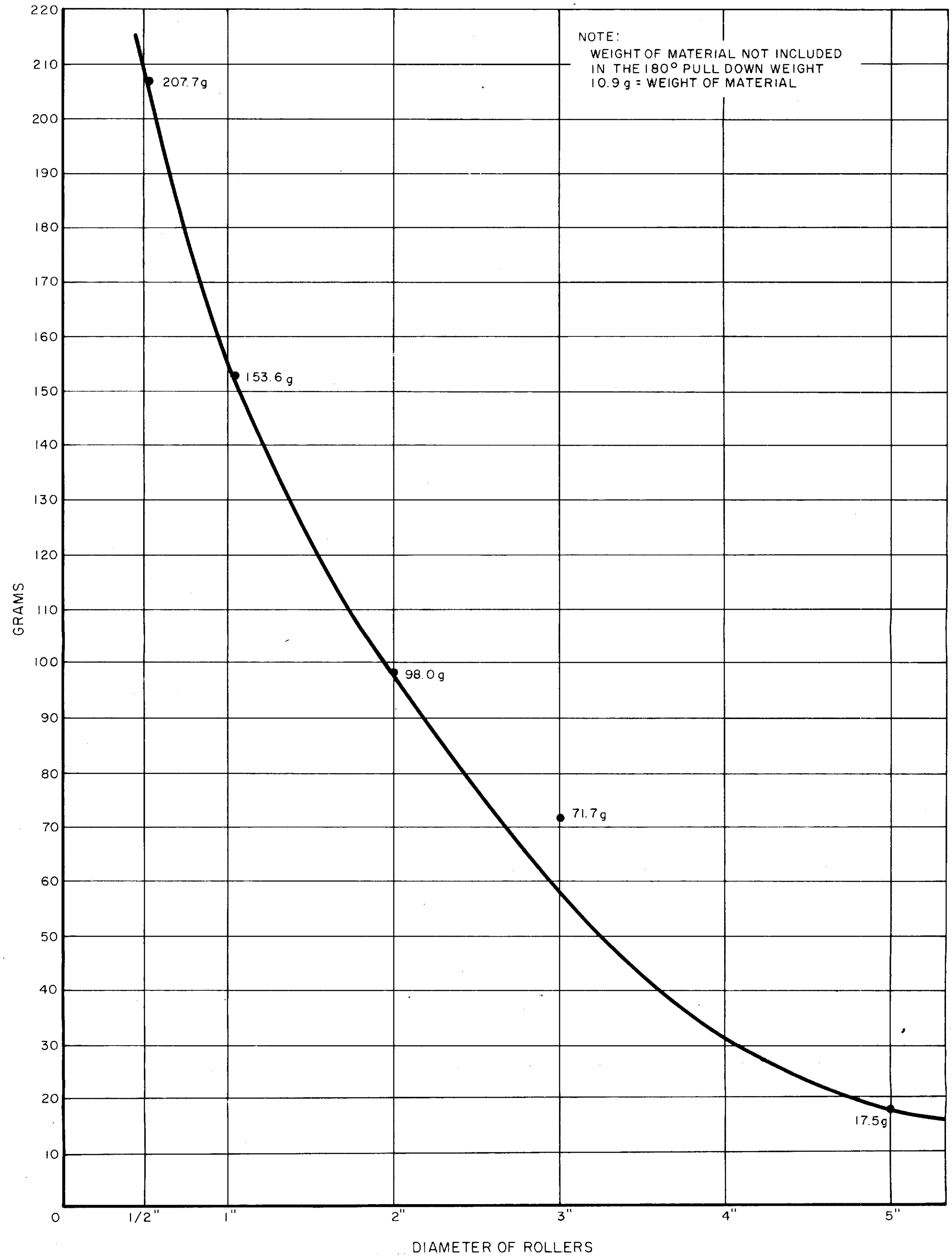

S.O. 1188 9-1/2" X 15" THIN-BASE FILM
2.57 MILS - THICKNESS OF MATERIAL
UNPROCESSED

Figure 1.

3

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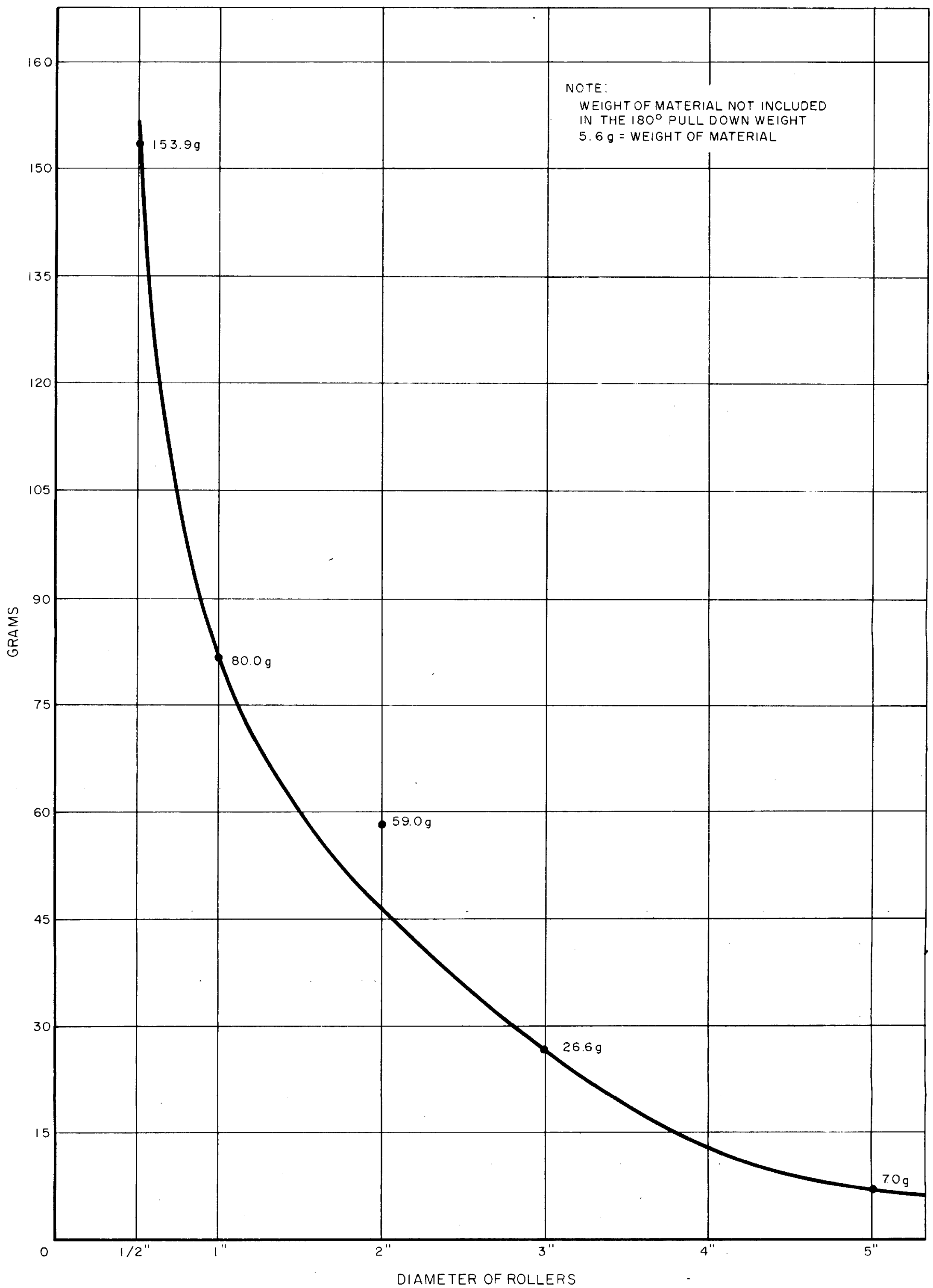

S.O. 1188 5" X 15" THIN-BASE FILM
2.57 MILS - THICKNESS OF MATERIAL
UNPROCESSED

Figure 2.

4

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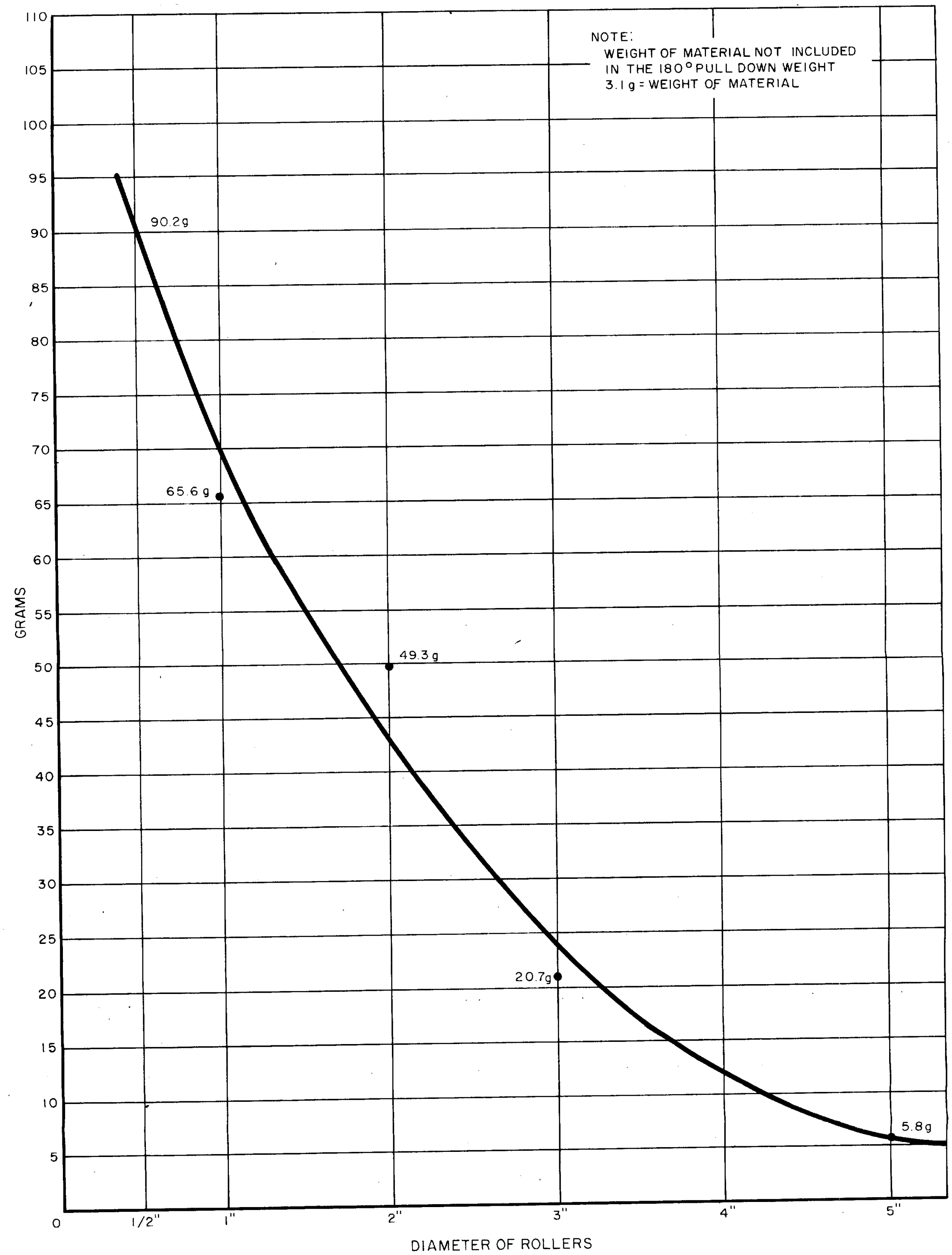
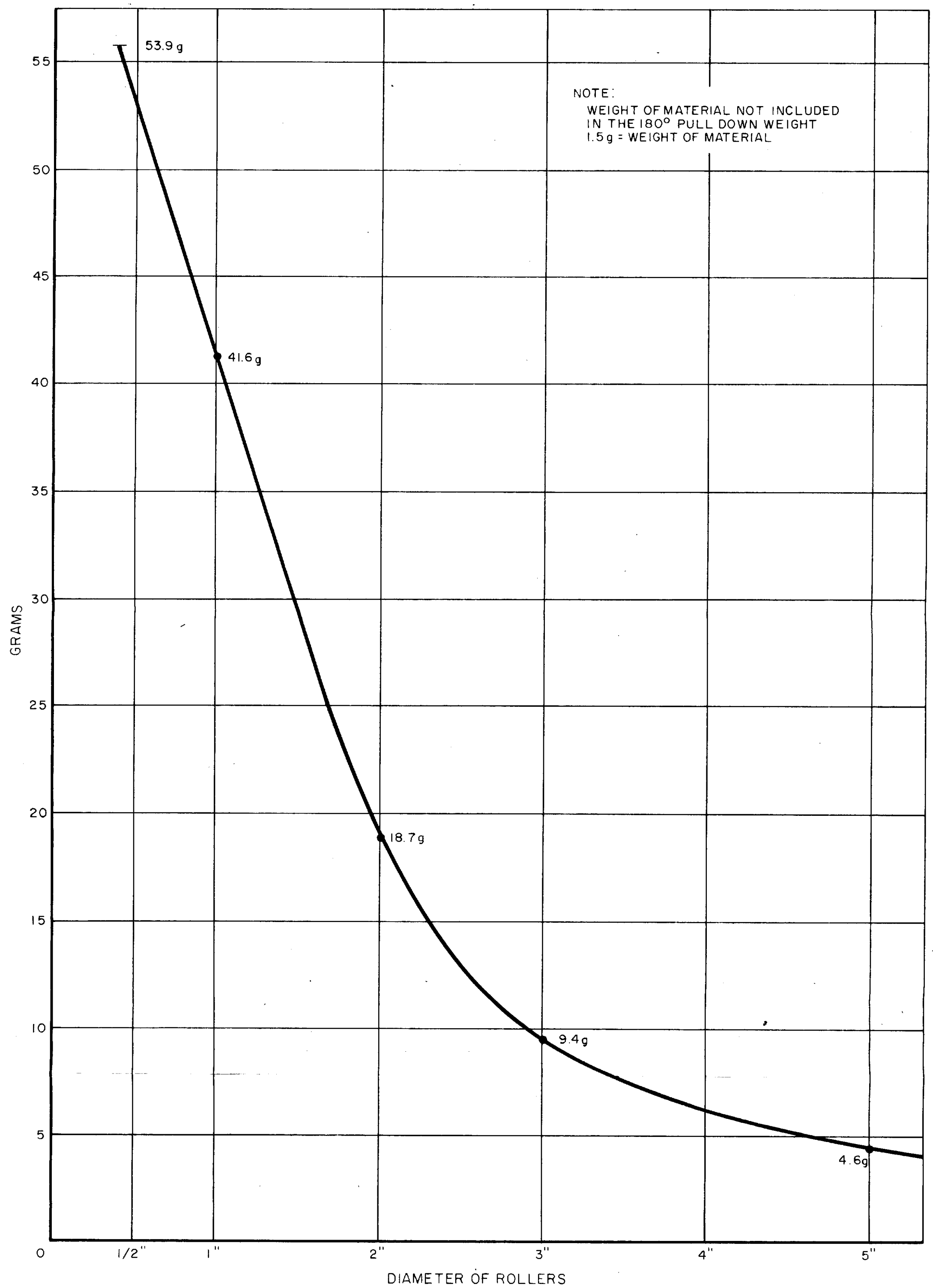


S.O. 1188 70 MM X 15" THIN-BASE FILM
2.57 MILS - THICKNESS OF MATERIAL
UNPROCESSED

STAT

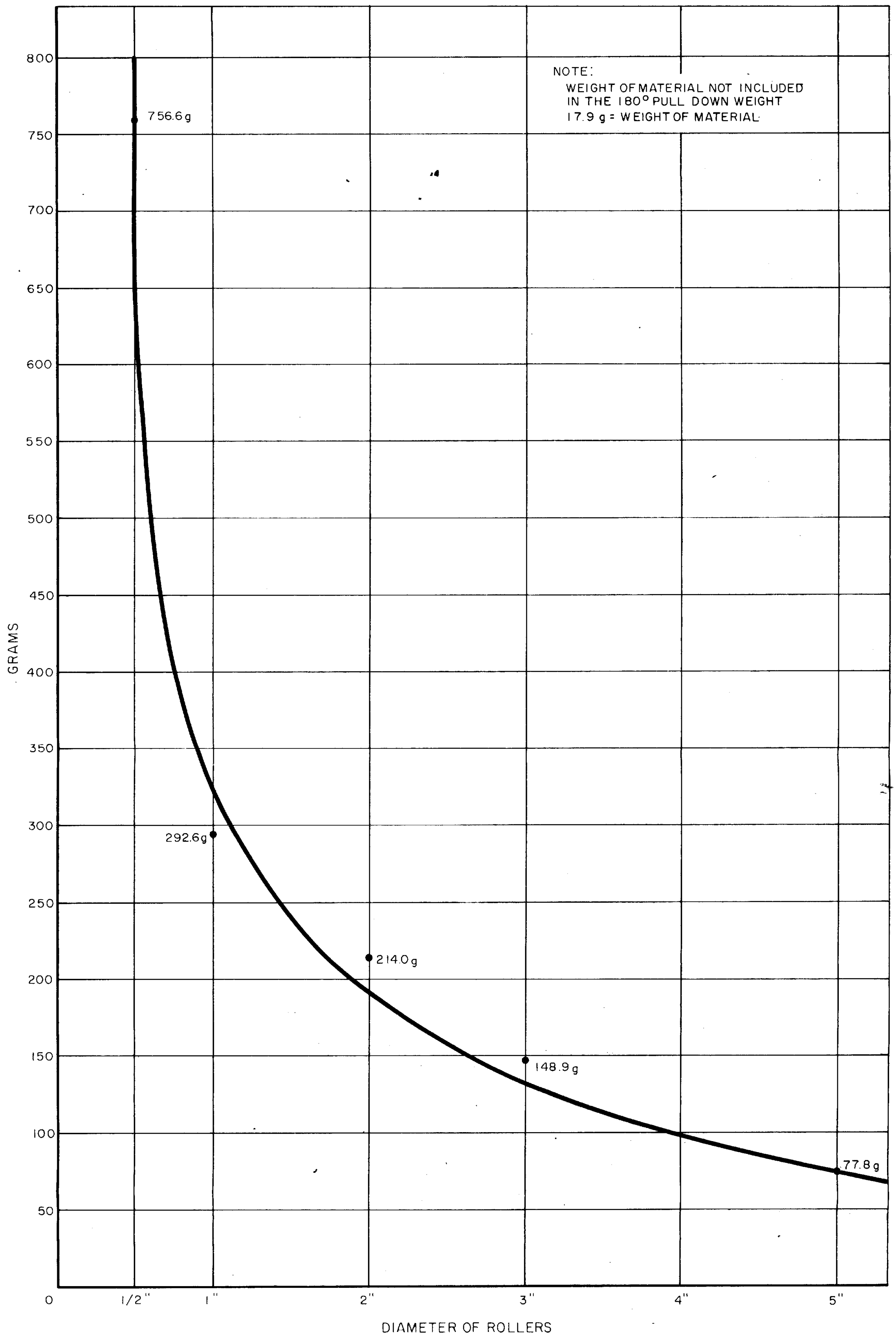
Figure 3.



S.O. 1188 35 MM X 15" THIN - BASE FILM
2.57 MILS - THICKNESS OF MATERIAL
UNPROCESSED

STAT

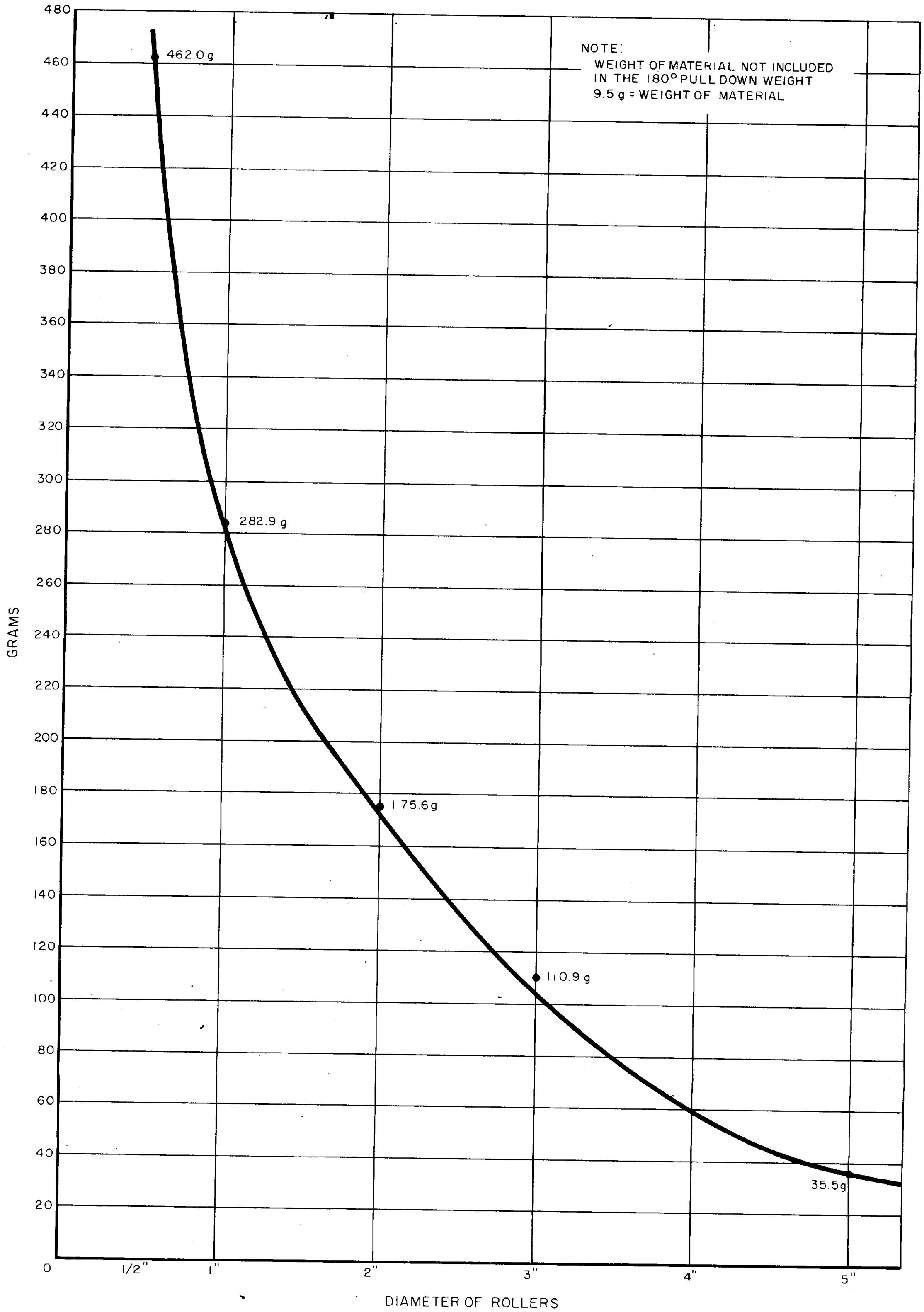
Figure 4.



TYPE 8430 9-1/2" X 15" MED-BASE FILM
5.6 MILS - THICKNESS OF MATERIAL (BASE ONLY)
UNPROCESSED

STAT

Figure 5.



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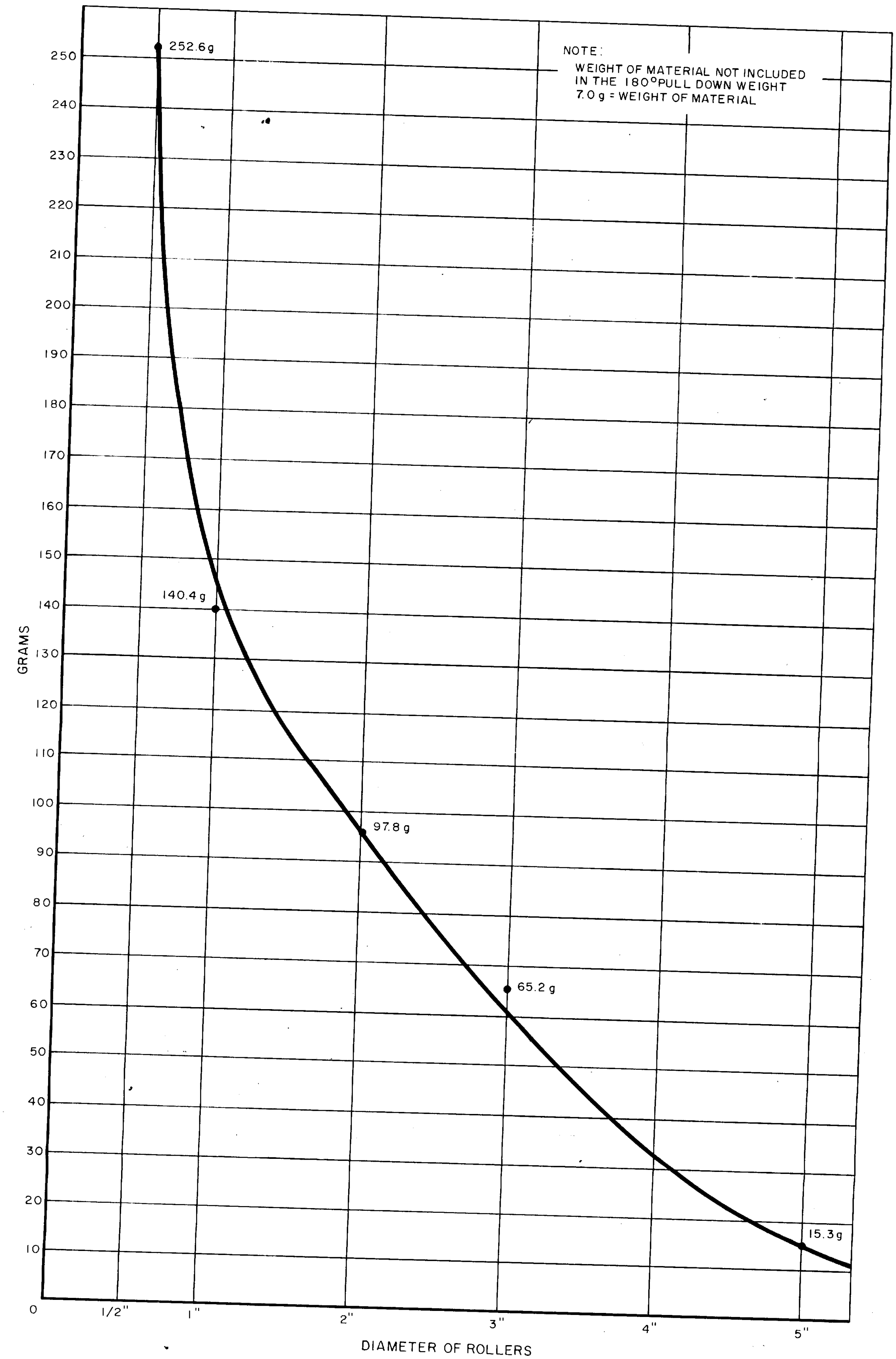
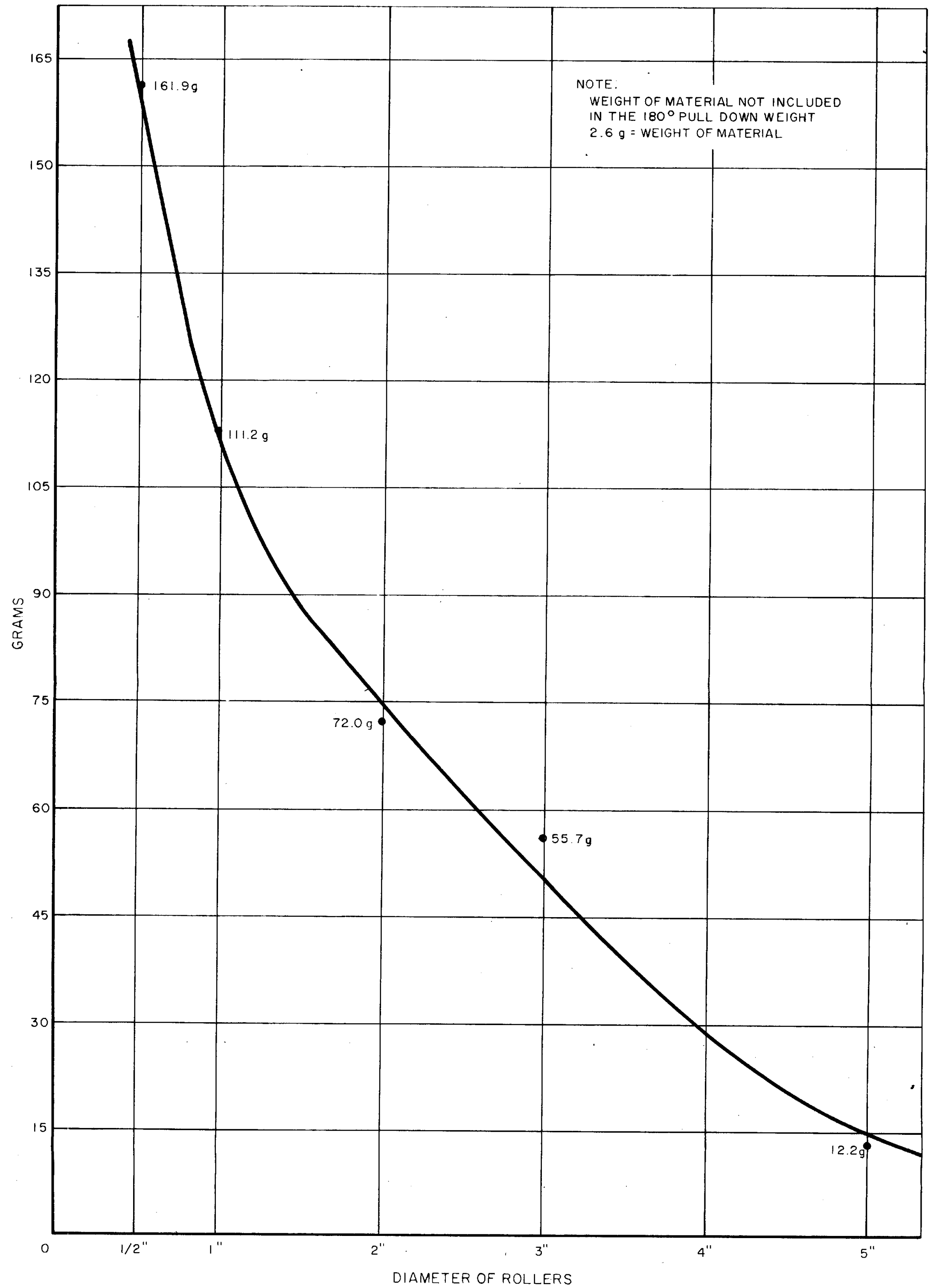
TYPE 8430 5" X 15" MED-BASE FILM
5.6 MILS–THICKNESS OF MATERIAL
UNPROCESSED

Figure 6.



TYPE 8430 70MM X 15" MED-BASE FILM
5.6 MILS – THICKNESS OF MATERIAL
UNPROCESSED

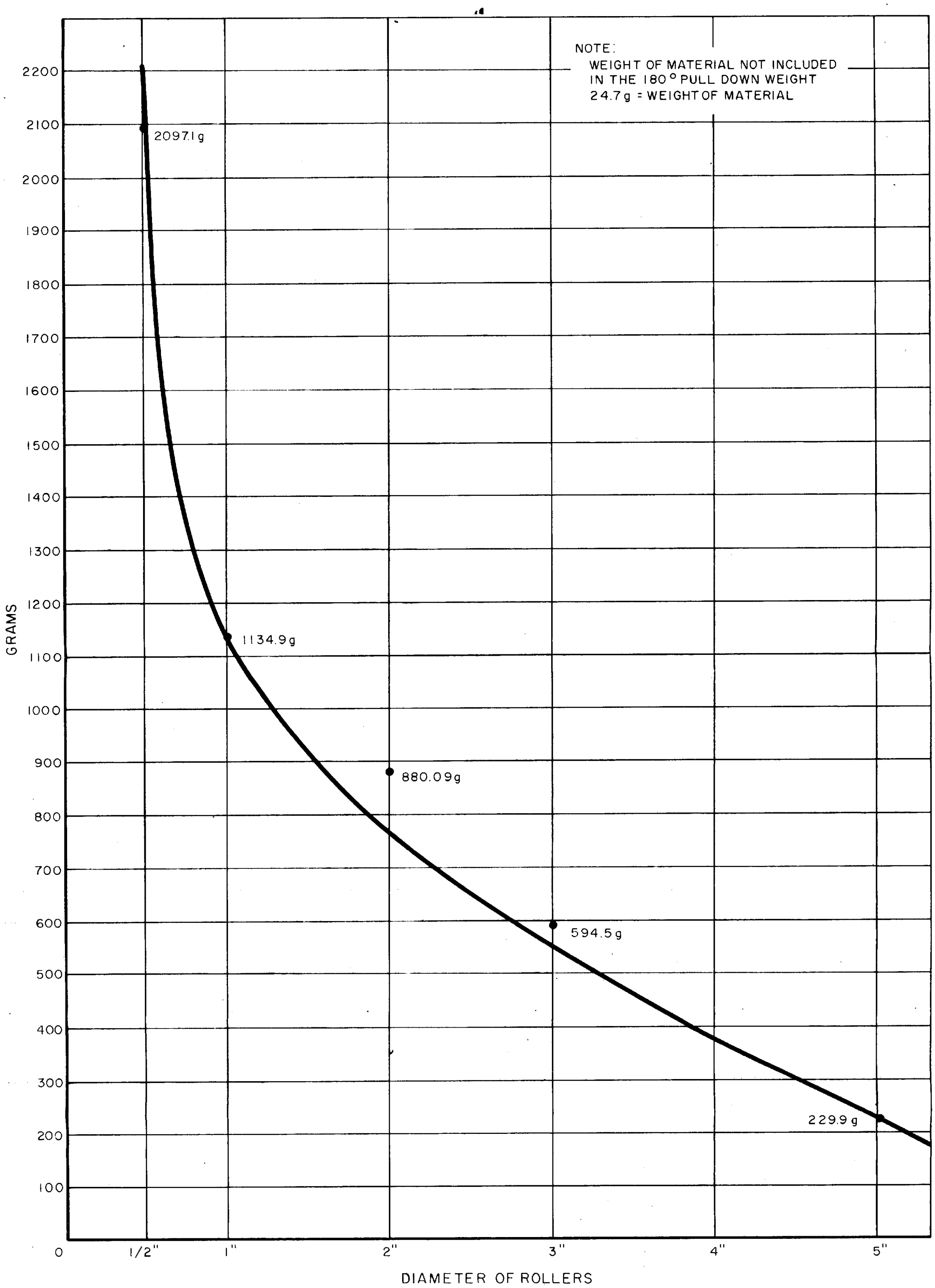
Figure 7.



TYPE 8430 35MM X 15" MED-BASE FILM
5.6 MILS-THICKNESS OF MATERIAL
UNPROCESSED

STAT

Figure 8.



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LEADER 9-1/2" X 15" HEAVY-BASE FILM
08.5 MILS - THICKNESS OF MATERIAL
UNPROCESSED

Figure 9.

UNPROCESSED

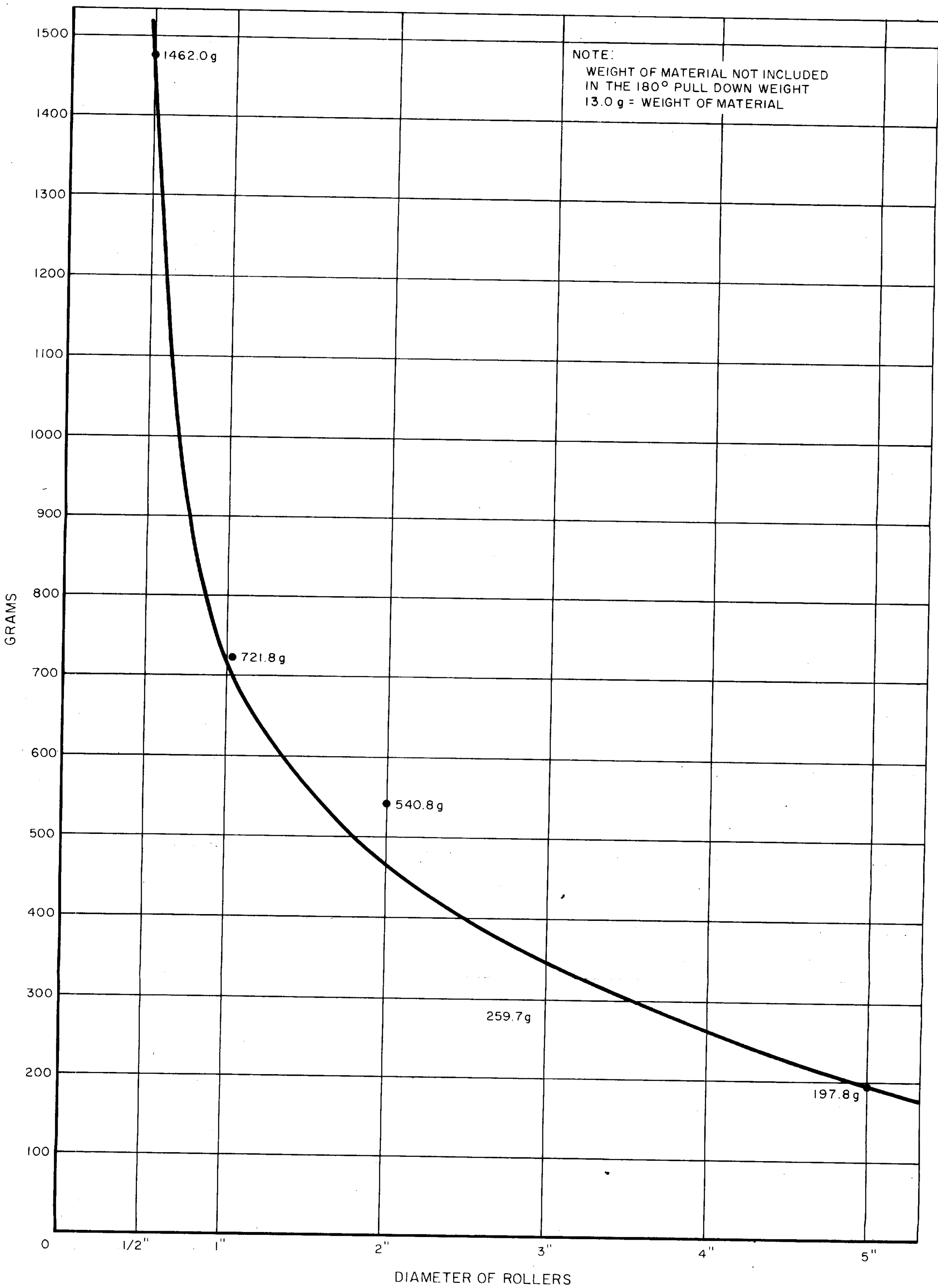

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Figure 10.

12

LEADER 5" X 15" HEAVY-BASE FILM
08.5 MILS - THICKNESS OF MATERIAL
UNPROCESSED

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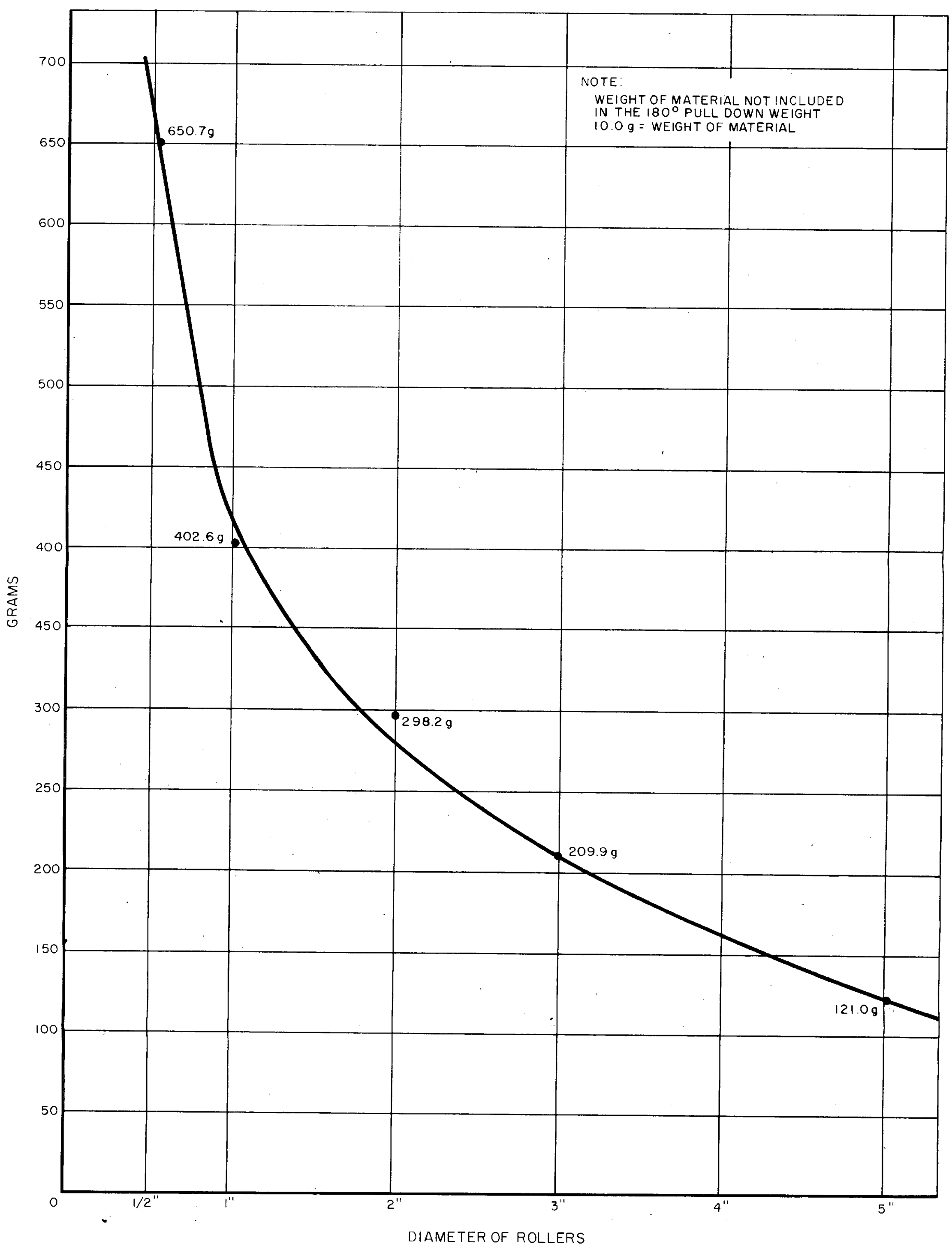

LEADER 70MM X 15" HEAVY-BASE FILM
08.5 MILS – THICKNESS OF MATERIAL
UNPROCESSED

Figure 11.

REPORT

STAT

HYDROMATIC LIQUID BEARING ASSESSMENT

STAT

Liquid Bearing

Hydromatic

STATINTL

February, 1965

FOREWORD

STATINTL

[redacted] submits this report in compliance with Item 3.1 of the Development Objective of Contract [redacted] This report should be read in conjunction with Report [redacted] of which it forms a part.

STAT
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Approved: [redacted]
Research Manager

STATINTL

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ABSTRACT

A representative number of liquid bearings were built, in which only one design parameter was changed. These were tested and graphed correlatively. Narrow bearings, center-feed types and self-centering designs were appraised. An efficient three-slot bearing was built, embodying the principles disclosed by the research. An outline of a possible mathematical approach to the analysis of the empirical data was presented.

 STAT

SECTION I

INTRODUCTION

1.1 BEARING CONCEPT

Prior to the innovation of the air and liquid bearing concept, all commercial film processing units transported film on a series of rollers (similar to the setup of a paper mill) or sprocket gears that matched the perforations in standard types of perforated film. These conventional machines use a preponderance of driven rollers, each of which must be rotated at precisely the same speed as every other roller. If they are not, one of two problems developes:

The first occurs when a variation of speed in the pressure roller groups in either the wet or dry end of the processing line causes the formation of a slack loop. This slack allows the film to slap against parts of the processor or to cohere and damage itself by abrasion. The second occurs when the film is suddenly shortened. This produces stretching, riffling, or even film breakage.

With sprocket gears instead of rollers, the film can lose tracking or its perforations can be torn. Any of these conditions, including the normal slippage over a smooth roller, results in film damage, intolerable in certain aerial surveillance and other irreplaceable original negative films.

The liquid and air bearing principle, conceived by [redacted] STATINTL
was a significant contribution to the state-of-the-art. Its development has advanced film processor design and has resulted in a new generation of equipment. By providing a fluid cushion on each bearing in the wet end of the machine and an air cushion in the drier section, the film could literally be floated through the complete processing cycle contacting only its drive capstan and takeup spool. The supporting fluid cushion was created by ejecting jets of liquid from within a cylindrical plenum. These jets, by impinging on the film passing over the bearing, created a firm liquid layer in the annular space. The air bearing is similar in principle.

Thus, in theory, the only frictional forces to which the film is subjected, are the very slight intermolecular drag coefficients and liquid viscosity factors at the boundary interfaces. In practice, however, it proved necessary to provide film guide flanges at either end to control the effluent. Although these flanges were placed far enough apart to allow for the width of film being used, there was still considerable edge friction due to erratic film tracking along its multiple-bearing pathway in the processor.

One of the serendipities of the design concept proved to be this: Because the liquid creating the cushion in each stage of the processor (developer, stop bath, fixer, etc.) was the chemical solution itself, greater penetration of the emulsion was achieved, resulting in reduced time of contact.

 STAT

1.2 PURPOSE AND OBJECTIVES

Viewing this brief historical background, it is not difficult to formulate a program for promising air and liquid bearing research. These are some of the areas needing improvement:

(1) Cushion stability - Provides a more stable cushion for film support, both for liquid (in-solution) and for air (inter-tank and drier).

(2) Horsepower requirements - Reduce these requirements by improved design in multiple-bearing installations. Measure the energy levels required to maintain firm cushions over the wide range of load conditions encountered by change of film width from 70mm to 9-1/2 inches and film thickness varying from 1.5 to 7.0 mils.

(3) Design considerations - Develop mathematical relationships among the complex hydrodynamic and fluid mechanic principles involved to enable engineers to predict bearing performance before fabrication.

(4) Mechanical efficiency - Check the effect of variable-slot openings as well as liquid and air feed arrangements.

(5) Correlations - Determine the interrelationship of velocity and flow rate for solutions and air and their optimization.

(6) Hole configuration - Determine whether or not slots are more efficient than holes.

(7) Film guide flanges - Eliminate these by designing a self-centering bearing.

(8) Construction economies - Try to reduce size, complexity, and cost.

(9) Installation - Investigate the design of self-contained, plug-in modular bearings and simplify alignment.

(10) Individual loading - Consider a feed-back mechanism to compensate for the progressive loading of the bearings from film entrance to exit.

Investigations in these areas comprises the objectives of this part of the overall processor improvement program.

 STAT

SECTION 2

TECHNICAL DISCUSSION

2.1 EQUIPMENT AND INSTRUMENTATION

The liquid bearing test rack (Figure 2-1) consisted of a 147.2 gallon stainless-steel hold tank with a rounded bottom and support STATINTL skirt, obtained on loan from [redacted] Its overall depth at the center was 39 inches and 37 inches at the edge. Its inside diameter was 33-3/4 inches. A wooden framework mounted over the tank supported the test bearings.

The liquid flow was provided by a 1-horsepower Corcoran pump, Serial No. 4322, Brg. Nash 1 and 2, rated at 36 gpm against a 45-foot head. The motor was a three-phase, 220-volt, 3.2 ampere, Type PF model rated at 3450 rpm, manufactured by the Doerr Electric Corporation. The pump inlet pipe was Kraloy/Chemtrol 1-1/2 inch IPS, schedule 80, PVC 2210, rated at 235 psi working pressure and 73.4°F temperature. It was provided with a removable plug for priming. At the discharge end, downstream from the flowmeter, the pipe was reduced to 1-inch IPS, schedule 80, PVC II, CS207-60, rated at 327 psi working pressure and 77°F temperature. Flow control was provided by a 1-inch Chemtrol ball valve, Type KC 101. All plastic fittings were manufactured by Tube Turn Plastics, Inc. The inlet to the test bearings was a short length of rubber hose with two standard adjustable stainless-steel hose clamps.

The flowmeter was a Brooks Rotameter Company instrument designed to measure gpm flow of liquids with a specific gravity of 1 (See Table 2-1 and Calibration Chart, Figure 2-2). The flowmeter serial number was D-7526 and its range was 5 to 55 gpm. Using the same data, we calculated the Reynolds numbers and graphed these against friction factors (both are dimensionless) for PVC pipe (Figure 2-3). The data for various commercial pipes and tubes was obtained from the literature (Reference
It is interesting to note how much less the coefficients of friction are for plastic than glass, supposedly the epitome of smoothness.

The pressure measurements were made on a United States Gauge Model 19061 "Supergauge". Its range was 30 inches of mercury vacuum to 30 psi pressure; its movement, connection, and bourdon tube were made type 316 stainless steel; its gearing was nylon. It was calibrated and certified by the W. R. Ladewig Company, Los Angeles, California on 7 October 1964. All temperatures were measured on a Taylor Instrument Company "Permafused", etched-stem Centigrade thermometer. It was a 381 mm long, gas-filled mercury type, ASTM 63C precision, No. 4173820, with a range of -8° to $+32^{\circ}$C. It read to 0.1°C with interpolation to 0.01°C.

Two other specialized instruments were built to record data for the research program. The first was a sensitive inclined mercury manometer (Figure 2-3a). It was constructed of accurate-bore Pyrex tubing, 122 centimeters long and 7mm in diameter. Its maximum reading was 40 inches scale, which reduced to 20 inches actual because of its 2:1 slope ratio. This was equivalent to 9.82 psi gage. It could be read to 1/100th of an inch and had an accuracy of 0.2 percent full scale when temperature-compensated. Each leg was provided with a glass tee and pinch clamps for bleeding the lines of entrapped air and for filling the instrument with water. The sensing pilot probe was constructed of a modified assayer's blowpipe.

The second device consisted of a vernier depth gage mounted on a 12-inch steel rule, graduated in 1/100ths of an inch (Figure 2-4). This device enabled a complete traverse to be made over the width of the film to determine cushion depths at different pressures and flows.

2.2 BEARING EXPERIMENTATION

2.2.1 Narrow Liquid Bearing Prototype:

The first series of experiments were performed on an experimental stainless-steel bearing (Figures 2-5, 2-6, and Table 2-2). Its overall length was 13.9 inches and the inside diameter of its throat was 1.0 inch. The outside dimensions of the plenum chamber were 11.57 inches long, 2.00 inches deep, and 0.52 inches side, with a wall thickness of 0.062 inch. The sides and bottom of the plenum were rectangular and the top semicircular. The bearing had a single slot 9.56 inches long and 0.067 inch wide.

The series of experiments and test data gathered in this report were confined to bearings having a single slot and to one bearing having three slots. As discussed in Section 1, patterns of holes in liquid bearings have inherent disadvantages such as excess power consumption, energy losses, inefficiencies, and streaking of film. It appeared obvious that a slot would obviate many of these drawbacks, particularly that of streaking due to local development.

All pressure profiles were made by sampling the slot width at 1/2-inch intervals with the pilot tube attached to the inclined manometer. The static pressures obtained were recorded, together with the inlet gage pressure, the flow in gpm, and the water temperature in degrees centrigrade. All air was bled from the test lines prior to measurement and repeatability to ± 1 percent accuracy was checked by duplicating runs on different days.

If a fluid is injected into one end of a closed straight tube of any constant cross-section and allowed to escape through a single long narrow slot, the shape of the resultant fluid fall is roughly parabolic (Figure 2-7). This effect is pronounced where the cross-sectional area is in the range of one square inch and gradually disappears as the area is increased. Thus, if the parabola were graphed in the first and fourth quadrants, in rectilinear coordinates with the base lying along the ordinate axis, the apex would fall on the abscissa. Considering the first quadrant only, the apex (or Y = 0 point) would appear at the upstream end of the slot and the highest point of the leg (or X = 0 intersection) at the downstream end. This latter was also the point of highest pressure as can be readily observed from a typical pressure profile plot.

Analytically, the effect can be explained by examining the pressure-flow relationship diagrammed. Apparently, the shock wave generated as the fluid enters is transmitted to a pressure gradient by impingement on the far end of the bearing. This unequal gradient, then gives the characteristic shape to the profile.

The plot diagrammed (Figure 2-6) also shows a sharp fall-off of pressure about midway on the slot. Since uneven pressure causes a tipping of the film and skidding away from the pressure area, this condition could not be tolerated. In order to compensate for the unbalanced pattern, a methacrylate plastic wedge was designed to fit inside the bottom of the plenum with its wide end downstream (Figure 2-5).

The wedge was 9.94 inches long, 0.41 inches wide, and tapered from 0.60 to 0.125 inch. It was drilled with 19 holes 0.196 inch in diameter; each hole was tapered with a No. 120 Ace reamer (0.498 to 0.119 inch in 3.53 inches). When this was mounted in the plenum so that the fluid was forced to flow from the small to the large ends of the tapered holes, the pressure profile was somewhat improved. However, there were still unacceptable eccentricities in the pattern. At this point, Cerrobend was added to the inside of the semicircular portion of the plenum to extend the throat of the slot to a 1/2-inch depth. Such a bearing, with a double plenum, had previously given an improved pattern. No significant change was noted; the general pattern was still subject to peaks and valleys (Figure 2-8 and Table 2-3).

Various attempts to equalize the flow were unsuccessful. These included masking the tapered slots with cellular polyurethane foam, filing lips on the upstream tapered nozzles in the equalizer, increasing the diameter of the upstream tapered nozzles, and changing the position of the equalizer one inch toward the upstream end of the bearing.

The test sequence was repeated on a similar type of bearing, differing in slot width. The slot, in this case, was 9.52 inches long by 0.120 inch wide. While considerably higher flow rates were necessary to duplicate the inlet pressure obtained with the narrow slotted bearing, the pressure profile without an equalizer was almost identical (Figure 2-9 and Table 2-4). The fact that no significant difference appeared in the plots obtained permits the following hypothesis.

Liquid bearings having an identical cross-sectional configuration and area will produce a characteristic pressure profile plot whose magnitude at constant inlet pressure is directly proportional to flow. Further experimentation with the narrow bearings was abandoned, not only because of the

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difficulty of equalizing the flow, but also because of a more significant drawback. This drawback proved to be that the extra force required to bend film and leader of heavier base thicknesses (5 to 7 mils) around the narrow radii used more energy (reflected in increased gpm) than was saved by the economies inherent in the bearings.

2.2.2 <u>Liquid Bearing Slot Data:</u>

To thoroughly explore the complex relationships between pressure profiles, throat/plenum area ratios, inlet pressures, and rates of flow, five experimental stainless-steel bearings were built. Slots were identical, 0.063 by 9-1/2 inches, and so were tube lengths. Only the inside diameter was varied. The nominal inside diameters were 1.000, 1.250, 1.500, 1.620 and 1.875 inches (Figure 2-10). The material was welded and drawn tubing selected because it is less expensive than the seamless and polished type and is normally stocked in [redacted] raw stores. These were "run-of-the-shop" jobs and no particular stress was placed upon dressing the milled slots after machining. STATINTL

Pressure profiles were measured on each bearing and then graphed (Figures 2-11, 2-12, 2-13, 2-14 and 2-15, and Tables 2-5, 2-6, 2-7, 2-8 and 2-9). The construction material, the machine finish of the slot, and the position of the seam relative to the slot appeared to be much more critical parameters than was supposed. When the empirical data obtained on all bearings were grouped on one graph, the differences became apparent (Figure 2-16).

The most and the least satisfactory bearings from the standpoint of economy of fluid flow were the two narrow prototypes discussed in Section 2.2.1. However, since the effective area available for the cushion support of film was only 47 and 42 percent respectively for these bearings, the results should not be misinterpreted. The five experimental stainless-steel bearings were grouped rather closely and differed mainly in their effective support areas. The fact that the curve for the 1.620 inch inside diameter bearing falls to the left of those for the 1.500 and the 1.875 inch inside diameter bearings supports the statement concerning the critical nature of some of the parameters.

The 1.500 inch inside diameter copper bearing discussed in Subsection 2.2.3 following, proved to be the most satisfactory compromise among the parameters.

2.2.3 <u>Self-Centering Liquid Bearings:</u>

In the list of experimental objectives, one important aim was to produce a liquid or air bearing that would tend to keep the film centered while it travelled through the processor, regardless of minor bearing misalignments. In the ideal configuration embodying this principle, the edge guides would be eliminated.

To attain this objective, a simple, single-slot bearing was built from smooth, polished copper. Its length was 20-1/4 inches; the inside diameter was 1.500 inches ± .005 inch, and the slot was 9-1/2 inches by 0.063 inch. The pressure profile was carefully measured (Figure 2-17 and Table 2-10) and the bearing was modified as discussed below.

Many years ago most machine tools were belt-driven by a series of overhead pulleys connected to a central power source. These pulleys were cylindrical, with a slight crown to keep the belts centered. Applying this principle directly to the liquid bearing problem, a unit was produced whose maximum liquid pressure was at the center. This was done by utilizing the parabolic principle previously discovered with the narrow bearings (Figure 2-7). By selecting a tube of such size that its internal cross-sectional area was equal to the annular area between its outside diameter and the inside diameter of the main bearing tube, the objective was accomplished. When the smaller tube was mounted on a solid bulkhead located midway on the slot and water was injected in one end, fluid flowing in the annular area formed a parabola toward the center, while fluid flowing through the inner pipe was forced to reverse its direction at the downstream end, forming a second parabola which met the first one at the center. The combined flow created a high-pressure area at the center of the slot and its waterfall pattern was extremely symmetrical.

The bearing was a failure. Any movement of the film off-center in either direction was immediately magnified and caused an acceleration toward that end of the slot. At this point, the reason became obvious. A liquid bearing has an extremely low coefficient of friction (versus the high coefficient for the center of a crowned pulley wheel); consequently, a vector diagram quickly reveals the unstable nature of such a bearing. To duplicate the self-centering action of a crowned pulley, the pressure profile has to be high at the outer ends and lower in the center portion.

Such a bearing was quickly designed and built (Figures 2-18, 2-19, and Table 2-11). A double plenum and split center-feed obtained the desired pattern. A small pressure tap was added (after the film-support tests were completed) to the throat of the center-feed inlet pipe to determine the pressure drop in a large loop of rubber hose that was used to feed the bearing. These readings are included as "actuals". Certain imperfections in the execution of the bearing, resulting from the necessary haste with which it was assembled, prevented top performance, but the principle was confirmed. Since center feed offers numerous design, construction, assembly, operating and maintenance problems in a typical processor, the next logical step was to duplicate the pressure profile with a single end-feed bearing.

This was accomplished by tapering the slot from the center outward. From a 0.010-inch width on the center line, the slot was expanded to a maximum of 0.040 inch at the outer ends in the 4-3/4-inch distance. This was done by a series of step cuts on a logarithmic progression rather than a continuous taper because the shop could not machine a continuous taper at the time (Figure 2-20). Because of the peculiar construction of the bearing, it did not prove feasible to measure its pressure profile, but its performance was most encouraging. When film was manually moved as far as 5/8 inch off center and then released, it would immediately recenter itself and remain stable. At this point, all of the experimental findings could be incorporated into one design.

2.2.4 Methacrylate End-Feed Self-Centering Liquid Bearing:

The bearing was built of clear methyl methacrylate tubing, cast and polished, having an outside diameter of 2-1/2 inches and an inside diameter of 2 inches (Figure 2-21). The outside of the tubing was tapered 1/8 inch in 5 inches toward the center over the area on which the film rides. The tubing was then split in half lengthwise and three slots were milled in one of the halves. The top slot was tapered from 0.010 inch (at the center) to 0.040 inch in 4-3/4 inches. The two side slots were each 0.020 inch, untapered. As illustrated, one was tipped to lead the film and the other to follow it slightly (rather than being machined on a radius line) to prevent cushion collapse under load. After milling, the inside edges of each slot were faired to reduce turbulence and coefficients of friction.

Since two of the slots were near the edge, and the plastic is inherently quite weak, the whole half-bearing was supported in a plaster of paris matrix for the machining operation. After machining, the bearing was assembled with a internal feed pipe in the same manner as previous metal models.

When first tested with the film supporting a total dry weight of 519.2 grams, a 1/4-inch cushion with an inlet pressure of 0.5 psi and a flow of 13 gpm was attained. However, the film tracking was not stable and tended to drift off on either side of the slots. Since the prototypes had been successful, a pressure traverse of the three slots was made. This confirmed the hypothesis that the side slots (combined area of 0.380 square inches) were overriding the beneficial self-aligning effect of the main slot (area of 0.190 square inches).

Four knife blades were constructed of 0.016 inch stainless steel shim stock and affixed to the periphery of the upper portion of the bearing so as to taper the side slots from a maximum of 0.020 inch at the outer ends to a minimum of 0.005 inch at the center. When retested with a continuous film loop supporting a spool (dry weight 354.5 grams), the bearing was self-centering. At an inlet pressure of 0.6 psi and a flow of 17 gpm, the cushion was fairly stable but there was some hunting as the film tracked. This may have been due to slight fluctuations in the line pressure introduced by the centrifugal pump. However, when pulled as far off center as 1-1/2 inches, the film would immediately move back to the center line.

The original 354.5-gram spool used had a 1.30-inch diameter spindle. This relatively narrow width allowed the adjacent sides of the film to cohere, somewhat, because of the Bernoulli effect introduced by the fast flow from the side slots in the confined area within the film loop. When this effect was minimized by the introduction of a heavier spool (four pounds dry weight) of larger spindle diameter (3.50 inches), a truer measurement could be made. The new bearing, at an inlet pressure of 0.4 psi and a flow of 10.5 gpm duplicated the lifting capacity (3.16-inch stable cushion) of its forerunner, the standard HTA-5 bearing, at a pressure of 1.5 psi and a flow of 26 gpm.

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When these figures are translated into horsepower, the requirement of the new design is approximately 1/10th that of the old. With the heavier load, however, the self-centering tendency was not as pronounced unless the film was spun over the bearing. It can be surmised, then, that self-centering would occur in a machine if the film were in motion. (The same is true of a crown pulley driving a flat belt.) In the case of the liquid bearing, however, it should be carefully checked out by further testing.

The prototype amply confirms the supposition that a liquid bearing can be designed so that it is self-centering without side flanges. However, extensive research, testing, and analysis still remains to be done before an optimal design can be created.

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TABLE 2-1
FLOWMETER CALIBRATION DATA AND REYNOLDS NUMBER CALICULATION

Flowmeter Reading gpm	Measured Flow gpm	T_2 °F	ρ gm/ml	ft^3/sec	V ft/sec	$1/V^2$ sec^2/ft^2	f	μ lb/ft sec	$1/\mu$ ft. sec./lb.	ρ $lb./ft^3$	R_ϵ
41.2	40.8	71.05	.99720	.0893	10.36	.00932	.00218	.000608	1645	62.25	1.11×10^5
39.6	40.8	72.55									
37.7	38.2	72.75	.99718	.0833	9.66	.01072	.00214	.000607	1649	62.25	1.04×10^5
35.8	38.2	72.85									
33.9	33.8	72.95	.99716	.0724	8.40	.01417	.00215	.000606	1651	62.25	9.05×10^4
32.1	31.2	72.45									
30.2	29.6	72.55									
28.3	26.9	72.65	.99716	.0618	7.17	.01945	.00228	.000605	1652	62.25	7.73×10^4
26.3	26.3	72.70									
24.4	23.7	72.85									
22.5	21.5	72.90	.99714	.0509	5.90	.02873	.00349	.000605	1654	62.25	6.37×10^4
20.6	19.7	73.00									
18.7	18.2	73.10	.99713	.0400	4.64	.04645	.00259	.000604	1655	62.25	5.01×10^4
16.7	15.8	73.20									
14.7	13.9	73.30	.99712	.0291	3.38	.08754	.00315	.000603	1657	62.25	3.65×10^4
12.8	11.1	73.45									
10.9	10.3	73.50									
9.0	7.9	73.55	.99710	.0182	2.11	.2246	.00332	.000603	1660	62.25	2.29×10^4
6.9	7.2	73.65									
4.9	4.5	73.75									
-	2.6	74.05	.99706	.0071	.823	1.476	.00417	.000600	1667	62.25	8.95×10^3

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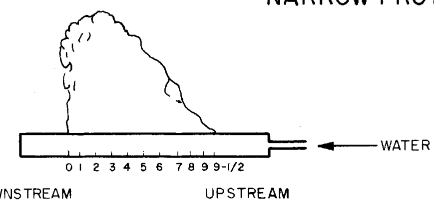
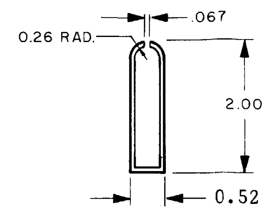
TABLE 2-2
PRESSURE PROFILE MEASUREMENT DATA FOR NARROW PROTOTYPE LIQUID BEARINGS



.062" WALL STAINLESS TUBE
(SIDES FLAT - TOP SEMICIRCULAR)
SLOT WIDTH = .067"



P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	12.5	19.90	1.89	1.91	1.79	1.99	1.84	1.99	1.87	1.93	2.03	1.99	1.43	.78	.46	.40	.44	.38	.38	.42	.48	.48
2.0	17.0	20.00	3.40	3.44	3.41	3.48	3.48	3.49	3.39	3.49	3.76	3.83	2.93	1.70	.64	.44	.44	.43	.41	.57	.64	.64
3.0	20.5	20.20	4.74	4.60	4.67	4.87	4.84	4.83	4.83	5.05	5.26	5.19	5.35	2.44	1.00	.54	.50	.42	.48	.66	.78	.73
4.0	23.5	17.55	5.91	5.99	6.01	5.88	6.00	6.05	6.01	6.29	6.47	6.79	5.30	2.50	.68	.57	.35	.32	.43	.65	.88	.81
5.0	26.0	18.10	7.07	7.10	6.94	6.86	6.98	7.14	6.80	7.10	7.70	7.73	6.27	2.79	.91	.54	.39	.39	.54	.77	.95	.82
6.0	28.5	18.40	9.04	9.03	9.34	9.08	8.45	9.11	9.15	9.20	10.07	9.88	7.58	3.34	1.14	.72	.64	.51	.61	.77	1.00	.92
7.0	31.0	20.40	10.18	10.13	10.53	10.66	10.76	10.29	10.47	10.92	11.36	10.99	8.09	3.48	1.20	.87	.54	.54	.60	.87	1.05	.92
8.0	32.5	20.50	11.28	11.29	11.23	11.55	11.99	11.47	11.56	12.03	13.22	12.13	8.99	3.94	1.05	1.05	.65	.48	.65	.97	1.23	.98
9.0	34.5	20.70	12.90	12.98	13.03	12.95	12.81	12.38	12.58	13.16	13.43	13.17	9.52	3.85	1.43	1.15	.88	.67	.64	.65	1.27	1.08
10.0	36.3	20.90	14.18	14.58	14.27	14.56	14.70	14.20	14.13	14.71	14.89	15.13	9.79	3.53	.91	1.11	.81	.66	.64	.68	1.34	1.06

TABLE 2-3
WITH CERROBEND AND METHACRYLATE WEDGE IN PLENUM

P_1 psi	Flow gpm*	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
0.5		19.60	.59	.52	.52	.46	.44	.42	.05	0	.05	.27	.32	.40	.41	.46	.51	.39	.25	.15	.10	.10
1.0		19.85	1.28	1.23	1.27	1.04	1.16	1.02	1.05	1.02	.45	1.07	1.24	1.04	1.03	.93	1.37	1.19	.80	.60	.42	.34
1.5		20.65	2.75	2.56	2.45	2.65	1.73	2.21	2.43	2.75	2.70	2.94	2.90	.88	2.62	2.07	2.81	2.97	2.02	1.49	.83	.95
2.0		20.95	3.43	3.55	3.25	3.63	2.35	3.09	3.27	3.75	3.69	3.76	3.87	3.17	3.63	3.15	3.87	3.29	3.02	2.17	1.68	1.66
3.0		21.10	4.81	5.21	5.06	5.54	3.51	4.78	3.81	5.49	5.53	5.46	5.61	5.81	5.34	5.41	5.58	5.52	3.81	2.59	1.89	1.90
4.0		21.20	6.77	6.99	6.39	5.94	4.15	5.99	6.73	6.81	6.79	6.31	6.05	6.70	6.70	6.66	6.97	6.15	5.54	3.82	2.79	2.80

*Chip in Rotameter made these readings suspect

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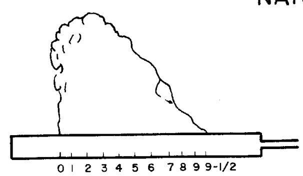
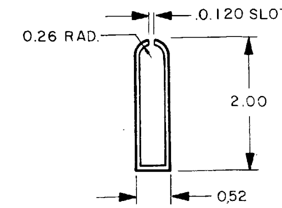
TABLE 2-4
PRESSURE PROFILE MEASUREMENT DATA FOR NARROW PROTOTYPE LIQUID BEARINGS



.062" WALL STAINLESS TUBE
(SIDES FLAT - TOP SEMICIRCULAR)
SLOT WIDTH = .120"



P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	19.0	21.40	2.01	1.98	2.02	1.97	2.02	1.95	1.97	1.93	1.99	1.96	1.91	.99	.43	.23	.20	.29	.27	.27	.30	.30
2.0	25.0	21.85	3.51	3.11	3.29	3.71	3.60	3.62	3.55	3.19	3.41	3.80	3.64	2.54	.78	.32	.28	.28	.29	.32	.33	.31
5.0	41.0	21.90	8.51	8.01	7.61	7.88	8.01	7.69	7.50	7.48	7.78	7.85	8.32	4.47	1.36	.51	.52	.33	.27	.26	.23	.27
6.0	46.5	22.05	10.80	9.97	10.20	10.25	10.08	9.98	9.83	9.62	9.89	9.98	9.76	2.65	.75	.53	.42	.33	.27	.25	.26	.30

TABLE 2-5

.065" WALL STAINLESS TUBE
(WELDED AND DRAWN)
INSIDE DIAMETER = 1.000"
SLOT WIDTH = .063"

P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	18.0	21.10	2.37	2.43	2.47	2.46	2.46	2.46	2.52	2.37	2.39	2.38	2.39	2.44	2.24	2.14	2.14	2.10	1.97	1.89	1.88	.49
2.0	24.9	20.92	4.54	4.54	4.53	4.53	4.62	4.62	4.58	4.47	4.38	4.31	4.35	4.19	4.17	4.20	3.74	3.88	3.95	3.78	3.73	3.54
3.0	30.8	21.60	6.96	6.93	6.56	6.97	6.88	6.81	6.71	6.89	6.92	6.60	6.73	6.73	6.56	6.59	6.22	6.21	5.66	5.51	6.18	6.10
4.0	34.0	22.00	8.39	8.45	8.39	8.74	8.76	8.73	8.70	8.58	8.70	8.15	8.17	8.12	8.15	7.36	7.31	7.73	7.02	7.54	7.26	6.04
5.0	37.7	22.40	10.85	10.74	10.72	10.64	10.84	10.80	10.68	10.59	10.37	10.14	10.47	9.33	9.54	9.61	9.61	8.76	8.53	8.26	8.81	7.76
6.0	41.5	22.80	12.76	12.69	12.78	12.78	12.88	12.78	12.41	12.51	12.67	12.73	12.67	11.88	11.01	11.15	11.15	10.88	10.96	10.41	10.21	10.08
6.75	45.8	22.40	15.00	14.75	14.90	15.17	15.39	15.17	15.04	15.06	14.90	14.65	14.07	13.90	13.51	12.75	12.75	12.72	12.05	11.95	10.11	10.53

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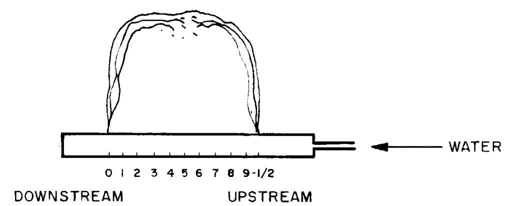
TABLE 2-6
PRESSURE PROFILE MEASUREMENT DATA FOR CIRCULAR CROSS-SECTION LIQUID BEARING



.065" WALL STAINLESS TUBE
(WELDED AND DRAWN)
INSIDE DIAMETER =1.250"
SLOT WIDTH = .063"

P psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	18.8	22.60	2.40	2.41	2.39	2.40	2.40	2.39	2.38	2.36	2.41	2.31	2.30	2.26	2.23	2.25	2.17	2.14	2.10	2.12	2.23	2.05
2.0	26.1	22.80	4.98	4.96	4.37	4.58	4.54	4.44	4.41	4.42	4.45	4.48	4.45	4.31	4.29	4.27	4.24	4.25	4.25	4.06	4.11	3.73
3.0	32.1	20.90	6.54	6.50	6.58	6.66	6.69	6.71	6.73	6.61	6.52	6.53	6.50	6.36	6.30	6.49	6.40	5.98	5.86	5.74	5.53	5.47
4.0	35.3	21.10	8.42	8.29	8.41	8.48	8.42	8.42	8.31	8.23	8.29	8.19	8.11	8.14	7.89	7.87	7.82	7.50	7.30	7.19	7.06	7.09
5.0	40.1	21.30	10.14	10.21	10.31	10.36	10.23	10.19	10.28	10.19	10.04	9.80	9.95	9.97	9.76	9.64	9.61	8.88	8.20	8.19	9.02	8.58
6.0	45.0	21.70	12.39	12.38	12.15	12.35	12.43	12.40	12.19	12.15	12.20	12.28	12.17	12.18	11.94	11.65	11.41	11.14	11.67	11.27	11.72	10.91

TABLE 2-7

.065" WALL STAINLESS TUBE
(WELDED AND DRAWN)
INSIDE DIAMETER = 1.500"
SLOT WIDTH= .063"

P psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	18.7	21.10	1.61	2.09	2.33	2.31	2.22	2.30	2.30	2.36	2.19	2.23	2.19	2.11	2.33	2.34	2.22	2.13	2.18	2.01	1.45	1.39
2.0	25.8	21.50	2.70	3.63	4.10	4.12	4.13	4.11	3.96	4.10	3.89	3.63	3.97	3.98	3.87	3.87	3.94	3.79	3.74	3.72	3.57	2.69
3.0	31.8	21.80	3.74	4.52	5.24	5.93	5.29	6.23	6.04	6.19	6.22	6.21	6.25	6.24	6.23	6.26	5.73	5.37	5.65	5.70	5.54	3.01
4.0	35.5	20.90	5.06	6.96	7.19	7.44	7.24	7.58	7.56	7.34	7.50	7.38	7.56	7.14	7.10	7.48	6.91	6.76	7.00	7.06	4.27	4.30
5.0	39.3	22.00	5.69	6.50	9.18	9.32	9.22	9.21	9.51	9.51	9.45	9.40	9.47	9.47	9.52	9.62	7.68	8.45	8.48	7.70	6.54	5.41
6.0	44.0	22.20	7.43	8.71	11.25	11.25	11.33	11.13	11.65	11.57	11.61	11.56	11.46	11.29	11.42	11.55	11.58	10.46	9.65	9.78	9.50	6.96

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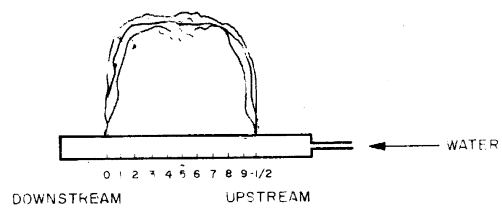
TABLE 2-8
PRESSURE PROFILE MEASUREMENT DATA FOR CIRCULAR CROSS-SECTION LIQUID BEARING



.065" WALL STAINLESS TUBE
(WELDED AND DRAWN)
INSIDE DIAMETER = 1.620"
SLOT WIDTH = .063"

P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	18.2	23.60	2.18	2.22	2.24	2.24	2.27	2.25	2.26	2.25	2.22	2.28	2.36	2.34	2.28	2.27	2.25	2.24	2.25	2.18	2.19	2.17
2.0	24.4	23.90	3.70	3.77	3.88	3.92	3.99	4.00	3.99	4.02	4.03	3.98	4.00	4.03	3.99	3.97	4.01	3.98	3.81	3.57	3.50	3.43
3.0	30.1	24.20	5.85	5.69	5.99	6.04	6.01	5.90	5.91	6.02	5.53	5.44	5.78	5.99	5.86	5.92	5.90	5.40	5.44	5.54	5.50	4.50
4.0	34.9	24.40	7.15	7.33	7.84	7.60	7.63	7.05	7.83	8.00	7.96	7.79	7.87	7.60	7.53	7.04	7.70	7.59	7.41	6.35	6.57	6.14
5.0	38.5	24.65	7.93	8.21	8.36	9.02	9.32	9.45	9.53	9.61	9.64	9.79	9.68	99.65	8.24	9.38	9.48	8.67	9.24	7.99	7.35	6.74
6.0	42.5	25.05	9.53	9.50	11.04	10.04	10.27	10.45	10.39	10.64	10.62	10.65	10.89	11.06	10.77	10.69	10.45	10.65	10.84	7.69	6.30	6.12

TABLE 2-9

.065" WALL STAINLESS TUBE
(WELDED AND DRAWN)
INSIDE DIAMETER = 1.875"
SLOT WIDTH = .063"

P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	19.6	22.7	2.40	2.35	2.35	2.35	2.43	2.40	2.41	2.41	2.45	2.48	2.46	2.44	2.40	2.41	2.35	2.40	2.40	2.42	2.42	2.43
2.0	26.6	22.9	4.32	4.28	4.34	4.31	4.41	4.45	4.45	4.47	4.46	4.48	4.41	4.42	4.57	4.45	4.41	4.29	4.25	4.25	4.24	4.24
3.0	32.8	23.1	6.36	6.42	6.42	6.42	6.55	6.57	6.56	6.60	6.58	6.54	6.45	6.40	6.35	6.45	6.92	6.92	6.36	6.32	6.49	6.23
4.0	37.4	23.4	8.10	7.96	8.04	7.58	8.43	8.46	8.46	8.41	8.42	8.25	8.39	8.40	8.33	8.30	8.23	8.00	7.83	8.19	8.04	7.62
5.0	41.3	23.2	9.51	9.41	9.61	9.45	9.48	10.07	10.00	9.99	9.96	10.03	10.01	10.00	10.05	10.03	9.96	9.75	9.98	9.96	9.84	9.51
6.0	46.8	23.4	11.62	11.93	11.65	12.17	12.22	12.37	12.41	12.36	12.41	12.39	12.37	12.32	12.35	12.28	12.23	12.01	12.02	11.85	12.08	11.63

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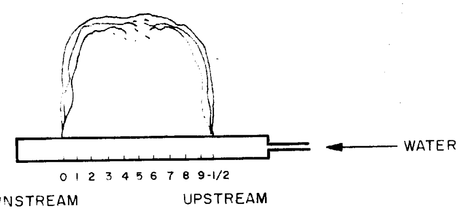
TABLE 2-10
PRESSURE PROFILE MEASUREMENT DATA FOR CIRCULAR CROSS-SECTION LIQUID BEARING



.065" WALL COPPER TUBE
(SEAMLESS)
INSIDE DIAMETER = 1.500" ±.005
SLOT WIDTH = .063"

P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
1.0	15.0	21.35	2.51	2.61	2.65	2.66	2.65	2.52	2.46	2.60	2.47	2.61	2.56	2.38	2.54	2.48	2.56	2.54	2.56	2.51	2.51	1.61
2.0	21.5	21.60	3.41	4.39	4.40	4.39	4.37	4.38	4.41	4.20	4.37	4.13	4.43	4.38	4.40	4.28	4.42	4.21	4.41	4.11	4.11	3.01
3.0	26.5	21.80	5.36	6.26	6.26	6.26	6.14	5.91	6.16	6.24	6.11	5.91	6.12	5.91	6.04	5.81	6.00	5.94	5.88	6.07	5.99	5.41
4.0	30.5	22.05	6.51	8.43	8.54	8.44	8.42	8.39	8.26	8.30	8.11	7.93	7.98	7.93	7.89	7.92	7.70	7.95	8.01	7.78	7.72	3.41
5.0	34.0	22.40	7.98	10.24	9.72	10.12	10.30	10.02	9.99	9.62	9.69	9.80	9.86	9.61	9.66	9.62	9.66	9.66	9.60	9.62	9.44	8.96
6.0	38.0	22.65	9.86	12.15	12.11	12.04	11.97	11.85	11.81	11.92	11.66	11.74	11.69	11.63	11.70	11.61	11.21	11.57	11.17	11.02	11.24	11.13
7.0	43.0	22.90	10.41	14.01	14.01	13.99	13.97	13.83	13.81	13.89	13.87	13.66	13.63	13.67	13.31	13.64	13.55	13.65	13.65	13.30	12.33	12.18

TABLE 2-11

.065" WALL COPPER TUBE
(SEAMLESS, DOUBLE PLENUM)
CENTER FEED
SLOT WIDTH: .063"

P_1 psi	Flow gpm	T_3 °C	0	1/2	1	1-1/2	2	2-1/2	3	3-1/2	4	4-1/2	5	5-1/2	6	6-1/2	7	7-1/2	8	8-1/2	9	9-1/2
0.5	9.0	19.50	1.13	1.15	.91	1.01	1.04	1.07	1.08	1.07	1.05	1.05	1.07	1.08	1.08	1.09	1.10	1.09	1.08	1.11	1.16	1.12
1.0	12.8	19.60	1.67	1.70	1.54	1.45	1.53	1.49	1.54	1.53	1.52	1.50	1.52	1.52	1.51	1.49	1.51	1.47	1.45	1.54	1.68	1.66

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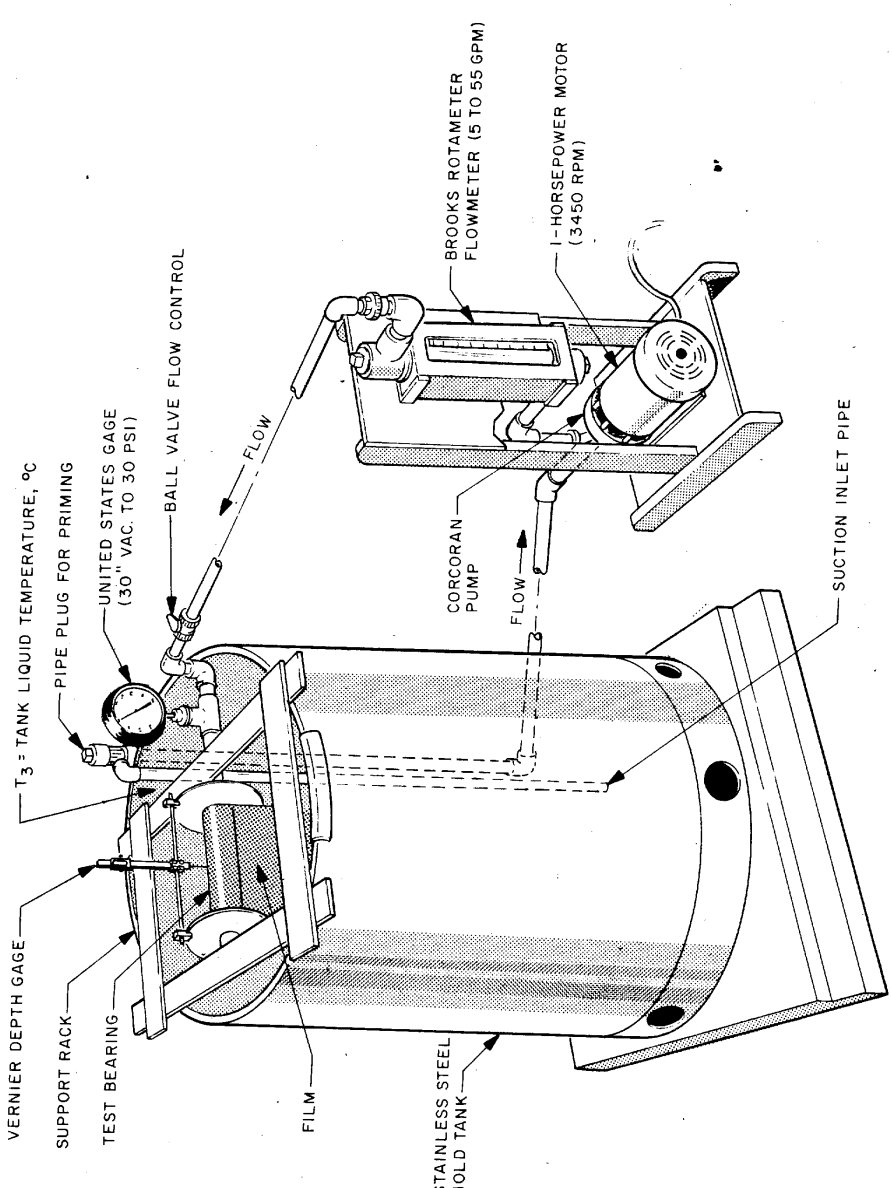

Figure 2-1. Liquid Bearing Test Rack

60073

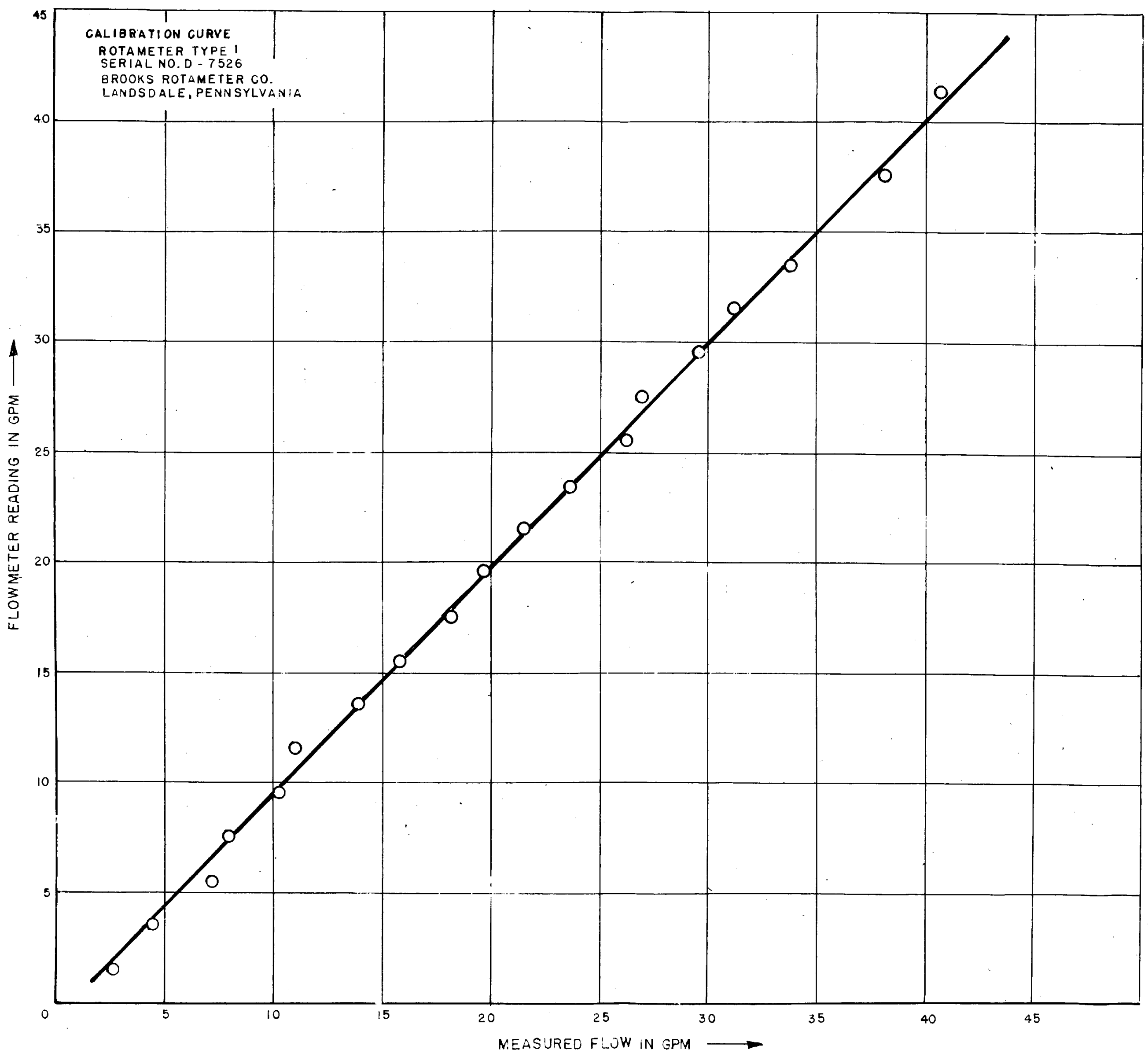

Figure 2-2. Rotameter Calibration Chart

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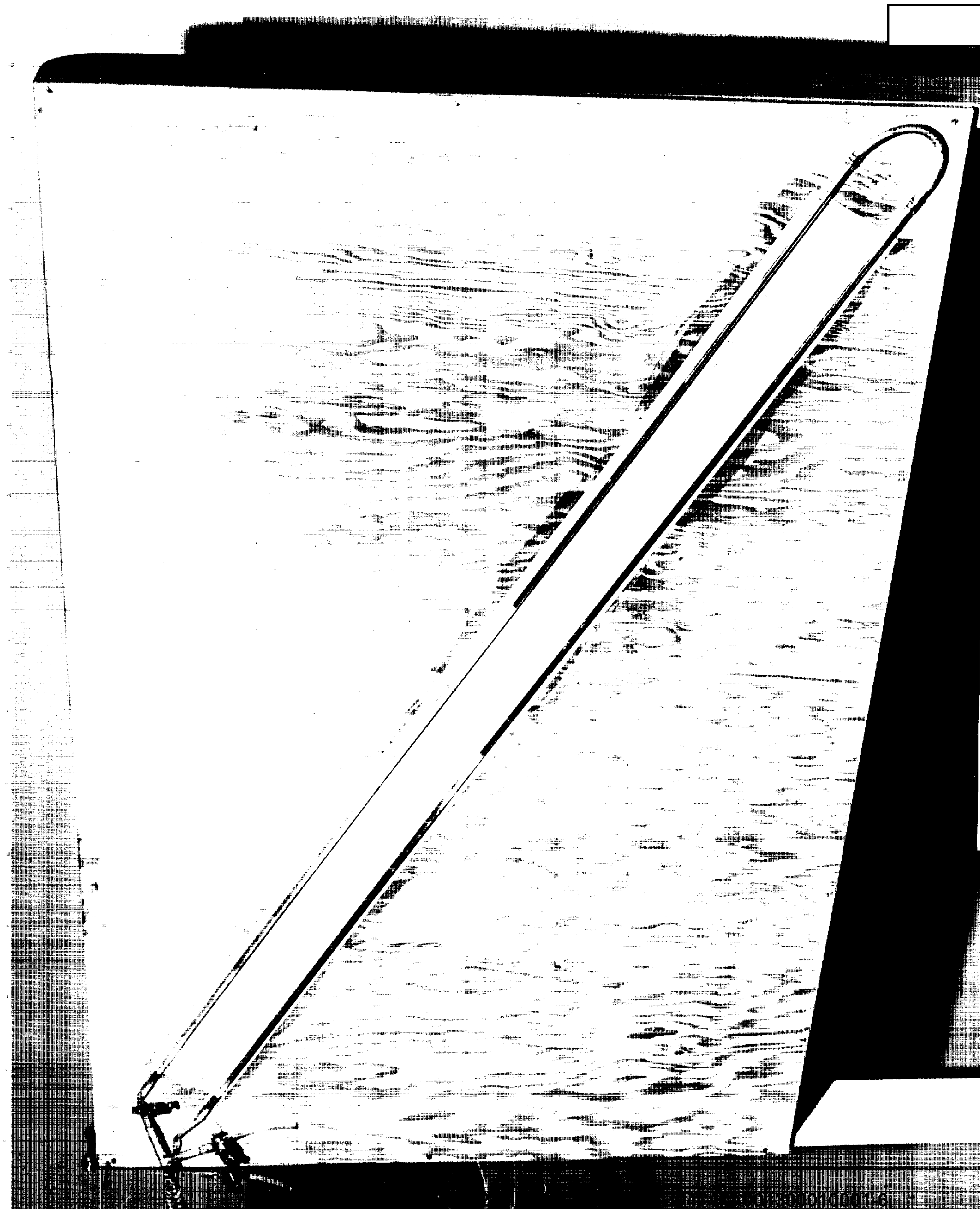

Figure 2-3a. Inclined Mercury Manometer

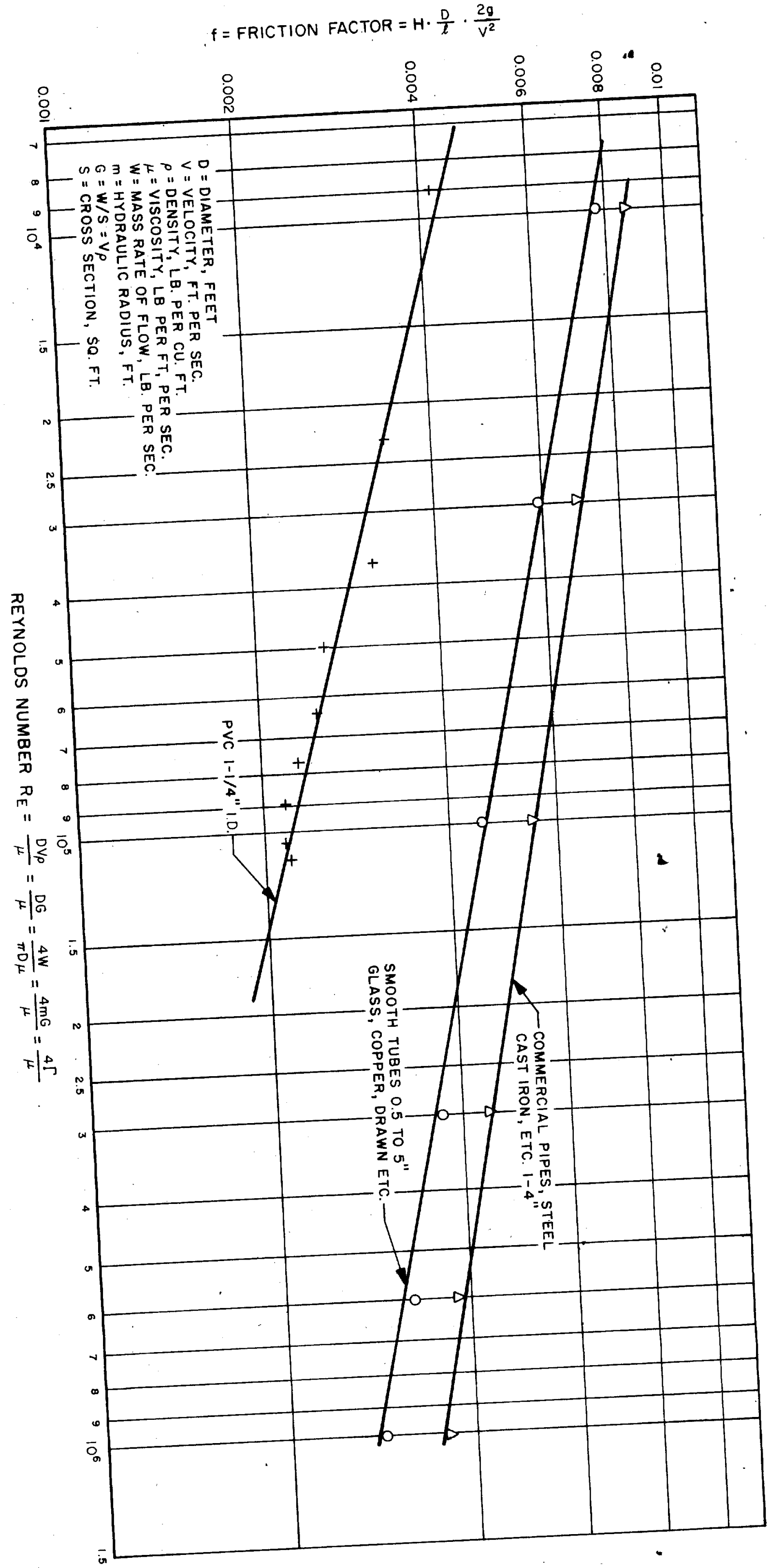

Figure 2-3. Friction Factors For Straight, Clean, Round Pipes

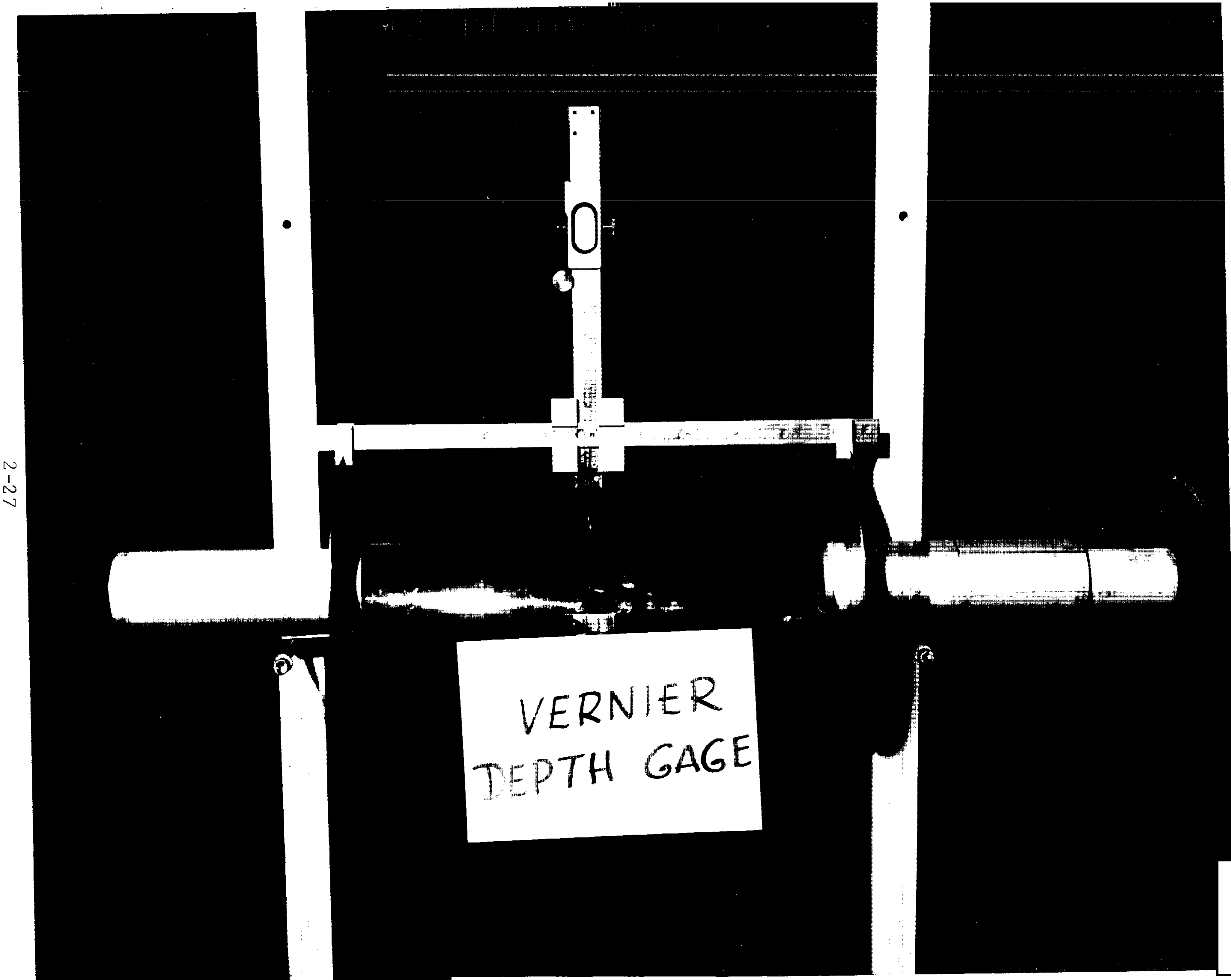

Figure 2-4. Vernier Depth Gage

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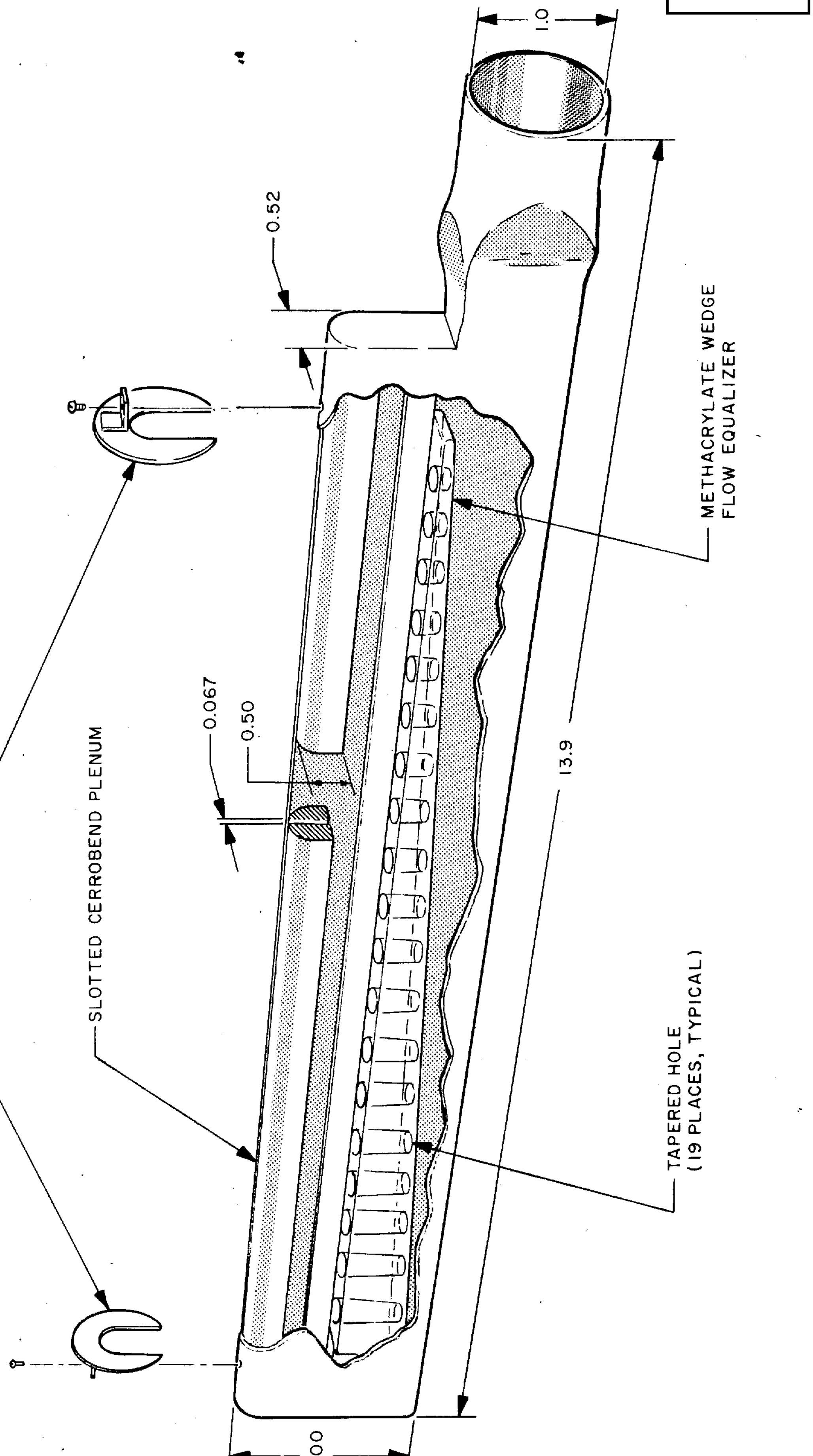

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Figure 2-5. Narrow Prototype Liquid Bearing

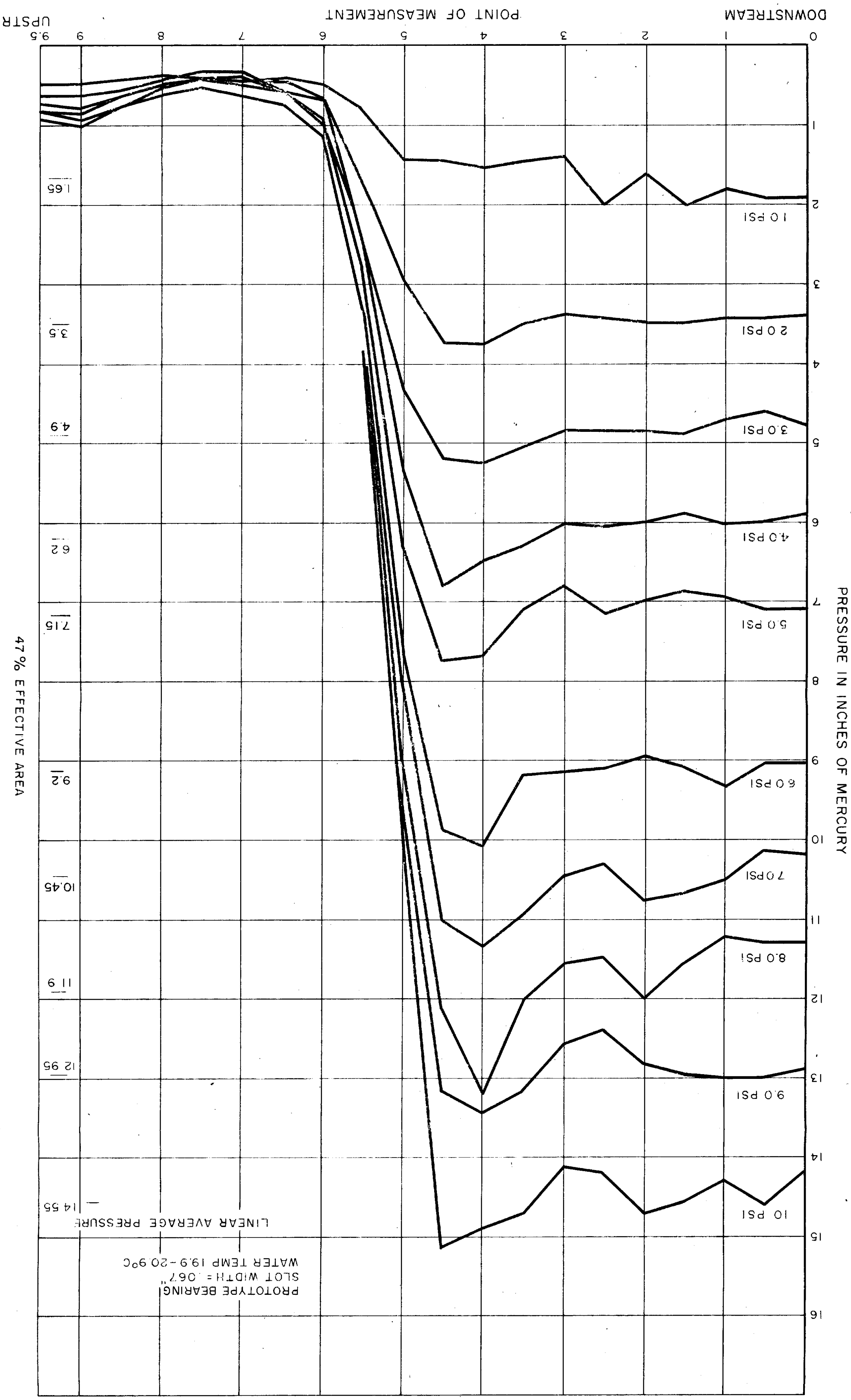

Figure 2-6. Narrow Prototype Liquid Bearing Pressure Profile (.067" Slot)

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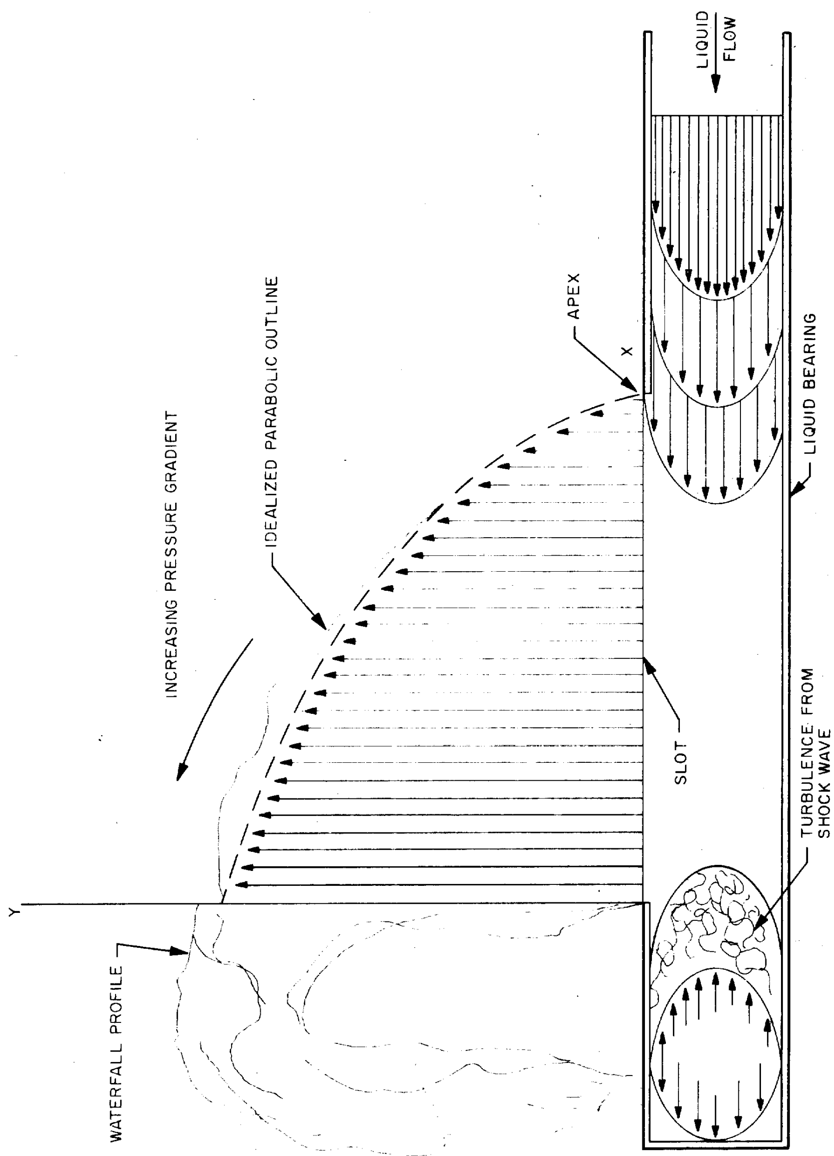

Figure 2-7. Analysis of Pressure-Flow Relationship

50077

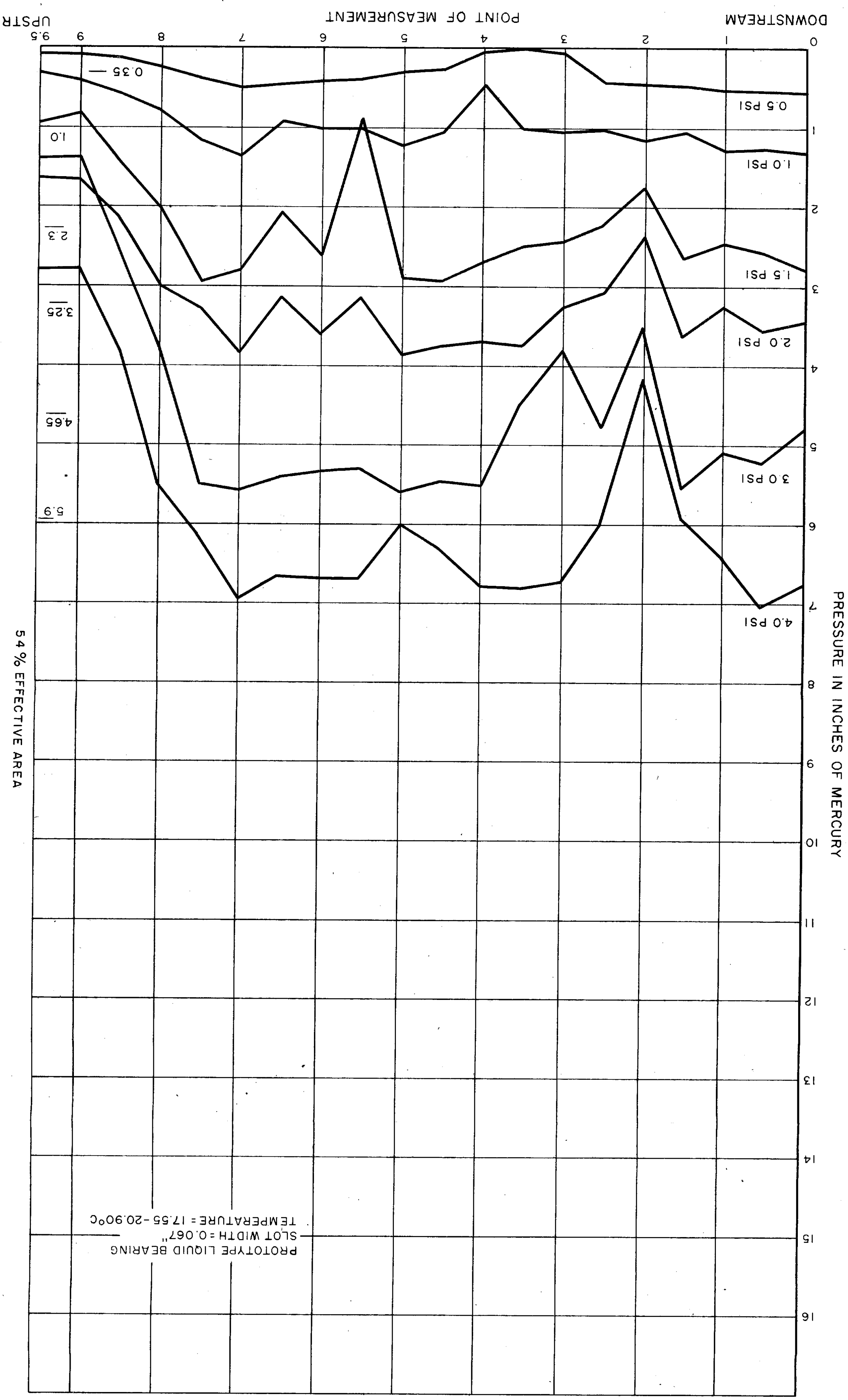

Figure 2-8. Narrow Prototype Liquid Bearing with Cerrobend Plenum and Plastic Wedge

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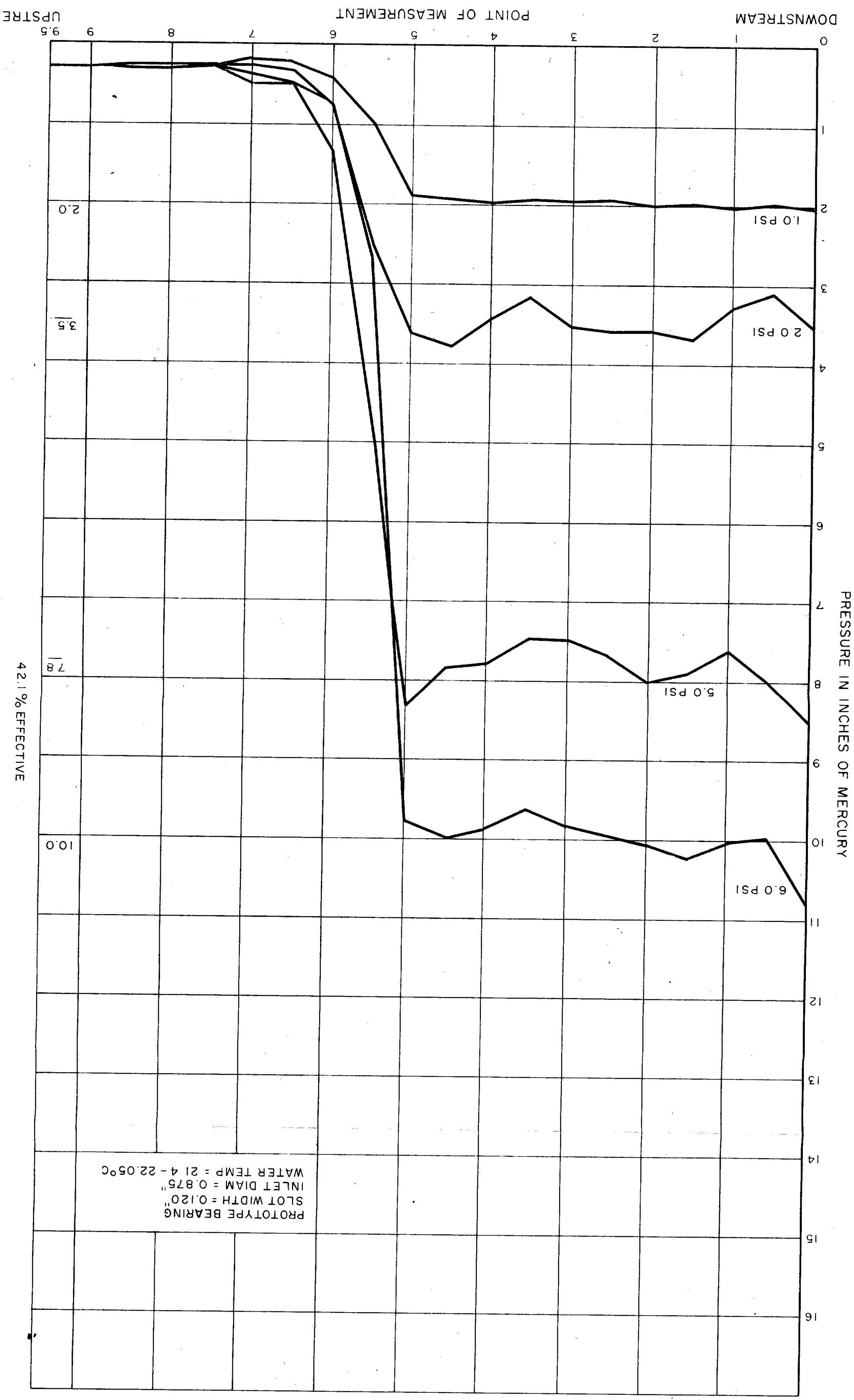

Figure 2-9. Narrow Prototype Liquid Bearing with Wide Slot

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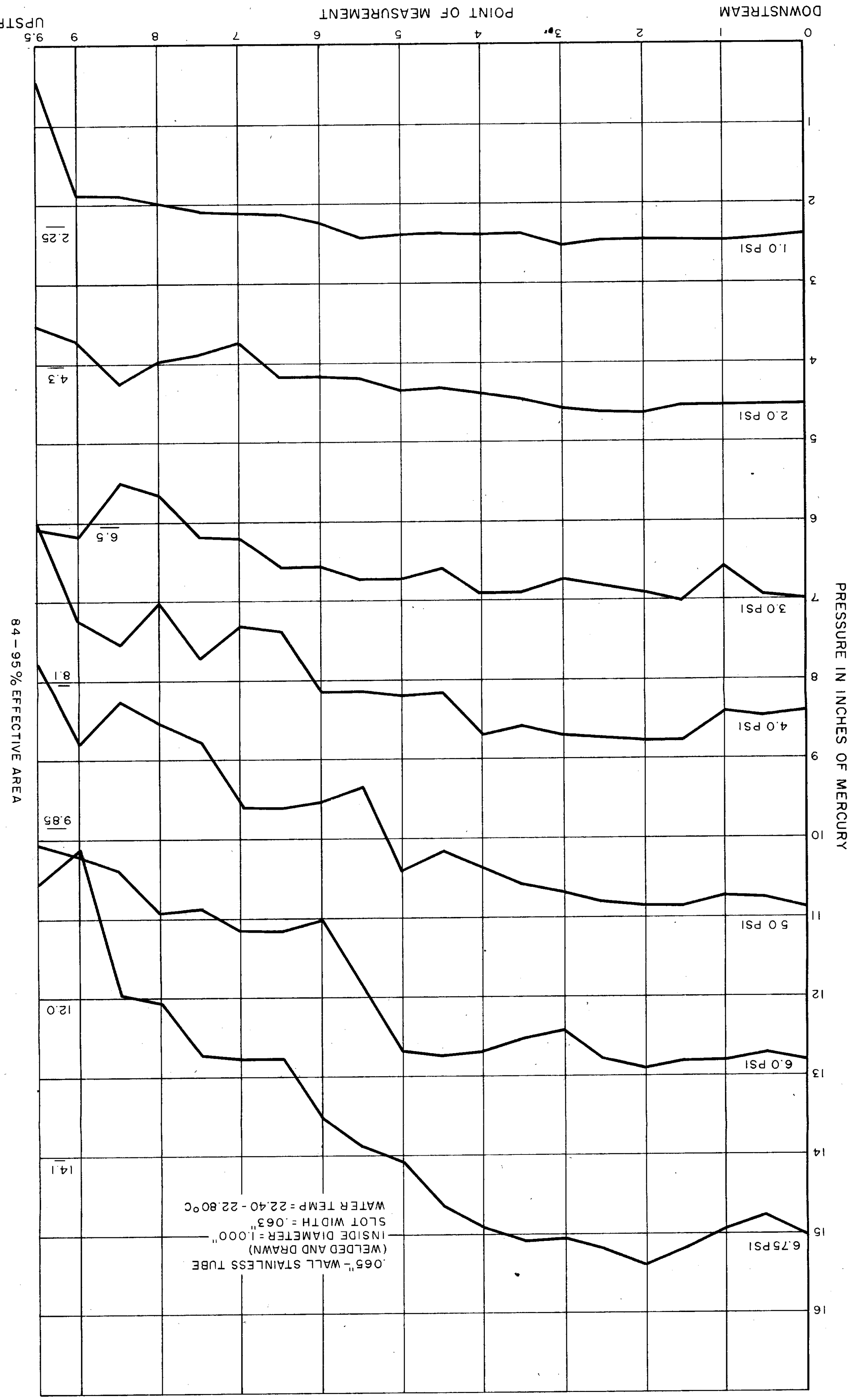

Figure 2-11. Pressure Profile for 1.000" I.D. Circular Stainless Bearing

2-39

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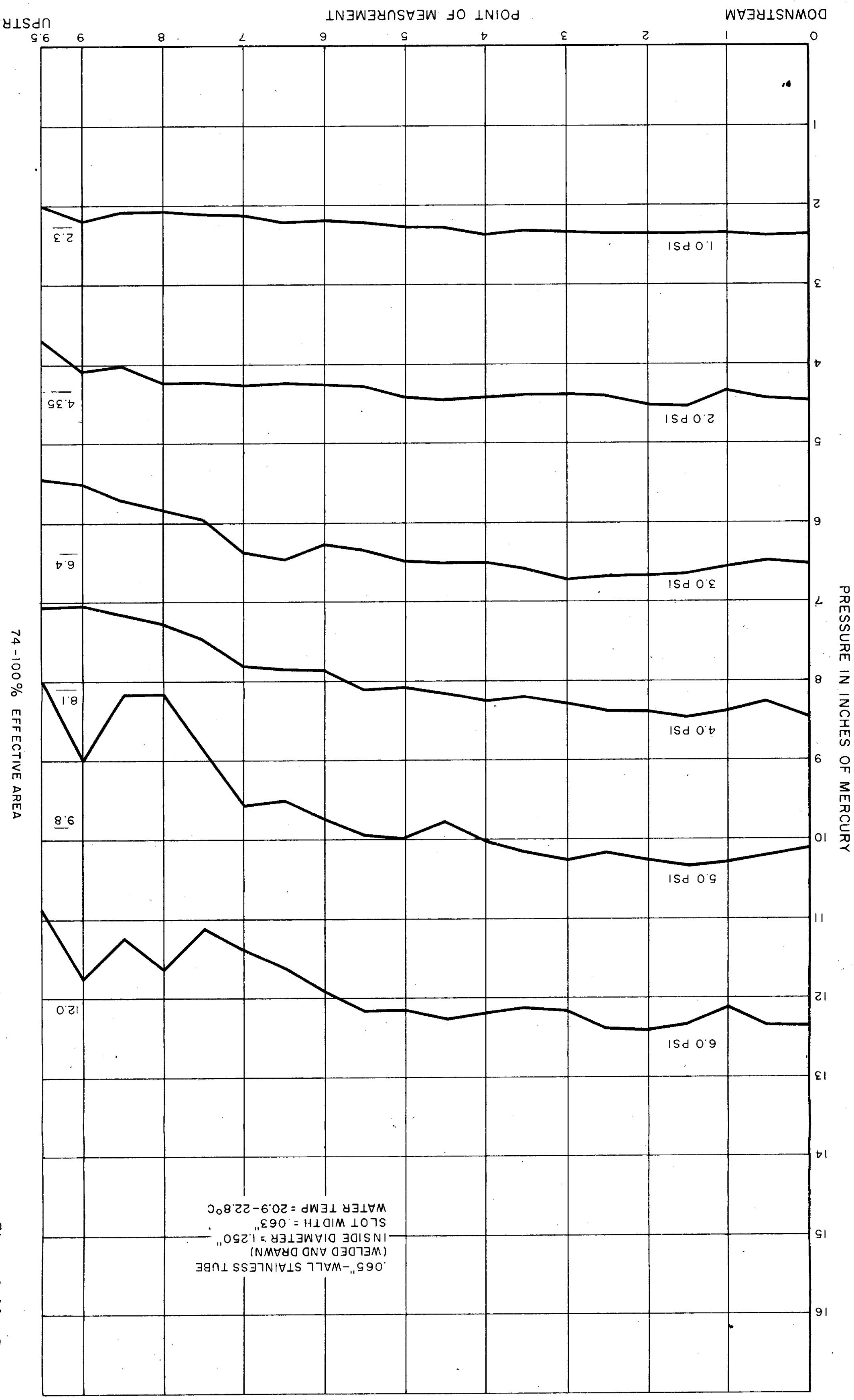

Figure 2-12. Pressure Profile for 1.250" I.D. Circular Stainless Bearing

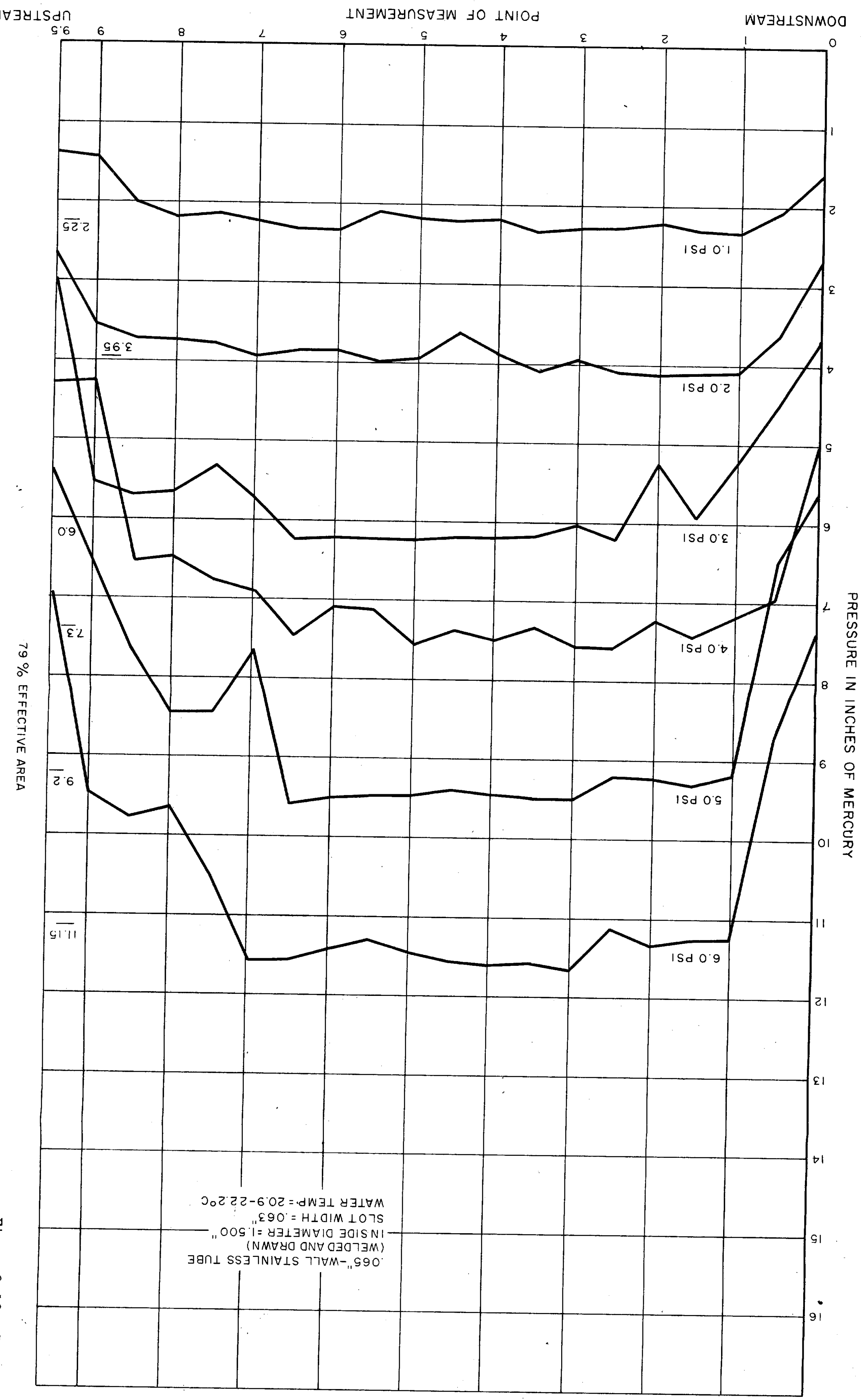

Figure 2-13. Pressure Profile for 1.500" I.D. Circular Stainless Bearing

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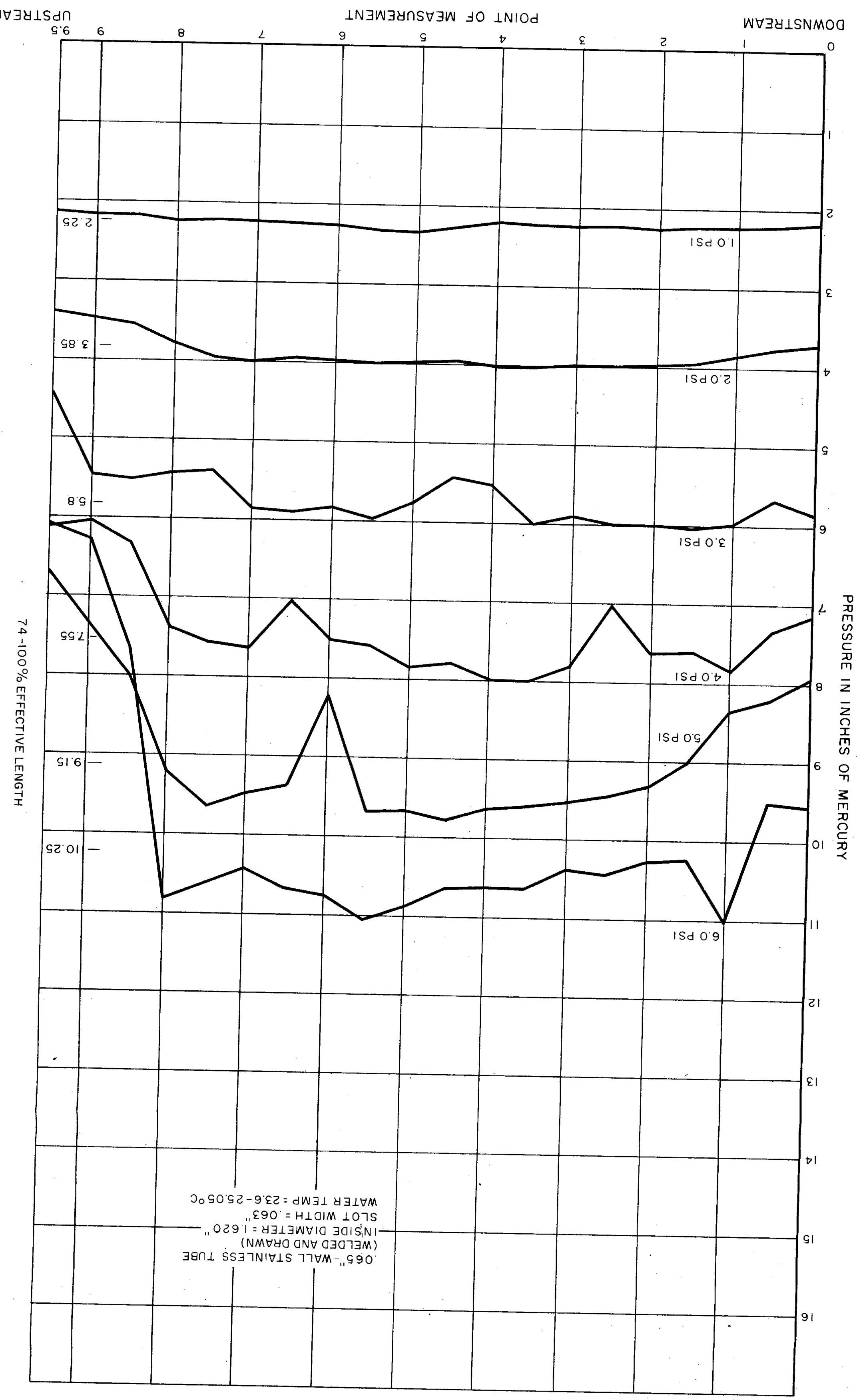

Figure 2-14. Pressure Profile for 1.620" I.D. Circular Stainless Bearing

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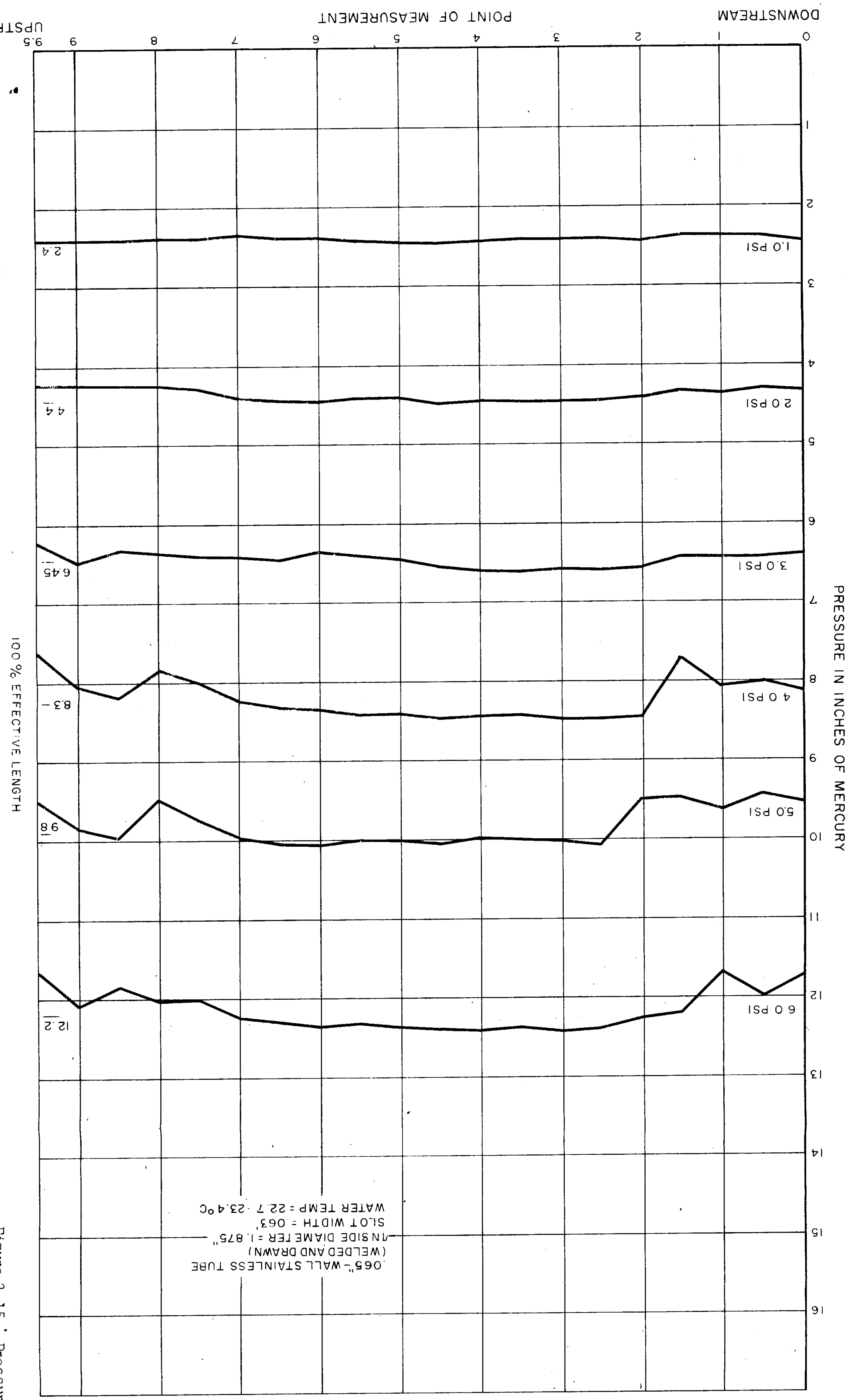

Figure 2-15. Pressure Profile for 1.875" I.D. Circular Stainless Bearing

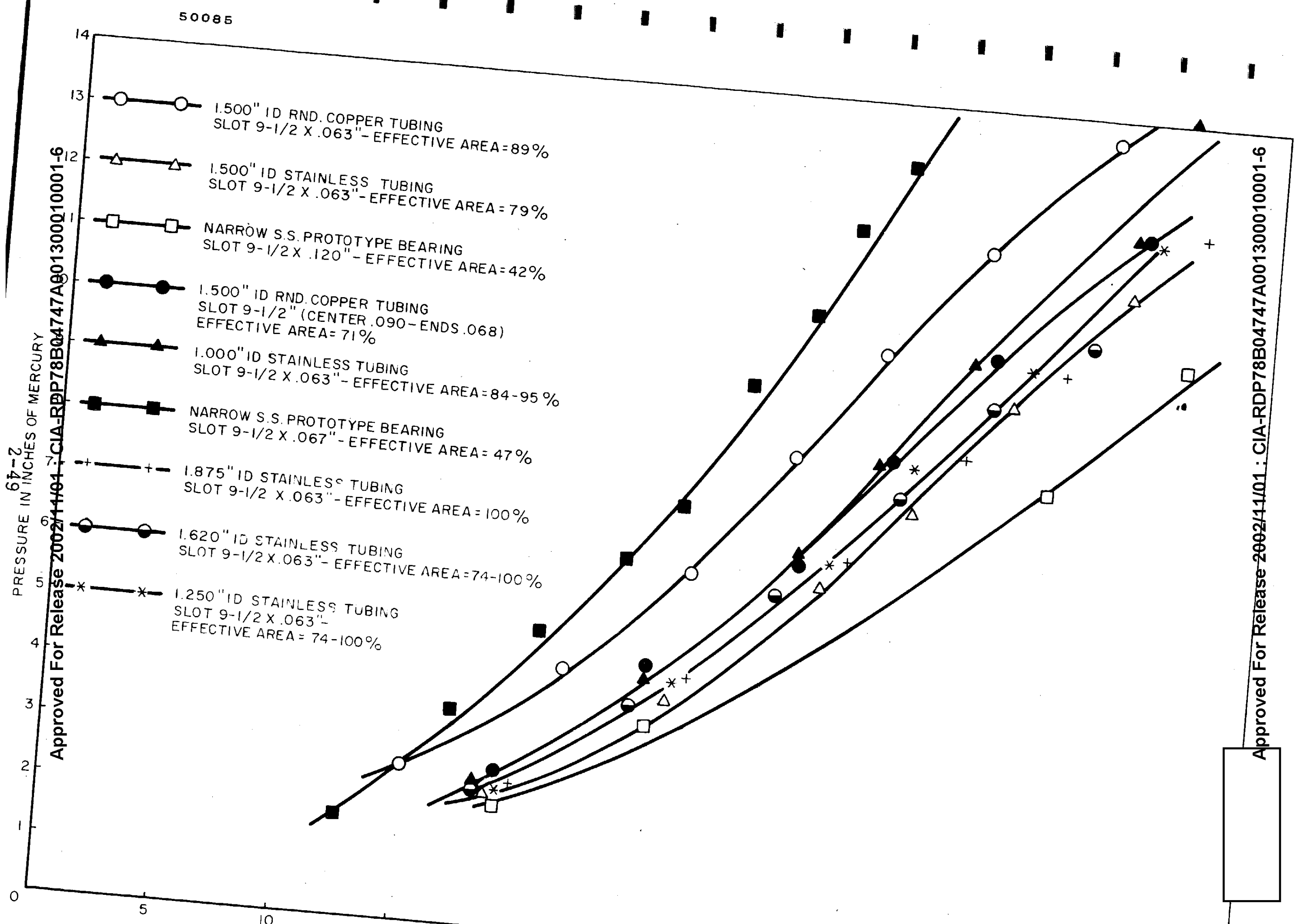

Figure 2-16. Pressure vs. Flow for Experimental Bearings

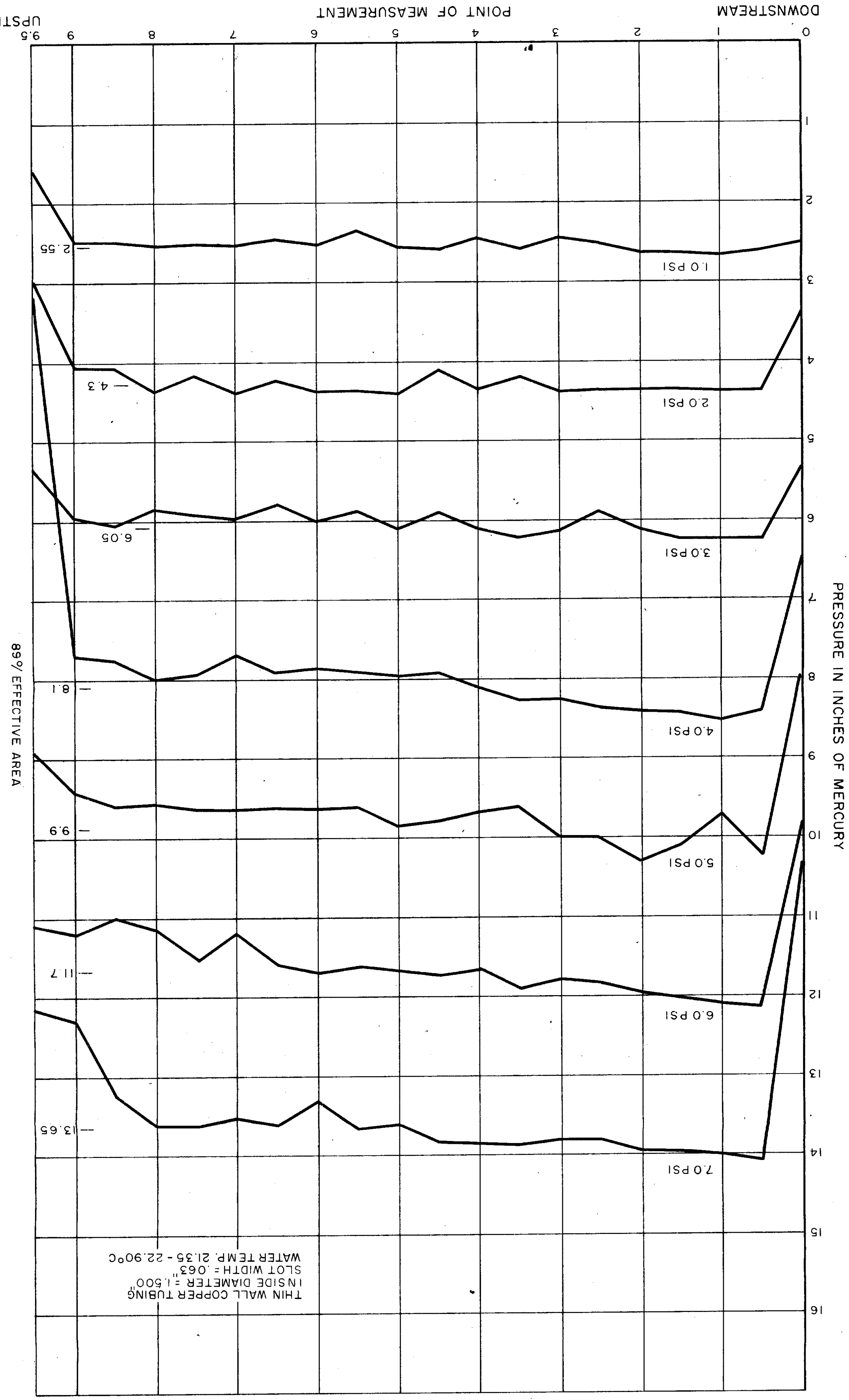

Figure 2-17. Pressure Profile for 1.500" I.D. Circular Copper Bearing

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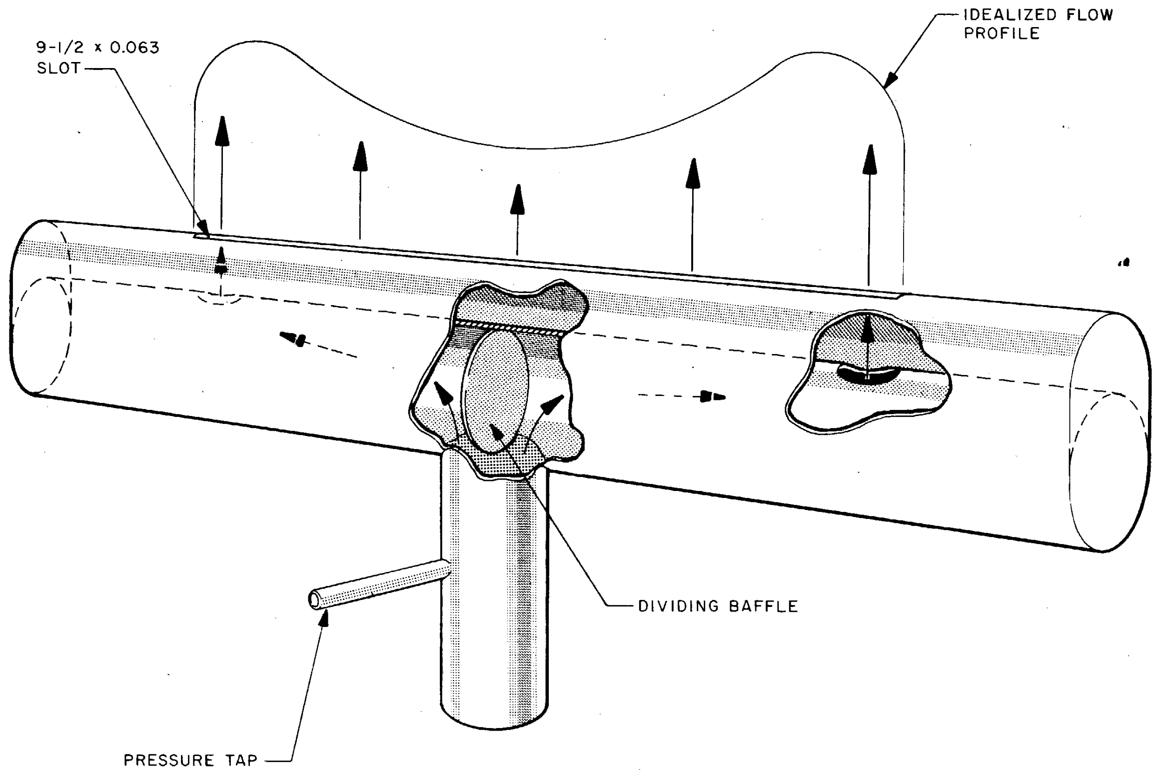

Figure 2-18. Double-Plenum Center-Feed Copper Bearing

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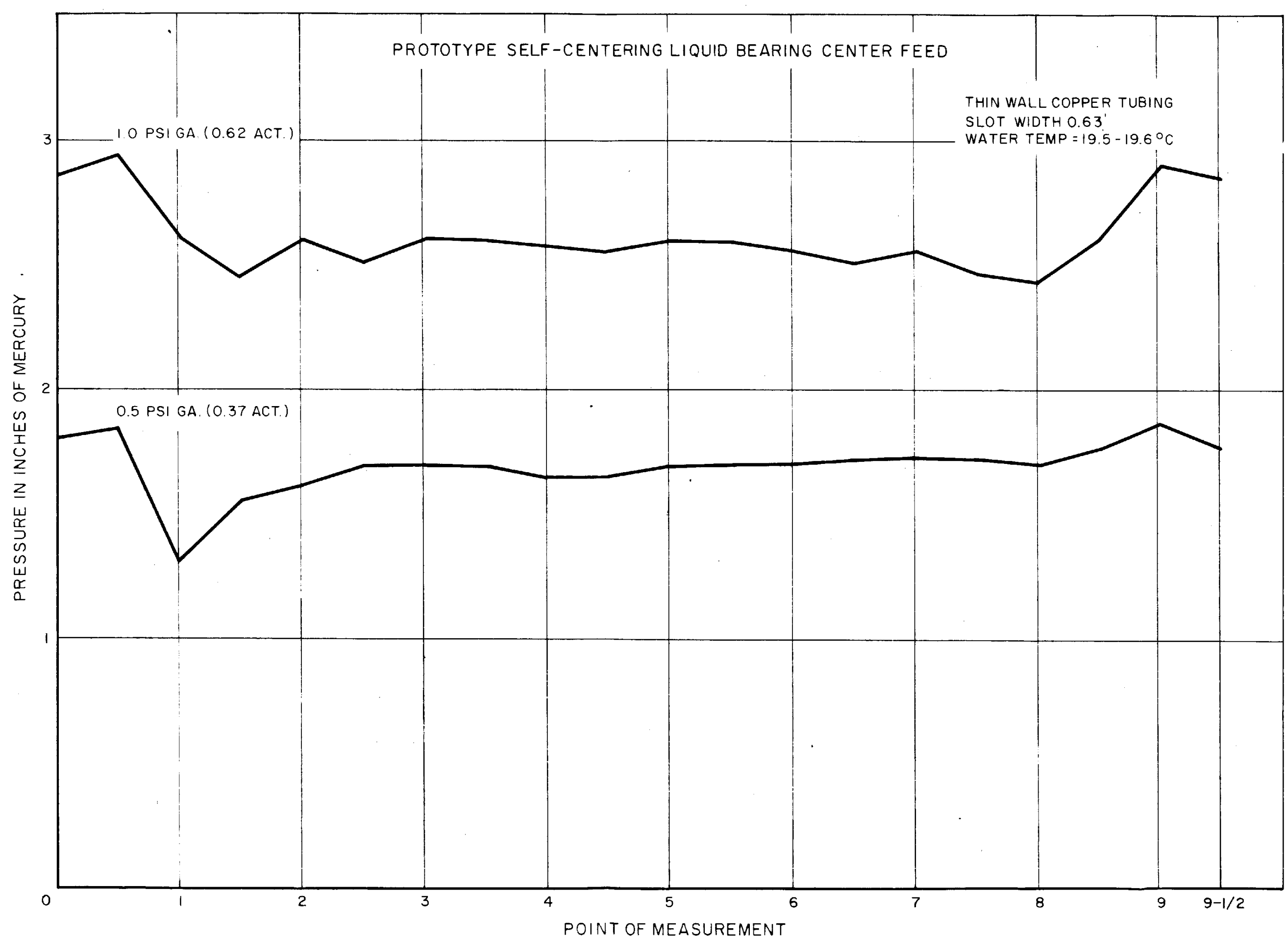

Figure 2-19. Pressure Profile of Center-Feed Copper Bearing

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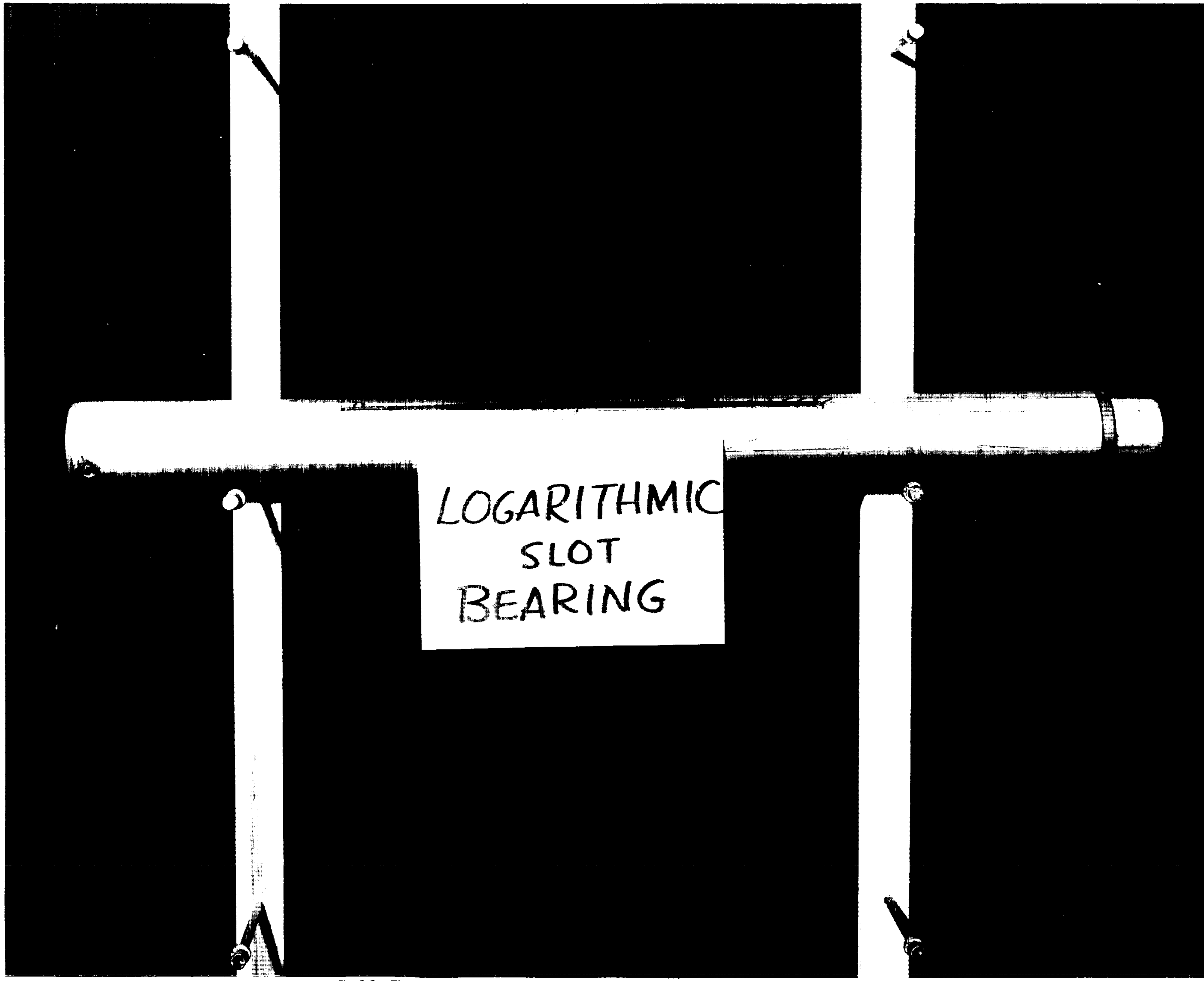

Figure 2-20. Logarithmic Slot Self-Centering Liquid Bearing

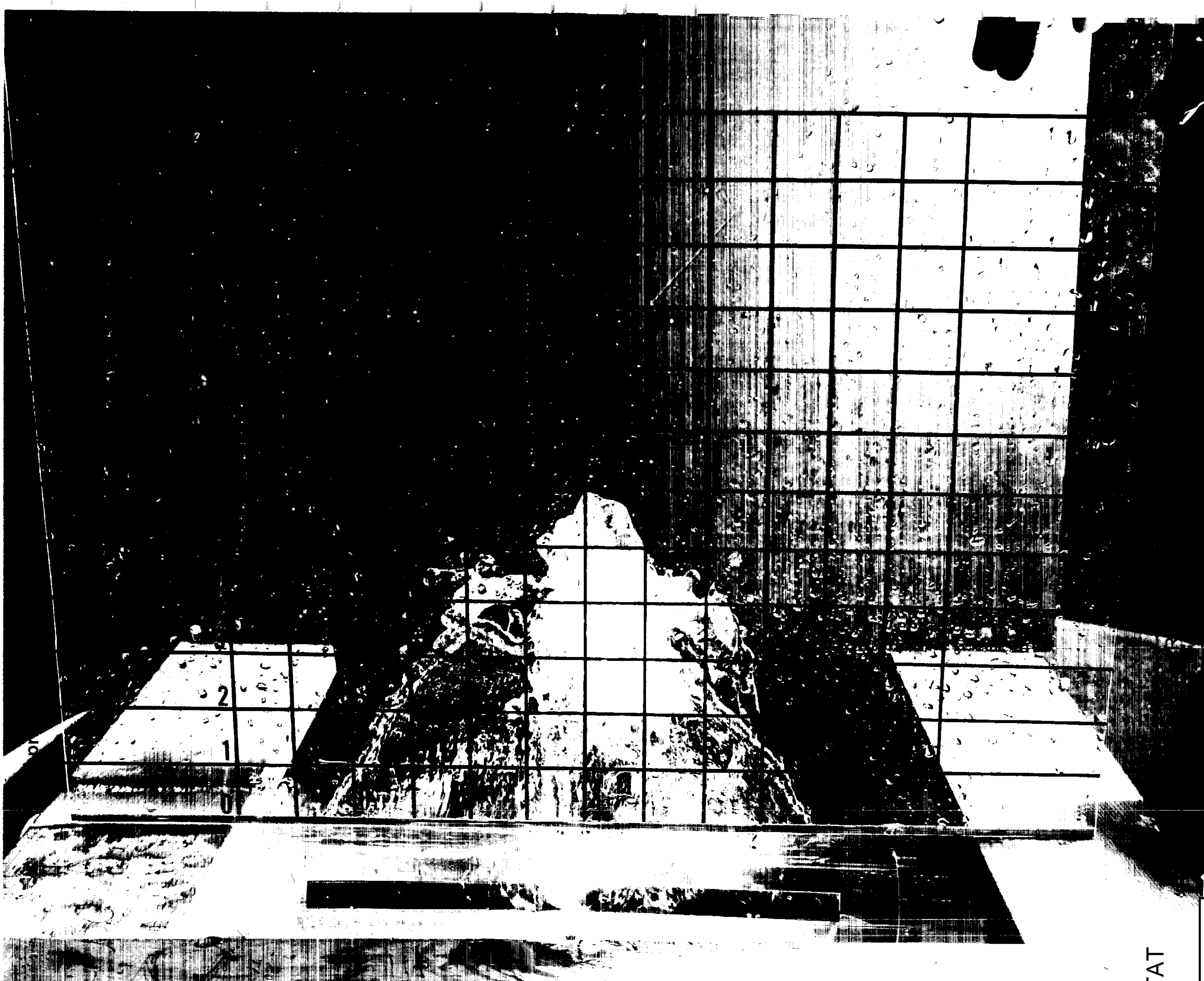
Figure 2-21. Methacrylate Self-Centering Bearing

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SECTION 3

CONCLUSIONS

3.1 NARROW LIQUID BEARINGS

Narrow liquid bearings, while economic from the standpoint of pressure and flow required to lift a specified load, had two serious limitations. The first was the difficulty of producing a desired pressure profile by means of internal wedge shaped equalizers, or other devices. The second was the fact that they would only be practical for extremely thin film and leader where the bending force required to conform to the small radius was negligible compared to the lifting force.

In narrow end-feed bearings in which the internal diameter was in the order of 1/10th the total bearing length, the pressure profile produced was parabolic in shape, with the high pressure point downstream from the feed. From this series of test results, the following hypothesis was formulated:

Liquid bearings having an identical cross-sectional configuration and area will produce a characteristic pressure profile plot whose magnitude at constant inlet pressure is directly proportional to flow.

3.2 END-FEED LIQUID BEARINGS

Single-slot, end-feed liquid bearings without provision for internal equalizing were impractical as film-supporting units for two reasons. In cross-section, the highest point of the cushion was at the top center and, from this point, it fell off rapidly on either side, virtually to zero at the 180-degree line (X-axis). The parabolic pressure pattern described earlier caused the film to tilt and move off the supporting slot. The skewed angle made accurate measurement of the cushion impossible.

When these empirical data were compared on a single graph (Figure 2-16), the significance of the construction material, the machine finish of the slot, the smoothness of the internal surface of the tubing, and the position of the seam relative to the slot became apparent. These respresent the critical performance parameters.

3.3 SELF-CENTERING LIQUID BEARINGS

A new principle of equalizing the flow pattern in a single-slot liquid bearing was developed. A concentric inner tube supported by a center bulkhead produced a symmetrical pattern. When this pattern was modified to a "bow-tie" pressure profile (high at the outer ends of the slot and lower in the center), the film exhibited self-centering tendencies.

All of the experimental findings were incorporated in a single bearing. This methacrylate unit proved to have an outstanding performance capability at modest flow rates and inlet pressures. Its power requirements for equal work loads were approximately 1/10th that of the old standard HTA-5 bearing. Furthermore, at certain load-pressure balances, it tended to advance the film which indicated that a lessening of the drive capstan load would be possible.

SECTION 4

RECOMMENDATIONS

4.1 FURTHER RESEARCH

Because of the advanced report date, there was insufficient time to make a thorough mathematical analysis of the empirical data. This should be done and can be accomplished with minimum additional testing.

On the "idealized" self-centering bearing, only one slot size relationship and only one film width, (9-1/2 inches) was examined. Other slots should be tested, along with various slot configurations. The narrower film widths must be checked to determine the behavior of the bearing and whether or not masking sleeves are needed.

No tests were performed to determine the effect on developing, fixing, etc., of film with this type of bearing. These tests should be carried out.

Since the self-centering action of the bearing under moderate to heavy loading only appeared to occur when the film was moving. This should be carefully verified. It would require several identical bearings to perform the test impartially, since any other type of bearings used on a test rack in conjunction with this one would introduce unknown parameters.

If all facets of the design prove feasible, considerable thought and design time should be devoted to the problems that would be encountered in producing the bearing in quantity. Expensive steel injection dies for plastic would be eliminated because of the small production runs required, but simple flexible molds would not be. Slot tolerances would be a prime factor in cast molding, assembly, and reproducibility.

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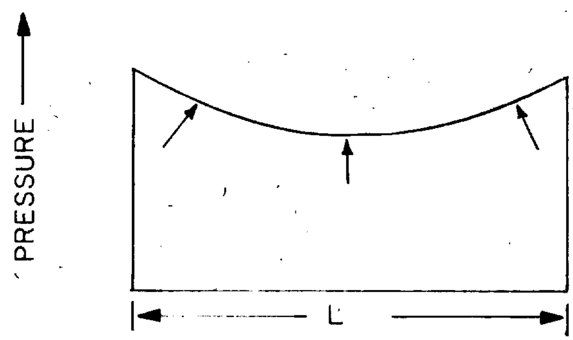
APPENDIX A

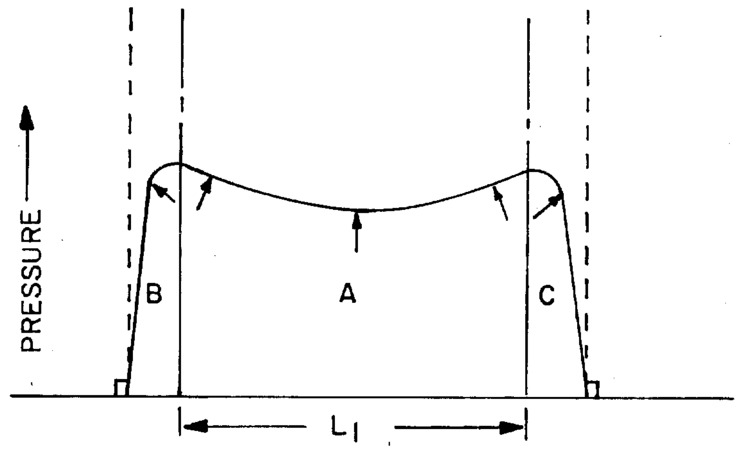
PRESSURE GRADIENT CHARACTERISTICS

To achieve automatic film centering, it is necessary that the pressure gradient of the fluid bearing be shaped so as to create a balanced condition when the film is centered and a restoring force proportional to displacement when the film is offset in either direction from the center position. A hyperbolic gradient such as that shown in Figure A-1 would provide this effect if the length "L" of the fluid bearing is great enough to accommodate all predictable offsets.



The abrupt transition from the hyperbolic characteristics to a zero-pressure area at either end of the length "L" is both impossible to realize in an open physical system and unnecessary in a practical sense. A composite characteristic such as that shown in Figure A-2 can be realized and will provide the necessary restoring force. The gradient characteristic should be considered as a composite of three segments, (designated "A", "B", and "C" in the figure. Section "A" represents an approximately hyperbolic characteristic, and is the area of interest. The length "L_1" of section "A" should accommodate the predictable offsets as described above. Sections "B" and "C" are transition regions that will contribute negative restoring force components if the film excursion extends beyond the useful region. "L_1", therefore, must include a safety margin to avoid such occurrence.

APPENDIX A (Continued)



The specific pressure gradient required for the hyperbolic portion of the gradient characteristic will be influenced by the horizontal and vertical loading forces on the film. These forces are largely a function of the film tension and speed of movement over the fluid bearing.

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STAT

REPORT

DETERMINING THE EFFECT OF HIGH PROCESSING TEMPERATURE AND SHORT PROCESSING TIME COMBINATIONS ON AERIAL FILM

STAT

Temp-Short time.

STATINTL

February 1965

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

ABSTRACT

Image Quality is one of the most important considerations in the field of photographic interpretation. Due to the lack of published data, little is known about this subject as far as the amount of image degradation that occurs when elevating the processing temperature.

The objective of this report is to evaluate the effects that high processing temperatures have on resolution, speed, fog, gamma, and granularity.

The information obtained from this report is essential as design criteria for liquid/air bearing processors. These criteria include transport speed in feet per minute, tank size, and temperature control systems.

LIST OF ILLUSTRATIONS

Figure	
1-1	Rocker Tray Thermos-Bottle Agitation System.
1-2	RPM Thermos-Bottle Agitation System.
1-3	[redacted] High-Resolution Contact Printer.
1-4	Preparation of Master Resolution Target.
2-1	Gamma Equalization at Different Processing Temperatures.
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2-6	
2-7	Time-Temperature for Constant Gamma and Their Relationship to Resolution, Fog and Speed.

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SECTION 1

1. INTRODUCTION

1.1 BACKGROUND AND OBJECTIVES

The sensitometric information pertaining to elevated-temperature processing of photographic aerial film is not available from data published by leading manufacturers of photographic materials. Therefore, it is essential to supplement this gap in the published literature by conducting a series of photographic experiments to determine the sensitometric characteristic and the degree of image degradation that occurs under these conditions: 1) When processing temperatures are elevated beyond those normally recommended by the manufacturer; and 2) when corresponding process times are reduced to maintain approximately equal development. The effect of these two variables on gamma, speed, fog level, and resolution was then evaluated.

The results of these tests should become a permanent adjunct to the design data necessary for liquid/air bearing processors.

1.2 PROBLEM OF UNIFORM PROCESSING

A literature survey was conducted to determine the best possible method of agitation during processing. Much investigative effort has been devoted to the subjects of time and temperature, but the problem of agitation has not been sufficiently explored even though a great deal of thought has been put into devising special agitation methods such as spray, nitrogen burst, and rocking trays. All these employ some means of consistent movement of the processing solution to remove the unwanted byproducts that diffuse from the emulsion during development.

Since consistency and reproducibility are of prime importance in experimental sensitometric processing of photographic materials, it is essential to achieve optimum agitation. To establish a good level of repeatability, many repetitive tests were made to meet the rigid standards adopted for this project. Lack of consistent results, however, made further search for a dependable standard necessary.

The method finally used was adapted from ASA, PH 2.5-1960, (Reference 1) which requires that during development, a thermos bottle be given an oscillating movement by rocking it back and forth about its middle axis to an angle of 45-degrees on either side. This rate is such that one complete cycle is made in about 1 second. At the same time, the vessel is revolved about its axis, the time for one revolution being five seconds.

The above agitation method was performed by hand and since normal human error was a factor, consistent results could not be maintained, particularly at the shortest processing times. A mechanical means of agitation was employed which produced good consistency and repeatability (Figure 1-1).

A newer instrument (Figure 1-2) has been constructed which promises to provide even better reproducibility, but it has not yet been thoroughly checked out.

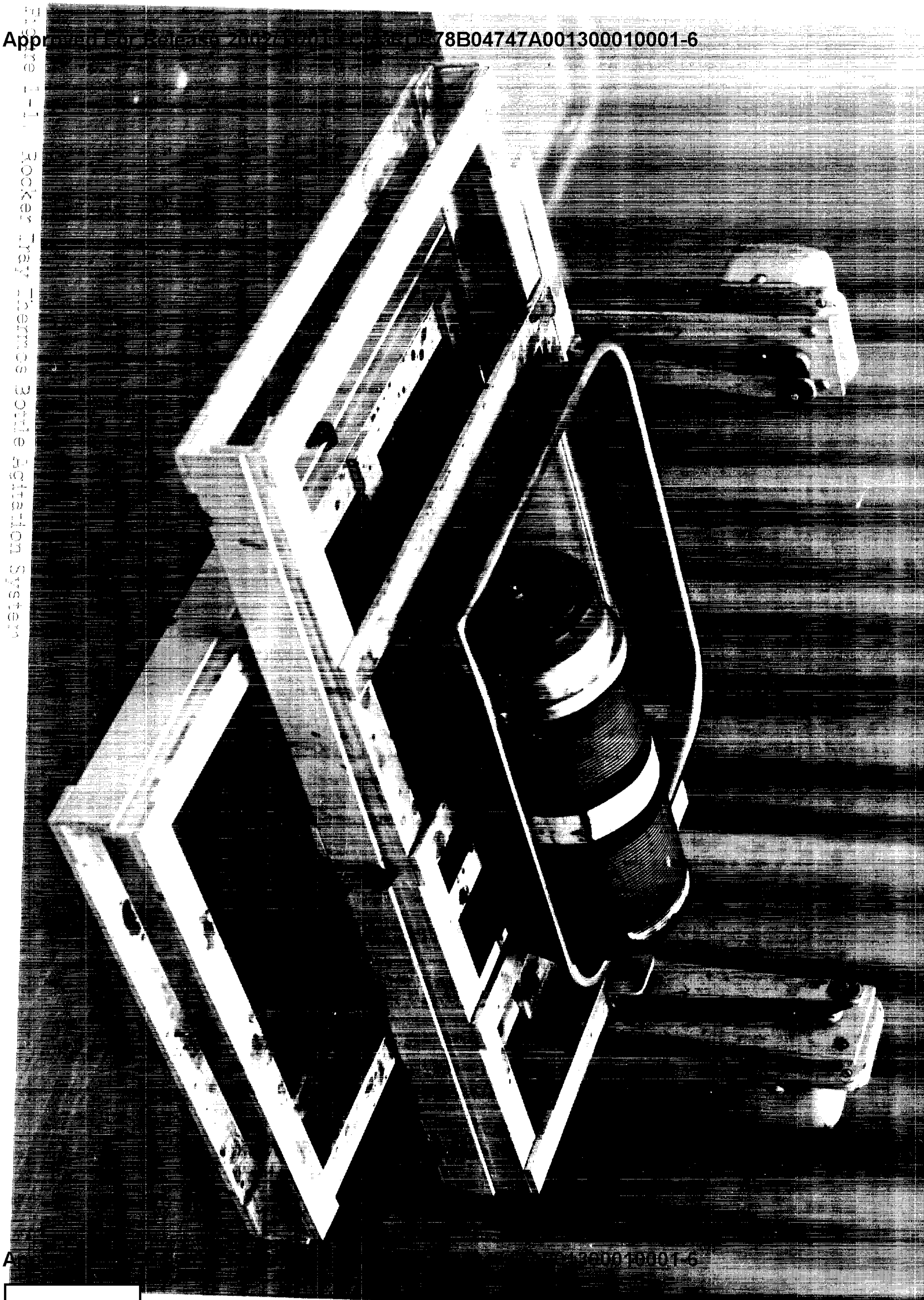

Figure 1-1 Rocker Tray Thermos Bottle Agitation System

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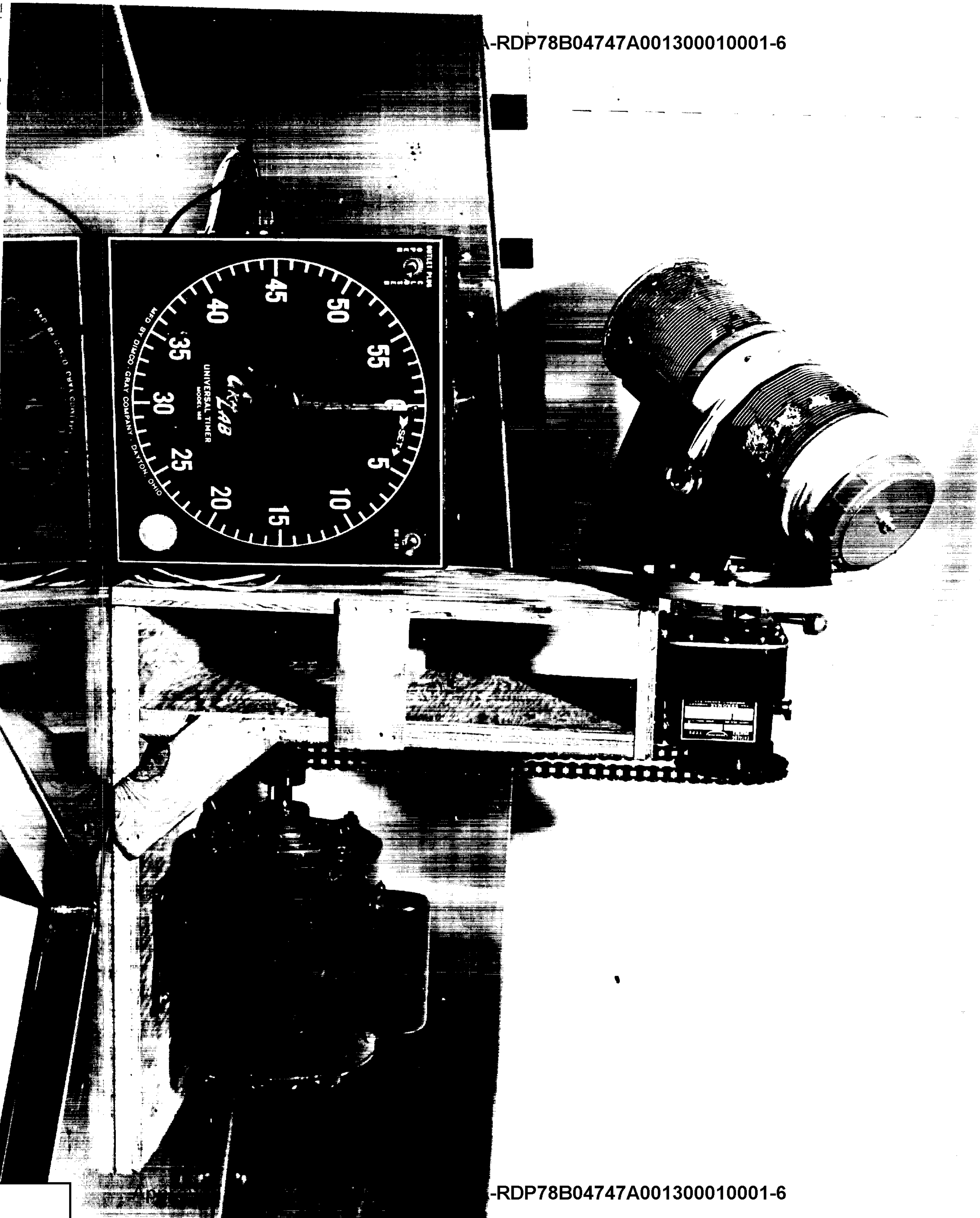

Figure 1-2. RFM Thermos Bottle Agitation System

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SECTION 2

2. TECHNICAL DISCUSSION

2.1 AERIAL FILM TESTED

Technical data on the photographic material tested during this phase of the project is listed below:

Kodak Panatomic-X
Aerial Film Type 4400
(Formerly Type SO-130)

Base: 2.5 mil Estar (polyester)
Emulsion thickness: 0.21 mils
Gel Backing Thickness: 0.18 mils
Sensitivity: Panchromatic, with extended red
RMS Granularity: Kodak Developer D-19. 052 (D_{net} = 1.0)
Resolving Power: 170 lines/mm at T.O.C 1000:1 (D-19)
65 lines/mm at T.O.C 1.6:1 (D-19)
Exposure index: Daylight 20
(Based on normal development of 8 minutes at 68°F in D-19.)

2.2 INSTRUMENTATION

Instrumentation used in this investigation is described in following subsections.

2.2.1 E.G. & G. Mark VII Sensitometer

A major advantage of this electronic flash sensitometer is that, in contrast to a tungsten source, the intensity and color balance remains essentially stable for an extremely long period of time. After 200,000 flashes, the total light output is decreased only 15 percent. Thus, experiments can be repeated under nearly identical conditions of exposure.

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STATINTL 2.2.2 High-Resolution Contact Printer

This printer features a point light source controlled by a variable transformer and timer, a folded light path to provide even intensity across the printing surface, and a pneumatic platen. Several control features are available including a voltmeter, variable diffusion, and variable light filtration (Figure 1-3).

2.2.3 Sensitometric Processor

This instrumentation employed a mechanical means of agitation which consisted of 2 tray rockers, one placed over the other in a criss-cross manner, with an 8410 processing tray placed on top of the two tray rockers. A thermos bottle was placed on its side in the tray, allowing it to rotate 180° from side to side. The rotation cycle was 27 per minute, or approximately twice the agitation recommended for tray processing.

The internal dimensions of this thermos bottle were 4 by 22 cm. The negative material tested was fastened to a plexiglass strip measuring about 3.5 by 20 cm. The plexiglass strip and the test film were inserted in the bottle and development performed as described above.

2.2.4 Welch Densichron

This densitometer is of the direct reading type which reads diffused densities. The aperture used throughout all photographic tests on this project was 1/16-inch in diameter.

2.3 TEST PROCEDURE

2.3.1 Sensitometric Exposing Data

Samples of 16mm Type 4400 film were cut in 12-inch strips and exposed on the E.G. & G. Mark VI Sensitometer using a Kodak No. 2

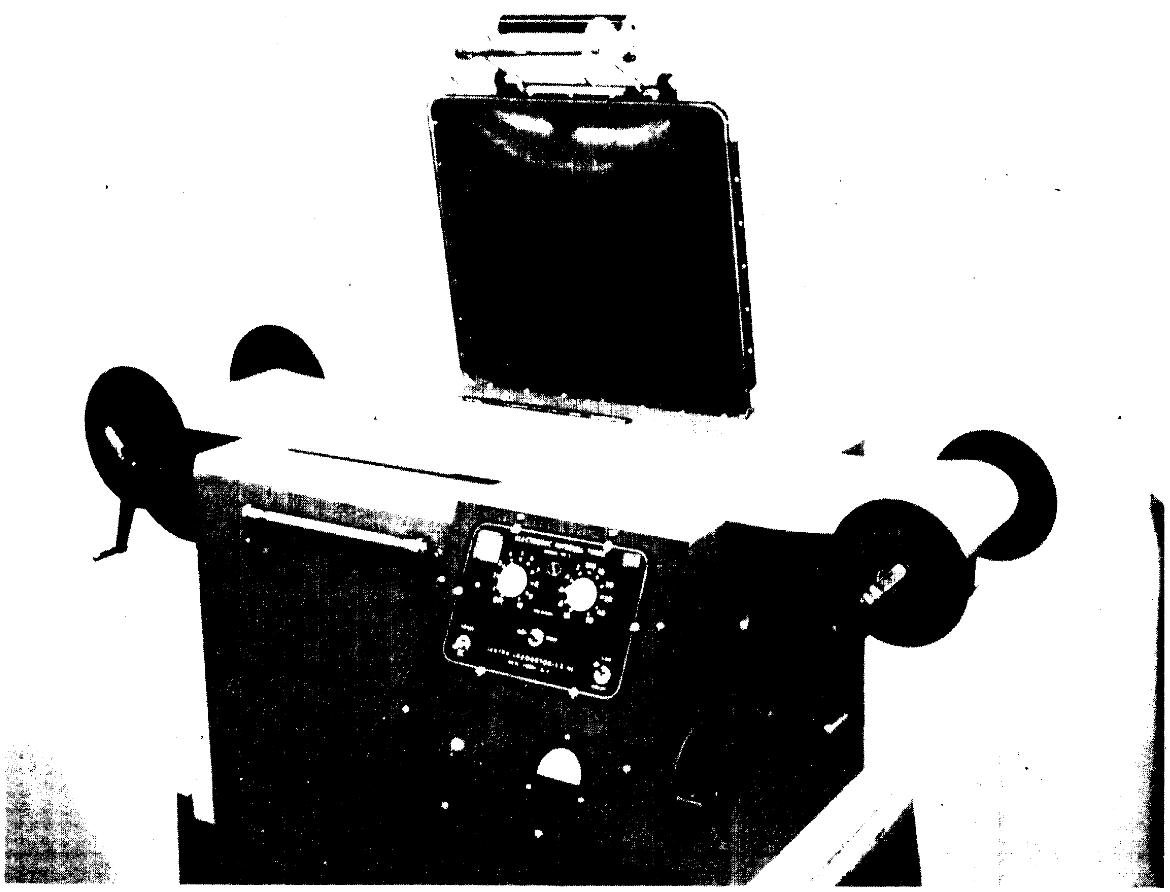

Figure 1-3. [redacted] High Resolution Printer

Photographic 21-step tablet as follows:

The compensating attenuator selected was attested as equivalent to a Neutral Density (ND) 1.7 filter by the manufacturer. The 10^{-4} second circuit speed on the Mark VI produced approximately 130 meter-candle-seconds. The charging time characteristics of the flash circuits were such that a charging time of 5 seconds was allowed between exposures to be certain of peak output.

2.3.2 Preparation and Exposing Data of Master Resolution Target

The resolution-target series was designed so that the optimum resolution for a particular time-temperature combination, using only one exposure, could be determined. This was done by superimposing a 50X reduction resolution target over each field of a 21-step tablet. This technique produced an ND-filter effect which controlled the intensity of the light source to each resolution target. The basic exposure used was 21 Volts at 0.6 seconds with a pneumatic platen pressure of 5 psi. Figure 1-4

2.3.3 Conditioning Film Before Processing

In the interval between exposure and processing, the samples were kept under conditions which were consistent from one test to another; namely, the latent image was allowed to stabilize at least 12 hours before processing. The film was encased in a film can at about 70°F during the aging step. The entire processing procedure for any one test was completed no later than 20 hours after exposure.

2.4 VARIATION OF PARAMETERS

The temperature was increased in 10°F increments from 68°F to 118°F. Concurrently, the developing time was decreased to provide a nearly constant gamma. Using this technique, the following relationships were evaluated:

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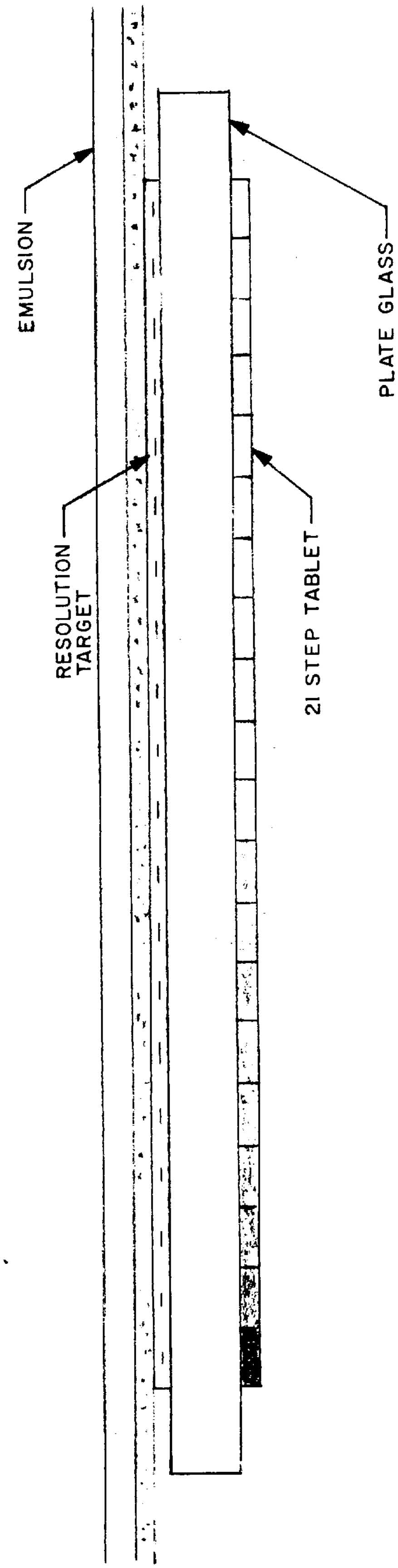

FIGURE 1-4: PREPARATION OF MASTER RESOLUTION TARGET

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1) The developing time-temperature combinations (by successive approximation) to produce a constant gamma.

2) The effect of increased temperature on fog level, resolution, speed, and granularity. (These latter tests are not yet complete and so, have not been included in this report.)

With each exposed sensitometric strip processed, a resolution strip was also processed to assure the same conditions and to preclude variables.

2.4.1 Additional Processing Control

In achieving a high degree of accuracy and repeatability in film processing, particular attention was focused on time ($\pm$0.5 second), temperature control ($\pm$ 0.1°C), volume of solution ($\pm$ 0.5 cc), pH and agitation for each film sample processed.

2.5 EXPERIMENTAL RESULTS

2.5.1 Gamma Equalization at Different Processing Temperatures

For a first approximation, a general rule-of-thumb was employed in this test series. It was to decrease the time 50 percent for every 10°F increase in temperature. The preliminary test proved the need for a fine adjustment of the processing times. The test procedure covered 27 processing times with the temperature increased in 10°F increments from 68° to 118°F. Nine of the tests followed the general rule-of-thumb outlined above, nine were 20-percent above, and the remaining nine 20-percent below the expected time.

This combination produced a predicted overlapping effect on the measured gammas. Using the gamma of the control as the normal, selections were made throughout the various time-temperature-combinations to maintain a consistent gamma (Figure 2-1).

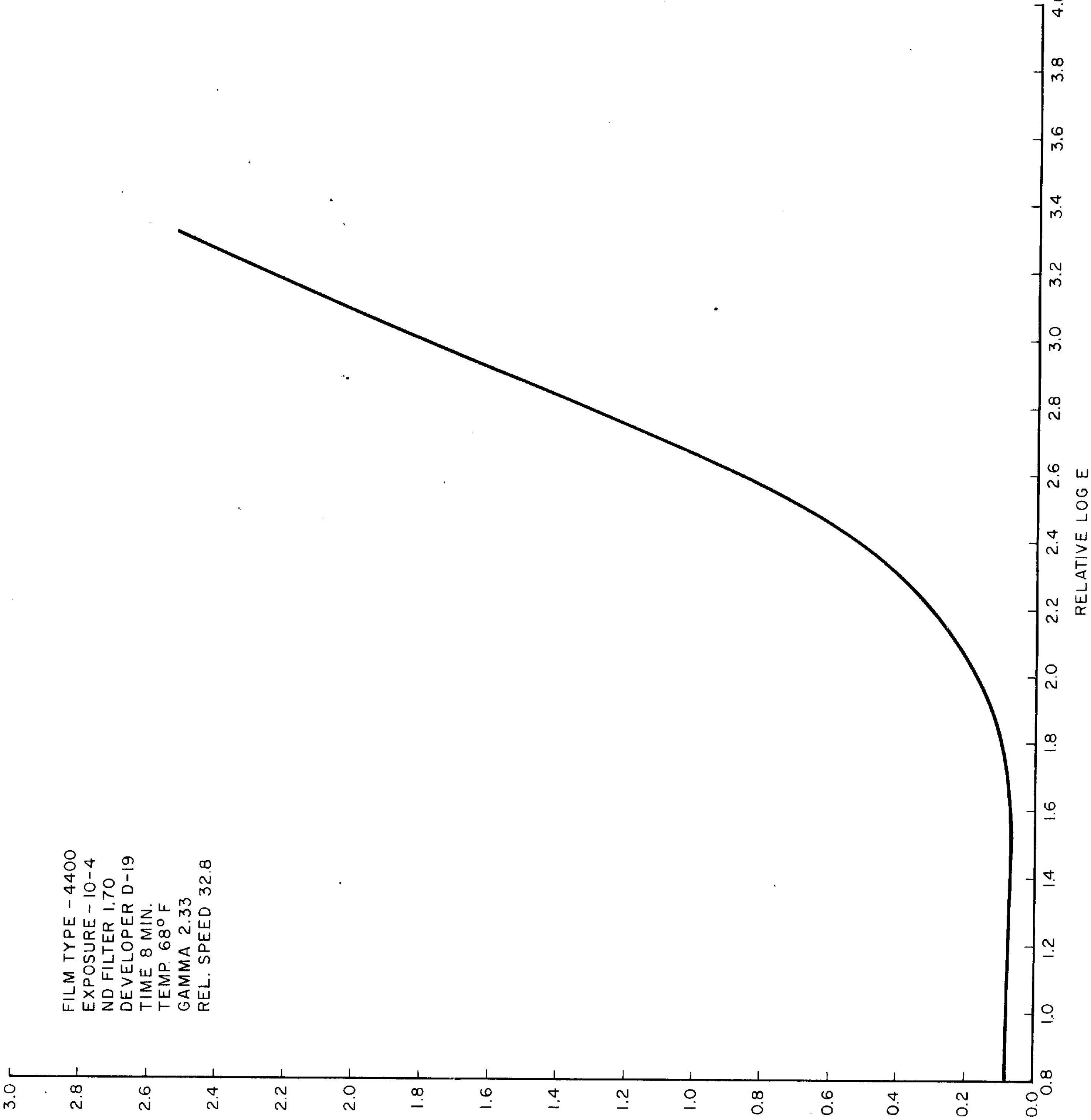

Figure 2-1. Gamma Equalization at Different Processing Temperatures.

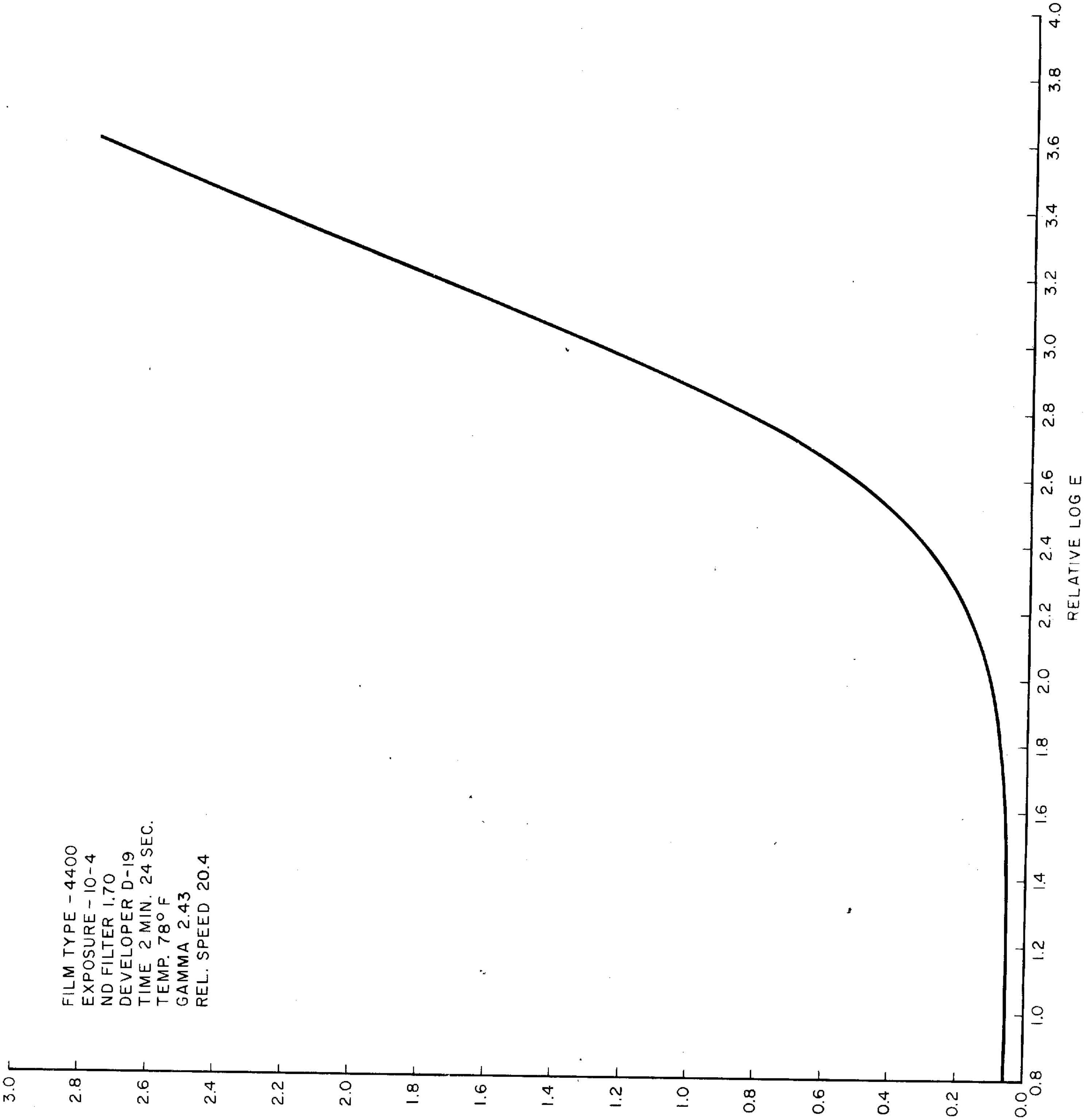

Figure 2-5. Gamma Equaliztion at Different Processing Temperatures.

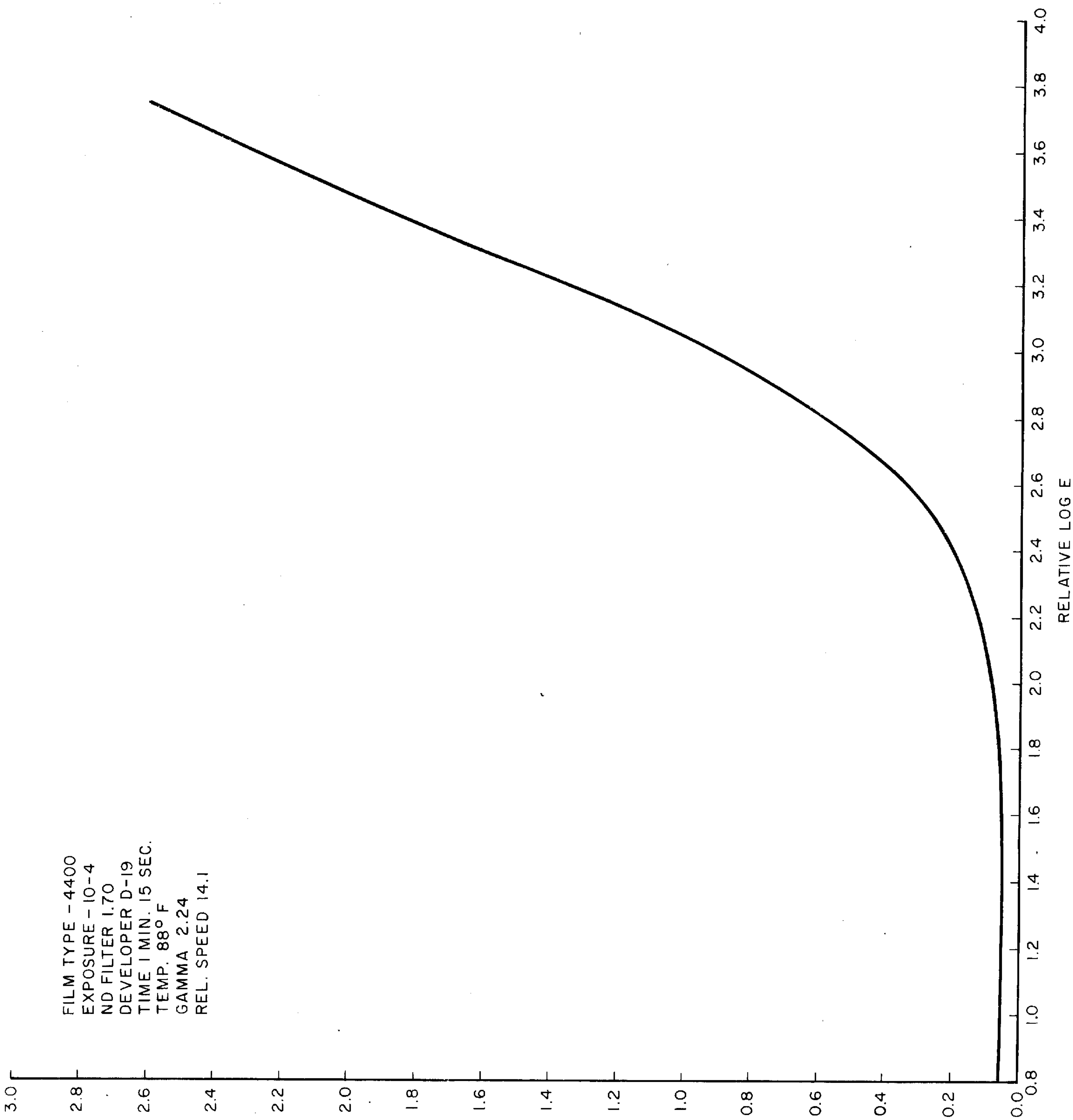

Figure 2-4. Gamma Equaliztion at Different Processing Temperatures.

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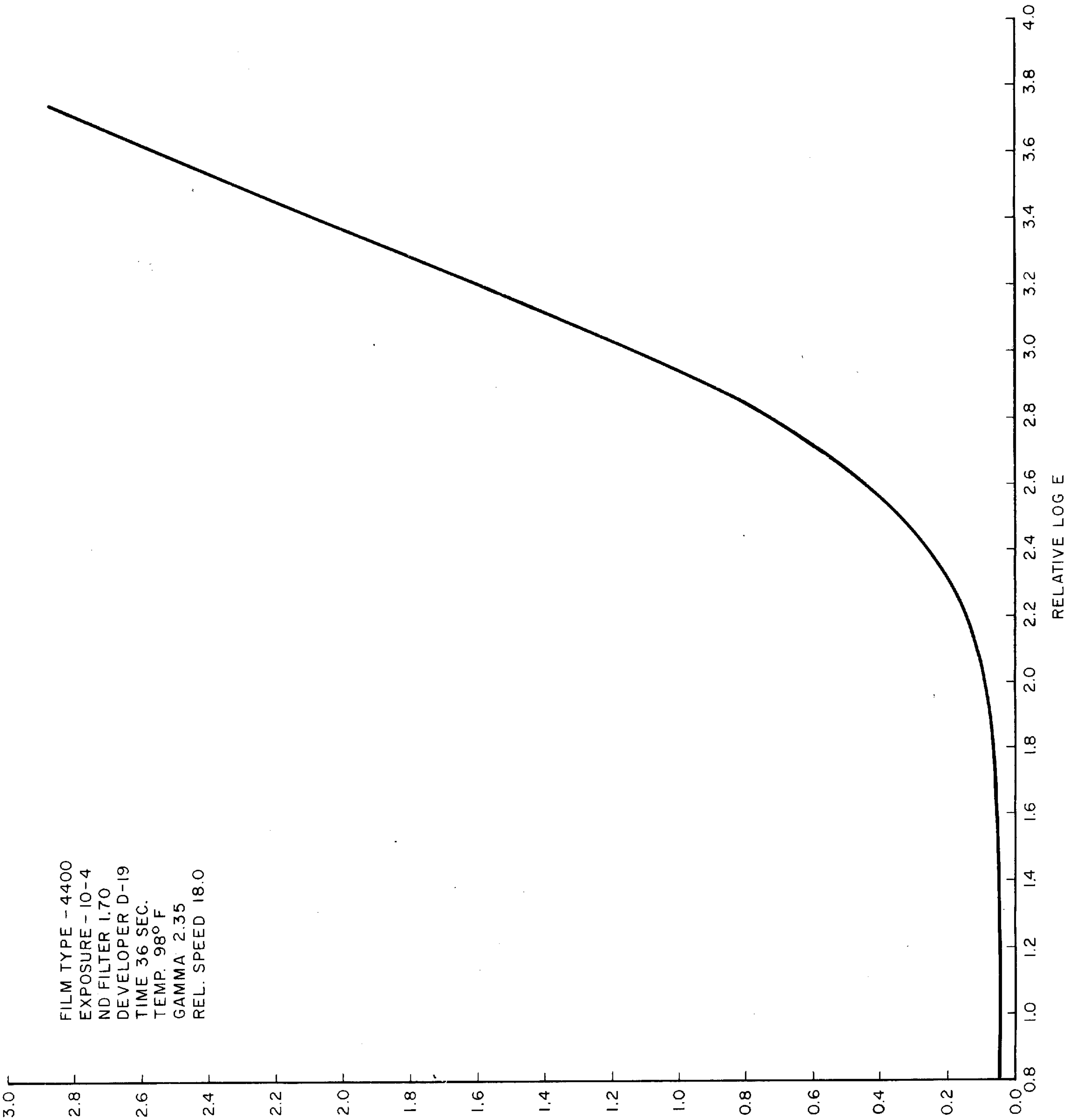

Figure 2-3. Gamma Equaliztion at Different Processing Temperatures.

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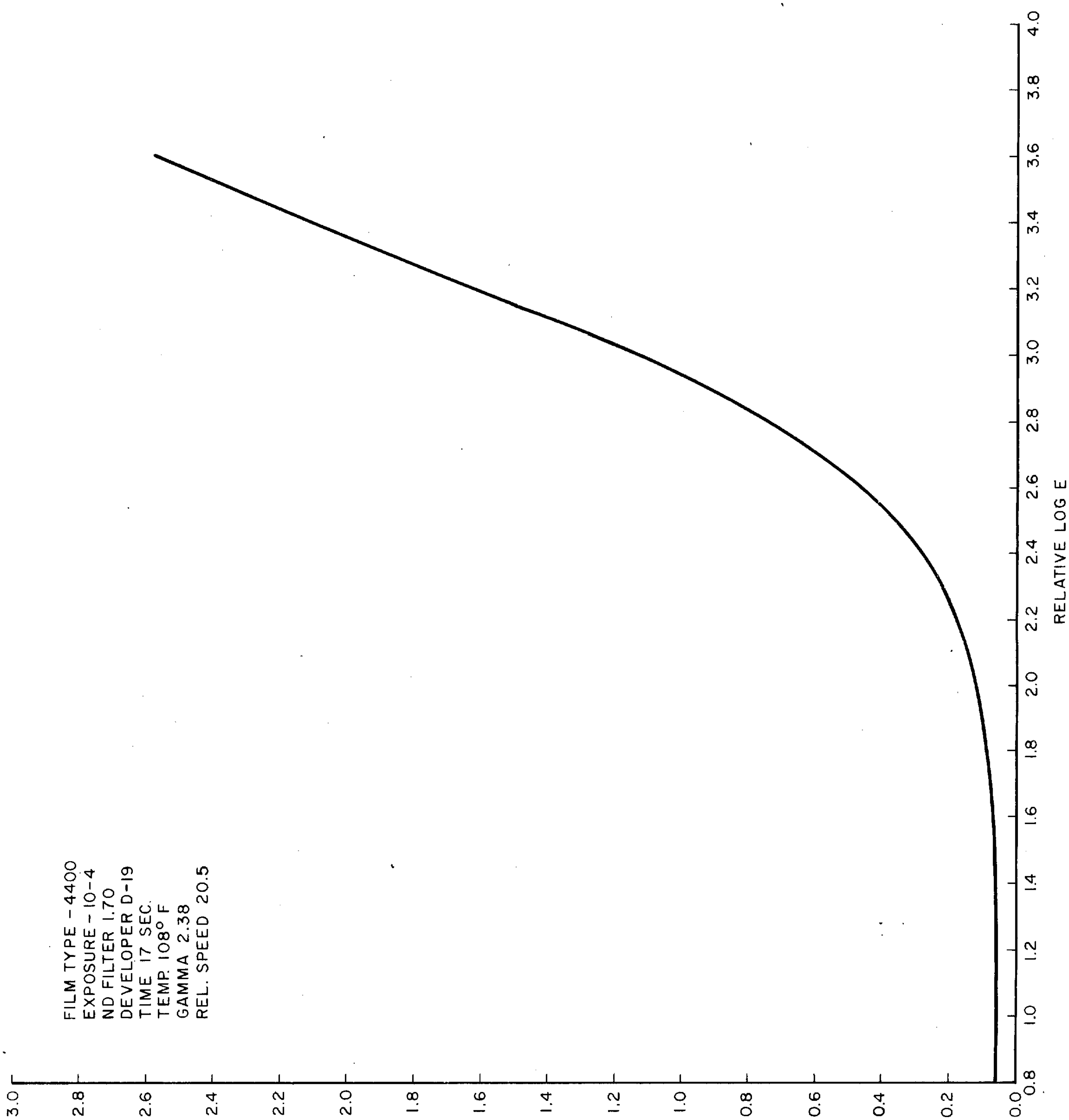

Figure 2-2 Gamma Equalization at Different Processing Temperatures.

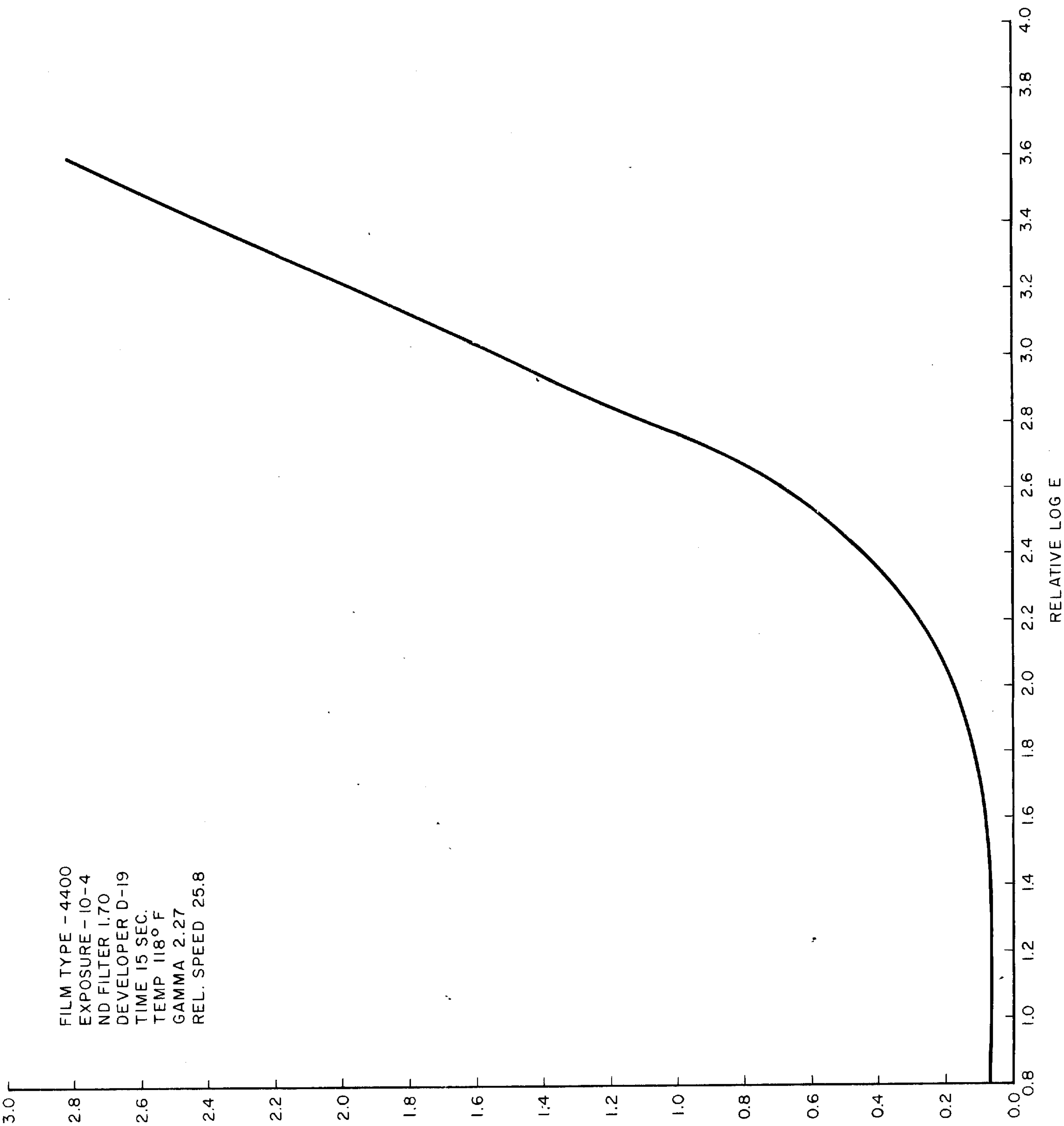

Figure 2-6. Gamma Equaliztion at Different Processing Temperatures.

2.5.2 Time-Temperature for Consistent Gamma and the Relationship to Resolution, Fog, and Speed

1) Time-Temperature Vs Resolution

The maximum resolution achieved was 50 lines/mm and remained the same throughout all nine time-temperature combinations ranging from 68° to 118°F. An interesting correlation was noticed; namely, that as the temperature increased and the time decreased, the exposure required to maintain the 50 lines/mm level was increased progressively across the 21-step tablet (Figure 2-2).

2) Time-Temperature Vs Chemical Fog

Contrary to the anticipated results, the fog level remained fairly constant, the average being base plus fog of 0.05. This was well within the allowable $\pm$ 0.01 tolerance (Figure 2-2).

3) Time-Temperature Vs Relative Speed

The following formula was used to calculate relative speed:

Speed = S_A relative sensitivity = $\frac{1}{E}$, where density = 0.6 above gross fog.

In Figure 2-2, it was noted that there is a sharp decline from the control to the third test; then the speed increased.

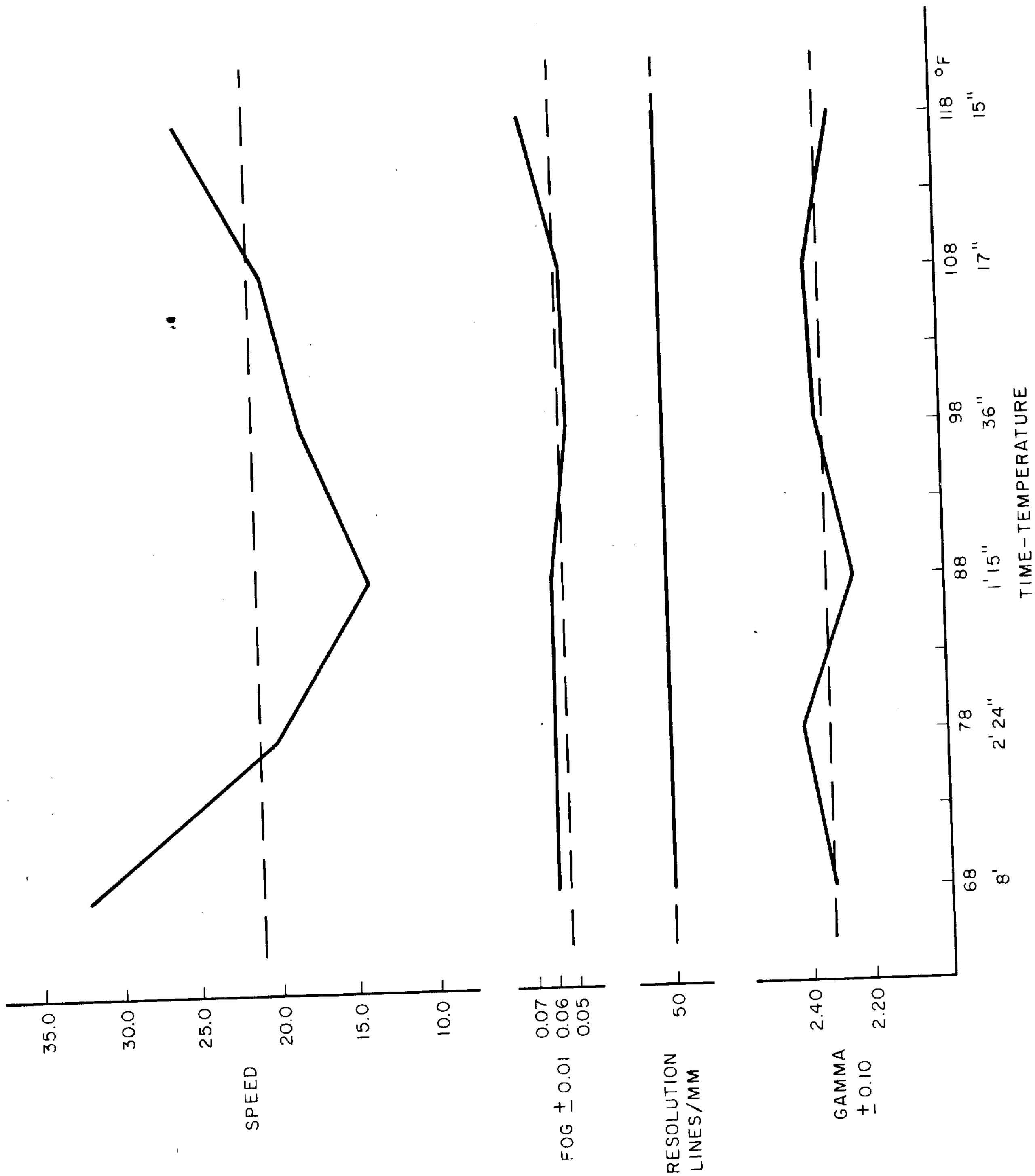

FIGURE 2-2: FILM TYPE 4400

3. SUMMARY

Measurements on film types 4404 and 8430 are still being made. Therefore, accumulated data are incomplete and will not be included in this report.

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

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REPORT

PERFORMANCE EVALUATION
OF A
POSITIVE-PRESSURE TRANSPORT CAPSTAN

STATINTL

February 1965

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FOREWORD

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[redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted] This report should be read in conjunction with [redacted] of which it is a part.

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Approved: [redacted]

ABSTRACT

An investigation was conducted on a new technique for generating and applying a vacuum to a rotating film-drive capstan. The diameter of the capstan was established as a result of the tests conducted under assignment [redacted]

1. INTRODUCTION

A program of investigation on a new technique for generating and applying a vacuum to a rotating film-drive capstan was conducted. This technique involves the generation of a vacuum by passing high-velocity air through a shaped annular orifice and connecting vacuum lines to an opening in the throat of this orifice.

Two capstans were fabricated for this program. The first one, 3 inches in diameter, was used to obtain most of the test data given in this report (Figure 1-1). A second capstan is a working model that will drive all film widths from 70mm to 9-1/2 inches in base thicknesses to 0.0012 inch. It is 6 inches in diameter so that undue forces will not be required for the sole purpose of bending film.

2. TECHNICAL DISCUSSION

The vacuum capstan is operated on the venturi principle to generate a vacuum in the rotating member of the capstan. Figure 2-1 shows a cross-section of a positive pressure capstan, illustrating the principle of generating a vacuum in an annular orifice.

Under positive pressure, air enters the capstan parallel to the axis of rotation. The air is deflected outward so that it exits from the capstan radially to the axis of rotation between a fixed and a rotating member. A circle of maximum restriction is found between the two members. This is shown at point B in Figure 2-1. The pressure at this restriction point is inversely proportional to a function of the velocity of the air passing through it.

Several openings were made around the circumference of the capstan at the throat of the annular orifice to create a multiplicity of vacuum ports. Each of these vacuum ports is connected to the surface of

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the rotating member of the capstan at the point where it is desired to drive the film. In this case, it is at the midpoint of the capstan so that all sizes of film can be driven.

The force holding the film to the capstan is dependent on the differential pressure on the two sides of the film and the area over which this differential pressure is applied. A greater driving force for the film can be created by increasing the area served by each vacuum port on the capstan. This can be done by providing a rough surface around each port or by providing a pattern of grooves connected to each port.

Most capstans are required to hold film over a fixed angle, which is related to the angle of rotation. This is usually 180 degrees or less, depending on the configuration of the machine in which they are used. It is necessary that the capstan release the film at a point beyond the film wrap. This is done with the positive pressure capstan by providing a vane on the fixed member, across the outlet of the orifice, so that back pressure is created to reduce the velocity through the throat at the desired release point. If a high enough back pressure is obtained, this will not only eliminate the vacuum but will also provide a positive pressure to separate the film from the rotating capstan.

A set of tests was conducted on the 3-inch capstan to determine vacuum pressure in inches of mercury as a function of the volume, or volume of air per minute used. These results are shown in Figures 2-2 and 2-3. Each is an average of four vacuum ports in one particular position; that is, having one relationship to the fixed member. These positions are indicated as A and B on the curves. Above a volume rate of about 6 cubic feet per minute, the curve remains linear for the volume rate of flow. The break in these curves is unexplained at this time.

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3. FUTURE PROGRAM

A program to compare the positive pressure 6-inch capstan to a negative pressure 6-inch capstan should prove of interest. All parameters associated with vacuum capstans would be compared.

The negative pressure capstan (Figure 3-1) to be used in this investigation was developed by [redacted] A cross-section view of this capstan is shown in Figure 3-2. STATINTL

STATINTL The [redacted] negative pressure capstan utilizes a vacuum pump with a capacity of 4 inches of water as a vacuum source. Vacuum from the pump is applied to the stationary rear section of the capstan through a flexible tube. The stationary section contains a vacuum restriction plate and a PVC backup plate that acts as a retainer for a Teflon bearing. Interchangeable Teflon plates can be provided to limit vacuum distribution to 180 degrees, 90 degrees, or other radii of film wrap.

The capstan has an annular slot that transfers air from the holes in its surface through the slot to the Teflon restriction plate. Numerous holes are arranged in a 70mm wide pattern in the center of the capstan, which is wide enough to transport 9-1/2-inch film. The capstan has a slight crown so that film seeks its center to provide automatic and precise tracking.

Cleaning and servicing is easily done by removing a single retaining knob and sliding off the rotating member. Provision is made for film transport in both dry and wet sections of the processor.

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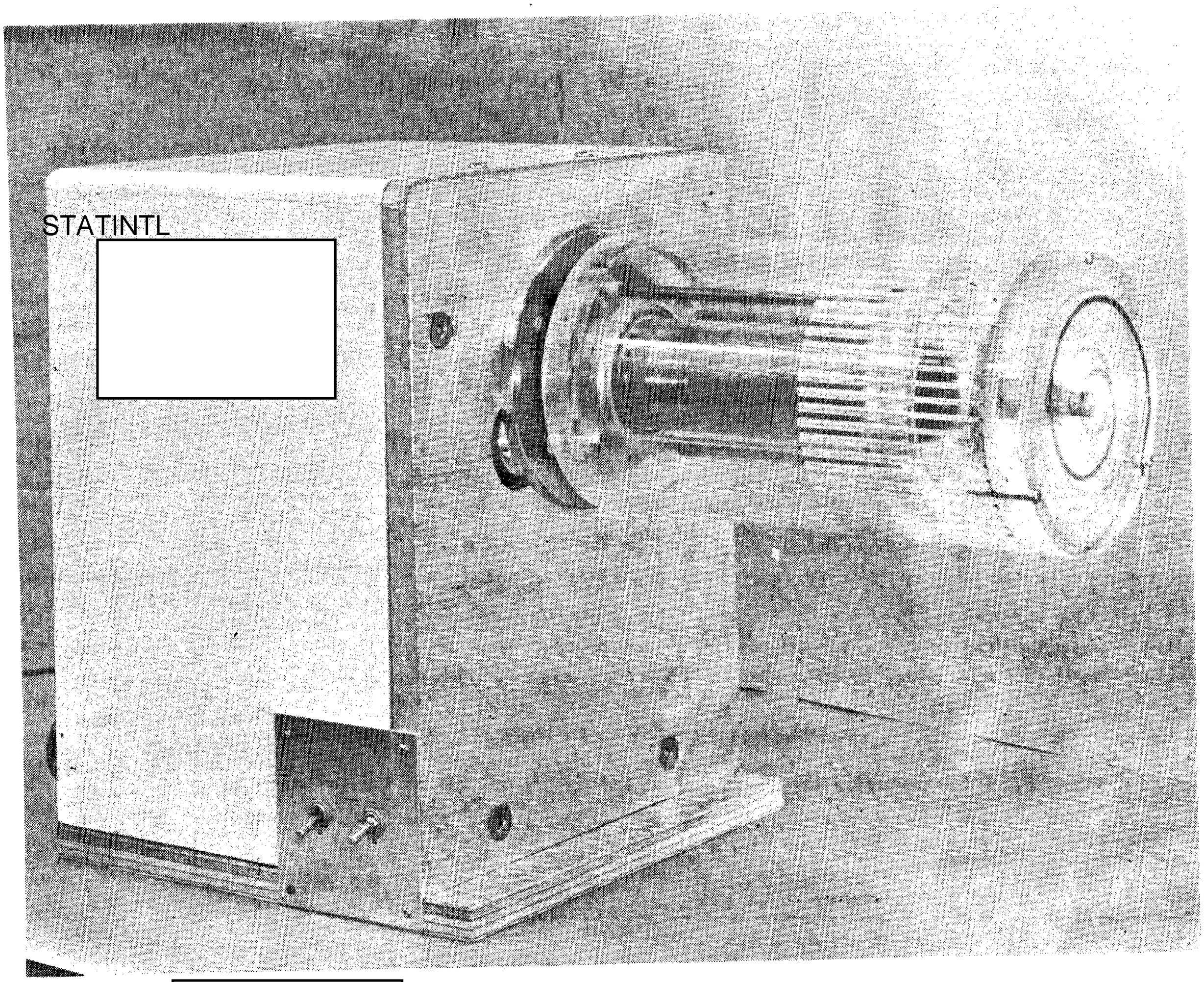

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Figure 1-1. [redacted] Vacuum Capstan Test Model

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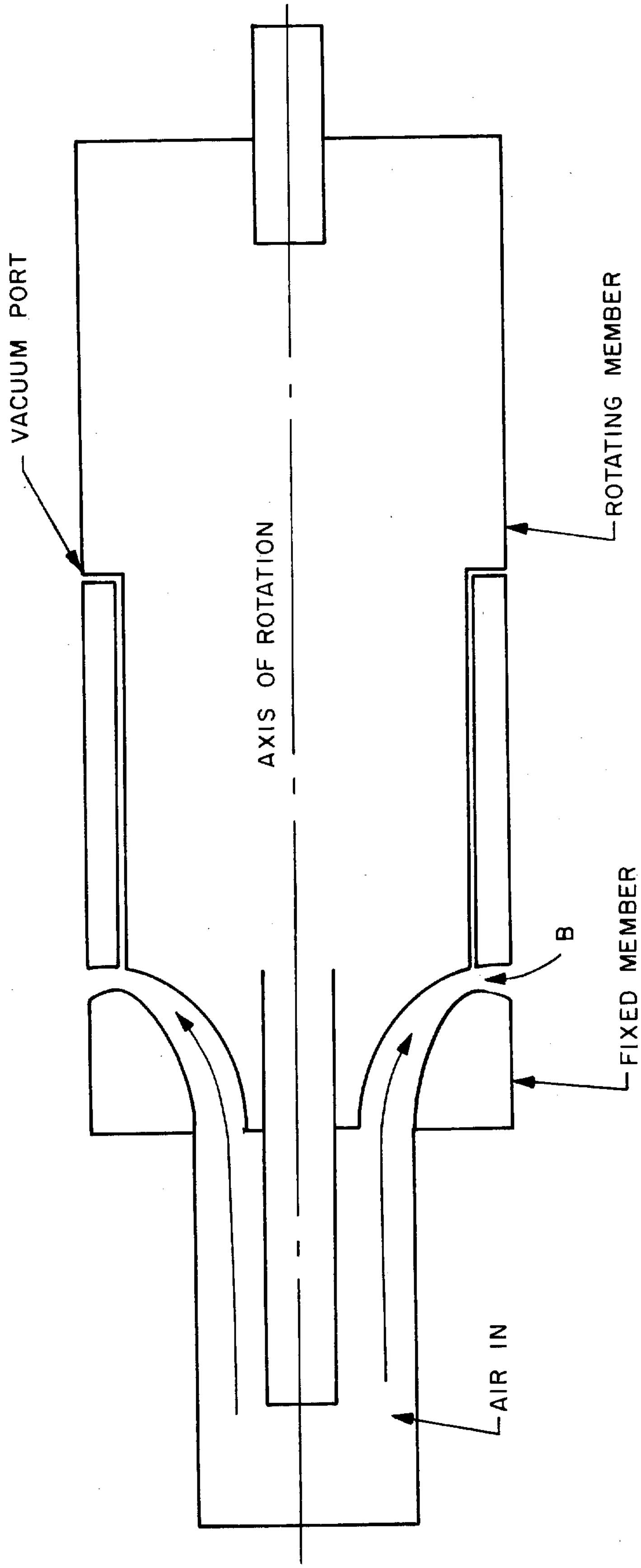

Figure 2-1. Positive-Pressure Vacuum Capstan

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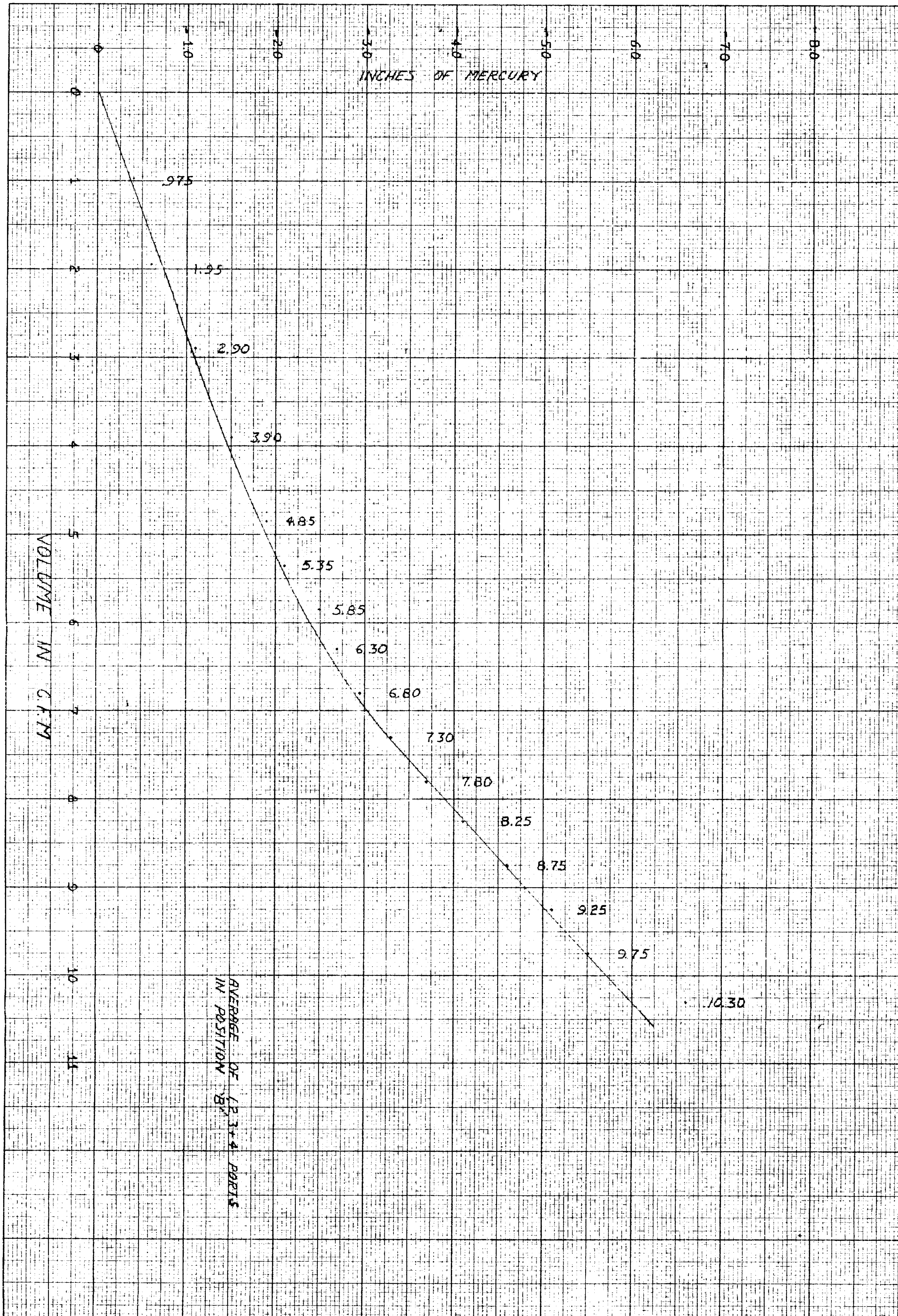

Figure 2-2.

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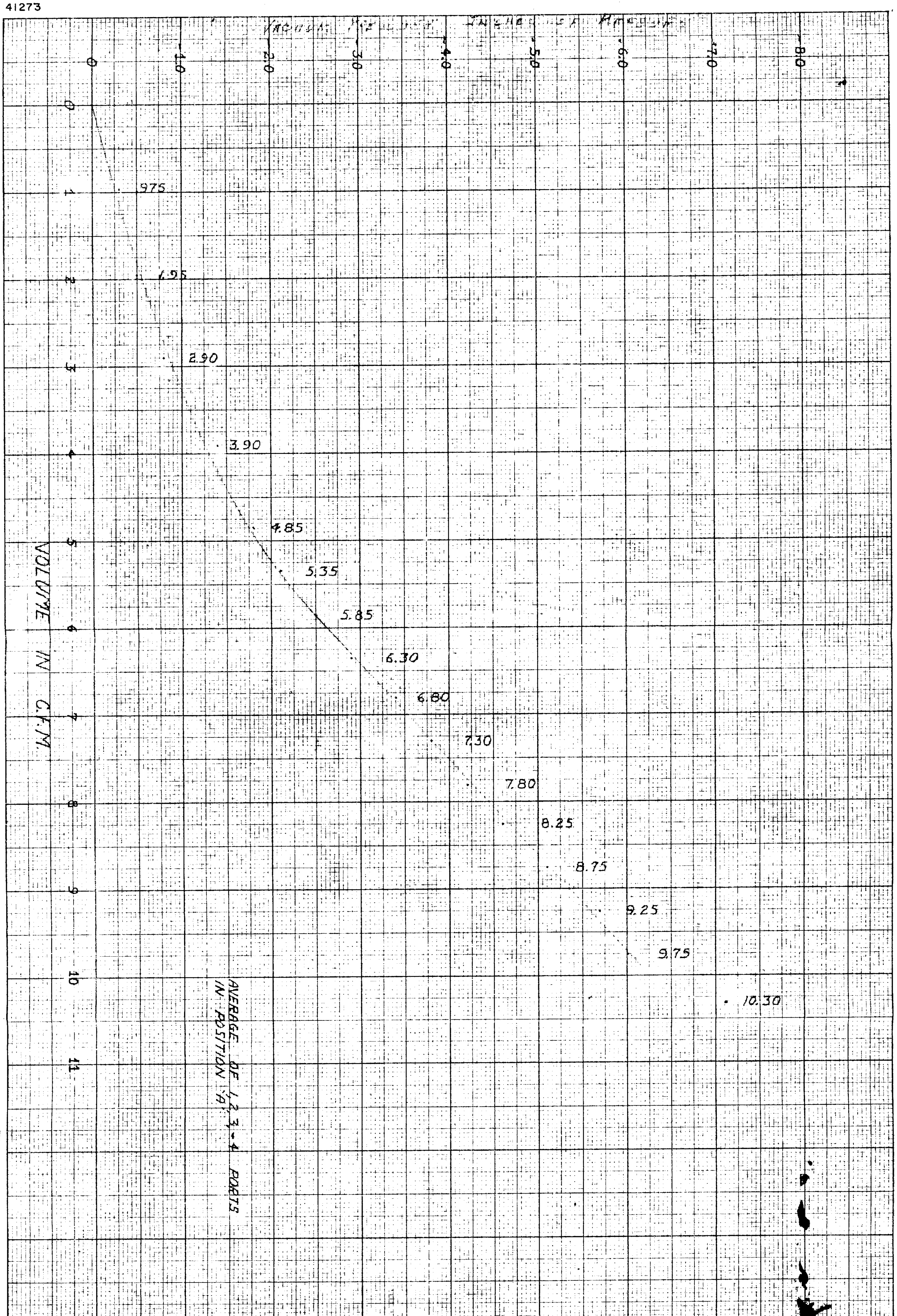

Figure 2-3.

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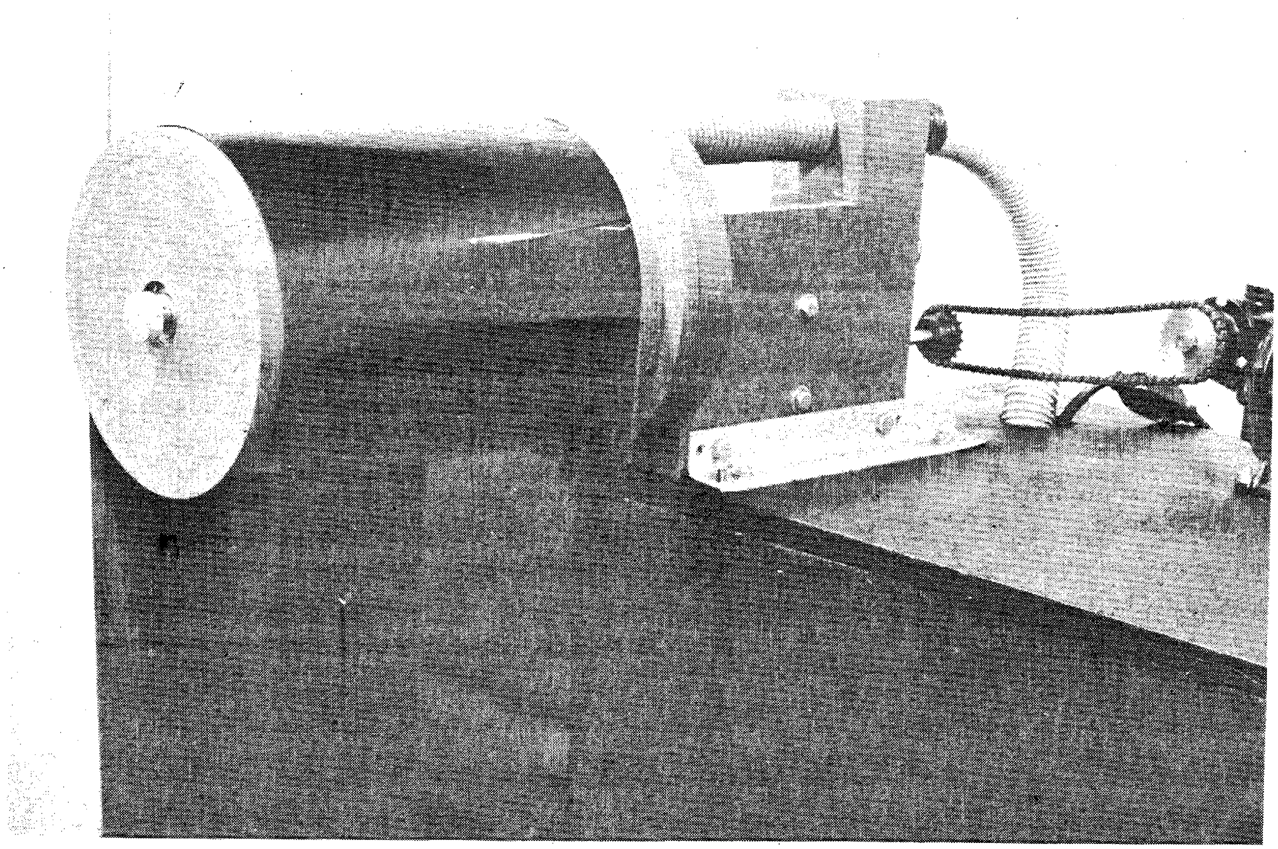

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STATINTL Figure 3-1. Vacuum Capstan Test Model

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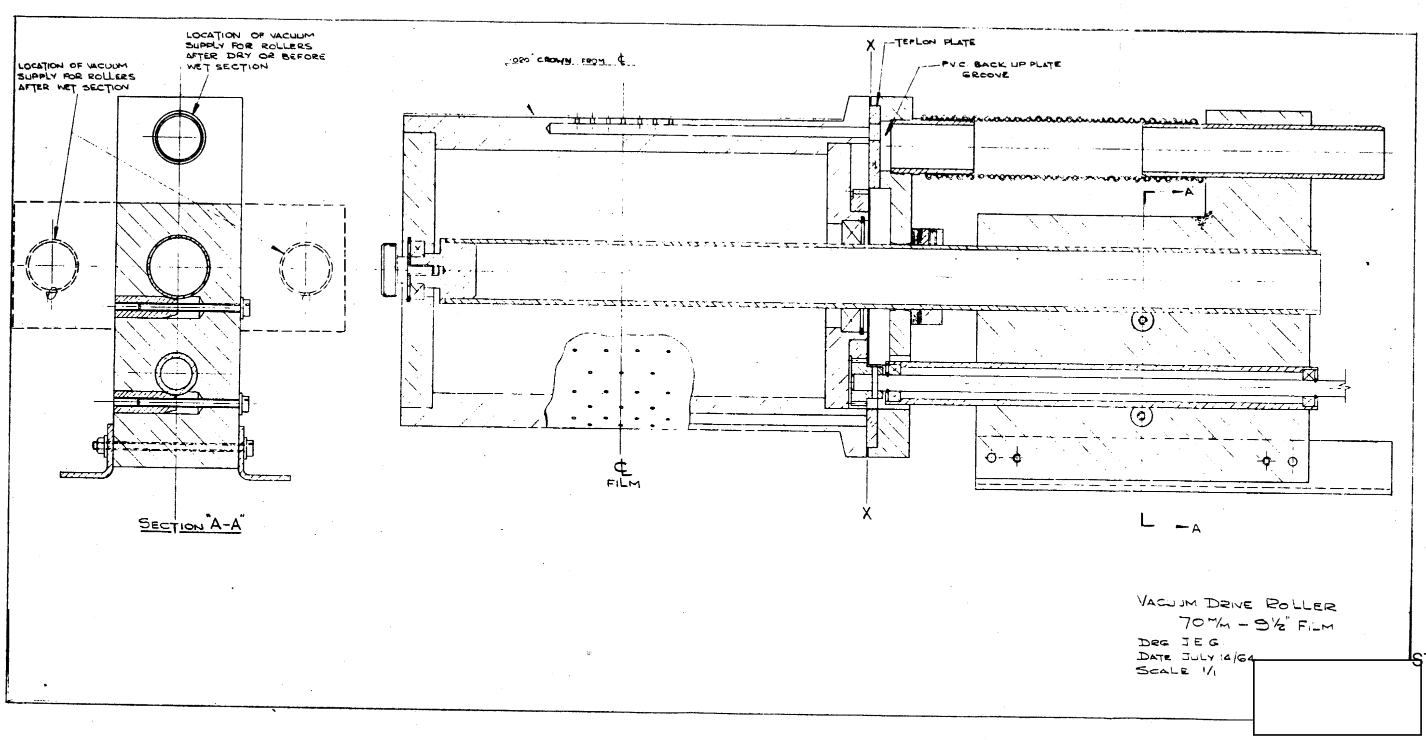

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Figure 3-2. Vacuum Capstan, Cross-Section View

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REPORT

Calculating The Efficiency of
a Tank With An Incorporated
Thermal Control.

Tank Thermal Control

STATINTL

February 1965

FOREWORD

STATINTL

[redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted] This report should be read in conjunction with Report [redacted] of which it forms a part. STAT STAT

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Approved: [redacted]

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

ABSTRACT

Among the major objectives of this Processor Development Program are those to reduce equipment size, to reduce power consumption, and to investigate the modular concept of processor design. In the HTA-5 Processor, the concept of separate service units in which all support equipment was mounted led to increased installation space and decreased efficiency. This assignment, in conjunction with assignment [redacted] investigates the possibility of eliminating the service units and separate temperature-control equipment as a step towards meeting these objectives.

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Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

INTRODUCTION

1. CONVENTIONAL TEMPERATURE CONTROL

One conventional method of regulating solution temperature in a processing tank,is to provide a control system in which solution is drawn from the tank,and pumped through a coil-type heat exchanger and through a small tank containing heating units. The solution is then returned to the processing tank through a specific recirculation system designed to eliminate striation (Figure 1-1).

Chilled water at a temperature of about 45°F is circulated through the jacket of the heat exchanger to extract heat from the solution as it passes through the coils. Should the solution already be below control temperature, its temperature is automatically raised by electrical resistance heaters as it passes through the small tank.

The temperature of the solution in the processing tank is continually monitored by a temperature probe mounted in the tank. The probe compares this temperature against that set in a Wheatstone bridge or other control circuit. An increase in the temperature of the solution in the tank causes a circuit to open a valve in the chilled water line to the heat exchanger. Conversely, a drop in the temperature of the solution will cause a circuit to close a solenoid switch and operate the heaters in the small tank. This equipment is both bulky and inefficient.

TECHNICAL DISCUSSION

2. INTEGRAL THERMAL CONTROL

To provide a built-in thermal control while permitting liquid bearings to be installed at first seemed to present a technically difficult problem. However, the concept of using the bearing pump for recirculation offers one approach to the problem (Figure 2-1). If the processing tank were divided into two sections by a heat-exchanging wall and the bearings were mounted in this wall, a new mode of operation could be attained. Then,solution from the bearing side of the tank could be pumped into the pressure side of the heat exchanger wall and thence returned through the bearings to the front section of the tank. Recirculation from one side of the tank to the other would provide a flow of liquid around the heat exchanger wall, with the added advantage that little energy would be lost to the ambient environment. To further improve thermal efficiency, the module (of which the tank would be an integral part) would be insulated to reduce heat loss to the atmosphere.

To maintain an 88°F solution temperature with a 65°F room ambient, a heating load of approximately 3530 BTU/hr is required. This does not consider the heat from the bearing pump which, for the purpose of this study is 3 horsepower, which is rated at 7635 BTU/hr. On the other hand, to maintain a solution temperature of 68°F in a 75°F room ambient with a 75°F solution replenishment temperature, 8720 BTU/hr would be required as a cooling load.

It is safe to assume that chilled water at 45°F and hot water at 160°F are available in most modern processing laboratories at flows of 1-3/4 and 3/4 gpm respectively and at 10°FTD. The tank size selected (Figure 2-1) provides a surface area of 4.95 square feet for heat exchange. This area can handle a 9000 BTU/hr load, which is very close to the required 8720 BTU/hr cooling load.

If the heat exchanger is used as the back wall of the tank, then the pulldown rate is approximately 1.4°F per hour. If the bottom surface of the tank is used as heat exchanger, the U factors could be a little higher. In this case, however, an area of only 3.1 square feet and 8350 BTU/hr would be available. The 3-horsepower pump could be used to obtain steady state cooling, but the pulldown rate would be only 0.5°F per hour.

The following calculations assume the use of a plate-coil wall comprised of 3/8 inch single-embossed small tubes. The pressure drop is high, but such devices could be employed in parallel circuits to reduce the losses. Several types of circuitry can be used for temperature control. On the assumption that both chilled and hot water are available in the quantity required, the system illustrated in Figure 2-2 offers a clean compact unit with a minimum of outside controls.

<u>Parameters:</u>

(1) Tank Capacity: 75 gallons.

(2) Length of Film in Tank: 44-1/2 feet.

(3) Approximate Film Transport Speed: 20 fpm.

(4) Development Time Required: 2 minutes.

(5) Bearing Diameter: 2 inches.

(6) Operating Temperature: 60° to 88°F.

(7) Bearing Flow (Assumed): 12 gpm.

(8) 13 Bearings per Tank: 156 gpm total.

(9) Replenishment Rate (Assumed): 5 to 10 gpm.

(10) Tank Insulation: 1 inch of Styrofoam.

(11) Ambient Temperature Range: 65° to 75°F at 50 percent RH.

Total Volume 38" x 32" x 14"
= 17024 Inches^3.
= 9.85 Feet^3 (use 10)

Gallons 9.85 x 7.48
= 73.6 gallons (use 75)

Weight 73.6 x 8.345
= 615 lbs. (use 625)

The following mathematical analysis assumes the use of stationary sleeve-type liquid bearings. However, assignment [redacted] STAT which considers a type of bearing incorporating a built-in pump, would influence the parameters given to the extent that no bearing pump would be required. Therefore, the 3-horsepower heat input load from the pump would not exist. Furthermore, the idea of a rear pressure compartment would not be required. A further analysis, using the bottom and sides of the tank, will be made when the processor module now under design STAT study (assignment [redacted] has advanced sufficiently.

To find heat losses during high temperature processing (88°F) assume all sides and bottom are insulated.

$$A_{si} = 2 \times \frac{38 \times 32}{144} + 2 \times \frac{18 \times 38}{144} + \frac{18 \times 32}{144}$$

$$= 16.9 + 9.5 + 4.0 = 30.4 \text{ ft}^2 \text{ (inside diameter)}$$

$$A_{so} = 2 \times \frac{39 \times 33}{144} + 2 \times \frac{19 \times 39}{144} + \frac{19 \times 33}{144}$$

$$= 17.85 + 10.3 + 4.35 = 32.5 \text{ ft}^2 \text{ (outside diameter)}$$

$$A_{s\ \text{top}} = \frac{4 \times 32}{144} = 0.89 \text{ ft}^2 \text{ (could be insulated too.)}$$

$$A_{s\ \text{open surface}} = \frac{14 \times 32}{144} = 3.1 \text{ ft}^2$$

Let $A_s = 33 \text{ ft}^2$

$A_{s\ \text{open}} = 3.1 \text{ ft}^2$

$U = 0.35$ BTU/hr - ft^2 - °F (1" Styrofoam)

$$Q_{\text{sides}} = UA_s\Delta t = (0.35)\ (33)\ (\overset{23}{\underset{\text{High Soln}}{88} - \underset{\text{low room}}{65}}) = 266 \text{ BTU/hr}$$

$U^1_{\text{top sens}} = 300$ BTU/hr - ft^2 @ 23°TD Still air cond. (Unvented)

$U^1_{\text{top sens}} = 300 \times 1.8 = 540$ @ 23°TD and 300 fpm (vented)

Assume normal B & W operation with <u>no</u> venting

$Q_{\text{top sens}} = 300 \times 3.1 = 930$ BTU/hr

 STAT

$Q_{Latent} = KU'' A_s$

$U'' = 95$ BTU/hr-ft^2 @ 88°F surf and 75° Room

f @ 65°F room = 1.4

$KU'' = 1.4 \times 95 = 133$

$Q_{Latent} = 133 \times 3.1 = 412$ BTU/hr

$Q_{Replen} = wc_p \Delta t$

Let W = 10 gph = 83.4 #/hr

$C_p = 1.0$ BTU/#-°F

$\Delta t = 88 - 65 = 23$°F (assume replen supply @ room temp.)

$Q_{Replen} = 83.4 \times 1.0 \times 23 = 1920$ BTU/hr

$$Q_{\substack{Total\\Heating}} = Q_{\substack{wall\\loss}} + Q_{\substack{top\\sens}} + Q_{\substack{top\\lat}} + Q_{replen} - Q_{pump}$$

Neglect = O
(Act = 7632 B/H)

$= 266 + 930 + 412 + 1920 = 3528$ BTU/hr

NOTE: If walls of tank were not insulated

$Q_{wall} = UA\Delta t$ where $U \cong 2.0$ or $6xU_{insul}$

$Q_{wall} = 2 \times 33 \times 23 = 1515$ BTU/hr

$Q_{tot} = 4777$ BTU/hr or $\frac{1249}{3528} =$ 35% incr w/o insul.

USE INSULATION

To find max heat gain or cooling loads for low temp processing (68°)

Let surf areas & U-factors = approx those stated previously

$Q_{\substack{wall\\gain}} = UA_s \Delta T = (0.35)(33)(75^7-68) = 81$ Btu/hr

$Q_{\substack{top\\sens}} = U^1 A_s \Delta T = 15 \times 3.1 \times 7° = 326$ Btu/hr

 STAT

$Q_{\substack{top\\lat}} = U^{11}A_s = 30 \times 3.1 = 93$ Btu/hr
Actually 0

$Q_{replen} = W c_p \Delta T = 83.4 \times 1 \times 7^{\circ} = 585$ Btu/hr

$Q_{pump} = 2544 \times hp = 2544 \times 3 = \underline{7632}$ Btu hr

$Q_{\substack{Total\\cooling}} = 8717$ Btu/hr

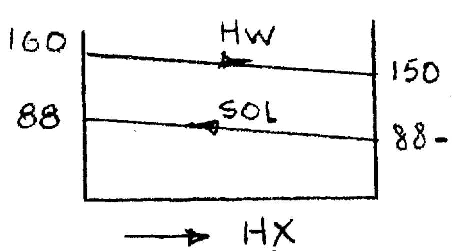
Assume hot & cold water can be supplied to HX at 160°F & 45°F respectively.

W_{HW} Reqd $= \dfrac{Q}{c_p\Delta T} = \dfrac{3600}{1.0 \times 10^{\circ} \times 500} = 0.72$ GPM

W_{CHW} Reqd $= \dfrac{8720}{10^{\circ} \times 500} = 1.75$ GPM

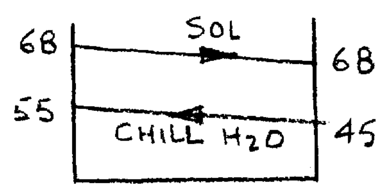
HX LMTD's where $\Delta T_{SOL} = \dfrac{Q}{W_{SOL}c_p} = \dfrac{8720}{156 \times 500 \times 1} \sim .11^{\circ}F$

HEATING



LMTD Heat = 67°

COOLING



LMTD Cool = 18°

NOTE: To determine surf area reqd, factors are LOAD (Q), U Factor, & LMTD

$Q = UA\ LMTD$

For given U

As reqd for heating $= f\left(\dfrac{Q}{LMTD}\right)$

$= \dfrac{3600}{67} = \dfrac{53.6}{U}$

& A_s reqd for cooling $= \dfrac{8720}{18} = \dfrac{484}{U}$

Therefore cooling requirement is max for A_s determination.

STAT

Surf area available in present tank design

$$A_s \text{ avail} \approx \frac{38x32}{144} = 8.45 \text{ ft}^2$$

if U = 100

then Q = UA LMTD = (100) (8.45) (18) = 15,200 Btu/hr <u>Cooling</u>

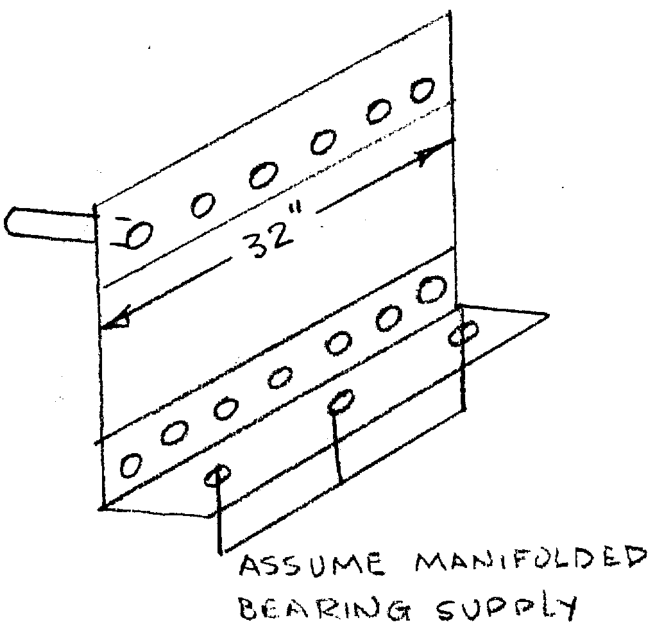
& Q = (100) (8.45) (67) = <u>56,500 B</u>/hr heating

Apparently plenty of surface is avail in present design for heating; cooling is ZX

Now check U-Factor assumed

$$U = \frac{1}{\frac{1}{h_{sol}} + \frac{t}{K_{ss}} + \frac{1}{h_{wat}}} \quad \text{where} \quad \frac{t}{k} = \frac{.06}{115} = .00052$$

$$\left(\frac{k}{t} = 1920\right)$$



At 12 GPM/bearing the bottom bearings will have a tendency to be short circuited from largest HX surface

$$A_s \text{ avail to bottom bearing flow} = \frac{3"x32}{144} = 0.665 \text{ ft}^2$$

$$V_{sol} = fps = \frac{cfs}{ft^2} = \frac{\overset{GAM}{156 \times 2.29 \times 10^{-3}}}{4 \times 32/100} = \frac{156 \times 2.29 \times 144}{128 \times 1000}$$

$V_{sol} = 0.402$ fps ; $\underline{h_{sol} = 400}$ & $h_{sol} = 200$ upper portion

Assume use of single embossed, small pass platecoil

$\approx 3/8"$ OD pipe; where $a_{cs} = 0.1 \text{ in}^2 = .000695 \text{ ft}^2$

$$V_{cw} = fps = \frac{cfs}{ft^2} = \frac{.25 \times 2.29 \times 10^{-3}}{.000695} = \frac{.25 \times 2.29}{.695} =$$

$V_{cs} = 0.82$ fps @ 0.25 GPM ; $\underline{h_{cw} = 250}$

$V_{hw} = 2.46$ fps @ 0.75 GPM ; $\underline{h_{hw} = 500}$

$$U_{cooling} = \frac{1}{\frac{1}{400} + .00052 + \frac{1}{250}} = \frac{1}{.0025 + .00052 + .004}$$

$$= \frac{1}{.00702} = \underline{142}$$

$$\& \; U_{heat} = \frac{1}{.0025 + .00052 + .002} = \frac{1}{.00502} = \underline{199}$$

Below lower bearings only

To find U-factor on wall between bearings

$$U_{cool} = \frac{1}{\frac{1}{200} + .00052 + \frac{1}{250}} = \frac{1}{.005 + .00052 + .004} = \frac{1}{.00952} = \underline{105}$$

$$U_{heat} = \frac{1}{.005 + .00052 + .002} = \frac{1}{.00752} = \underline{133}$$

$$Q_{cool} = Q_{\text{low as}} + Q_{\text{Hi as}}$$

$$= \left[(142)(.665) + (105)\left(\frac{32+28}{144}\right) \right] 18^\circ \text{ LMTD}$$

Getting close to reqmnt

$$= \left[94.5 + 40.5 \right] 18 = 9{,}000 \text{ Btu/hr cool}$$

 STAT

& $Q_{heat} = \left[(199)(.665) + (133)(4.29)\right] 67''$ LMTD

$= \left[132 + 570\right] 67 = \underline{47,000}$ Btu/hr Heat

---

To get above cooling or heating, find flow rates!

$W_{chw} = \frac{9000}{10^{\circ} \times 500} = 1.8$ GPM

NOTE: Then $V_{chw} = \frac{1.8}{.25} \times .82 = \underline{5.9 \text{ fps}}$ (Good)

$W_{hw} = \frac{47,000}{10'' \times 500} = 9.4$ GPM

NOTE: $V_{hw} = \frac{9.4}{1.8} \times 5.9 = \underline{30.8 \text{ fps}}$ (Too Hi)

$\Delta p \geq$ —— Assuming 26 passes X2-1/2' long

$\Delta p = 7X.42$ psi x $(26 \times 2.5 + 50 \times 3)$

$= 7 \times .42 \times 212 = 625$ psi

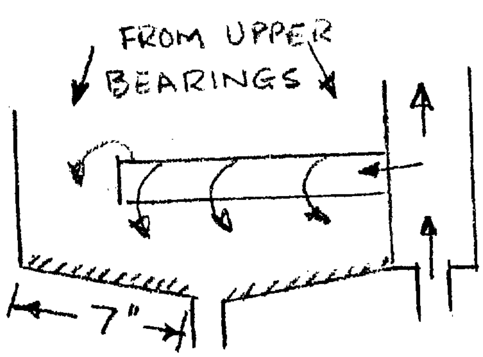
@ 1.8 GPM for cooling

$\Delta p = 7 \times .025$ psi/ft $= 1.75$ psi/ft $\times 212 = \underline{370 \text{ psi}}$

Way too high

This HX will need parallel flow paths and increased flow rates to maintain steady-state velocity & h factors.

Try HX at bottom of tank (cleaning tank may be a prob.)



A_s avail $= \frac{14 \times 32}{144} = \underline{3.1 \text{ ft}^2}$

$V_{sol} = \frac{156 \times 2.29 \times 144}{4 \times 14 \times 1000} = 0.92$ fps

$h \approx 450$

$U_C = 150$ for cooling

$U_H = 200$ for heating

8720 Reqd: not enough surface ! !

$Q_{cool} = (150)(3.1)(18) = 8,350$ Btu/hr

$Q_{heat} = (200)(3.1)(67) = 41,500$ Btu/hr

Q avail by A_s

Add pump HP to cooling loads

$W_{sol} = 156$ GPM

$P = 20$ psi (?)

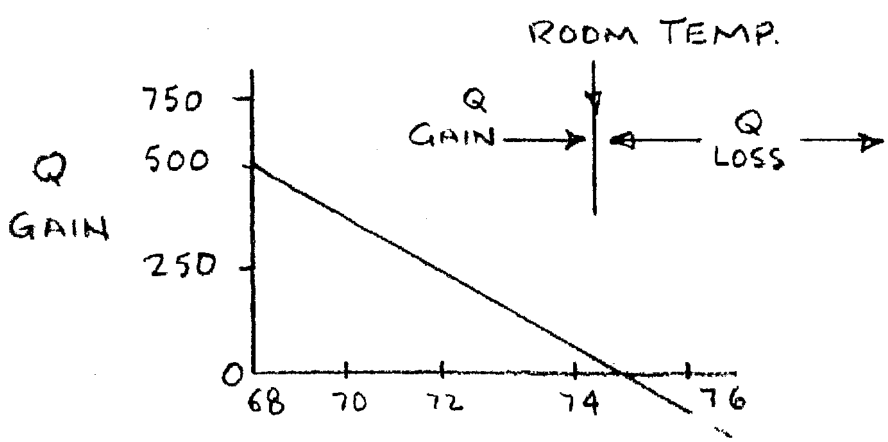
$$bhp = \frac{GPM \times P}{1714 \times Neff} = \frac{156 \times 20}{1714 \times .6} = 3.02 \text{ hp}$$ (1st try a 3HP pump is antic.)

$\therefore\ Q_{pump} = 3 \times 2544 = \underline{7632 \text{ Btu/hr}}$

What is pulldown rate w/ 9000 B/hr HX?

$Q_{wall\ \&\ top\ gains} = 0$ @ 75° solution W/ 75° room

$= 500$ B/hr @ 68° sol. W/ 75° room



Assume $Q_{replen} = 0$ since system not in oper.

let $Q_{pump} = 7630$ B/HR

$\therefore$ $Q_{total\ gain} = 8130$ B/HR @ $75^{\circ}F$ start

$Q_{avail} = 8130 + mcp\ \Delta T/hr = 9000$

$$\Delta T/hr = \frac{870}{625 \times 1} = 1.4^{\circ}F/hr$$

Pulldown from $75 \rightarrow 68^{\circ} = \frac{7}{1.4} = \underline{5 \text{ hrs}}$

Using 8350 Btu/hr bottom type HX

What is pulldown time?

$Q_{start} = Q_{pump} = 7630$

$Q_{end} = Q_{pump} + Q_{gain} = 8130$

$\Delta T/hr = 8350 = 7630 = \frac{720}{625} = 1.15^{\circ}F/hr$ @ start

$$\frac{7}{1.15} = 6+ \text{ hrs}$$

$\Delta T/hr = 8350 - 8130 = \frac{220}{625} = 0.35^{\circ}/hr$ @ end

$$\frac{7}{.35} = 20 \text{ hrs}$$

$\overline{\Delta T/hr}$ actual $\approx$ Avg $= \frac{6+20}{2} = \underline{13 \text{ hrs}}$

STAT

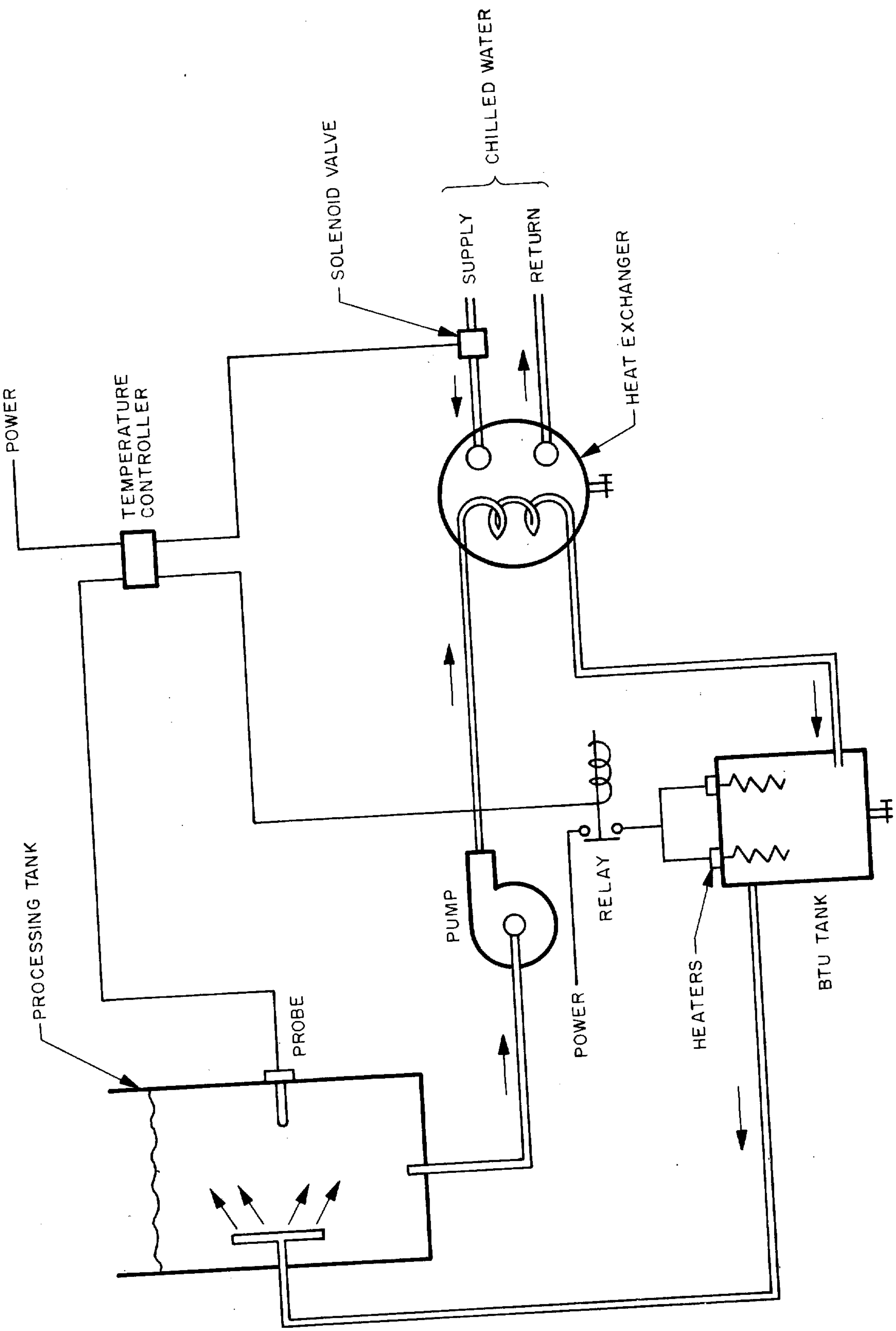

POWER
TEMPERATURE CONTROLLER
SOLENOID VALVE
CHILLED WATER
SUPPLY
RETURN
HEAT EXCHANGER
PROCESSING TANK
PROBE
PUMP
RELAY
POWER
HEATERS
BTU TANK

FIG. 1-1

STAT

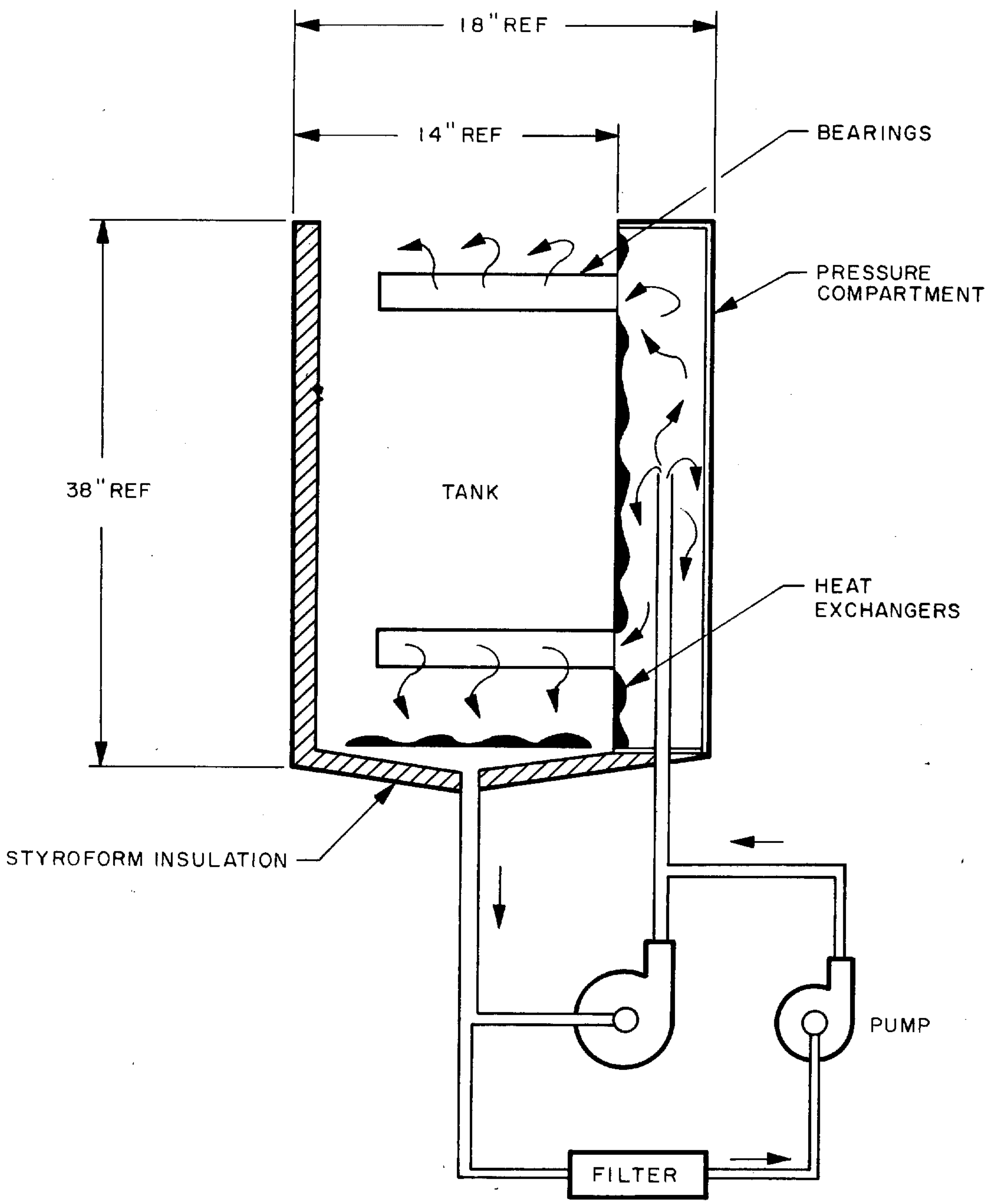

18" REF
14" REF
BEARINGS
PRESSURE
COMPARTMENT
38" REF
TANK
HEAT
EXCHANGERS
STYROFORM INSULATION
PUMP
FILTER

FIG. 2-1

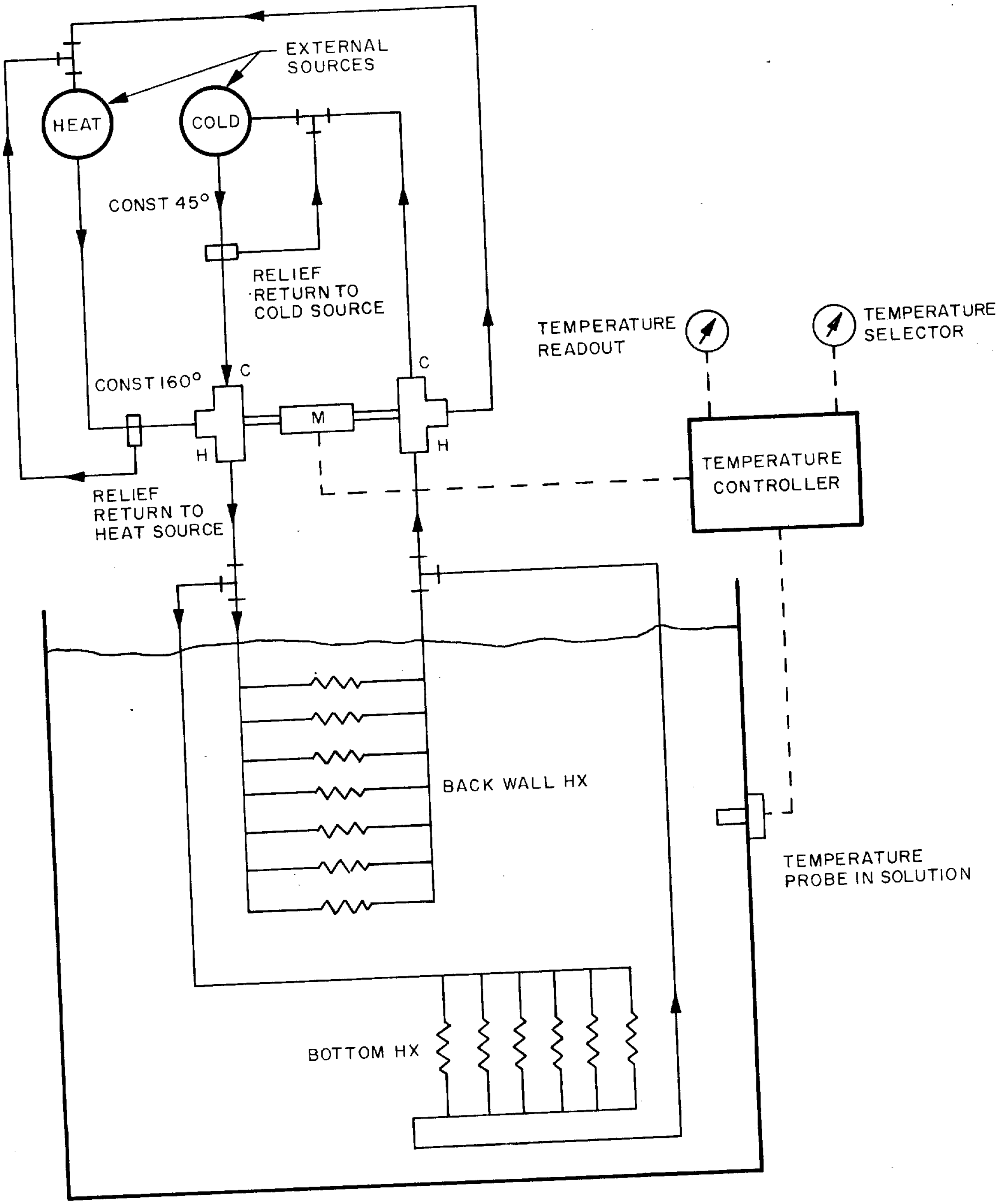

EXTERNAL SOURCES
HEAT
COLD
CONST 45°
RELIEF RETURN TO COLD SOURCE
CONST 160°
C
C
H
H
M
RELIEF RETURN TO HEAT SOURCE
TEMPERATURE READOUT
TEMPERATURE SELECTOR
TEMPERATURE CONTROLLER
BACK WALL HX
BOTTOM HX
TEMPERATURE PROBE IN SOLUTION

FIG. 2-2

STAT

REPORT

Evaluating Requirements of A Clean Room Liquid Bearing Module

STAT

Liquid Bearing
Module - Requirements for

STATINTL

February, 1965

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Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

 STAT

FOREWORD

STATINTL

[redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted]. This report should be read in conjunction with [redacted] of which it forms part.

STAT

STAT

STATINTL

Approved: [redacted]

Research Manager

ABSTRACT

Some of the design parameters which would form the basis for the design of a modular film processor compatible with clean room environments are presented. There is indication that such an approach is feasible, presenting possibility of designing flexible processors, adaptible to wide ranges of processing specifications.

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

1. INTRODUCTION

[redacted] processors have been constructed to handle all types of processing techniques over the full range of processing specifications common to the art. However, each processor has been designed and constructed for a particular type of process. Some of the more advanced processors we have constructed have included means of controlling many of the specified parameters, providing considerable flexibility and precise control of the resultant output. It is considered that, concomitant with research in processing parameters, and their effects on maintaining the quality and information content of film material, consideration should be given to a new approach to the structure of a processor adapted even more flexibly not only to control of parameters in a given processing sequence but to extend the use of a processor to all ranges of photographic emulsions. An approach to this philosophy of processing is a modular processor in which major variations can readily be introduced by changes in the number of solution tanks,and the relative times and temperatures of the emulsions contact with the solution in each stage of the process.

2. TECHNICAL DISCUSSION

2.1 General

The concept of the modular processing machine can be considered both from the point of view of the entire machine and of the component modules. In essence, there will be a basic structure including a loading module, a drying module, a base and control console. Between the load module and the drying module, will be two or more solution modules and the inter-module air bearing transfer units.

Certain basic conditions are established which must be met by the processor. These include:

Compatability with clean room operation;

Applicability to all film widths from 70mm to 9-1/2 inches;

Adaptability to the maximum number of different emulsions and processing procedures;

Controllability of temperature and all other processing parameters;

Use of liquid and air bearings for the film throughout the processing.

In addition, several criteria were assumed as preliminary design constants. These included:

Ambient clean room temperature of 68°F with sufficient capacity to eliminate requirements for exterior heat and humidity sinks.

Maximum processing solution temperature of 120°F (48.9°C).

Nominal film speed through processor 20 ft/min.

Maximum time in any solution at maximum speed - 2 min.

On the basis of the above criteria, preliminary design factors for the modular processor is outlined in the following paragraphs. It should be noted that several of the elements described are tentative and subject to modification as the result of detail design consideration and further experimentation. It is conceived that the processor as a whole will offer considerable flexibility. Adding or removing modules as required will accommodate all present and almost all advances in the art of photographic processing. In addition, each processing module in itself will be independently adaptable to modification of function or operation parameters as required to meet wide ranges of processing specifications.

A block diagram of the basic modular processor is shown on Figure 2-1.

2.2 Base and Installation:

2.2.1 Base -

The base of the modular processor will serve to support the modules in accurate alignment, while permitting the interchange and addition of components without requiring elaborate procedures to maintain the accuracy of alignment required for rapid transfer of the film through the processor. The proposed base design would be a massive rail structure including precise flat and inverted "V" pads on opposite sides, as indicated in the cross section diagrams, Figures 2-2 or 2-3.

Flat tracks would be located along the base on which retractable casters would permit raising the modules clear of the pads so that they could be rolled onto or off the base as required for a given processing sequence.

The base would also carry a master conduit containing all power lines and connections between the control console and the modules. Suitable umbilical connectors would permit direct plug-in connection of each module to the master conduit as it is installed.

2.2.2 Installation -

In most cases, installation of the base in a clean room would require permanent installation, carefully leveling and setting into the floor. Under such clean room conditions, a drain trough would be required, built into the floor, and suitable sealing means would be a part of the installation, including flush covers over the portions of the base not used in a given processing sequence assembly. This will also serve to isolate the drain trough from the clean room area.

Main solution tanks would perferably be outside the clean room area with suitable piping and pumping arrangements to bring the solution under the floor into the processing modules. Rapid disconnects would be used to connect the modules to these supplies.

In addition, overhead reservoirs would be used to contain replenishment and buffering solution which might be required to maintain constant processing parameters.

The modular processor would also be adaptable to less permanent installation without extremely stringent clean room conditions. For such a case, the base would be equipped with leveling screws and portable troughing would be provided for drainage and installation of the connections to the processing supply.

2.3 Control Module:

Two alternative control modules are considered, differing in location and connection to the processor, but identical in function. These control modules will contain all meters, control switches, parameter computers and all necessary valve controllers will be designed in accordance with accepted human engineering practice for optimum use by the operator. It may be located either in intimate association with the drying end of the processor or outside the clean room area. In either case, suitable protected cable connections will be provided to the master conduit.

Where the control console is outside the clean room area, the cable will be brought to it under the clean room floor. If the console is associated with the dry end, additional interlocks will be provided so that all illuminating lights on any of the meters or controls will be out if one of the light tight covers of any of the modules is opened.

2.4 Solution Module:

The solution modules will be basically insulated stainless steel tanks in which the processing solutions and liquid bearings are arranged to provide path lengths varying from approximately 480 inches to as short as approximately 76 inches, to provide for various times in given processing solutions. The maximum length provides for two minutes processing at 20 ft/minute film speed. In processes where specifications called for a greater duration than this, the film speed will be reduced as required, or one or more duplicating modules containing the same solution will be inserted in the modular process. There are, however, considerations other than ability to contain the processing solution and vision of the required path length to be considered in the design of the modules.

2.4.1 Mechanical Structure -

An important feature of the modular processor will be the accurate alignment of all the components. Each module will be constructed as a rigid structural framework resting on carefully aligned pads shaped to mode with the pad on the base in accurate alignment to the required tolerances. All bearings within the solution will be referred to the structural members of this framework and precisely aligned to the pads. Thus, minimal specific alignment procedures will be required in adding additional modules or removing any one module and replacing it with another. In addition, pins and sockets will also be provided on suitable vertical members to insure pairs of modules accurately locking to each other.

The structural framework will be also support any pumps, motors, power supplies as well as the stainless steel tank lining and the exterior shrouding, (this last will be provided with suitable soft plastic base skirts to provide the required contact with the flooring as required for clean room operation).

2.4.2 Tank Lining and Shrouding -

Since it is proposed to have a common modular structure for all tanks in the processor, this may involve solutions ranging in pH from 2.0 to 14.0 and solution temperatures up to 120°F. It is considered advisable to use only selected stainless steels in contact with the solutions. This dictates the material for module linings. In general, the tank section of the module will be divided into two compartments; a reservoir or plenum and the active processing compartment through which the film will pass over the liquid bearings. These compartments will be deep drawn or smooth welded with all corners suitably rounded for ready cleaning except at such orifices at which the contour will be controlled to regulate rates of flow. A small sump section will be provided in the processing section from which the dump drain and any pipe connections for bearing or filter pumps will originate. The tank liner will, of course, be rigidly attached to the structural framework of the module using only connective members which will minimize conductive transfer of heat from the tank lining to the structural member. This tank section will be surrounded by external shrouding which will merge into the total modular shrouds. A minimum space of 2 inches between the external shrouding and the tank liner will be filled with epoxy foam, foamed in position to provide a very high degree of thermal insulation between the contents of the tank and the ambient atmosphere. The material used for the shrouding may be either stainless steel or a plastic laminate, depending upon which is determined most suitable for clean room operation.

The only access to space within the shroud but below the tank proper, in which will be located any required pumps, power supplies, and electrical connections, will be through a single flush-access door on the back of each module.

A removable top cover will be provided, again using stainless steel for their inner surface and the same exterior shrouding material, with the space between filled with foamed in place epoxy. The mating edges between the top of the tank and the cover will be designed to form a light trap in which all changes in direction are curves suitable to maintaining clean. Provision will be made in this top cover for the proper installation of the modular air bearing structures which will be used for intertank transport of film.

2.4.3 Liquid Bearing -

The solution modules contemplate two alternative types of liquid bearings for the support of the film through the processor. Each tank will contain 13 bearings in an array; 6 across the top, and 7 across the bottom. Threading over all provides for the maximum film length of approximately 480 inches. Both type bearings will be designed to maintain centering of any width of film from 70mm to 9-1/2 inches. They also will be designed to permit modification of the loop over the bearing from 180° to 90°, as would be required for variations in threading path to accommodate shorter lengths of film in a given tank module. However, these two types of bearings operate on quite different principles and possess quite different hydrodynamic characteristics and, hence, other elements of the modules will be affected by the final determination as to which type of bearing will be used. This decision will depend upon further detailed analysis and experimentation.

2.4.3.1 Jet Tank Bearings -

The standard liquid bearing which has been incorporated in other [redacted] processors, designated as jet type bearing, supports the film by the efflux of liquid through shaped apertures in the bearing structure. Although there are several variations of the aperture configuration being considered, the general principles of this type of bearing is fairly consistent and requires the efflux of approximately12 gallons per minute of liquid for proper support of film. The pump required to produce this volume of flow is quite substantial. In tanks using these bearings, the structure will be essentially as shown in the cross section diagram, Figure 2-2. The bearings will be supported by a relatively massive plate directly supported by the structural members of the module. The tank lining will merge with and be smoothly connected to the base of this plate. This will provide a continuous separator between the plenum and processing sections of the tank, interconnected only through the orifices in the bearings and the bearing pump connections.

The bearing pump will be selected to provide the necessary head to insure the movement of solution through all the bearings with sufficient velocity. In order to equalize the support provided by each bearing, two design elements are included. The heat exchanger within the plenum will be designed to also serve as an equalizing baffle so that none of the bearing ports will receive the full force of the stream from the bearing pump. In addition, provision will be made for inserting suitably sized and shaped orifices at each bearing intake to adjust the relative efflux through each bearings as required to maintain proper bearing support at each station under all threading paths.

In solution modules using this type of bearing it will also be necessary to employ a supplementary filtering pump. This will bypass some of the solution around the bearing pump, returning it to the plenum after passing through a filter which will remove all solid particles larger than one micron. The capacity of the filter pump and filter will be such that the entire tank full of solution will pass through the filter in approximately 15 minutes. Pressure affects of the filter pump will add essentially negligible boost to the bearing pump.

2.4.3.2 "Rotatron" Bearings

A new type of liquid bearing is being considered for use in this modular tank. It consists of a rotating fan structure which supports the film by the pumping action produced in the liquid. A diagramatic cross-section of a tank using these bearings is shown as Figure 2-3. The development and use of this type of bearing will simplify the structure of the solution module in several significant factors. It will be possible to reduce the mass of solution required in a given tank module and to eliminate several external pumps. It also modifies and simplifies the internal structure of the tank. These bearings again will be supported by a rigid wall supported by the structural members of the module but with sealed shafts passing through the plenum driven by motors mounted within the shrouding, outside the tank. This partitiion now can also support the heat exchanger and a filter element. The only additional circulating pump would be a completely.immersed centrifugal pump within the plenum section of the tank module. Since it would not be necessary to maintain a considerable pressure difference between plenum and processing sections of the tank, the capacity of this pump would be sufficient only to insure the suitable rate of passage over the heat exchanger and through the filters.

2.4.4 Heat Exchanger -

One of the important features of the modular processor will be an independent control of temperature in various solutions used in a given process specification, as well as the ability to accurately maintain the solution temperatures at the values called for by the specification. It is, therefore, necessary to have within each module a heat exchanger suitable to maintain this specified temperature against an ambient controlled clean room temperature of 68°F. In most cases, even though the solutions are pre-conditioned in the supply tanks to a specified temperature it will be necessary to add or remove additional heat. Most rapid process specifications call for elevating the temperature to relatively high values. Consequently, heating capacity to maintain a temperature up to 120°F will be provided. However, considerable heat will be dissipated to the solution by the pumps, and, therefore, provisions must be also made for withdrawing additional heat to maintain the solutions at temperatures lower or close to the ambient. It is intended to employ a plate-type heat exchanger for this purpose in which heat will be added as required directly by resistive heating elements within the heat exchanger plate. This plate will be designed to also contain either tubular structures for liquid refrigerant or thermo-electric cooling elements.

In general, the required heat absorbing capacity of the heat exchanger plate will be much less than the required heating capacity to meet the specified extreme control temperatures. It is considered that adequate cooling capacity could be achieved with either a small compressor type refrigeration unit or a thermo-electric heat exchanger unit mounted in the space beneath the tank section of each solution module. It would then dissipate the removed heat to the space directly above the drain trough.

Heat replacement requirements, however, will be quite large even though the most efficient type of thermal insulation will be used around the tanks. One of the largest heat loss sources, which must be compensated for by the heat exchanger, arise from evaporative losses which result from the air bearing intertank connection. At the maximum temperature of 120°F this loss may be as large as 1/2 pound of water per minute. This would require an input of 500 BTU/per minute to maintain the required 120°F constant solution temperature if no other losses were present.

2.4.5 Sensors and Controls -

Provision will be made in each tank module for installation of proper sensors and controls to maintain the specified processing parameters in all solutions. Some of these, such as solution temperature and solution depth controls, will be common to all modules and will be built into the module in a manner compatible with their continued proper function.

Other sensors and controllers would be required for particular solutions, as for example: pH monitoring, residual soluble silver detection, sulfur detection, etc. For such sensors and monitors, provisions will be made for ready installation as required. In all cases, the leads from these elements, as well as the power leads for each module, will be included in a common umbilical cable which will plug into the master conduit.

2.5 Air Bearing Transfer Modules:

Between each module there will be a transfer station in which the film must be supported above the height of the module cover. This will be accomplished with the use of small modular air bearing components. These will engage in provided sockets in the module's mechanical structure. The air bearings used for the transfer will be based on one of the various types of air bearings designed by

STATINTL [redacted]. Each module will be a self-contained unit consisting of the air bearing proper, its blower and plenum. For the transfer from or to a solution module, a lip of the air bearing will be below the solution surface constraining the lifting force of the air under the film, limiting film flutter. However, it should be noted that the flow rate of air through a conventional air bearing is of the magnitude of 150 cu. feet per minute and in such a case, this volume of air will impinge on solution surface. At maximum solution temperature of 120°F, this volume of air will absorb water vapor and, hence, promote evaporation at up to 1/2 pound per minute. The heat loss from the solution due to this rapid evaporation rate results in major heat loss in each module.

In some arrays, where a large number of tank modules may be used for a given specified process, film drive from the takeup end of the processor may not satisfactorily maintain travel of the film through the entire processor without excess tension. In such a case, one of the air bearing modules will be replaced with a vacuum capstan interchangeable with the air bearing module. The capstan will be carefully synchronized to the main drive to assist the passage of the film. It should be noted that if "rotatron" liquid bearings are used, these impart to the film a tendency to move and additional drive assistance will not be required. In fact, by having one or more counter-rotating bearings in each module it may be possible to carefully control the tension of the film while maintaining very uniform motion through the entire processing array.

2.6 Load and Drying Modules:

The load and drying modules will be common to all processor assemblies. Consideration may be given to permanently mounting the drying module as a component part of the base structure.

On the other hand, the load module as the lightest component, particularly when empty, will be used as a file closer. That is, it will swing out of position to permit mounting or removing solution modules as required. The load module will be designed to accept all standard size reels and magazines up to 4,000 feet capacity in all widths from 70mm to 9-1/2 inches.

The drying modules will primarily be conventional forced-air and heat dry labyrinth, vented to the exterior in a conventional manner. It is, however, conceived that within the same modular form, modified drying techniques such as vacuum assisted and desiccant assisted drying techniques may be incorporated as they become further developed.

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3. CONCLUSIONS AND RECOMMENDATIONS

The preliminary considerations of the elements which would be involved in the design of a modular processor leads to the belief that such a unit would be feasible and a desirable addition to the art of processing photographic materials. The preceding discussion has considered very superficially many of the factors which would have to be further studied in considerable detail before producing definitive design of such a processor. It is recommended, therefore, that study of adapting various control parameters and techniques which previously have been incorporated in other [redacted] processors, be considered for incorporation in such modules. Further research should be conducted on the most efficient means of implementing the heat exchange requirements. Design studies and experimentation should continue to determine the advantages of the various types of liquid bearings and air bearing transfer modules to such a modular concept of processor design.

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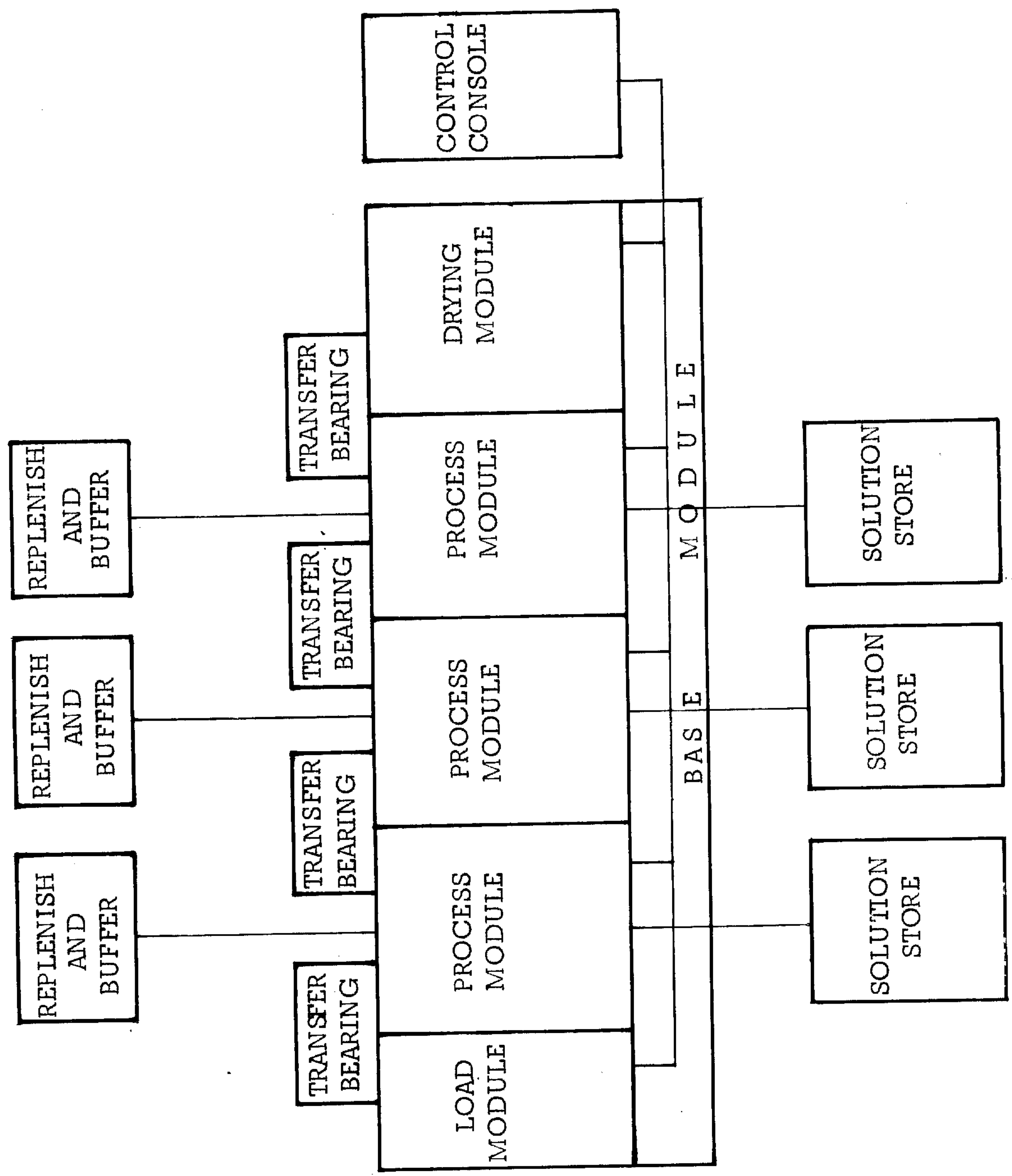

Figure 2-1. MODULAR PROCESSOR - BLOCK DIAGRAM

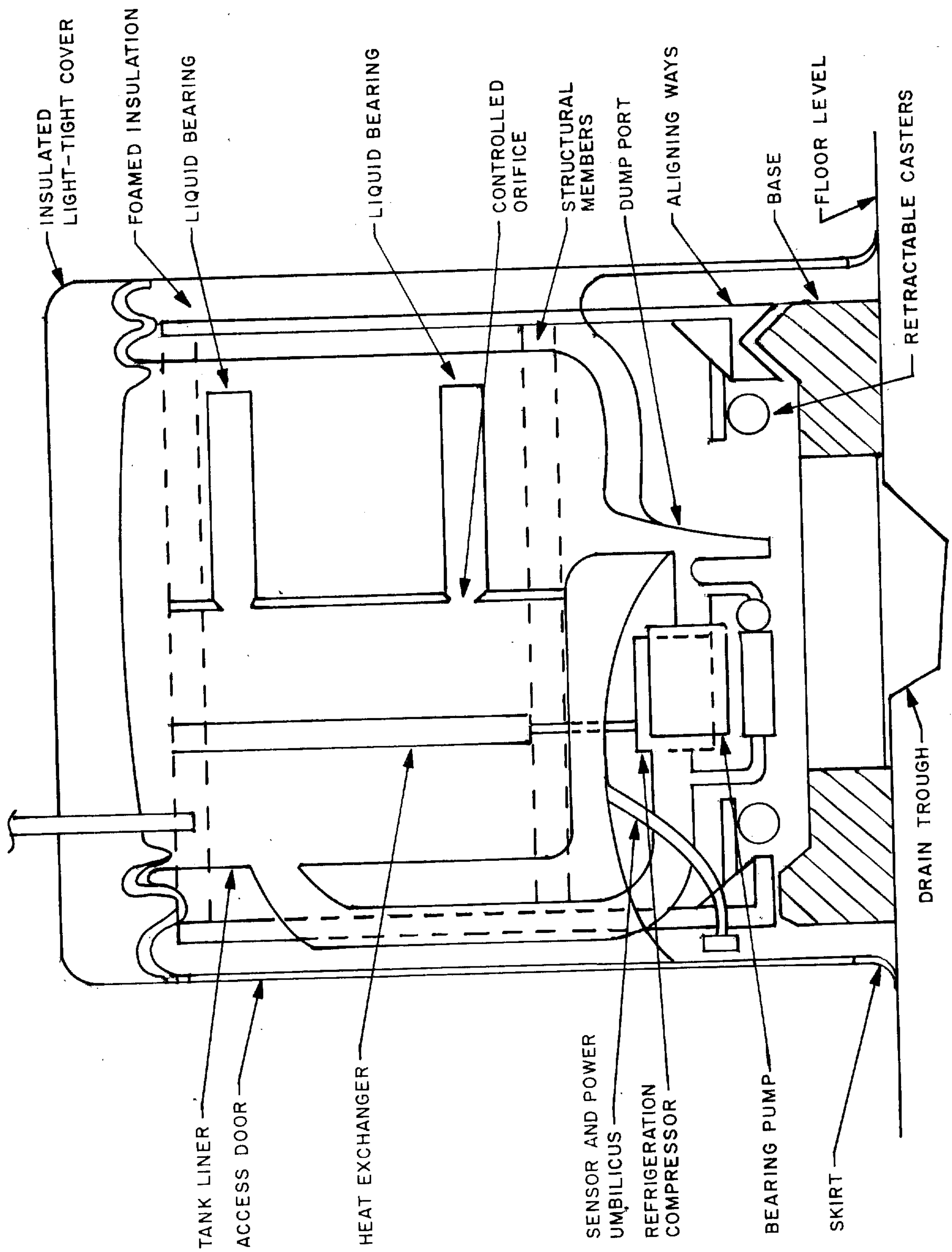

Figure 2-2. DIAGRAMMATIC SECTION-SOLUTION MODULE WITH CURRENT JET-TYPE BEARINGS

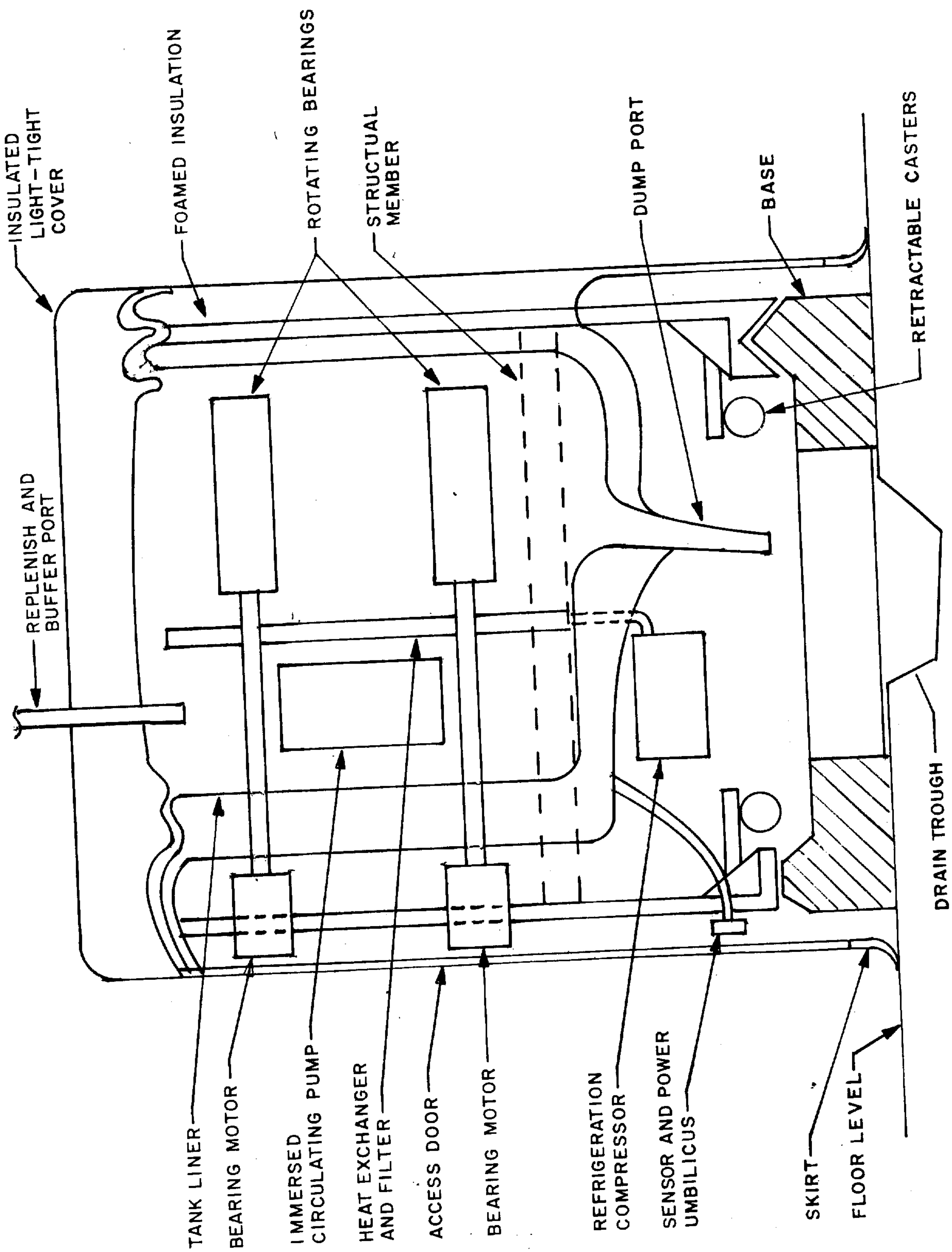

Figure 2-3. DIAGRAMMATIC SECTION-SOLUTION MODULE WITH "ROTATRON" BEARINGS

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REPORT

TO DETERMINE THE COEFFICIENT OF FRICTION OF FILM

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Coefficient of Friction of Film

STATINTL

February 1965

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

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FOREWORD

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[redacted] submits this report in compliance with Item 4-2 of the Development Objectives of [redacted]. This report should be read in conjunction with Report [redacted] of which it forms part.

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Research Manager

ABSTRACT

This assignment was issued to determine one of the major contributory factors to film tension and therefore, bearing loads and capstan torque. Other factors examined elsewhere are the forces required to bend film over the bearings and those devices required to provide tension for tracking.

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1. INTRODUCTION

In conventional processing machines, the film is moved over a series of driven and idler rollers and clutches to control the tension of the film between rollers. The skin friction of film in these machines is of such a low value when compared to the torque at each driven roller that it can be ignored. Since the air or liquid cushion over which the film is transported is virtually frictionless in a liquid/air bearing processor, the tension in the film is generated from another source. In the HTA-5 this tension was evidenced only during transport; there was no tension in the film at rest.

Preliminary work in a swimming pool with 20-foot lengths of film indicated that for 20 feet per minute, a 3-ounce tension was generated by drag forces. This figure converts to 24 ounces for the length of film in the HTA-5, or a bearing load of 3 pounds. The cushion depth of the last bearing in the HTA-5 was measured while film was being transported at a speed of 20 fpm. A similar bearing was mounted on a test stand, and using the processor blower, with the output adjusted to an identical manometer pressure, the cushion was loaded until an equivalent depth was obtained (Figure 1-1). A total weight of 3-3/4 pounds was required. If the additional bending forces are considered, this weight is compatible with the 3-pound load extrapolated from the swimming pool tests.

2. TECHNICAL DISCUSSION

2.1 COEFFICIENT OF FRICTION

To obtain accurate figures of the coefficient of friction of film, a tow tank was considered; however, to keep the percentage of errors low

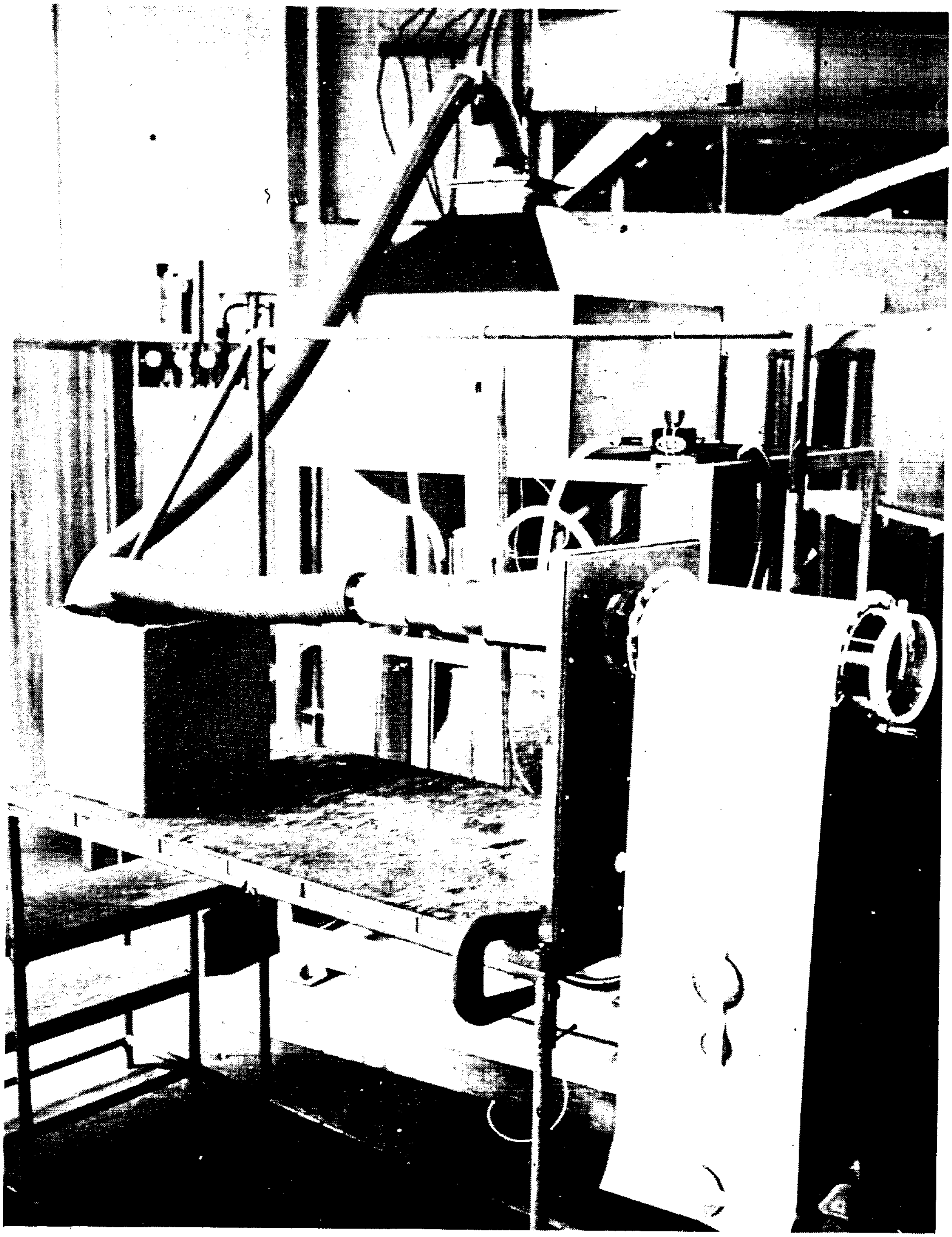

Figure 1-1. Air-Bearing Test Setup

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in comparison with total friction values, a long length of film should be towed. A tank large enough to permit such a test would have to be at least 120 feet long and therefore impractical and too costly. As a practical alternative, a floating test rig was designed and built (Figure 2-1). The float was constructed of 1/4-inch thick commercial plywood and filled with styrofoam sheets.

The test gear mounted on the float consisted essentially of an electric drive motor with a speed control. The motor was used to drive a pulley of exactly 1-foot circumference. A small takeup motor and reel were also used.

The length of test film was attached to a monofilament nylon line which then passed under the float and over a roller to the pulley. The line was wrapped one complete turn around the pulley and secured to the takeup motor reel. At each revolution of the pulley a microswitch closed to operate a digital counter. A stopwatch was used to make an accurate adjustment of the film wind-in rate.

The test rig was attached through a dynameter to a fixed point so that the drag pull on the line when the film was wound in could be read as a reactionary force on the test-rig anchor point (Figure 2-2).

2.2 RESULTS

Tests have been obtained from trial use of the test rig at a marina where power and facilities were available; however, due to winds, passing vessels, and a strong, variable tide, these figures are too inaccurate to be used. An inland body of still water has been located and a new series of tests will be started as soon as the present work load permits.

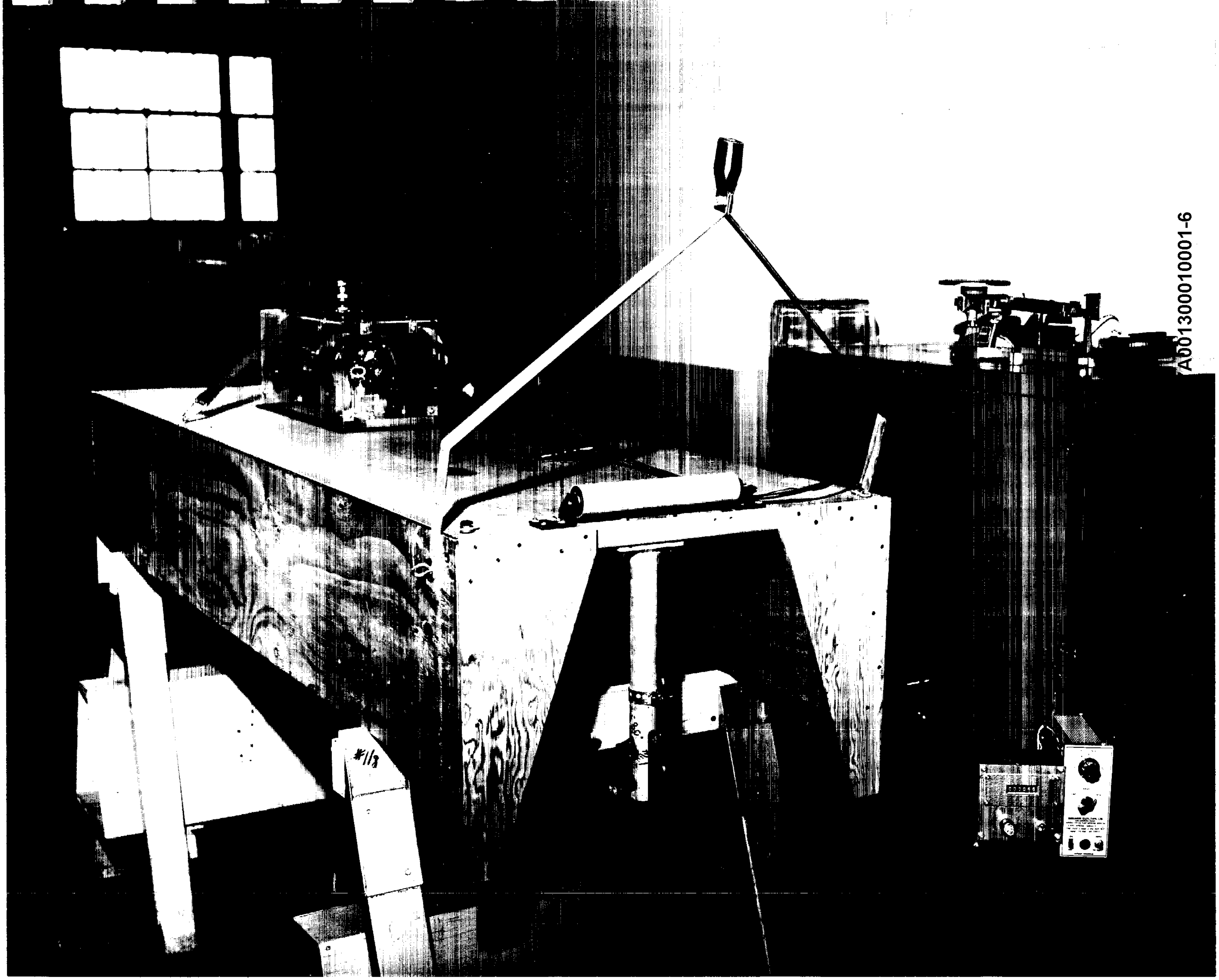

FIG. 2-1 Floating Test Rig

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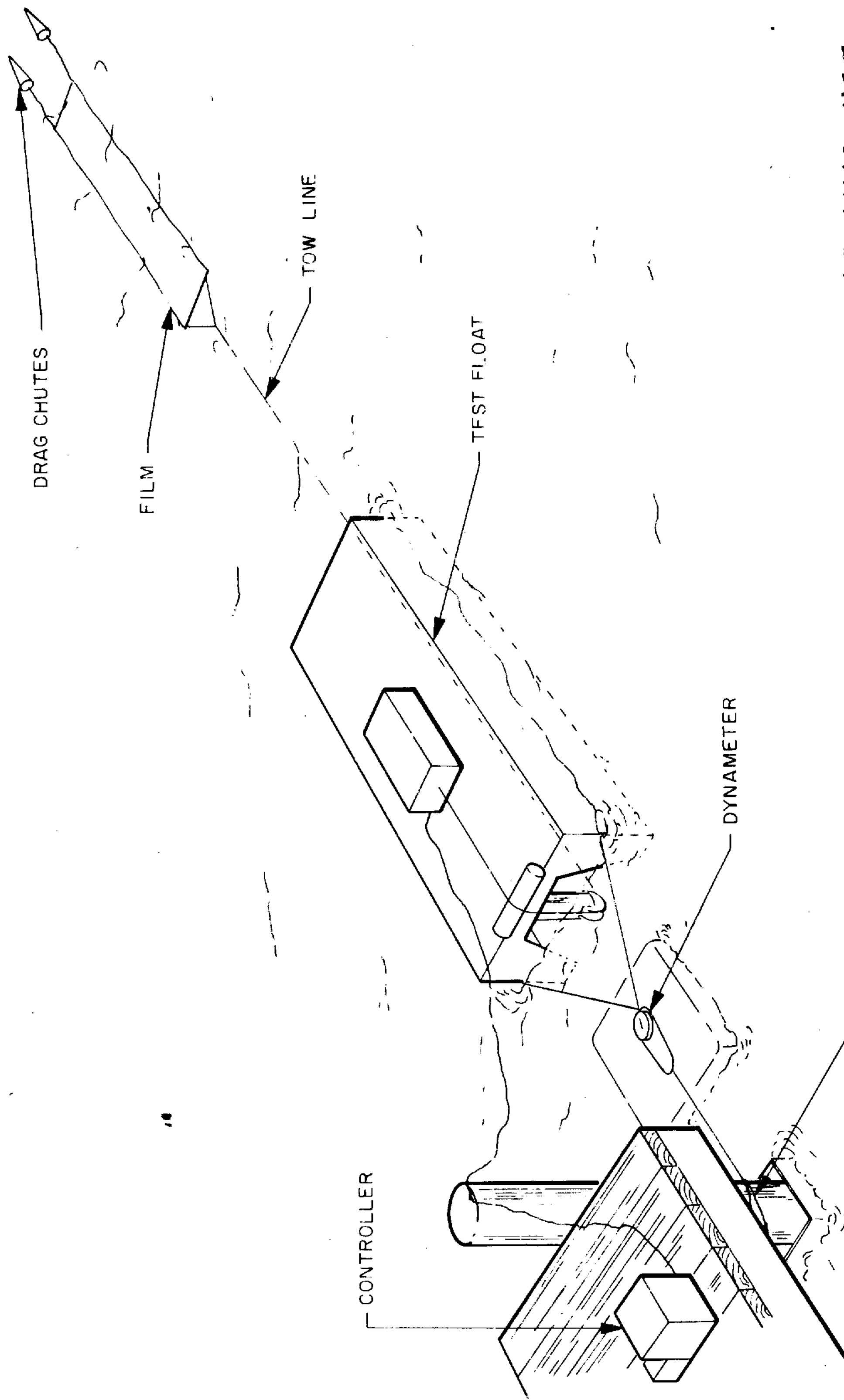


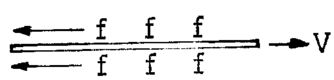
DIAGRAM SHOWING USE OF FLOATING TEST RIG.

FIG. 2-2

The skin function values obtained from this test series will be used as the C_F factor in the following formula applicable to smooth flat plates. To this formula will be integrated a constant for the bending force of the film around the bearing diameter used, times the total of bearings in the processor.

3. MATHEMATICAL FORMULA

When any body is moved through water at a velocity V there is resisting force F exerted by the water on the body moving through it. In the case of film or thin flat plates of little or no frontal area, this force is known as skin friction.



This force may be calculated from the following equation:

$$F = C_F \frac{\rho A V^2}{2} \qquad (1)$$

where

F is the skin friction force in lbs, C_F is a dimensionless coefficient, ρ is the mass density of the liquid in slug/ft^3, A is the area in sq ft, and V is the velocity in ft/sec

Note ρ of water at 20°C
is 1.937 slug/ft^3
or $\frac{1.937 \text{ lbs f} \quad \text{sec}^2}{\text{ft}^4}$

The drag coefficient C_f is mainly proportional to the Reynolds number

$$R, N, = \frac{VL}{v} \qquad (2)$$

where

V is velocity in lb/sec

L is length

v is kinematic viscosity in ft^2/sec

v for water at 20°C is 1.08×10^{-5} (this number is very temperature-sensitive).

For smooth, flat, plates the coefficient C_F has been approximated as follows:

$1 \underline{L} RN \underline{L} 5 \times 10^5$

$$C_F = \frac{1.327}{\sqrt{R.N}} \qquad (3)$$

$5 \times 10^5 \underline{L} R.N \underline{L} 5 \times 10^6$

$$C_F = \frac{0.074}{\sqrt[5]{R.N}} - \frac{1700}{R.N.} \qquad (4)$$

and for RN 5×10^6,

$$C_F = \frac{0.455}{\left[\log_{10} (R.N.)\right]^{2.58}} \qquad (5)$$

However, it is not known how well these approximations apply to film in water, developer, and fix solutions.

There are several things to note carefully.

The entire area of the film - both sides included - that is in solution between drive points must be considered.

The drag varies according to the square of the velocity. Double the speed and get 4 times the drag.

In air-liquid bearing machines, the load on bearings is due almost completely to the fluid friction drag. Only that part of film tension due to tensioning devices is not caused by fluid friction. The tension is cumulative from tank to tank and is greatest near the tail end of the machine.

It is instructive to compute an example according to the formulas above to understand why the above discussion is pertinent and why additional research is needed.

The HTA-5 had 160 feet of 9-1/2-inch wide film submerged from head end to tail end. At 20 feet/min the drag force was, according to formula

$$F = C_F \frac{\rho A V^2}{2}$$

First compute R.N. to evaluate C_F

$$R.N. = \frac{VL}{v} = \frac{\frac{20}{60} \times 160}{1.08 \times 10^{-5}}$$

$$= \frac{0.33 \times 1.60}{1.08} \times 10^7 = 4.9 \times 10^6$$

since this is very close to 5×10^6 both (4) and (5) forms of C_ρ will be evaluated.

$$C_F = \frac{0.074}{\sqrt[5]{R.N.}} - \frac{1700}{R.N.} \qquad (4)$$

$$= \frac{0.074}{\sqrt[5]{49 \times 10}} - \frac{1700}{4.9 \times 10^6}$$

$$= \frac{0.074}{21.8} - \frac{1700}{4.9 \times 10^6}$$

$$3.4 \times 10^{-3} - 0.348 \times 10^{-3} = 3.05 \times 10^{-3}$$

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or

$$C_F = \frac{0.455}{\left[\log_{10} (R.N.)\right]^{2.58}} \tag{5}$$

$$C_F = \frac{0.455}{\left[\log_{10} 4.9 \times 10^6\right]^{2.58}} = \frac{0.455}{[6.690]^{2.58}}$$

$$= \frac{0.455}{137} = \frac{4.55 \times 10^{-1}}{1.37 \times 10^{2}} = 3.31 \times 10^{-3}$$

We shall use the slightly larger (10 percent) value of 3.31 for computation of F.

$$F = \frac{3.31 \times 10^{-3} \times 1.937 \ (160 \times 0.70 \times 2) \times \left(\frac{20}{60}\right)^2}{2}$$

$$= \frac{3.31 \times 1.937 \times 2.53 \times 1.11}{2} \times 10^{-2}$$

$$= 9.0 \times 10^{-2} = .09 \ \text{lbs.}$$

However, in the HTA-5, drags of almost 3 pounds were experienced.

Some of this was the accumulator tension and some of this was higher fluid friction than the equation predicts, thus pointing up the need for experimental work to get the C_F for film instead of smooth flat plates.

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VERTICAL SPACING OF LIQUID BEARINGS

STATINTL

February 1965

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ertical Spacing of Liquid Bearings

FOREWORD

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[redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted] This report should be read in conjunction with [redacted] of which it forms a part.

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Approved: [redacted]

Research Manager

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ABSTRACT

The primary objective was to determine a suitable vertical spacing for liquid bearings. In order to meet this requirement, a mockup test stand was constructed. Two upper rollers and one lower roller were mounted on the test stand to simulate a film transport system. Type 4400 film was selected and was prewet to simulate processing conditions. Small increments of weight were applied to one end of the film until the force required to maintain a safe operating separation between the sides of the film loops was determined.

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1. INTRODUCTION

Tests were conducted to determine the maximum vertical bearing spacing required to eliminate any damaging effects to film resulting from solution turbulence caused by liquid bearings. The following liquid-bearing sizes were investigated: 1-1/2, 1-3/4, and 2 inch diameters, plus a 1/8-inch liquid cushion height.

2. TECHNICAL DISCUSSION

2.1 MATERIALS AND EQUIPMENT

Principal materials and equipment used for experiments and tests are listed below:

1) Mockup test stand with an adjustable frame which controlled the vertical positioning of the upper liquid bearings to a maximum height of 3 feet (Figure 2-1).

2) 1-1/2, 1-3/4, and 2-inch diameter rubber and PVC rollers, plus a 1/8-inch liquid cushion height.

3) Type 4400 film, 9-1/2 inches wide by 12 feet in length

4) Small lead weights

5) Force gage indicator, Hunter Spring Co., Model No. L-20, Serial No. 1170, zero to 20 pounds.

2.2 EXPERIMENTAL PROCEDURE

Twelve-foot lengths of Type 4400 film were soaked in 90°F water for five minutes to approximate the flexibility of film undergoing standard processing. The film was then threaded on the roller array. A 180-degree film wrap on each bearing was precisely maintained throughout the test series to automatically determine the correct horizontal spacing regardless of bearing diameter.

To simulate film agitation or the fluttering action inherent with liquid bearings, a small air blower was held about one inch from the surface of the film. This proved inadequate for simulating the desired effect. To achieve the desired effect, small weights were applied to one end of the film until the weight was sufficient to remove the slack between loops on the adjacent rollers. Once this weight was established, a force gage was slowly pressed against the surface of the film until it touched the adjacent loop. A reading of 12.8 ounces was recorded. This 12.8-ounce reading was considered a constant. In each subsequent test, regardless of the diameter or vertical separation of the rollers, weights were added to one end of the wet film until the force gage read 12.8 ounces (Figure 2-2).

2.3 RESULTS

Using vertical heights of 24 and 36 inches and applying the same 12.8-ounce force, the amount of weight required to maintain safe separation between the loops of film varied according to the diameter of the roller used and the height selected. Typical results are given below:

1) 36-inch vertical height:
   1788.3 gr for 1-3/4-inch roller
   1634.5 gr for 2-inch roller
   1567.3 gr for 2-1/4-inch roller

2) 24-inch vertical height:
   1324.3 gr for 1-3/4 inch roller
   1170.5 gr for 2-inch roller
   1119.8 gr for 2-1/4-inch roller

3. CONCLUSIONS AND RECOMMENDATIONS

3.1 VERTICAL SPACING

While the test conducted under these simulated conditions indicated that a vertical distance of 36 inches between liquid bearings would be feasible (using any of the bearing diameters tested and assuming that the proper tension is applied to the transport system) it is by no means conclusive.

The static condition of the test rack cannot fairly reproduce the dynamic effects encountered in an operating processor. Other experiments performed in conjunction with this phase of the contract clearly demonstrated the powerful cohesive forces generated by the Bernoulli effect of liquid moving between the film loops.

It is strongly recommended, therefore, that the test be repeated with moving film, different loads, and different types of film before definitive results can be assured.

Predictably, for each of the two vertical bearing spacings, the smaller diameters required greater force than the larger. This agrees with our research on the forces required to attain certain specified bend radii. The percentage increase in force was approximately linear with the same increase in diameter at a constant vertical spacing and, again, approximately linear with increase in vertical spacing (diameter remaining constant). This is shown graphically in Figures 3-1 and 3-2.

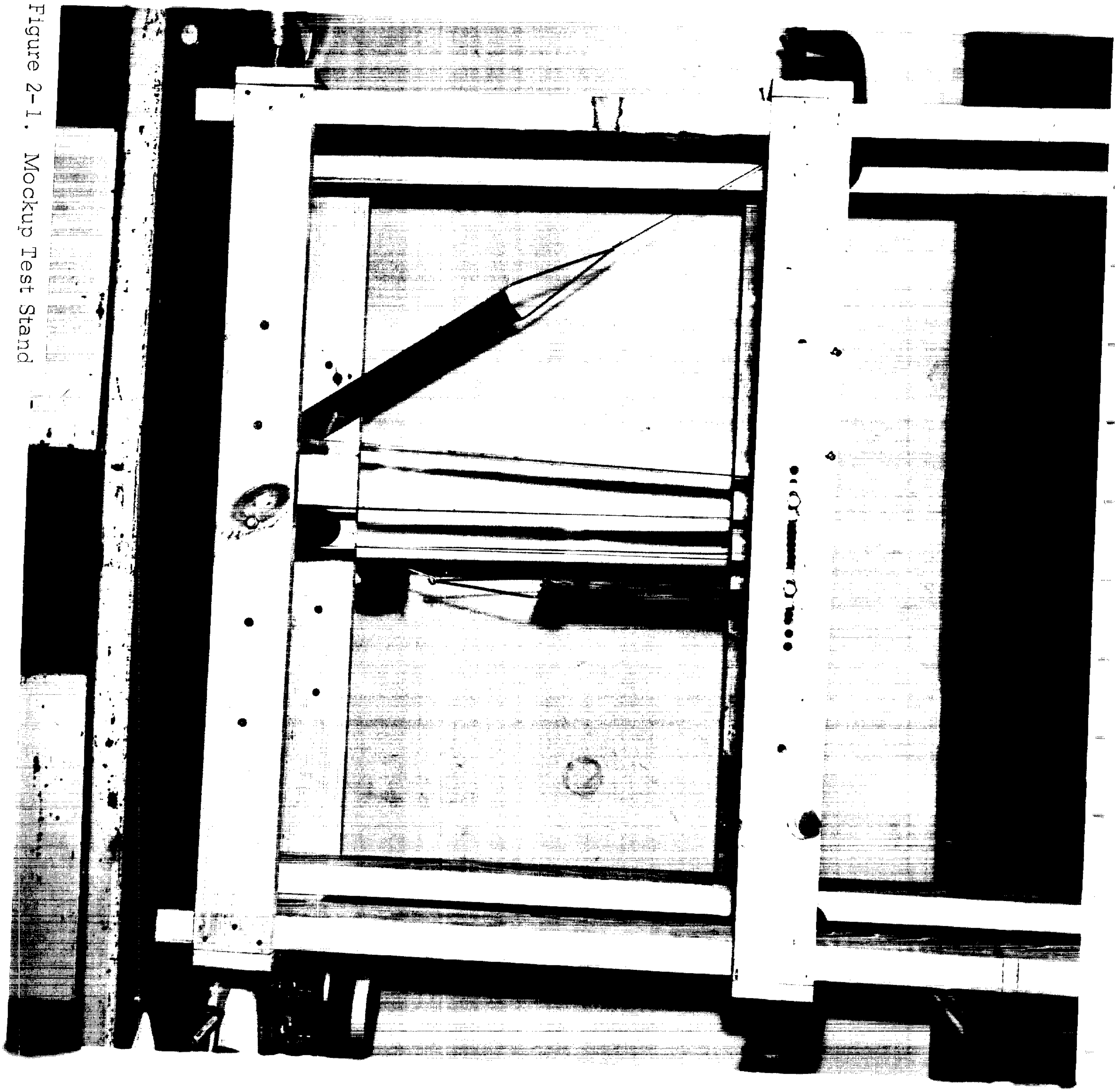

Figure 2-1. Mockup Test Stand

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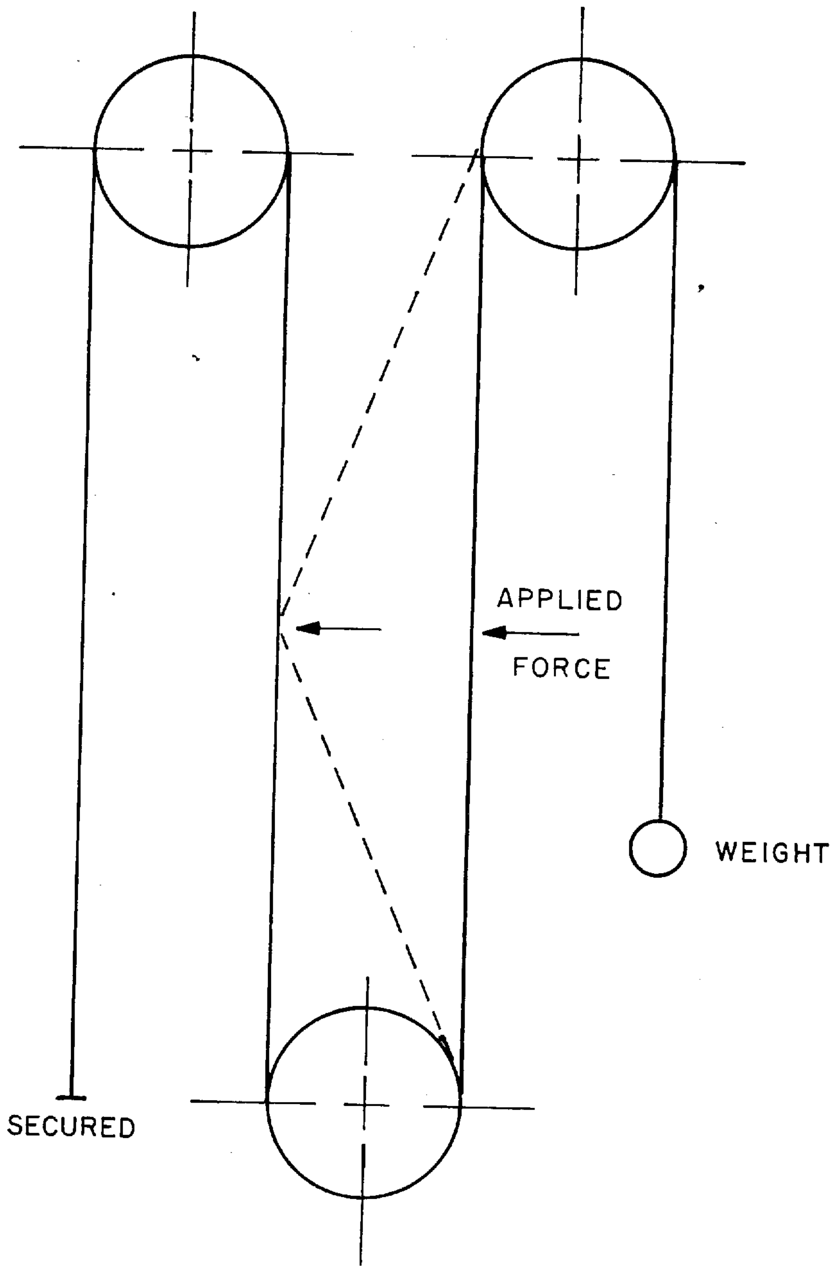

Figure 2-2. Schematic Test Apparatus

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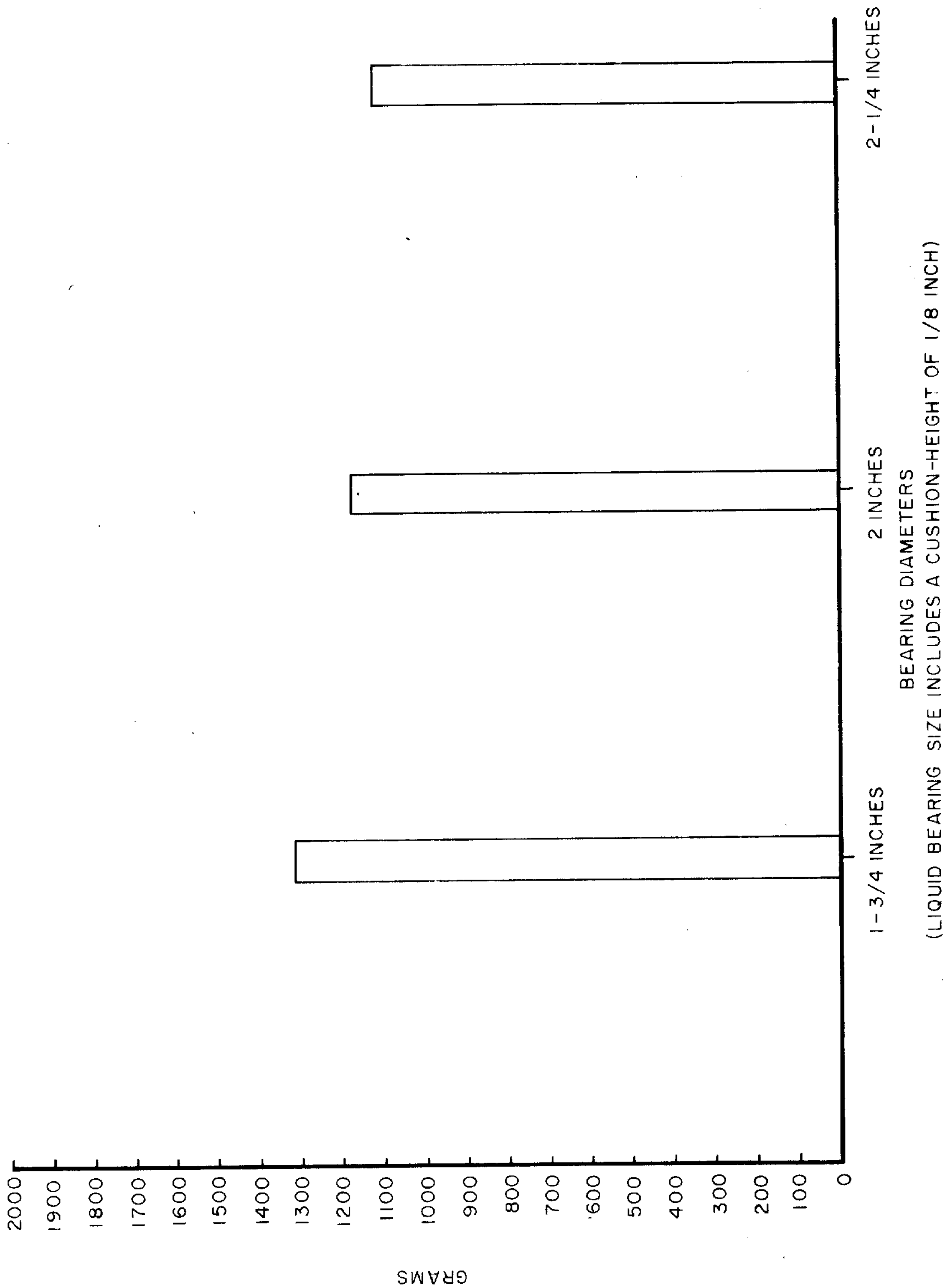

Figure 3-1. 24 Inch Vertical Height

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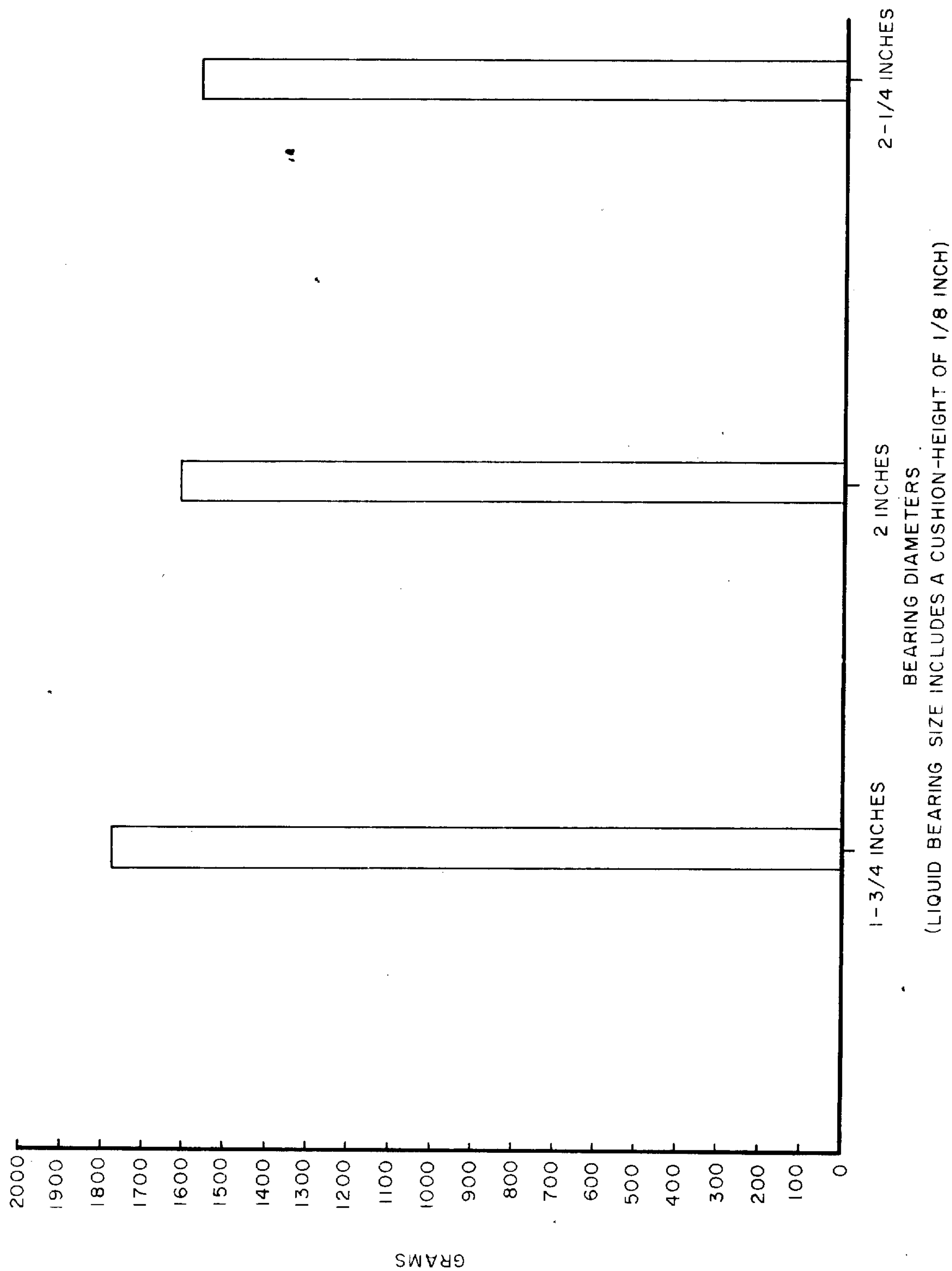

Figure 3-2. 36 Inch Vertical Height

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Air Bearings

Intertank

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DESIGNING AND TESTING INTERTANK AIR BEARINGS

STATINTL

February 1965

FOREWORD

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ABSTRACT

The two objectives of this task are to study the possibility of designing an improved, plenum-type air bearing that is particularly suitable for a clean room environment and, making a radical departure from this type of bearing, to design one that incorporates its own blower fan.

1. INTRODUCTION

The type of air bearing presently used consists of a tube into which air is pumped and then released through slots or holes in the periphery. Adjustable flanges are provided on each side of the film to permit the release of air between the bearing and the film at a controlled rate, thereby maintaining a cushion over which the film is transported. This bearing requires a high flow of air, necessitating a blower of about 10 horsepower for all the air bearings.

An undesirable result of the high air flow is the release of aerosol spray when the air impinges on the liquid present on the film. To overcome this problem, a plenum-type bearing was constructed.

2. TECHNICAL DISCUSSION

2.1 PLENUM-TYPE BEARING

A study made of the requirements for an improved bearing showed that the following requirements should be considered:

1) Completely enclose the film in its passage between tanks.

2) Reduce the air flow required by enclosing the film in a tunnel, the entrance and exit of which would be below the liquid level in the tanks.

3) Provide air release holes that would center the film in the tunnel by equalizing the air pressure on both sides of the film; also, by closing the release holes to control the air pressure, possibly permit the passage of all widths of films without the use of adjustable flanges.

4) Reduce the aerosol spray by controlling the release of air.

Figure 1 illustrates the design selected to meet the above requirements. It is, in effect, a tunnel which carries the film from one tank to another. A satisfactory air cushion was obtained with a load of 1 pound after the air flow on each side of the film had been balanced by increasing the air to the underside of the film and decreasing it against the opposite side of the film.

It was found that the film oscillated at a high frequency at the entrance and exit slots. Additional air release holes were added in the film path, which reduced the oscillations but did not eliminate them. When the air bearing was positioned over a tank of water with the film entrance and exit slots under the surface, film oscillation was damped. Because of the favorable result of this test, a method of extending the sides of the film path further into the solution will be developed and testing will be resumed.

At the time of this report, further tests are scheduled to plot air pressure and flow against the load lifting capability of the bearing.

2.2 SELF-POWERED AIR BEARING

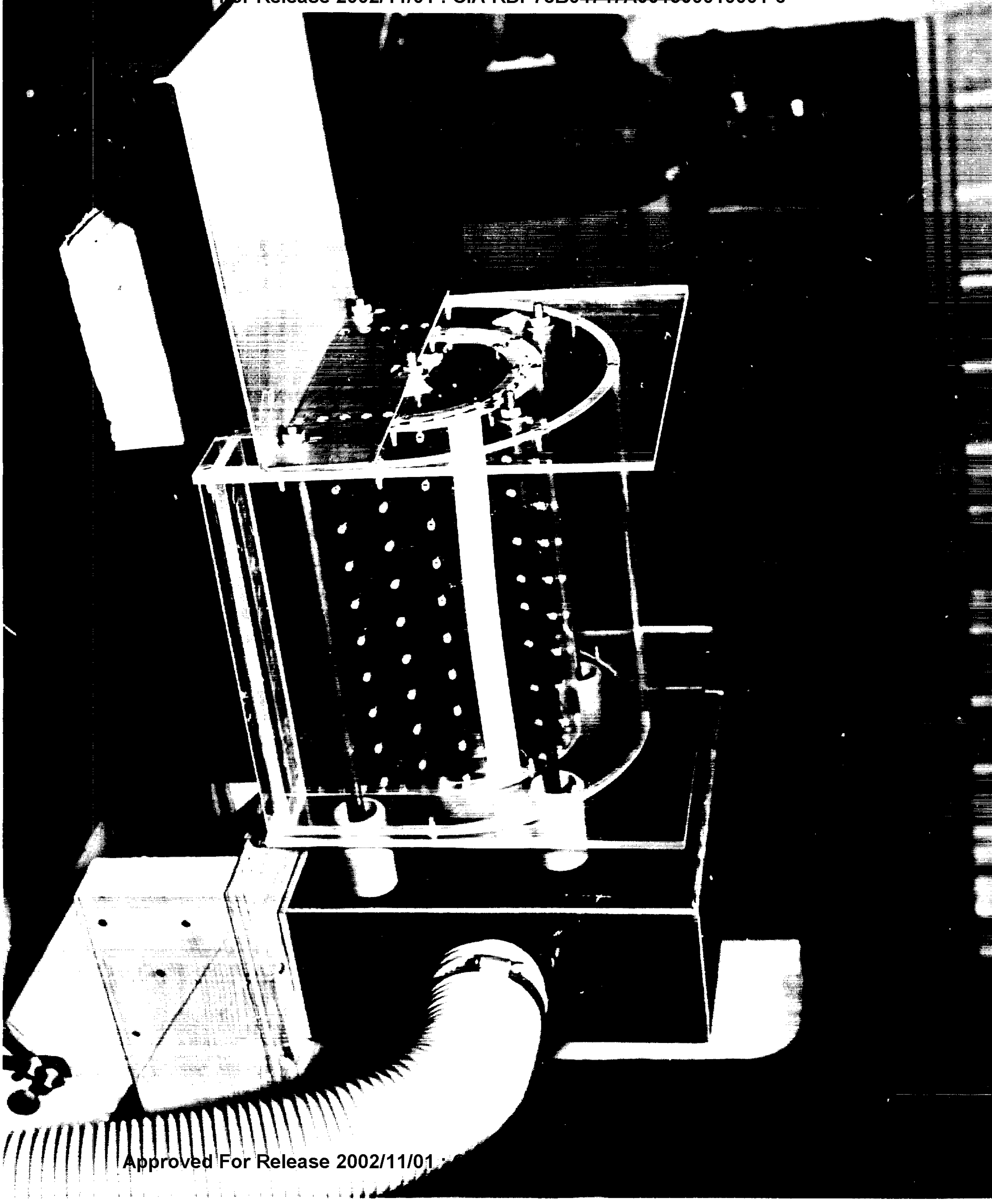
As an extension of the work being carried out on the plenum-type bearing, a design study is in progress on a self-powered "rotatron" air bearing in which a fan will be incorporated to generate the air pressure and flow necessary to provide a transport cushion.

The use of a standard centrifugal blower, squirrel-cage wheel as a built-in power source presents difficulty because, to obtain optimum efficiency at the given design speed, the width of the wheel should not generally exceed 0.6 of the diameter. Various methods of avoiding this restriction can be employed, such as the use of two wheels, but achieving even flow becomes a serious problem over the length of the 9-1/2-inch bearing required. A new type of fan developed in Europe, the transverse-flow fan, shows promise for this application and is now becoming available here. A conventional centrifugal blower draws in air axially and discharges it radially. The transverse flow fan draws air inward radially and discharges it outward radially through a different section of the fan periphery.

The advantage of incorporating this type of fan in an air bearing is that higher static pressures are obtainable for the same rotor diameter (a pressure coefficient of 1.8 to 5.5 as against 0.60 to 1.10) over a length restricted only by structural considerations, such as housing strength. A rough mock-up bearing, using an available wheel 1.8 inches wide, indicated that a blower can be part of the bearing. The main problems appear to be the air flow distribution over the required 9-1/2-inch length, which can be solved by using a transverse flow; and flutter of the film on one side of the bearing cage, caused by the direction of rotation of the wheel. Experimentation with flow baffles will be undertaken as a possible solution to this problem.

Detailed design of a self-powered bearing is scheduled in the period remaining in the contract. Construction of a test model will be undertaken if this can be accomplished within the present scope of the contract.

Figure 1-1. 4



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THE DESIGN, CONSTRUCTION, AND TESTING OF A LIQUID BEARING INCORPORATING A BUILT-IN PUMP

Liquid Bearing w/built-in pump

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February 1965

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FOREWORD

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[redacted] submits this report in compliance with Item 4.2 of the Development Objectives of [redacted] This report should be read in conjunction with [redacted] of which it forms part. STAT STAT

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Approved: [redacted]

Research Manager

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ABSTRACT

The purpose of this assignment is to design, construct, and test a liquid bearing of a radical departure from the type employed previously. The bearing is required to be self-powered and to incorporate its own fluid pump. At this time, only preliminary tests to prove the soundness of the design principle have been conducted, but these tests will be continued through the unexpired term of the contract.

Approved For Release 2002/11/01 : CIA-RDP78B04747A001300010001-6

1. INTRODUCTION

The liquid bearing design described in this report and now known as the "Rotatron" was developed to overcome the many disadvantages inherent in the liquid bearings now being used.

A liquid bearing is a bearing on which a load is supported by a liquid. In the HTA-5 processor, liquid bearings are used to support film while it is transported through the various tanks of solutions used in the developing process. These liquid bearings support the film on "jets" of liquid (developer, wash water, etc.) which is pumped into each bearing and subsequently through a combination of orifices in such a manner that the film is forced to ride upon these jets, thus, the film contacts only the fluid. The load on the bearings is accumulative and the force increases as the rate (fpm) of processing is increased; at any speed, the load on the final bearing is the greatest. Consequently, the speed is governed by the maximum tension that can be tolerated by the film. It is obvious that as the load increases, the jets must provide a greater support force.

1.1 SCOPE

The Rotatron Bearing was designed to:

1) Lower the unit pressure on the film
2) Provide a definite degree of transport assist
3) Be independent of other bearings
4) Eliminate conventional plumbing
5) Obviate the need for circulating pumps
6) Provide a simple method of cushion control
7) Provide a simple means of film tension regulation
8) To "float" the film around the bearing center

9) Cause the film to self-center on the bearing

10) Be easily adaptable to the module-type processor

11) Eliminate the harmful effects inherent in conventional mechanical bearing systems.

2. TECHNICAL DISCUSSION

2.1 PHYSICAL DESCRIPTION

Essentially, the prototype Rotatron is a squirrel-cage type axial vane pump with 12 blades set at 30 degrees from the radial and with 4-inch diameter. The blades are approximately 7/8 inch wide and 10-1/4 inches long; they are supported by a solid 1/8-inch plate in the center with rings at each end. The hubs are three spokes arranged conically from the end supporting rings converging to the shaft (driving) blocks, which are attached to a 3/8-inch diameter shaft by two set-screws. The pump is "caged" in a helix of 1/8-inch diameter wire, one-half of which is left-hand wound, the other half right-hand wound, on a pitch of 1/4 inch. Approximately 120 degrees on the circumference is covered by a 1/16-inch plate which extends for the entire length of the blades. The helixes are welded to this plate which is curved to fit their outside diameter of 4-3/8 inches. Two helix-retaining end rings are also welded to this curved plate. The rings rest in contoured cradles on the mounting plate thus making it possible to rotate the cage (and plate) through 360 degrees. The cage is secured by two metal straps held in tension by thumbscrews. A pair of ball bearings at each end of the shaft support the pump. The entire unit is constructed of corrosion-resistant steel. The base support, ball bearing supports, and helix-cage cradles are made of aluminum alloy.

2.2 TEST SET-UP

To facilitate testing under realistic conditions, the pump was mounted on the side wall of a specially constructed tank, three walls of which were plate glass. The tank was approximately 24 inches square by 42 inches deep. Water level in the tank covered the bearing to a depth of approximately 4 inches. The Rotatron was driven by a 1/8 hp, 1725 rpm, electric motor through "V" belts, various pulleys (four in the system), and a 2:1 speed reducer, the final driveshaft entering the tank through a watertight sleeve bearing, connecting to the pump by a rubber flexible-coupling. This pulley and "V" belt system was used only for testing. The Rotatron design will be driven directly by a motor designed to operate at speeds within the requirements of the individual bearing.

2.3 TEST PROCEDURE AND RESULTS

The pump was first run without any film to study the flow pattern. The calculated pump speed was approximately 445 rpm. The switch was operated and before the pump was up to speed, water was cascaded over the top of the tank. Immediately the motor was shut off. It was obvious that the pump was operating too fast.

2.3.1 Flow Pattern

With the pump speed reduced to approximately 150 rpm the flow pattern was considerably higher in the center. This was due to the ram effect (inertia) of the liquid coming from both ends. The pump cage had been installed so that the helixes would cause the liquid to converge. The cage was reversed so that the helixes diverged from the center. The flow pattern immediately changed, dropping in the center and building up on both ends (Figure 2-1).

FIG. 2-1

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The next step was to provide a loop of 9-1/2-inch thin-base mylar leader (approximately 30 inches on centers) loaded with a regulation spool (1-1/2 lbs) at the bottom end of the loop. At 150 rpm, a very adequate cushion was provided, but the loop immediately slid off the side of the bearing. Two metal clips, approximately 1-1/2 inches long, were attached to the end rings at top center to keep the film on the bearing. The restraining force was so slight that measurement was not possible with available equipment. The force was estimated at less than 1/4 ounce.

2.3.2 Loop Instability

With the film restrained from floating off the bearing, the loop "loped" as it came over the top. The divergence from a symmetrical (stable) cushion was maximum at the 3-o'clock position. This loping caused the bottom of the loop to bounce at a frequency of about 60 cps. An attempt was made to determine the cause of this phenomenon, and it was found that the pressure was building up until the spool at the bottom of the loop had been displaced vertically by the increase of the depth of cushion; at this point, the pressure fell again. The weight of the spool was increased to 4 pounds, (the equivalent of the load on the last bearing in a film processor running 9-1/2-inch aerial negative at a speed of 20 fpm), to determine if this would affect the frequency. The additional weight required an increase of pump speed (185 rpm) to preserve the cushion depth of approximately 3/8 inch. The loping tendency persisted at about the same frequency. A functional analysis of the phenomenon led to the elimination of the pulsating "lope."

2.3.3 The Stable Cushion

The test set-up in use, with the weighted spool hanging in the loop, provide little or none of the damping that would exist if the film continued around other bearings. To provide a degree of damping, two 1-pound weights were suspended to the hubs of the spool by rubber bands. The pump speed was increased to 250 rpm to maintain the cushion depth. The damping action provided by the suspended weights minimized the pulsing tendency but did not eliminate it entirely. The frequency, however, was irregular and the magnitude less. This led to the thought that the film was reacting to a jet pressure effect caused by the greater volume of liquid emanating from the pump in the center. The greater spool weight, 6 pounds, now required a greater mean pressure against the total projected area of the film. This undoubtedly meant that there was better pressure distribution and that the release (flow) of the liquid out from the film was such that the pressure distribution was more even throughout the bearing area. Though the cushion was now quite stable, it was eccentric, the cushion being deepest in the area where the pulsating (3 o'clock) was taking place. The cage was rotated counterclockwise (contra pump-rotation) thereby increasing the pressure on the entering side of the bearing and, conversely, decreasing the pressure on the exit side. This allowed the liquid cushion depth in this area to decrease, which, at the same time increased the unit pressure in the area causing the reappearance of the loping of the loop.

The necessity of reducing the higher pressure in the center area became quite obvious. To accomplish this, a width of perforated aluminum sheet was wrapped around the helix cage midway between the end rings. The perforations constituted approximately 50 percent of the area of the sheet, the width of which was approximately one half the length of the cage. The obstruction to the flow decreased the depth of cushion but did stabilize it. The pump speed was increased to 275 rpm to restore the depth of cushion. It now appeared that the restriction was too great over too wide an area. The ends of the screen were trimmed out (Figure 2-1) to feather the flow while flow in the immediate center was further restrained by wrapping a 1/2-inch wide piece of tape on each side of the 1/2-inch wide metal retaining clamp. Thus, the center of the cage was closed off completely by this 1-1/2-inch wide band. The resulting flow pattern can be clearly seen in Figure 2-1. (The water level was lowered to just cover the cage for this photograph.) This type of flow pattern (which is constant around the periphery of the bearing), was one which, it was thought, would tend to keep the film centered on the bearing. The theory was that the film would tend to settle in a "trough" of the liquid. Contrary to the crowning of a pulley which keeps a belt in place, the high-pressure center of the jet type liquid bearing tended to cause the film to "skid" off. This characteristic proved to be true with the Rotatron type also. The pulley had friction working in its favor while the liquid bearing was provided with a continuous waterfall down which the film could migrate. Furthermore, it was reasoned that this migration was aided by flow direction of the liquid as it "escaped" from under the film, the area increasing as the film moved off center.

To offset this migratory tendency, two 1/2-inch wide strips of metal were wrapped around the cage at the extreme ends thus reducing the effective bearing width to 9 inches. The effectiveness of this modification was not sufficient to establish any conclusion; however, the film had a tendency to move from one side to the other between the two clips.

With the installation of the perforated screen, the loop over the bearing became quite stable and concentricity was readily controlled by positioning the cage bottom plate to equalize the pressure distribution. Increasing the pump rpm, which increased depth of cushion, also changed the location of the maximum pressure area necessitating a change of the position of the cage bottom plate to restore concentricity.

2.3.4 Film Transport Assist

At various times during the tests, the film loop travelled in the direction of pump rotation. This action continued until the splice came to either the spool or the bearing. The splice had been made by butting the edges of the film and applying 1 mil mylar tape (1 inch wide) to both sides of the film. This relatively stiff splice (water temperature 66°F) increased the normal bending forces.

The film loop was removed from the tank and allowed to remain at room temperature for approximately 1 hour. A 4-pound spool was placed in the loop; the pump speed reduced in rpm and the loop restored to the tank over the bearing. The pump was started and the film then moved in a clockwise direction at the rate of approximately 1 fps, rotating the hanging spool in so doing. This action continued until the film became stiffened by the cold water.

During this observation, the film tended to remain quite steady on the approximate center of the bearing. As was stated earlier in this report, the two cage helixes were wound in opposite directions for the purpose converging or diverging the liquid flow; however, it appears that the influence of the 1/8-diameter wires plus the low pitch of the helix winding is insufficient to satisfy the requirements of this type of vector flow control, although all indications are that this is within the realm of possibility.

2.3.5 Load Capability

Figure 2-2 shows the load capability of the experimental rotating-type liquid bearing. Its capability appears to be limited only by the strength of the film. It also appears that the speed of film processing need no longer be limited to the film strength because each bearing of the Rotatron design can contribute to the film transportation, thus reducing the accumulative load.

The cushion depth appears to remain quite constant over a speed variation of ±10 percent. Each bearing speed can be independently controlled since each is individually driven. By the same token, each bearing is easily removed for servicing, replacement, and tank cleaning.

Experimental testing will continue during the remainder of the contract. Many methods of increasing the potential use of this type of bearing are apparent. After completion of tests, runs will be made to obtain data on rpm against transport speed, load (film tension), and cushion depth; the resulting data will then be correlated.

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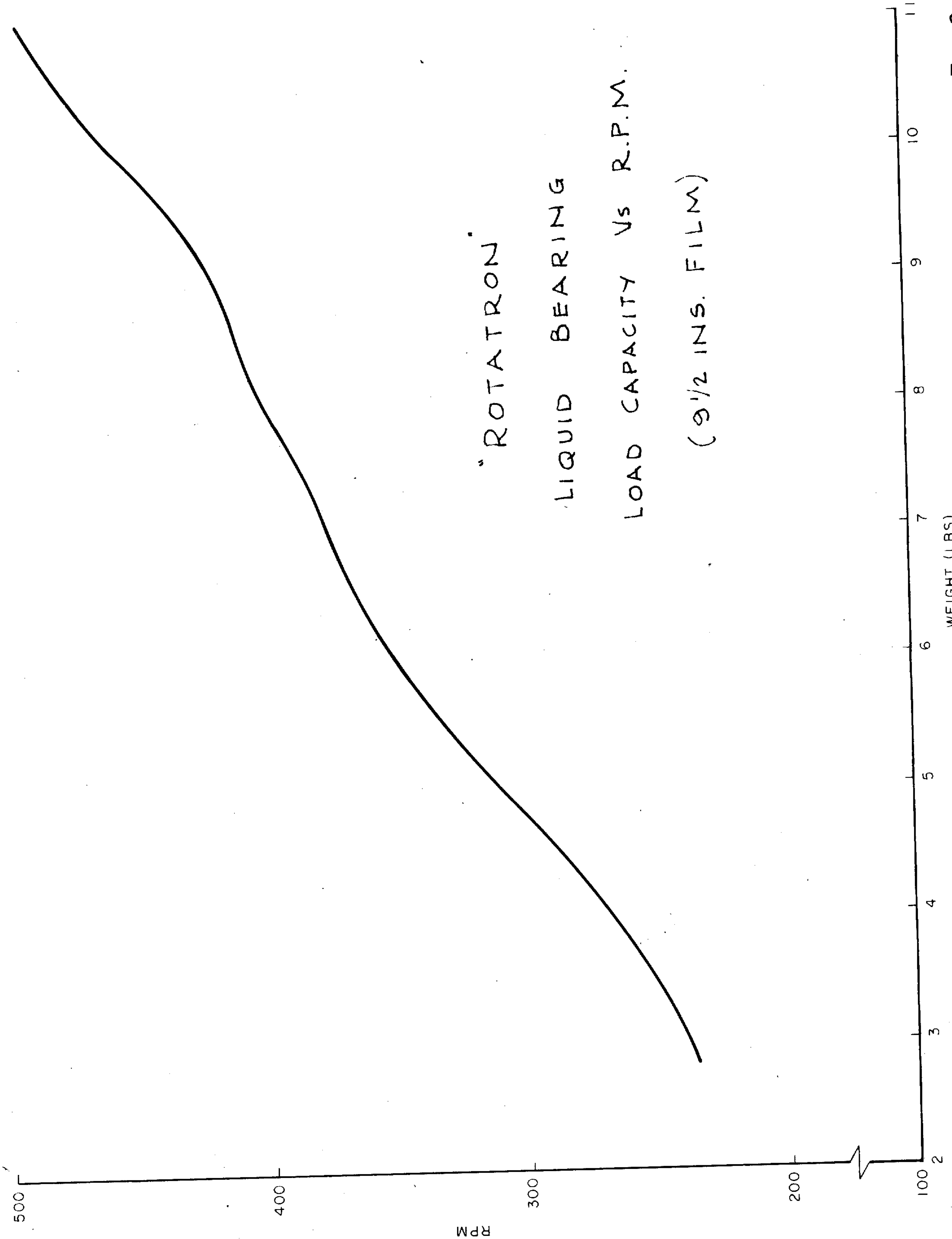
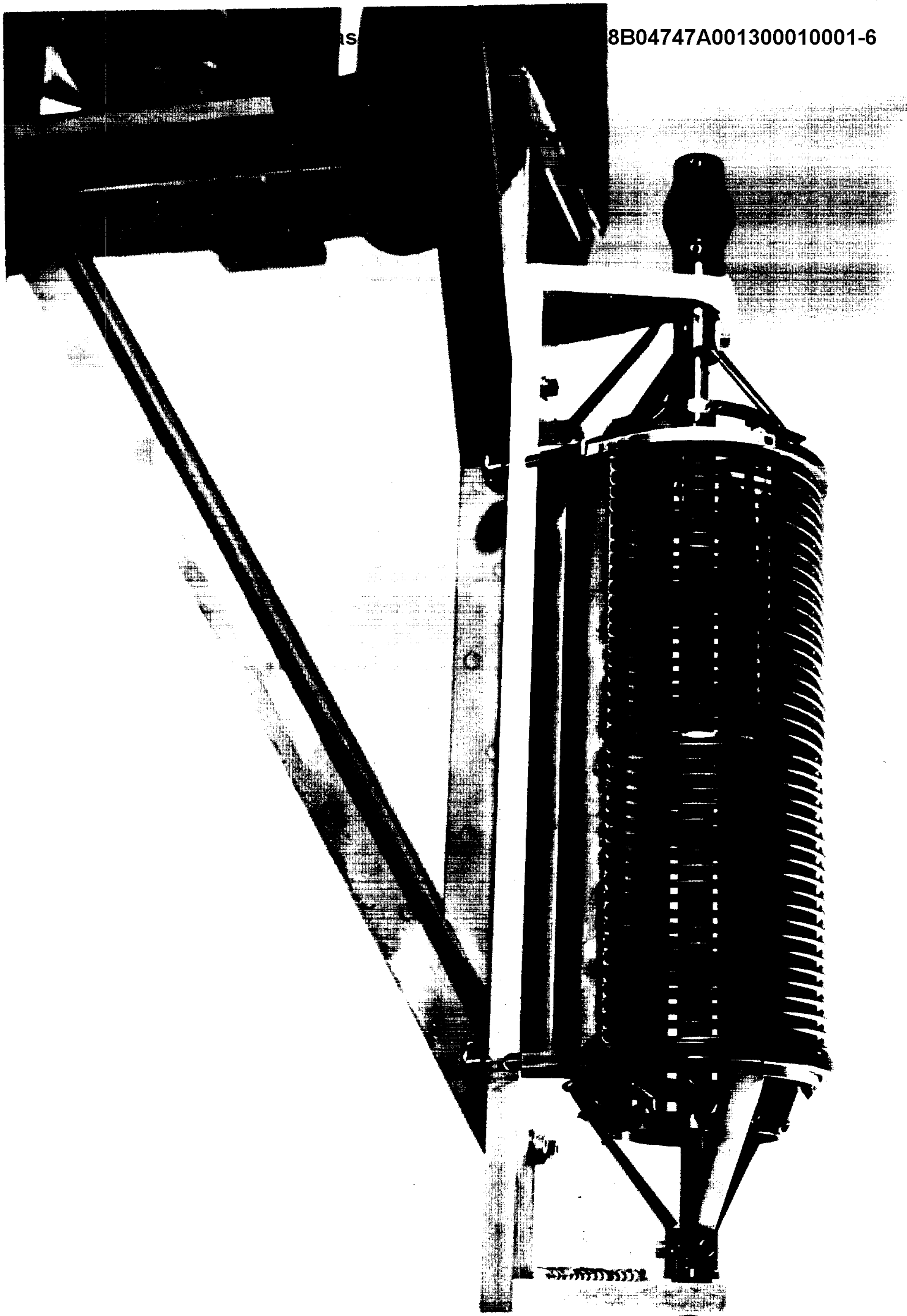

"ROTATRON"
LIQUID BEARING
LOAD CAPACITY Vs R.P.M.
(9½ INS. FILM)
WEIGHT (LBS)
2
3
4
5
6
7
8
9
10
11
RPM
100
200
300
400
500

FIG. 2-2



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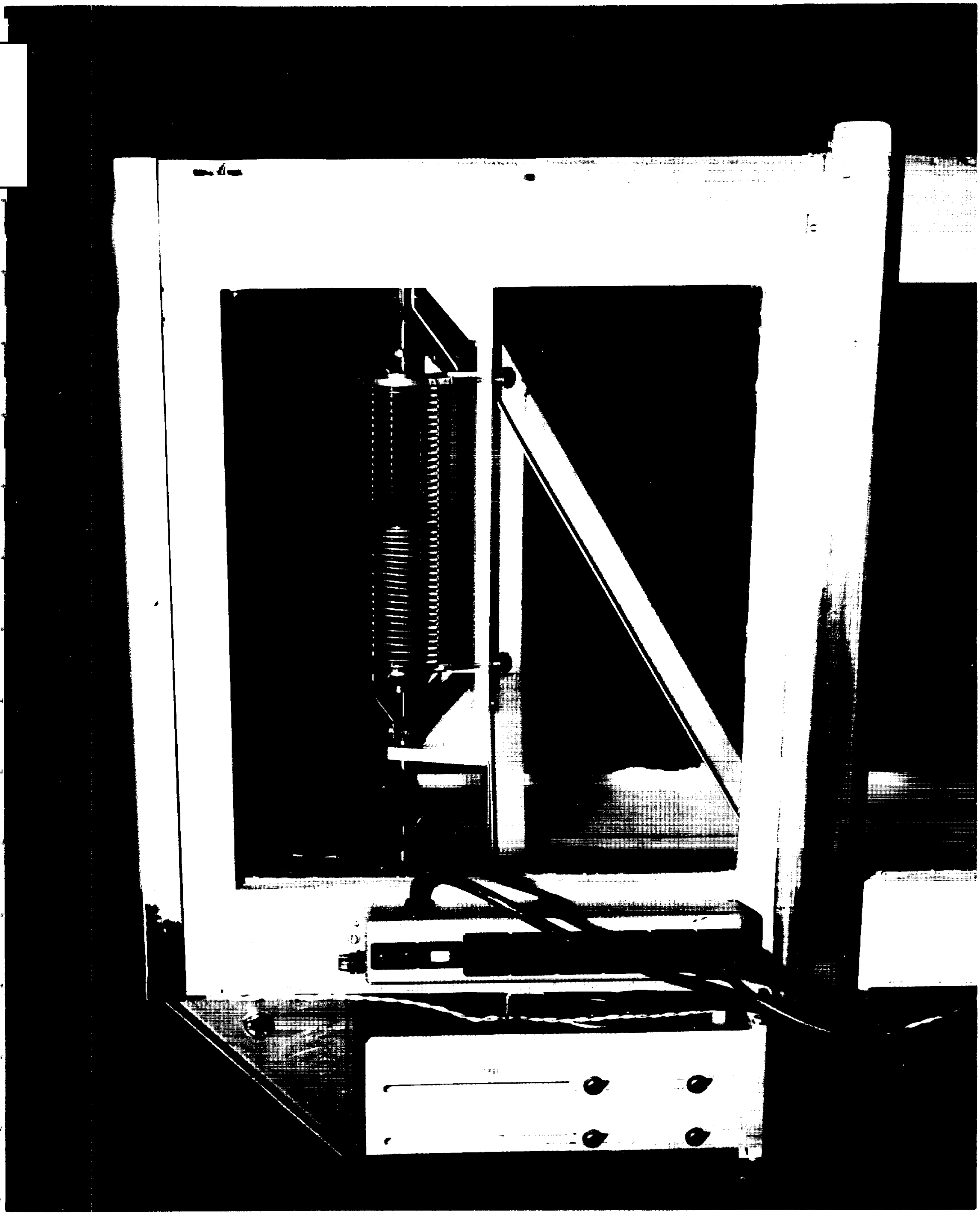

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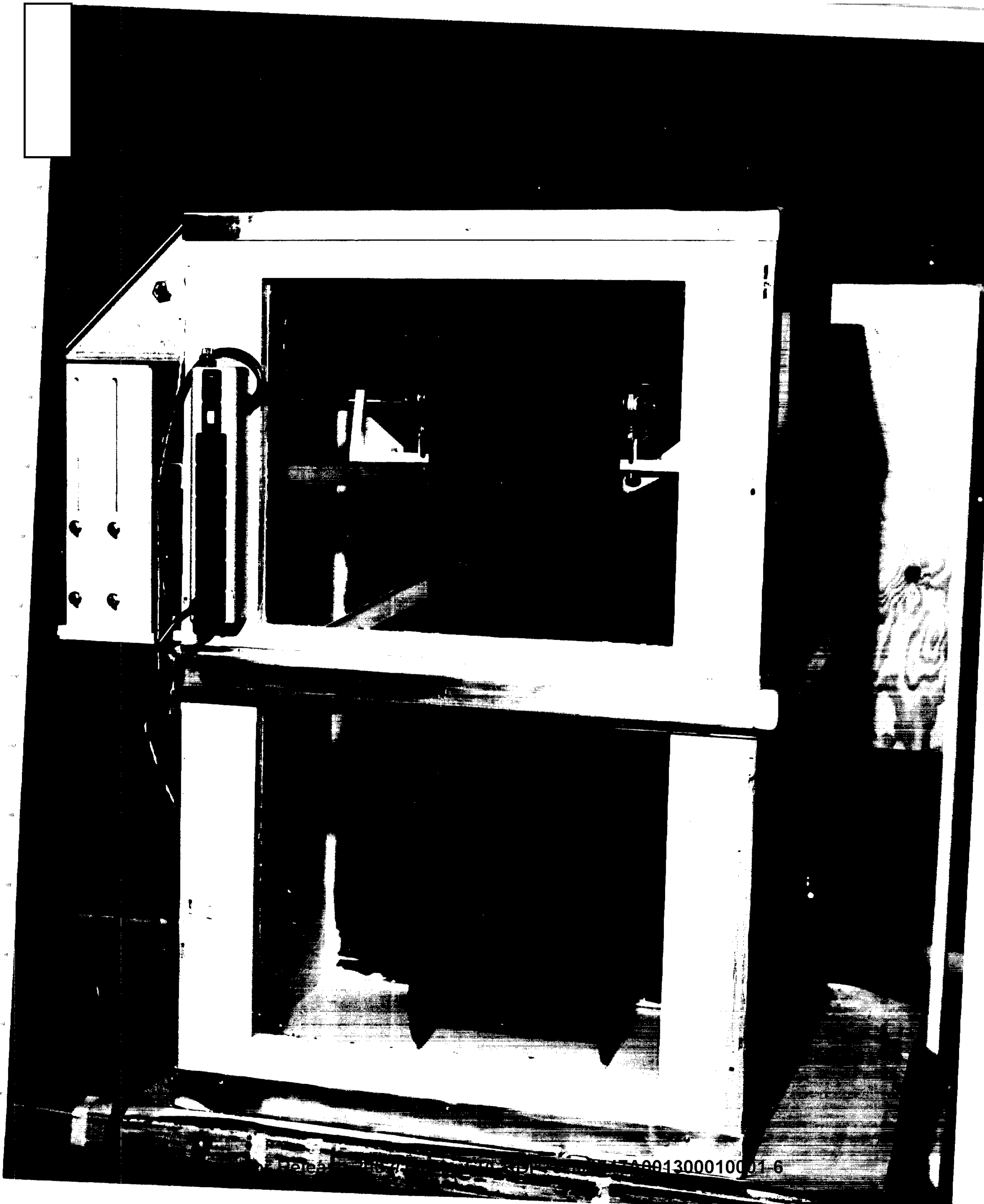

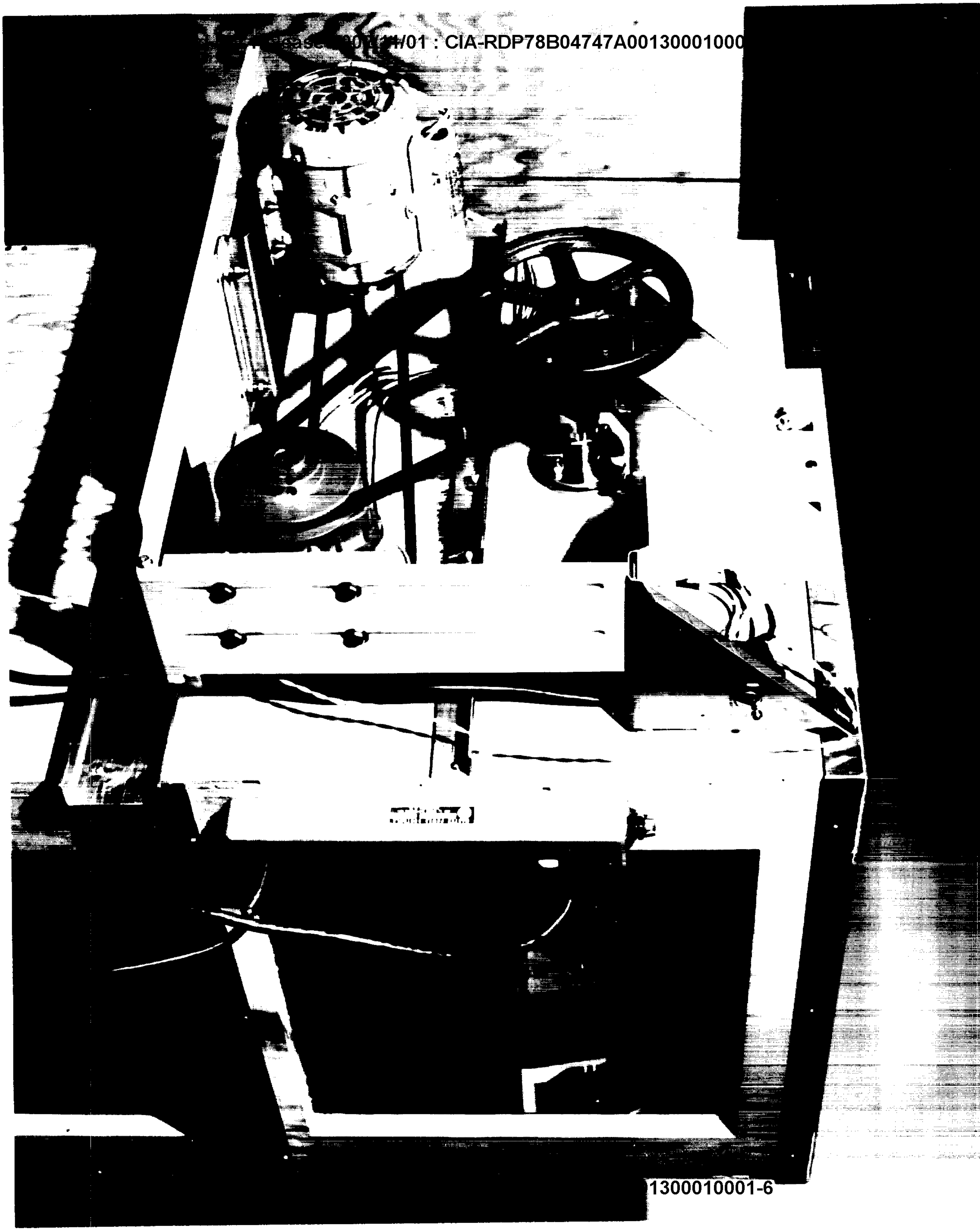

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