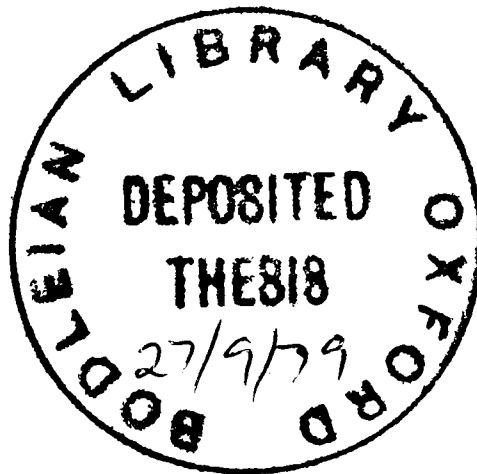


EXPERIMENTAL STUDIES OF
VISCOUS EFFECTS ON CAVITATION

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The work reported in this thesis falls into three distinct, though intimately related parts. Part I is concerned with the production of a variable temperature two-phase flow test facility for studying single and two-phase flows around arbitrary two-dimensional bodies. It includes the initial design study required to define an operational envelope for the facility, and the design and constructional features of its main components and ancilliary systems. The design of a two-phase flow vapour ventilation experiment is described, together with the operating procedure for the facility.

Part II reports single and two-phase experiments with a two-dimensional circular cylinder, the two-phase flows being generated by inducing natural cavitation. The cavitation inception flow regime was found to be strongly influenced by viscous effects. Three forms of incipient cavitation were observed, two attached or very close to the cylinder surface, and one of a detached nature, occurring well downstream in the wake of the cylindrical test body. These inception modes have been related to the fully wetted viscous flow around the cylinder in the Reynolds number range $10^5 < R_d < 10^6$. The development of cavitation from these three aforementioned incipient states was investigated. Viscous effects were found to influence both the limited and developed cavitation flow regimes. For development of cavitation at Reynolds numbers corresponding broadly to the supercritical range for fully wetted flow, a critical cavitation number was found at which the apparent free stream lines of the flow changed from a concave to a convex disposition. For the experimental configuration used, this critical cavitation number was independent of Reynolds number, for Reynolds numbers above the critical value. With development of cavitation at Reynolds numbers corresponding broadly to the subcritical range for fully wetted flow, no such gross changes in flow pattern were observed, the displacement of the apparent free streamlines of the flow between the limited and fully developed cavitating states being minimal.

Part III contains recommendations for modifications to the experimental facility and suggestions for further studies arising from the results reported herein.

To
JEAN AND VICTORIA

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NOMENCLATURE

A_c	Effective heat transfer area of cavity
A_d	Flow area of heat exchanger duct at tube bundle outlet
A_m	Projected area of test body normal to the flow
A_t	Flow area of heat exchanger tube bundle
B	Wire gauze open area ratio
C	Power dissipation of tunnel circuit without the heat exchanger
C_A	Non-dimensional cavity surface area
C_D	Drag coefficient
C_{DO}	Drag coefficient of isolated gauze wire
C_f	Skin friction drag coefficient
C_p	Pressure coefficient
C_{pb}	Base pressure coefficient
C_{pf}	Fluid specific heat at constant pressure
C_{ps}	Pressure coefficient at laminar separation
C_q	Non-dimensional ventilation flow
C_R	Contraction ratio
C_W	Wall pressure coefficient
$C_{W \min}$	Minimum wall pressure coefficient

d	Cylinder outside diameter; Tube inside diameter
D	Tube outside diameter
D_m	Contraction diameter at the inflexion plane
D_1	Contraction inlet diameter
D_2	Contraction outlet diameter
f	Skin friction factor
F_r	Froude number
G_p	Primary heat transfer fluid mass velocity
G_s	Secondary heat transfer fluid mass velocity
H	Turbulent boundary layer shape factor; Contraction outlet height
h_c	Area mean cavity heat transfer coefficient
h_{fg}	Latent heat of vaporization
h_p	Primary side heat transfer coefficient
h_s	Secondary side heat transfer coefficient
ΔH_v	Vapour pressure depression in a developed cavity
K	Auxilliary variable defined by Equ. I.3.
k	Pressure loss coefficient
k_f	Fluid thermal conductivity
L	Streamwise length

L_c	Cavity length
L_k	k^{th} estimate of heat exchanger power absorption
M_o	Initial estimate of heat exchanger mass flow rate
M_k	k^{th} estimate of heat exchanger mass flow rate
n	Exponent defined by Equ. I.17
N_u	Nusselt number
P	Pressure
P_b	Base pressure
P_e	Peclet number
P_{IN}	Diffuser inlet pressure
P_{OUT}	Diffuser outlet pressure
P_r	Prandtl number
P_s	Static pressure of onset flow
P_{STAG}	Stagnation pressure
P_v	Vapour pressure
P_w	Wall pressure
ΔP	Pressure loss
q'	Dynamic pressure
$\dot{q}$	Heat flux

$\dot{Q}_c$	Heat flux across cavity - liquid boundary
$\dot{Q}_v$	Heat flux to ventilation flow
R_c	Chord Reynolds number
R_{CR}	Cylinder critical Reynolds number
R_{CRIT}	Flat plate transition Reynolds number
R_d	Reynolds number based on cylinder or tube diameter
R_e	Reynolds number
R_L	Streamwise length Reynolds number
R_o	Reynolds number based on X_o
$R_{\theta s}$	Reynolds number based on boundary layer momentum thickness and stream velocity at laminar separation
$R_{\delta s}$	Reynolds number based on boundary layer displacement thickness and stream velocity at laminar separation
t_v	Reduction factor of v component of turbulent energy
S	Dead water length of laminar separation bubble
S_f	Saturated liquid specific entropy
S_{fg}	Fluid specific entropy of vaporization
S_t	Strouhal number
T_c	Cavity temperature
T_{CR}	Critical temperature (thermodynamic)

T_R	Temperature non-dimensionalized to critical value
T_{SE}	Bulk mean temperature of secondary heat transfer fluid
T_u	u component of turbulent energy
T_v	v component of turbulent energy
T_∞	Onset flow temperature
T_1	Heat exchanger inlet temperature
T_2	Heat exchanger outlet temperature
T_{2k}	k^{th} estimate of heat exchanger outlet temperature
T_2 (converged)	Converged value of heat exchanger outlet temperatures
U	Main stream velocity
u	Fluctuating velocity in main stream direction
U_T	Design value of test section velocity
U_∞	Onset flow velocity
U_s	Stream velocity at laminar separation
v	Fluctuating velocity normal to main stream
V_L	Liquid volume
$\dot{V}_e$	Entrainment volumetric flow rate
V_v	Vapour volume
$\dot{V}_w$	Vapourization rate from cavity wall

(xx)

$\dot{V}_*$	Vapour entrainment and ventilation volumetric flowrate at thermal equilibrium
W	Contraction outlet width
W_e	Weber number
X	Streamwise length coordinate
X_o	Streamwise distance from virtual origin of turbulent boundary layer to diffuser inlet
X_1	Streamwise displacement from contraction inlet
X_2	Streamwise displacement from contraction inflexion plane
ΔX	Heat exchanger tube elemental length
X_*	Value of X when diffuser pressure recovery coefficient = $(n-2)/(n+1)$
Y_1	Contraction radius in inlet section
Y_2	Contraction radius in transition section
β	Vapour to liquid volume ratio = V_v/V_L
δ_0^*	Turbulent boundary layer displacement thickness at diffuser inlet
δ_s	Boundary layer displacement thickness at laminar separation
μ	Fluid dynamic viscosity
μ_b	Fluid dynamic viscosity at bulk mean temperature
μ_f	Fluid dynamic viscosity at film temperature
μ_w	Fluid dynamic viscosity at wall temperature

ν	Fluid kinematic viscosity
θ	Bulk mean temperature in Equ. I.25; Angular coordinate of contraction profile in Eqs. I.44 and I.45
θ_s	Boundary layer momentum thickness at laminar separation
ρ	Fluid mass density
ρ_f	Liquid mass density
ρ_g	Vapour mass density
σ	Cavitation number
σ_c	Cavitation number based on cavity pressure
σ_{CH}	Choked flow cavitation number
σ_{CR}	Critical cavitation number
σ_i	Inception cavitation number
σ_{ib}	Cavitation number at Band type inception
σ_{is}	Cavitation number at Spot type inception
σ_{iw}	Cavitation number at wake type inception
σ_T	Design value of test section cavitation number
σ_v	Cavitation number based on vapour pressure
ψ	Free-streamline separation angle measured from frontal stagnation point
ω	Thermodynamic parameter defined by Equ. I.2

P A R T I

SPECIFICATION, DESIGN AND COMMISSIONING OF THE
OXFORD UNIVERSITY HIGH TEMPERATURE
CAVITATION TUNNEL

CHAPTER I

INTRODUCTION

There are many different forms of cavitation testing facilities, the configurations and working fluids of which reflect the particular aspects of cavitation to be investigated. They may be broadly classified as general purpose facilities, which are capable of being adapted to accommodate a wide variety of experiments, and special purpose facilities designed specifically to investigate cavitation in a given type of device or flow configuration. The former group includes open and closed circuit cavitation tunnels, with a variety of working sections, and towing tanks and channels. Special purpose facilities include equipment for testing cavitation resistance of materials, cavitation noise, thermodynamic effects, water entry hydroballistics and many forms of test rigs for evaluating the cavitating performance of items such as pumps, valves, flowmeters and propellers. Since the first part of this thesis reports the design of a general purpose closed circuit cavitation tunnel, the following outline is limited to facilities of this type. Detailed reviews of many of the existing civilian cavitation facilities may be found in References 1 and 2.

Continuously-running closed circuit cavitation tunnels are by far the most numerous type, the larger ones invariably using water as the working fluid, for obvious practical reasons. The most important part of a tunnel is the working section, which consists of a contraction to

produce a uniform outlet flow, followed by a test section to accommodate the experimental body. Schematics of some basic types of working section are shown in Fig. I.1. The form of test section used in a given investigation depends on the type and size of the test body and the flow conditions to be simulated. Confined or closed test sections (Fig. I.1a) give rise to wall interference effects which limit the size of the test body which may be used. This problem can be ameliorated by using open working sections with or without a diffuser (Fig. I.1b). With this arrangement the free streamlines of the jet issuing from the contraction may deform under the influence of the test body blockages, thereby reducing wall interference effects and allowing larger test bodies to be used than with a closed working section. Open working section closed circuit tunnels having a diffuser tend to suffer from flow instabilities associated with the entrainment of low velocity fluid from the flooded chamber into the diffuser inlet. This instability may manifest itself as undesirable pulsations in the cavity flow under investigation^[2]. Since this situation gets worse with longer open type working sections, the length of these is usually shorter than those used in closed working section tunnels. Thus open working sections are not well suited to experiments on bodies with large streamwise dimensions. A further variant is the slotted-wall working section shown in Fig. I.1c, in which the test section is bounded by longitudinal bars separated by slots which communicate with a surrounding flooded chamber. The efficacy of this design depends on sizing the bars and slots such that no^{net} lateral flow occurs through the slots, thereby producing a flow at the

diffuser inlet similar to that of a closed working section but having the blockage characteristics of an open type one. In the free-jet design shown in Fig. I.1d the possibility of flow instability in the diffuser is eliminated by dispensing with it altogether, with the jet discharging into a variable pressure, gas filled chamber. This arrangement is particularly suited for studying very low cavitation number flows since the gas pressure may be reduced to the working fluid vapour pressure. A disadvantage of this design, particularly with large tunnels, is that the jet dynamic pressure is completely dissipated, requiring large power input to the circulating pump. Free surface type working sections, as depicted in Fig. I.1e, are usually of rectangular or square cross section with the flow guided by walls at the sides and below, with a gas space above. This arrangement is suitable for investigating submergence effects in cavity flows.

An important consideration in the design of closed circuit cavitation tunnels is the provision of a means of controlling the gas content of the flow entering the working section. During cavitation experiments, air dissolved in the working fluid tends to come out of solution in the cavitated region, producing an increase in free air content which may be recirculated back to the working section inlet. In free jet tunnels, air may also be entrained by the jet at the free-surface, further exacerbating the problem. It is well known that air content influences both the limited^[3,4] and developed^[5] states of cavitation. In tunnels with no free-surface entrainment or ventilation air supply, the air content may be controlled by de-aerating the working fluid prior to a given experiment. This method

however, reduces both the free and dissolved air content. In certain scale model experiments this might prove disadvantageous since the full scale flow to be simulated might have a high dissolved and a low free air content. An alternative approach is to incorporate a resorber into the tunnel circuit. Fig. I.2 shows a schematic of the California Institute of Technology High Speed Water Tunnel, which has a resorber system, and is widely regarded as being one of the best cavitation tunnels in existence, from the viewpoint of flow quality and free air content control. The resorber is essentially a flow path in the tunnel circuit providing a relatively high pressure, long residence time environment in which free gas bubbles up to a specified size are absorbed into the liquid. The required pressures are generated partly by the circulating pump and partly hydrostatically, and the residence time is achieved by making the resorber of large volumetric capacity. These requirements mean that resorber structures are large in flow area and extend for considerable depths below the tunnel working section level. The main advantage of resorbers is that they enable working section flows with low free air content at variable total air content. The ideal gas content control system comprises some form of de-aerator to vary the total air content, combined with a resorber to control free air content, as used on the National Maritime Institute No. 2 Water Tunnel [6].

An essential requirement for cavitation tunnels is the provision of a means of varying and controlling the pressure in the test section. This may be done using a liquid or gas pressurizing medium. With the liquid system,

tunnel working fluid is supplied to or abstracted from the circuit under manual or automatic control, the high pressure fluid being obtained from some form of pump. The pressure may be regulated by adjustment of the pump by-pass flows, as with the Pennsylvania State University 1.5 inch High Speed Cavitation Tunnel^[7] or by using some form of hydraulic spool valve^[6]. With gas pressurization, the gas is applied to the liquid at a free-surface, the liquid beneath which connects with the tunnel working fluid. A disadvantage of this method is the direct contact between gas and liquid which allows the possibility of pressurizing gas entering the working fluid. This problem may be circumvented by the insertion of a flexible, impervious membrane or bellows between the gas and liquid phases^[8]. A further variant is the use of sealed floats to achieve the same end^[9]. Free-surface gas pressurizing systems are particularly suitable for variable temperature tunnels, since the vessel containing the free-surface may also be used to accommodate temperature-induced changes in working fluid volume^[10].

With a general purpose cavitation tunnel, it is highly desirable to have means of adjusting and controlling the test section flow velocity, since many of the fundamental parameters of fluid flow are functions of this variable, (for example, Reynolds, Froude and Weber numbers). The greater the velocity range achievable by a tunnel, the more versatile it is. High velocities cause correspondingly high stagnation pressures which with larger tunnels would require circuit components of prohibitive cost and size. Thus the maximum design velocity is governed by a trade-off between cost and performance. Most general purpose

tunnels have maximum test section speeds of up to 30 m/sec., though one existing tunnel can achieve 100 m/sec.^[7]. Continuously variable speed adjustment is clearly better than the discrete type, particularly for investigating velocity dependent transitions between different flow regimes.

For investigating flows with significant viscous effects, variable temperature capability provides an attractive alternative to extending the operational velocity range, since the viscosity of liquids is strongly temperature dependent. Further, variable temperature tunnels allow the study of heat and mass transfer, or so-called thermodynamic effects, in developed cavity flows, due to the temperature dependence of the thermodynamic properties of the working fluid. It is clear from the foregoing that the ability to vary working fluid temperature is a very desirable feature for a cavitation tunnel and it is equally desirable in a single phase hydrodynamic tunnel.

The working section, circulating pump and air content and pressure control systems constitute the main components of a general purpose cavitation tunnel. The design of the flow ducting joining them together is governed by considerations of economy and hydrodynamic efficiency. The requisite design rules are similar to those pertaining to closed circuit subsonic wind tunnels, with the added proviso that under operational conditions, cavitation should only occur within the experimental flow in the test section. Since the design of subsonic wind tunnels is so well documented^[11,12,13], no further mention is made in this section of the design methods for diffusers, turning vanes

and other flow conditioning devices. Rather, the relevant rules will be outlined as required in their application to the design of the Oxford University High Temperature Cavitation Tunnel in subsequent chapters.

CHAPTER I.1

I.1 ESTABLISHMENT OF TUNNEL DESIGN PARAMETERS

An outline specification was evolved for the cavitation tunnel, with the object of incorporating as many as possible of the desirable features noted in the Introduction. The completed specification thus required that the tunnel should:

1. Be of closed circuit form.
2. Have a two-dimensional test section with transparent viewing windows on all four faces bounding the flow.
3. Have a variable-temperature capability in order to provide Reynolds number variation at constant velocity and to enable the study of thermodynamic effects in cavitating flows.
4. Have a continuously-variable test section velocity up to 60 m/sec. This was taken as the original design value, but was subsequently reduced to 30 m/sec. at a later stage in the design process.
5. Be capable of using a variety of working fluids for the investigation of fluid scale effects in general, and thermodynamic effects in particular.
6. Have effective systems for the control of gas content, pressure and velocity.

When designing a cavitation tunnel, the test fluid is normally prescribed, and the fluid properties, together with the required range of test section velocity, cavitation number and fluid temperature govern the detailed design of the tunnel. When designing a tunnel to

accommodate a variety of working fluids, however, the selection of the test fluids must be made together with the definition of the tunnel operating envelope in order to arrive at a design which is practical in terms of technical feasibility and cost.

I.1.1. THERMODYNAMIC DESIGN CONSIDERATIONS

The first stage in the fluid selection process was the estimation of the potential thermodynamic effect achievable with the prospective test fluids over a wide temperature range. The fluids selected for this exercise were hydrogen, nitrogen, carbon dioxide, ammonia, 4 saturated paraffins (C_nH_{2n+2}), water and 6 of the 'Freon' family of fluids. In order to quantify the potential thermodynamic effect obtainable with these fluids, their 'vapour pressure depression' was calculated on the basis of the thermodynamic model of Stephanoff^[14]. The vapour pressure depression (ΔH_v) is related to the vapour-to-liquid volume ratio (β) by the expressions,

$$\Delta H_v = \frac{\rho_g}{\rho_f} \cdot \frac{h_{fg}^2}{(1 - \frac{\rho_g}{\rho_f})} \cdot \frac{\beta}{C_{pf} \cdot T_v} = \frac{\rho_g}{\rho_f^2} \cdot \frac{s_{fg}}{(\frac{ds_f}{dP_v})} \cdot \beta \quad I.1$$

The variable 'B' is discussed more fully discussed on page 91.

These expressions are implicit functions of ΔH_v [15,16]

since the fluid property terms on the right hand side of Equ. I.1 must be evaluated at the cavity pressure (free stream vapour pressure minus the vapour pressure depression). Thus the relationship between ΔH_v and β , at a given free stream vapour pressure, is nonlinear, as shown by the characteristic decreasing slope plots of ΔH_v versus β shown in Fig. I.3. The nonlinear nature of the ΔH_v versus β

relationship renders it unwieldy for the purposes of comparing the thermodynamic effect of various fluids over a wide temperature range, for which a single figure of merit, for each fluid at a given temperature, is more appropriate. To achieve this end, the slope at the origin of the ΔH_v versus β relationship was used as an index of the potential thermodynamic effect that might be exhibited by a fluid at a given free stream saturation temperature, i.e.,

$$\omega = \left. \frac{\partial \Delta H_v}{\partial \beta} \right|_{\beta=0} = \frac{\rho_g^2}{\rho_f} \cdot \frac{h_{fg}^2}{\left(1 - \frac{\rho_g}{\rho_f}\right)} \cdot \frac{1}{C_{pf} T_v} = \frac{\rho_g}{\rho_f^2} \frac{S_{fg}}{\frac{dS_f}{dP_v}} \quad \text{I.2}$$

in which the fluid properties on the right hand sides of the equation are evaluated at the free stream temperature. Values of ω were computed for all the prospective working fluids over a temperature range of approximately 50% to 95% of their critical temperatures. Since ω changes by several orders of magnitude over the temperature range considered, the ancilliary variable,

$$K = \log_{10}(10^3 \omega) \quad \text{I.3}$$

was introduced to bring all the computations into a reasonable range. Fig. I.4 shows K plotted against dimensionless temperature T_R . The temperature ranges are equalized for all the fluids by normalizing the temperatures onto the critical value, T_{CR} , for each fluid. The plots for the various fluids are all of the same form, increasing in value monotonically with T_R over the temperature range considered. For chemically simular fluids, K and hence ω increases with decreasing molecular weight at a given value of T_R . This trend is broadly followed by all the fluids,

although some of the plots have larger slopes and thus intersect those of other fluids, as, for example NH_3 and H_2O . Fig. I.5 shows K plotted against absolute temperature and the salient feature of this plot is that it indicates that a significant thermodynamic effect is achievable with a number of working fluids above a temperature of 300°K .

I.1.2 TUNNEL OPERATIONAL TEMPERATURE RANGE

The choice of the low temperature operational limit had a profound influence on the design and manufacture cost of the cavitation tunnel. It was decided at the outset to utilize the 60 KW hydrostatic drive of an existing cold water cavitation tunnel^[17]. Since the High Temperature Cavitation Tunnel (HTCT) would have to be charged with fluids which might be costly, one of the main design requirements was to minimize the total circuit volume, and an initial target value of 0.3 M^3 was aimed at. With such a large power dissipation (60 KW) into this small fluid volume, it was clear at the outset that without a heat exchanger, the resulting change in fluid temperature would preclude stable experimental conditions. Thus it was necessary to provide a heat exchanger capable of dissipating the total circulating pump input power of 60 KW. In order to operate the tunnel at sub-ambient temperatures, a refrigeration plant would be required, the cost of which would increase with decreasing rig operating temperature. It was clear that major simplifications in tunnel design and reductions in production costs would accrue if the heat exchanger could be cooled with mains water. This measure would preclude the testing of fluids such as H_2 , N_2 , NH_3

and the lower molecular weight hydrocarbons, and would only be tolerable if a significant thermodynamic effect could be obtained with fluids at temperatures above ambient. As noted in Section I.1.1, Fig. I.5 indicates that a significant thermodynamic effect can be obtained with a number of fluids above a temperature of 300° K. Consequently, a mains water cooled heat exchanger was specified, allowing the HTCT to operate at a minimum temperature of approximately 300° K. It was subsequently decided to reject the heat from the heat exchanger coolant flow in a falling-film air cooled secondary heat exchanger. This means that, in practice the minimum operating temperature of the HTCT depends on the ambient air temperature.

At the upper end of the tunnel operating temperature range, the main design problems encountered were those of the structural integrity of certain components in the circuit. 'O'-ring seals were selected to seal all joints in the circuit, in order to prevent the ingress of air into the working fluid when operating under sub-atmospheric pressure conditions. The type of 'O'-rings selected (Viton) are limited to a maximum operating temperature of 475° K. The test section was required to have glass viewing windows on all four faces bounding the flow, and the design of these would present substantial problems with fluid temperatures in excess of 475° K. Moreover, with reference to Fig. I.5, a significant thermodynamic effect can be obtained with a variety of fluids below a temperature of 475° K. In particular, water shows a considerable ΔH_v at this temperature, which is important since water is usually the reference substance in experiments in scaling thermodynamic effects between different fluids.

Table I.1 shows the vapour pressure depression parameter, $\left. \frac{\partial \Delta P}{\partial \beta} v \right\} \beta = 0$, obtainable for some of the prospective test fluids at temperatures less than or equal to 475° K.

Where the critical temperature of a fluid is in the region of 475° K, the vapour pressure depression parameter is quoted at a reduced temperature, T_R , value of 0.95, due to uncertainties in the fluid property data in the region of the critical point. The values of $\left. \frac{\partial \Delta P}{\partial \beta} v \right\} \beta = 0$ shown in Table I.1 are given in units of pressure (bars), whereas the data presented in Fig. I.4 are based on units of specific energy (J/KG). The pressure units are used in the former case, since this is the quantity which could be measured in any quantitative experiment. The Table indicates the order of the maximum vapour pressure depressions attainable with water and three of the 'Freon' fluids. The corresponding data for the cryogenic fluids are not presented since the selection of a heat exchanger rejecting heat at ambient temperature precluded the cryogenics as test fluids. The data for the hydrocarbon fluids are likewise omitted from Table I.1 since it became clear, following consultations with the appropriate authorities, that safety regulations would not allow their use as working fluids at any available sites for locating the HTCT at the Oxford University Engineering Laboratory. Thus on the basis of the data presented in Fig. I.4, Fig. I.5 and Table I.1, the 'Freon' compounds and water constitute desirable test fluids in that they exhibit large vapour pressure depressions, particularly over the upper portion of the tunnel operational temperature range (370 to 475° K). Further, the thermodynamic properties

Table I.1 Salient thermodynamic properties and vapour pressure depressions attainable with certain test fluids with a maximum tunnel operating temperature of 475° K.

Fluid	Vapour pressure depression $\left. \frac{\partial \Delta P_v}{\partial \beta} \right\} \beta=0$	Temp. °K	Pressure bar	Reduced temp.	Atmospheric boiling point °C	Critical pressure bar	Critical temp. °K
H ₂ O	1.52	478	17	0.738	373		
Freon 114	4.9	400	23.2	0.955	277	32.6	413
Freon 113	3.1	461	23.3	0.955	321	34	487
Freon 11	4.67	444	29.6	0.943	297	44	470

of these fluids are well documented, and the following design and operational simplifications accrue from their use:

- a. Non-flammable and non-explosive, thus greatly easing the constraints on tunnel location and the deployment of accident containment systems.
- b. Low level of toxicity, thus reducing the hazard to personnel consequent upon a fluid spillage from the containment structure.
- c. Commercially available in the required quantities and purity at a reasonable price.
- d. Chemically compatible with commonly used engineering materials.

I.1.3 TUNNEL OPERATIONAL PRESSURE RANGE

The maximum tunnel pressure was determined by:-

- a. The maximum test section velocity.
- b. The maximum test section cavitation number required.
- c. The vapour pressure and temperature characteristics of the working fluids.

It was initially decided that a maximum test section velocity, U_T of 60 m/sec. should be achievable, at a corresponding maximum test section cavitation number σ_T (based on free stream vapour pressure) in the region of 2. Using the data of Table I.1 the values of σ_T attainable at the corresponding conditions were calculated and are shown in Table I.2, for a stagnation pressure P_{STAG} of 69 bar. At this stagnation pressure and $U_T = 60$ m/sec., reasonable

values of σ_T result, and the maximum tunnel design pressure was specified as 69 bar.

After further consideration of the hydrodynamic design of the circuit the following factors emerged:-

a. With a maximum value of $U_T = 60$ m/sec., it was thought that severe erosion of the test section walls might occur, particularly at the end-span locations of a two-dimensional cavitating body^[18]. This might greatly reduce the working life of the test section. It was thus decided that a lower speed test section be used initially ($U_T = 30$ m/sec.), but that the tunnel be designed with the capability of being uprated to $U_T = 60$ m/sec. as required at some later date.

b. A circulating pump of the type required and with a suction pressure capability of 69 bar could not be obtained commercially within the program cost constraints, the closest to this requirement being a unit with a suction pressure capability of 37 bar.

In the light of the above factors, it was decided to retain the circuit design pressure specification of 69 bar, but to reduce the test section design velocity to 30 m/sec. and the maximum tunnel operating pressure to 35 bar. The test section cavitation numbers thus achievable, corresponding to the conditions of Table I.1, are shown in Table I.2. Reasonable values of σ_T are attainable with water and Freons 113 and 114, but not with Freon 11, due to its higher vapour pressure.

At the lower end of the pressure range, it was deemed prudent that the tunnel should be capable of operation at sub-atmospheric pressures, both for the

Table I.2 Test section cavitation number (σ_T) based on
free stream vapour pressure, obtainable at
various test section velocities and stagnation
pressures.

FLUID	σ_T AT: $-u_T=60$ m/sec $P_{STAG.}=69$ bar	σ_T AT: $-u_T=30$ m/sec $P_{STAG.}=35$ bar
H ₂ O	2.16	3.27
Freon 114	1.46	1.46
Freon 113	1.41	1.41
Freon 11	1.15	0.005

purpose of de-aerating the working fluid and for operating as a conventional cold water cavitation tunnel. A summary of the circuit design parameters for the low speed test section configuration is shown in Table I.3.

I.1.4 WORKING FLUID GAS CONTENT CONTROL

As noted in the Introduction, one possible solution to the problem of gas content is the use of a de-aerator to control total air content and a resorber to control free gas content. An initial feasibility study for a resorber system, on the basis of the design parameters given in Table I.3, indicated that a resorber to meet these pressure and temperature requirements would be far beyond the cost ceiling of the program. Further, the capacity that it would add to the tunnel would invalidate the requirement of a total circuit volume in the region of 0.3 m^3 . It was thus decided to make provision for the removal of dissolved gas by boiling the tunnel contents prior to a given experiment with no subsequent control of free gas content.

Table I.3 Summary of tunnel design parameters

Circuit design pressure	68 bar
Maximum operating temperature	473 ^o K
Minimum operating temperature	300 ^o K
Maximum test section velocity	30 m/sec
Maximum operating pressure	37 bar
Minimum operating pressure	vacuum conditions
Working fluids	initially H ₂ O and Freons. Any fluid compatible with 18/8 stainless steel, soda lime glass, 'O' ring seals and circulating pump mechanical seals; also safety regulations covering location.

CHAPTER I.2

HYDRODYNAMIC DESIGN OF THE HIGH TEMPERATURE CAVITATION TUNNEL

I.2.1 GENERAL CONSIDERATIONS

Having fixed the operational pressure, temperature, test section velocity range and prospective working fluids, it was possible to commence the hydrodynamic design of the circuit. It had been decided at the outset to utilize an existing 60 KW hydraulic power supply to drive the tunnel circulating pump, and thus the circuit was designed around this input power. In order to cater for the requirement of a maximum test section velocity of 60 m/sec. it was clear that a very high proportion of the test section dynamic pressure would have to be recovered in order to give a test section of reasonable size. As previously mentioned, preliminary estimates indicated a circuit capacity in the range 0.3 to 0.4 m³. Assuming the latter value, then at the full power input of 60 KW, with water as working fluid, the bulk working fluid temperature would rise at approximately 0.05° C per second. It was thus clear that a heat exchanger would be required in order to give stable temperatures for a reasonable running period. Further, when operating, tunnel working fluids could be at temperatures well above their atmospheric boiling points. Under accident conditions, the heat exchanger would be required to reduce tunnel temperature to a value less than the atmospheric boiling temperature of a given fluid.

To facilitate the hydrodynamic and thermal design of the tunnel circuit a digital computer simulation was made of the complete hydraulic path, the idea being that the requisite parameters such as pipe lengths and diameters could be varied to arrive at an 'optimum' design. In order to produce the simulation, the tunnel configuration shown in Fig. I.6 was adopted. The layout is very similar to that of a typical low speed wind tunnel, with the exception of the heat exchanger and pump, and the complete working section (cascade bend inlet to diffuser outlet) was designed on the basis of subsonic wind tunnel practice, the objective being to produce a plane, low-turbulence flow in the test section. The recommendations of Bradshaw and Pankhurst^[13] were used as the basis of the working section design.

I.2.2 DESCRIPTION OF TUNNEL CONFIGURATION

Referring to Fig. I.6, the working fluid is driven round the circuit by a centrifugal circulating pump, the discharge line of which contains a flow straightener to assist the removal of flow disturbances generated by the pump. The flow is turned into the working section leg by a cascade bend, the discharge from which enters a turbine type flowmeter. The location of this flowmeter is not ideal, since it would be preferable to have a clear pipe run upstream and downstream of the unit. The only other possible location for it would be in the pump suction leg, where it would be prone to cavitation, with consequent loss of calibration, particularly when operating the test section at low cavitation numbers. From the flowmeter

outlet, the circuit increases in internal diameter, by way of a conical diffuser section, to the settling section, the flow area expansion being assisted by fitting conical splitter vanes in the diffusing section. The settling section contains a honeycombe type flow straightener followed by a number of gauze turbulence damping screens. The contraction at the settling section outlet also serves as a transition from the circular settling section to the rectangular test section. In order to circumvent the difficulties of providing a rectangular to round transition at the test section outlet, the diffuser has a rectangular cross section over its whole length, the area transition being accomplished at its outlet end where any flow separation is less important due to the lower flow dynamic pressure at this location.

The choice of heat exchanger type and configuration posed considerable difficulties since it was required to operate with a variety of fluids and to abstract a variable heat load at fluid input temperatures anywhere between 300° K and 475° K . Further, its working fluid containment structure must withstand the full tunnel operating pressure of 69 bar. To avoid the mechanical complications accruing from the use of an intermediate secondary coolant, the bypass type scheme shown in Fig. I.7 was adopted. With this scheme, the heat exchanger is cooled by mains water and has a bypass loop, the proportion of the total flow passing through the exchanger being governed by the setting of the bypass valves. Thus for operation at the lowest attainable circuit temperature, all the flow is diverted through the heat exchanger. The heat exchanger interacts with the rest of the circuit

hydraulically since the pressure drop across it depends on the bypass to heat exchanger flow rate ratio, which itself depends on the heat load to be dissipated and hence the total circuit pressure drop. One advantage of the bypass configuration is that the efficacy of the scheme is not dependant on the accuracy of the heat transfer predictions. As long as the exchanger can handle the maximum thermal load at the lowest operating temperature, it can dissipate this same load at a higher temperature by suitable adjustment of the bypass control valves. The heat exchanger and bypass fluid streams rejoin at a T-bend to enter the pump suction line. It was assumed that any temperature gradients persisting in the fluid at the end of the suction line, due to incomplete mixing therein, would be eliminated during transit through the circulating pump.

I.2.3 CIRCUIT PRESSURE DROP COMPUTATIONS

To determine the required duty of the circulating pump, the circuit pressure drop, from pump discharge flange to suction flange was computed, the pressure drop equations used being given below.

a. Circular pipe line sections

Due to uncertainties in the nature of the inlet flow conditions to the various circular section pipe runs, pressure drops were computed on the basis of fully developed turbulent pipe flow, i.e.

$$\Delta P = \frac{fLq}{d}$$

I.4

where q is the local dynamic pressure

L is the pipe length

d is the pipe inside diameter

f is the friction factor, given by^[19],

$$f = \frac{0.136}{R_d^{0.25}}, \quad 5 \times 10^3 < R_d < 10^5 \quad \text{I.5}$$

$$f = \frac{1}{(1.82 \log_e R_d + 1.64)^2}, \quad R_d > 10^5 \quad \text{I.6}$$

b. Corner turning vanes

The configuration of the turning vanes is in accordance with the recommendations of Ref. 13, from which the pressure drop was computed from the pressure drop coefficient,

$$C_p = \frac{1.2}{(R_c)^{0.25}} \quad \text{I.7}$$

where R_c is the vane chord Reynolds number.

c. Turbine flowmeter

The pressure loss coefficient for the flowmeter was taken from manufacturers data and is given by,

$$\Delta P = 0.15q \quad \text{I.8}$$

where q is the flowmeter inlet dynamic pressure.

d. Settling section inlet diffuser

In order to err on the pessimistic side, the pressure drop in the 30° included angle settling section inlet diffuser was computed neglecting the effect of the conical splitter vanes, the pressure loss being taken as [20],

$$\Delta P = 0.5q$$

I.9

where q is the inlet dynamic pressure.

e. Settling section

The settling section pressure drop was assumed to be that due to the honeycomb flow straightener and the turbulence damping screens only. The pressure loss coefficient K for the honeycomb was taken as^[13], $K = 0.5$. For the turbulence damping screens, the loss coefficient is given by Hoerner^[20] as,

$$K = C_{DO} \frac{(1-B)}{B^2} \quad \text{I.10}$$

where B is the gauze open area ratio and C_{DO} the isolated wire drag coefficient in a uniform flow of velocity U/B (U being the velocity of the screen onset flow).

f. Contraction

In a well-designed contraction, there are no regions of separated flow and the boundary layers should be thin due to the favourable pressure gradient in the direction of flow. Pope^[21] gives an expression for the pressure loss in terms of the contraction outlet dynamic pressure,

$$\Delta P = 0.32 \frac{fLq}{d} \quad \text{I.11}$$

where L is the contraction length

d is the contraction outlet hydraulic mean diameter

q is the outlet dynamic pressure

f is an 'overall' friction factor

g. Test section

The pressure drop through the rectangular test section was evaluated by considering each of the walls as one side of a flat plate at zero incidence. The pressure drop contribution of an experimental body was allowed for by assuming that it could be represented by a two-dimensional body of thickness 10% of the test section height, and with a drag coefficient of 0.5. The wall drag coefficients were computed from the Prandtl-Schlichting skin friction formula^[22],

$$C_f = \frac{0.455}{(\text{LOG } R_L)^{2.58}} - \frac{A}{R_L} \quad \text{I.12}$$

where R_L is the test section length Reynolds number and

$$A = R_{\text{CRIT}} [0.074 R_L^{-0.2} - 1.328 R_{\text{CRIT}}^{-0.5}], \quad \text{I.13}$$

for each pair of opposite walls. R_{CRIT} , the transition Reynolds number was arbitrarily assumed to be 0.5×10^6 .

h. Diffuser

With a maximum available power input of 60 KW and a projected maximum working fluid flowrate of $0.11 \text{ m}^3/\text{sec.}$, the corresponding allowable pressure drop around the circuit is limited to 5 bar. With some of the Freon fluids running in the test section at 60 m/sec. , the dynamic pressure at the diffuser inlet could be as much as 28 bar. Thus a change of 1% in the diffuser pressure recovery coefficient,

$$C_p = \frac{P_{\text{OUT}} - P_{\text{IN}}}{(\text{inlet dynamic pressure})} \quad \text{I.14}$$

would represent a pressure loss equivalent of 5% of the available circuit pressure drop. It was thus clear that caution would have to be exercised in the design of the diffuser. In order to minimise the possibility of flow separation, the maximum included angle between each pair of walls was limited to 5 degrees. The diffuser pressure recovery coefficient was estimated by a method due to Stratford^[23], based on an analysis of the maximum static pressure rise that a turbulent boundary layer can sustain without separating. The expressions used to compute the pressure coefficient C_p as a function of diffuser length X , were,

$$C_p = 0.645 \{ 0.435 R_o^{1/5} [(\frac{x}{x_o})^{1/5} - 1] \}^{2/n} \quad \text{for } C_p < \frac{n-2}{n+1} \quad \text{I.15}$$

$$C_p = 1 - \frac{A}{(x+B) \cdot (\frac{1}{1+H/2})} \quad \text{for } C_p > \frac{n-2}{n+1} \quad \text{I.16}$$

The diffuser was preceded by the test section walls, which were each assumed to be one side of a flat plate at zero incidence. x_o is the distance from the virtual origin of the turbulent boundary layer on the test section walls, assuming a transition Reynolds number of 0.5×10^6 , to the diffuser inlet, and R_o the Reynolds number based on this length. The exponent n is given by,

$$n = \log_{10} R_o \quad \text{I.17}$$

H is the boundary layer shape factor, assumed as 2.6. A and B are constants evaluated from the condition of continuity of C_p and $(\frac{dC_p}{dx})$ at $C_p = \frac{n-2}{n+1}$, i.e. from Equ. I.15

$$X_* = X_o \left[\left\{ \left(\frac{n-2}{n+1} \right) \frac{1}{0.645} \right\}^{\frac{n}{2}} \frac{1}{0.435 R_o} \right]^{1/5} + 1 \right]^5 \text{ at } C_p = \frac{n-2}{n+1} \quad \text{I.18}$$

from which

$$B = X_* \left| \frac{3n(n+2)}{2(n-1)} \left[1 - \left(\frac{X_o}{X_*} \right)^{\frac{1}{5}} \right] - 1 \right| \text{ and} \quad \text{I.19}$$

$$A = \frac{3}{n+1} \left\{ \frac{3n(n+2)}{2(n-1)} \left[1 - \left(\frac{X_o}{X_*} \right)^{\frac{1}{5}} \right] \right\}^{\frac{1}{1+H}/2} \quad \text{I.20}$$

Reference 23 shows that Equs. I.15 and I.16 may be expressed in terms of the diffuser dimensionless length based on inlet boundary layer displacement thickness, i.e.

$$\frac{L}{\delta_o^*} \text{ where } L = (X - X_o) \text{ and } \delta_o^* = \left(1 + \frac{2}{n} \right) \cdot 0.036 X_o R_o^{-0.2} \quad \text{I.21}$$

When compared to experimental data for a number of diffusers over a wide range of L/δ_o^* , the foregoing method overestimated the recovery coefficient by about 5%. For the purpose of estimating the circuit power absorption, the diffuser pressure recovery coefficient was computed as,

$$C_p = C_{p/\text{THEORETICAL}} - 0.05 \quad \text{I.22}$$

i. Heat exchanger

The pressure drop through the heat exchanger and its associated bypass line is a function of the bypass to heat exchanger flow rate ratio. The bypass ratio depends on the working fluid temperature setpoint, the heat transfer characteristics of the heat exchanger, the transport properties of the working fluid and the thermal load to be dissipated. This latter quantity is proportional to the total pressure drop around the circuit. Because of

this interaction between the heat exchanger and the rest of the circuit, the heat transfer and pressure loss equations were solved simultaneously. Fig. I.7 shows a layout of the heat exchanger and bypass system. The bypass flow appropriate to the desired circuit operating temperature is set by the butterfly type bypass valves, a fine control valve being used for very low heat exchanger flows which might be beyond the rangeability of the butterfly valves. The tube bundle is circular in section with the tubes brazed into the circular tube plates at each end. One of the major constraints on the tube design was to maintain the thermal stress in the tube wall at a reasonable level, particularly at the inlet end of the tube bundle, where, at the maximum tunnel operating temperature, the temperature difference between the primary and secondary fluids could be as much as 170° K.

The tube bundle heat transfer and pressure drop were computed on the basis of the flow being turbulent at all locations within the tubes. When operating at high circuit temperatures and hence low heat exchanger flow rates, the axial temperature profiles along the tube bundle are very steep, decreasing from 475° K at the inlet end down to values as low as 320° K at exit. Conversely, when operating the tunnel at its lowest temperature, all the working fluid passes through the heat exchanger and the tube flow approaches isothermal conditions. The latter case is simple to analyse, but the former is more difficult due to the large temperature gradients in both the radial and axial directions.

In order to accommodate the axial variation in fluid transport properties, the heat transfer and pressure drop

equations were solved for elemental lengths Δx of a typical tube (Fig. I.7b), and the results summed over the active tube length to give the total heat transfer and pressure drop. Within each elemental length Δx , a linear axial variation of fluid temperature was assumed. Primary side heat transfer coefficients were computed from the relation (24)

$$\frac{h_p}{C_p G_p} \left(\frac{C_p \mu}{K} \right)^{0.66} \left(\frac{\mu_w}{\mu_b} \right)^{0.14} = \frac{0.023 (1 + d/L)^{0.7}}{\left(\frac{d G_p}{\mu_b} \right)^{0.2}} \quad \text{I.23}$$

where h_p is the primary side heat transfer coefficient

C_p is the specific heat at the primary side film temperature

G_p is the mass velocity of the flow within the tubes

μ is the dynamic viscosity

K is the thermal conductivity

d is the tube inside diameter

L is the distance from the tube inlet

Suffixes b and w denote bulk and wall conditions respectively. The secondary side heat transfer coefficient was estimated from (24),

$$\frac{h_s D}{K} = 0.33 \left(\frac{C_p \mu}{K} \right)^{0.33} \left(\frac{D G_s}{\mu_f} \right)^{0.6} \quad \text{I.24}$$

where h_s is the secondary side film coefficient

D is the tube outside diameter

f denotes evaluation at film conditions

G_s is the mass velocity of the secondary fluid

The heat transfer between primary and secondary fluids, for each axial length element was computed from,

$$\Delta Q = \frac{\pi \cdot D \cdot \Delta X (\theta - T_{sB})}{\frac{d}{Dh_p} + \frac{D}{2K_w} \log_e \frac{D}{d} + \frac{1}{h_s}} \quad I.25$$

the terms of which are defined in Figs. I.7b, I.7c and I.8. The pressure drop for each axial length element was computed from,

$$\Delta(\Delta P) = \frac{f \cdot x \cdot G_p^2}{\rho_f \cdot d \cdot 2} \quad I.26$$

where f is the local friction factor as defined by Equations I.5 and I.6, the Reynolds number being evaluated at the bulk mean temperature of the element. Tube entrance and exit losses were estimated using a loss coefficient of 0.2 for the inlet^[20], and for the outlet,

$$K_{OUT} = \left(1 - \frac{A_t}{A_d}\right)^2 \quad I.27$$

where A_t is the total flow area of the tube bundle
 A_d is the flow area of the duct immediately after the tube plate. The pressure drop across the butterfly valves, as a function of vane angle, were taken from manufacturers' data.

The pressure loss equations for the hydraulic circuit external to the heat exchanger, and the pressure loss and heat transfer equations for the heat exchanger were programmed for solution on a digital computer. Block diagrams of the programs used in this exercise are shown in Figs. I.9, I.10 and I.11. The master program POWER was used to compute the pressure drop and flow power absorption of the complete circuit. This program basically solved the previously quoted pressure loss and heat transfer equations for a range of working fluid temperatures and test section velocities. At each temperature, the tunnel performance

was simulated for a range of test section velocities starting at the lowest required value (taken as 2 m/sec. for computational purposes). The velocity U_T was incremented by the program until the point was reached where the circuit power absorption reached 60 KW, the maximum power available from the hydrostatic drive. This velocity was taken to be the maximum attainable test section velocity at the given temperature, for the particular fluid. Due to the aforementioned interaction between the heat exchanger and the remainder of the tunnel circuit, an iterative method was used to effect a solution of the heat exchanger pressure drop and heat transfer equations, using a subprogram named HEATU, which was called as required by the master program POWER. Fig. I.10 shows a block diagram for this program, which basically solves the heat transfer and pressure loss equations for successive elemental lengths of a typical tube.

Experimentation with the program showed that increasing the number of length elements beyond 19 produced little change in the computed primary fluid axial temperature distributions, even at the highest fluid inlet temperature (475° K) and thus this number of length elements was used throughout the subsequent computations. In order to start the heat transfer and pressure drop calculations, an initial approximation of the flow rate through the heat exchanger was required. This was taken as,

$$m_o = \frac{\dot{Q}}{C_p(T_1) \cdot T_1 - C_p(T_{sB}) \cdot T_{sB}} \quad \text{I.28}$$

Thus an initial estimate was obtained by neglecting the flow power absorption of the heat exchanger, and assuming no temperature difference between primary and

secondary fluids at the tube outlet location. Equ. I.28 gave a surprisingly good initial estimate of the final converged value of the primary fluid mass flow rate. This was so, since the assumed initial thermal dissipation, being set equal to the power absorption of the circuit excluding the heat exchanger, was always less than the converged value of the thermal dissipation (this being the sum of the power absorption for the external circuit and the converged power absorption of the heat exchanger). Conversely the converged value of the primary fluid outlet temperature ($T_{2,CONVERGED}$ in Fig. I.9) was always greater than the bulk mean secondary temperature T_{sB} . These two effects appeared to offset each other to a large extent, resulting in a good initial estimate for M_o from Equ. I.28. With this value of M_o the heat transfer equation (Equ. I.25) was solved for each axial length element ΔX , using Eqs. I.23 and I.24 to evaluate the appropriate heat transfer coefficients. An iterative solution was used to determine the heat transfer and hence the radial and axial temperature distributions for each length element ΔX . This computation was performed by a subprogram named HITER which was called by the subprogram HEATU as required. As a first approximation to the temperature distribution within a new length element, the element outlet temperature was assumed equal to the converged outlet value of the previous element (isothermal flow). The temperature differences across the primary, tube wall, and secondary thermal resistances were assumed equal, as a computational starting value (see Fig. I.7c). Using this linear radial temperature distribution, the primary and secondary side heat transfer coefficients were computed and Equ. I.25 solved to obtain a second approximation to the radial

temperature distribution. For the i^{th} iteration on the j^{th} length element, the element outlet temperature $T_{j+1, i+1}$ was obtained from the appropriate form of the steady flow energy equation,

$$T_{j+1, i+1} = T_j - \frac{\Delta Q_{j, i+1}}{M_k \cdot C_p \cdot (\theta_i)} \quad \text{I.29}$$

from which the bulk mean temperature $\theta_{j, i+1}$ of the element is,

$$\theta_{j, i+1} = \frac{1}{2} (T_j + T_{j+1, i+1}) \quad \text{I.30}$$

The iteration procedure was continued until the element outlet temperature changed by less than 0.1%, at which point computational control was returned to subprogram HEATU. The above iterative method was used for successive length elements of the tube bundle until the tube outlet location was reached. The subprogram HEATU then checked whether the computed tube outlet temperature, $T_2(K)$, had changed by more than 0.1% of the previously estimated value, $T_2(K-1)$. If it had, the primary fluid mass flow rate was updated according to the relation,

$$M_{K+1} = \frac{C + L_K}{C_p(T_1) \cdot T_1 - C_p(T_{2, K}) \cdot T_{2, K}} \quad \text{I.32}$$

where M_{K+1} is the K^{th} updated estimate of the tube bundle mass flow, C the external circuit power absorption and L_K the K^{th} value of heat exchanger power absorption. The mass flow (K) iterations were repeated until convergence of L_K , M_K and the primary fluid temperature distributions were obtained, at which point subprogram HEATU returned program control to the master program POWER. Program POWER then incremented the test section velocity U_T by 1 m/sec. and the circuit flow power absorption and heat

exchanger thermal dissipation were again computed for this velocity. When the computed circuit power absorption reached 60 KW, the maximum power available from the hydrostatic drive, the program reset the velocity to 2m/sec. and incremented the working fluid temperature by 5 K and repeated the simulation of the tunnel performance. The requisite fluid properties were written as sub-programs in the form of polynomials in temperature, and were called by the programs POWER, HEATU and HITER as required.

These programs were written in Fortran 4 language and executed on an I.B.M. 1130 digital computer. Print outs of these programs have not been included in this thesis, since the programs were long and contained no unusual features, the computer being used merely as a rapid calculator. The program POWER was used interactively, using the computer console keyboard to adjust the geometric parameters of the tunnel, such as test section size, pipe diameters and lengths and heat exchanger configuration. These simulations continued until an 'optimum' circuit design was evolved, based on engineering judgement. These simulations were performed for both the high speed ($U_T = 60$ m/sec.) and the low speed ($U_T = 30$ m/sec.) test section configurations. The heat exchanger design parameters were also adjusted to ensure that the unit would dissipate the tunnel power absorption with water, Freon 113, Freon 114 and the higher molecular weight saturated paraffins as working fluids. Fig. I.12 shows a typical simulated tunnel power absorption curve for the high speed test section configuration, for water at a desired operating temperature of 325° K.

As previously noted, the tunnel performance was computed for test section velocities up to that corresponding to a circuit power absorption of 60 KW, the maximum power available from the hydrostatic drive. In practice the tunnel could never attain a power absorption of 60 KW, since this would imply a circulating pump efficiency of 100%. The pump efficiency was not included in the hydrodynamic simulation since its value was not known a priori, indeed one of the purposes of the simulation was to obtain data to permit selection of a suitable pump. Since it had previously been decided to construct the tunnel initially with a low speed test section, the circulating pump duty point was based on this configuration. Table I.4 shows the nominal maximum operating condition for water and Freon 113 at a temperature of 300° K, this giving the maximum circuit pressure loss at a given test section velocity and hence volumetric flowrate. The maximum attainable velocity for Freon 113 at this temperature was computed to be 28.8 m/sec., this being limited by the power available assuming a circulating pump efficiency of 80%. The pump head rise and flowrate were specified using the water data of Table I.4, with a margin of 10% allowed on the computed circuit pressure loss to compensate for any underestimate of it. The pump design duty was thus specified as,

Head rise 25.9 m of water
flowrate 7.1 m³/min

The pump selected was a British Labour type 70 MHC mixed flow horizontal suction machine with a suction pressure and temperature rating of 37 bar and 475° K respectively and obtaining the above duty when running at 1500 r.p.m.

Table I.4 Circuit pressure loss and volumetric flowrate for water and Freon 113 at 300° K

Test section configuration	High speed test section 54 mm x 36 mm		Low speed test section 76.2 mm x 50.8 mm	
	Water	Freon 113	Water	Freon 113
Test section velocity m/sec	61	50 *	30.5	28.8 *
Circuit pressure drop. bar.	3.79	4.14	2.93	4.14
Working fluid volumetric flow-rate m ³ /sec.	0.118	0.1135	0.118	0.1115
* Limited by available pump power, assuming pump efficiency of 80%.				

CHAPTER I.3

GENERAL DESCRIPTION OF CIRCUIT COMPONENTS

The following circuit dimensions were evolved from the hydrodynamic design exercise and tunnel performance simulations outlined in the previous section, and the HTCT was ultimately constructed according to these.

a. Pipework connecting major tunnel components

The pump suction and discharge pipes, the heat exchanger bypass pipe and all adjoining bends and tees are of 127 mm inside diameter pipe. Hence the turbine flow-meter and butterfly valves are also of the same bore.

b. Settling section

The diameter of the flow duct inside the settling section is 305 mm and the overall length 706 mm, this latter dimension being dictated by the requirement to accommodate a flow straightener and ten wire gauze turbulence damping screens spaced at a centre distance corresponding to 500 gauze wire diameters^[13].

c. Contraction

The contraction is 305 mm inside diameter at inlet and 76.2 mm high and 50.8 mm wide at outlet, giving a contraction ratio of 18.85. A detailed description of the design method used for this item is given in Section I.4.4.

d. Test section

The low speed test section is 76.2 mm high, 50.8 mm wide and 280 mm long. It is rectangular in section throughout, no specific provision being made to compensate for

boundary layer growth by tapering out the walls, on account of the manufacturing problems involved.

e. Diffuser

The diffuser is rectangular at all cross sections and is so designed that, by insertion of an extension piece at the inlet end, it can be used with a high speed test section (54 mm high by 36 mm wide). Fig. I.13 shows a line diagram of the diffuser internal profile. The diffuser outlet dimensions were based on an outlet flow area equivalent to that of a 127 mm bore pipe, and with a maximum wall divergence of 5° included angle.

f. Heat exchanger

The heat exchanger tube bundle configuration which emerged from the computer simulation is shown in Fig. I.14. The large number and thin walls of the tubes was dictated by thermal stress requirements, the details of which are given in Section I.4.8, which also outlines the shell and tube-plate design details.

CHAPTER I.4

DESIGN AND CONSTRUCTIONAL FEATURES OF THE MAJOR COMPONENTS OF THE HIGH TEMPERATURE CAVITATION TUNNEL

All parts of the circuit constituting the pressure containment structure were designed to withstand a maximum pressure of 69 bar and a maximum temperature of 475° K. Certain components of the circuit were designed in accordance with the A.S.M.E. VIII Code for the Design of Unfired Pressure Vessels, where their detail design was covered by this code. All components generally were designed in accordance with its recommendations in respect of working stress levels. In order to make the circuit chemically compatible with as wide a range of working fluids as possible, all the metallic parts of the circuit were specified to be made from 18/8 stainless steel. To prevent the ingress of air when operating under vacuum conditions, 'o' ring type seals were specified at all joints.

I.4.1 PIPE COUPLINGS

The circuit has 17 joints in the 127 mm nominal bore pipework, and the use of conventional flanged couplings for all these would have produced a very heavy assembly, due to the size of the flanges required to meet the maximum design pressure of 69 bar. In particular, the settling section and heat exchanger shell required the provision of joints with a bore of approximately 300 mm. The use of flanged couplings in these components would have been extremely expensive and would have drastically

increased the weight of the tunnel over that ultimately achieved. 'Gray lock' quick release type couplings were used wherever possible, and details of a typical 127 mm bore coupling of this type are shown in Fig. I.15. The coupling consists of a pair of flanges with tapered outer faces which engage with a corresponding female taper in the clamp element. The flange neck reduces down to the pipe diameter with a 14 degree transition section and the front flange faces are relieved to ensure that the bearing portions of the faces remain in contact under load. Flange alignment is provided by a loose sleeve which engaged with recesses in the flanges. The alignment sleeve also forms the inner face of a triangular section 'o' ring seal housing. The clamp assembly consists of two semicircular split rings hinged by a pin joint. The assembly is drawn up with a nut and swing-bolt at the top of the clamp. The swing-bolt housing is so formed that when the nut is slackened, the bolt can be swung clear of the housing and the clamp removed completely from the joint. Since only one nut has to be tightened to make the joint, assembly is much quicker than with a conventional flanged joint. The couplings used on the settling section and heat exchanger are similar in principle, but have a spigot arrangement in place of the alignment sleeve, and a three-segment clamp for ease of assembly. The joints adjacent to the turbine flowmeter and butterfly valves are of the bolted flange type since these items were only available with flanged connections.

I.4.2 CORNER TURNING VANES

The corner vane assembly is shown in Fig. I.16. The cascade of vanes was hydrodynamically designed in

accordance with Ref. 13, which recommends the use of sheet metal vanes with a pitch to chord ratio of 0.25, the trailing edge at zero incidence, and a leading edge incidence of 4 to 5 degrees. The vane thickness was determined by providing a sufficient vane section modulus to withstand the bending loads resulting from the momentum change of the working fluid in passing through the cascade. The vane details are shown in Fig. I.16. They were produced by slitting a 76 mm diameter tube in four positions at right angles with a cutter of the appropriate diameter to produce the required 86 degree vanes. The corner cascade body was designed to a pressure rating of 69 bar and because of the awkward shape caused by the reinforcement required at the intersection plane, was produced from a casting. The difficulties of fixing the vanes in the correct position inside the casting were overcome by using vertical vane mounting plates, the slots being dimensional so as to correctly position the vanes on assembly. After a trial set up the vanes were tack-welded to the mounting plates and the mounting plates tack welded to the casting.

I.4.3 SETTLING SECTION

A longitudinal section through the settling section assembly is shown in Fig. I.17. The 127 mm bore flange on the inlet end couples with the outlet flange of the turbine flowmeter. The 30 degree conical diffuser is an undesirable feature hydrodynamically on account of the possibility of flow separation occurring in its entrance region and the consequent disturbance persisting downstream to affect the test section flow. To minimise this possibility, three concentric conical splitter vanes were

fitted inside. The outlet end of the diffuser has a gas collection dome welded on top of it, with a communicating hole drilled through the diffuser wall to allow gas access to the dome. The dome is provided with two outlet ports, one to accommodate a safety blow-off valve and the other for a de-gassing line. The parallel portion of the settling section is formed from rolled and welded plate, and is joined to the diffuser by a quick-release type pipe coupling, alignment being achieved by a spigot on the parallel section, which also houses an 'o'-ring seal. Because of the way in which it was manufactured, the parallel part of the settling section was considerably eccentric, on account of welding distortion. To avoid machining it internally over its whole circumference, with consequent thinning of the vessel walls, four guide strips were tack-welded into the bore, these then being machined concentric to the vessel axis (see detail, Fig. I.17). The flow surface was formed by 318 mm outside diameter and 306 mm inside diameter annular rings located externally by the aforementioned guide strips, and housing the flow straightener and wire gauze screens with their associated spacers. These annular housing and spacer rings are a sliding fit on the guide strips and are tensioned after assembly by a ring of compression springs located in the spacer at the outlet end (see detail Fig. I.17). The flow straightener consists of cells made from 9.5 mm outside diameter by 0.1 mm wall thickness tube brazed together inside one of the annular spacer rings. The length of this flow straightener is 51 mm, giving a cell width to length ratio of 5.5, which corresponds to that recommended in Ref. 13.

In order to design the damping screens an intensity value of the u and v components of the turbulence upstream of the screens has to be assumed. These were arbitrarily assumed to be 10% ($T_u = T_v = 0.1$). Bachelor^[27] shows that the reduction factors for the u component of turbulence energy T_u , are greater than those for the v component, T_v , for screen resistance coefficients of, $0 < K < 4.5$. Moreover, the effect of the contraction is to reduce T_u more than T_v . Thus if the screen design is based on reducing T_v in the test section to a reasonable load, the consequent reduction in T_u should be greater. A further requirement of turbulence damping screens is that the open area ratio be greater than 0.6 to avoid unsteadiness associated with instabilities in the exit flow from the screens^[13]. With a gauze of 40 mesh, 39 s.w.g. (0.132 mm wire diameter), the Reynolds number at a volumetric flowrate of 0.118 m³/sec. of water at 300° K is about 4000. The drag coefficient of an equivalent isolated wire in a uniform stream, C_{DO} , is therefore of order one. Hoerner^[20] gives the screen resistance coefficient K as,

$$K = C_{DO} \cdot \frac{(1-B)}{B^2} \quad \text{I.32}$$

where B is the screen mesh open area ratio, and for a 40 mesh 39 s.w.g. gauze, is 0.627. The corresponding screen resistance coefficient, K, is, from Equ. I.32, 0.95. Using the experimental data presented by Bachelor^[27], the reduction in T_v for K = 0.95 is,

$$r_v = 0.67 \quad \text{I.33}$$

Thus if ten screens are used with an assumed value of

$T_v = 0.1$ upstream of the screens, then the corresponding value of T_v downstream of the screens is,

$$T_{v2} = T_{v1} \cdot r_v^{10} = \frac{0.1}{54.8} = 0.001825 \quad \text{I.34}$$

For the contraction, Bachelor quotes a reduction in T_v of,

$$r_v = \frac{\sqrt{0.75}}{C_R} \quad \text{I.35}$$

where C_R is the contraction ratio and is 18.85 in the present exercise. With this value, $r_v = 0.2$ in Equ. I.35. The value of T_v downstream of the contraction is thus,

$$T_v = 0.00185 \times 0.2 = 0.000365 \quad \text{I.36}$$

As will be shown in Section II.2, the actual value of T_u , as evidenced by the relationship between the base pressure and Reynolds number for a two-dimensional cylindrical test body, was considerably higher than that estimated above, having a value in the region $T_u = 0.01$. This large difference between the actual and estimated values of T_u may be partially accounted for by the fact that the screen design was based on the assumption of homogeneous turbulence. This assumption is unlikely to be true under actual tunnel running conditions, since the wakes shed from both the turbine flowmeter blades and the conical splitter vanes at the settling section inlet almost certainly persist to the screen inlet locations. However, the design method outlined above did at least give the screen design some scientific basis.

The gauzes are positioned by clamping them between the annular rings as shown in the detail of Fig. I.17. In order to maintain the stress level in the screens at a safe value they were allowed to sag by 5% of their diameter

(15 mm). The screen spacing was determined by the length required for the decay of the turbulence generated by the screen wakes. The value of 500 wire diameters quoted in Ref. 13 was used, giving a screen spacing of 66 mm.

I.4.4 CONTRACTION

The hydrodynamic design of the contraction was not straightforward, since the cross section changes from a circular one at inlet to a rectangular one at the outlet end. The first possibility considered was that of making an area transition at the end of the settling section and designing a contraction of rectangular shape at all cross sections. This possibility was rejected, however, since for a given settling section diameter, it would reduce the contraction ratio by some 30%. Moreover, the design of the transition section would be almost as difficult as the design of a transitional contraction. The flexible membrane technique for designing transitional contractions [28] was initially considered but rejected on account of the experimental work involved. It was decided to adapt some form of axisymmetric design method, since the main parameters of the contraction could then be readily varied to produce an appropriate design. Several approximate methods of computing the axisymmetric potential flow in contractions have been reported [29,30], in which a centre line velocity is assumed and the resultant stream lines computed by series solution, one of the streamlines being chosen as the contraction wall. These methods involve a large amount of computation, and since a transitional contraction is somewhat different from an axisymmetric one, it was reasoned that a simple design method would give as

good a result as a more involved one. One such design technique is that of Rouse and Hassan^[31], who solved the axisymmetric potential flow equations by the electrolytic bath method for a wide variety of contraction shapes. The objective of their method was to produce wall profiles for which the outlet velocity 'overshoot' or pressure coefficient 'undershoot' was minimised. The wall profiles were defined by cubic polynomials and the contraction design data arising from their study were presented as shown in Fig. I.18. The graph shows the maximum recommended value of X/L , the dimensionless length up to the contraction inflexion plane plotted against coordinates of inlet to outlet diameter ratio and length to inlet diameter ratio. Rouse and Hassan stated that to minimise the local pressure drop at the contraction outlet, the contraction should be designed with parameters which lie to the right of the boundary line in Fig. I.18. Since this technique had been used to design a contraction for a cascade tunnel, with good results^[32], it was modified for use with transitional contractions.

With the axisymmetric design problem, third order polynomial curves are fitted to the inlet section, up to the inflexion plane, and to the outlet section, the coefficients of the cubic equations being determined by the known boundary conditions, as shown in Fig. I.19.

If the wall polynomials are,

$$Y_1 = A + Bx_1 + Ex_1^2 + Dx_1^3, \quad Y_2 = a + bx_2 + cx_2^2 + dx_2^3$$

I.37

Axisymmetric section up to inflexion plane	Transition section down- stream of inflexion plane

and the boundary conditions are,

$$x_1 = 0, y_1 = \frac{D_1}{2}, \frac{dy_1}{dx_1} = 0 \quad x_2 = 0; y_2 = \frac{D_m}{2}, \frac{dy_2^2}{dx_2^2} = 0 \quad \text{I.38}$$

$$x_1 = X; y_1 = \frac{D_m}{2}, \frac{d^2y_1}{dx_1^2} = 0 \quad x_2 = (L-X); y_2 = \frac{D_m}{2}, \frac{dy_2}{dx_2} = 0 \quad \text{I.39}$$

Solving the polynomials with these boundary conditions gives,

$$y_1 = \frac{D_1}{2} - \frac{3}{4} \frac{(D_1 - D_m)}{X^2} x_1^2 + \frac{(D_1 - D_m)}{4X^3} x_1^3 \quad \text{I.40}$$

$$y_2 = \frac{D_m}{2} - \frac{3}{4} \frac{(D_m - D_2)}{(L-X)} x_2^2 + \frac{(D_m - D_2)}{4(L-X)^3} x_2^3 \quad \text{I.41}$$

with a common slope at the inflection point of,

$$\left. \frac{dy_1}{dx_1} \right|_x = \left. \frac{dy_2}{dx_2} \right|_0 = -\frac{3}{4} \frac{(D_1 - D_2)}{X} \quad \text{I.42}$$

Since D_1 , D_m and D_2 lie on a straight line

$$\frac{D_1 - D_m}{X} = \frac{D_m - D_2}{(L-X)} = \frac{D_1 - D_2}{L} \quad \text{I.43}$$

The above equations apply strictly to an axisymmetric contraction where, for any given contraction ratio, D_1 , D_2 , D_m , X and L have unique values. The equations are applied to a transitional contraction in the following manner. Referring to Fig. I.20, instead of the contraction having a unique outlet diameter, it has a variable one, $D_2(\theta)$, where

$$D_2(\theta) = \frac{W}{\cos \theta}; \quad 0 < \theta < \tan^{-1}\left(\frac{H}{W}\right) \quad \text{I.44}$$

$$D_2(\theta) = \frac{W}{\sin \theta}; \quad \tan^{-1}\left(\frac{H}{W}\right) < \theta < \pi/2.$$

However, D_1 , D_m and D_2 are constrained to be on a

straight line in Rouse and Hassan's method, and D_m and D_1 are constant since the inlet section is axisymmetric. These conditions can be satisfied if the transition section length $(L-X)$ is allowed to vary according to the relation,

$$(L-X) = \frac{(D_m - D_2(\theta)) X}{(D_1 - D_m)} \quad \text{I.45}$$

Thus the coefficients of the polynomial representing the transitional section walls are functions of θ . $(L-X)$, the length of the transitional section is also a function of θ , and is maximum for a minimum value of D_2 . This has the effect of making the contraction two-dimensional beyond a certain point, as shown in Fig. I.21, the location of the point depending on the outlet aspect ratio H/W .

A unique value of D_2 is required to determine the coefficients of the axisymmetric inlet polynomials. This is taken as the area mean outlet diameter of the contraction, since the method is based on potential flow.

For the 306 mm inlet diameter and 72 mm equivalent outlet diameter of the present tunnel contraction the diameter ratio D_1/D_2 is 4.33 and lies just off the top of the design plot of Rouse and Hassan (Fig. I.18). The design point chosen is shown by the arrowed asterisk. Since extrapolation of the data was required, a conservative value of $(X/L)_{\max}$ was chosen.

Some experimentation had to be performed to find the right range of angular increment $\Delta\theta$ which would define the contraction walls sufficiently to allow manufacture of the contraction within reasonable tolerances. In order to reduce the computational effort, the polynomial equations were programmed onto a digital computer. The contraction shape was defined numerically and it was found that an

angular increment of 5 degrees was adequate to define the cross sectional shape, but in general this will depend on the inlet to outlet diameter ratio, length to inlet diameter ratio and the outlet aspect ratio. Axial length increments of 16 mm (2.6% of the overall length) were found adequate to represent the variation in shape down the contraction, although this will in general be subject to the same factors as the angular increment. The actual cross sectional areas at each axial length increment were also computed and compared with the area of the corresponding axisymmetric contraction based on the area equivalent outlet diameter. The areas corresponded to within $\pm 3\%$ at all axial stations, but again, in general, correspondence depends on outlet aspect ratio etc. Cross sectional profiles at 10% transition length increments are shown in Fig. I.22, for one quadrant of the contraction. It may be seen that most of the area transition occurs in the last half of the transition section length, which corresponds to $1/3$ of the total contraction length. This conforms to Bradshaw's recommendation that with a transitional contraction, the area change should be accomplished in the last third of the contraction length^[33].

I.4.5 CONTRACTION MANUFACTURE

The contraction was made in two portions, being split at the inflexion plane as shown in Fig. I.23. Both parts were made from 18/8 stainless steel forgings, the inlet section being hollow forged and the outlet section solid. The inlet section wall profile was produced by copy turning from a template.

In view of the difficulties of getting any form of cutting tool inside the transition section bore, it was

decided to spark machine the wall profiles. A wooden male master pattern of the contraction shape was produced from laminated plywood. Since the important dimensions of the transition section are the inflexion plane diameter and angle, the outlet end sizes and the overall length, these were fixed independently of the wooden pattern by setting it on a metal former as shown in Fig. I.24. The important contraction dimensions were thus defined and the plywood laminations could be blended out accurately onto the end pieces. After making the wooden pattern, a plaster of paris female mould was taken from it, the metal former inserted into the mould, and epoxy resin poured into the cavity through holes in the inflexion plane steel plate. From the male pattern thus produced, graphite electrodes were copy-milled. Dimensional accuracy at the transition inlet and outlet were also obtained by conically boring the inlet end of the forging, to a diameter of 266 mm and a semi angle of $14^{\circ} - 32'$. The outlet end was spark machined with a 76.2 mm by 50.8 mm rectangular graphite block. The internal profile was then spark machined to blend with the pre-machined inlet and outlet sections. After joining the inlet and transition sections, the joint line was polished out by hand. Fig. I.25 is a photograph taken from the inlet end of the contraction looking down the axis. Fig. I.26 shows the complete contraction assembly. The small pipe fitting near the inlet end of the contraction was provided to accommodate a thermocouple for measuring the working fluid temperature.

I.4.6 TEST SECTION

Details of the test section construction are shown in Fig. I.27. A major design requirement was the provision

of four windows in the test section to enable the use of optical experimental methods. The design finally evolved gives line of sight through more than 90% of the test section depth. Since any glass adjacent to the test section flow would be prone to cavitation damage during tunnel operation it was deemed highly undesirable for such glass to bear any of the loading consequent upon the high maximum tunnel operational pressure. A dual window scheme was thus adopted in which the flow is guided by a pressure-balanced inner flow guide window and the pressure load taken by armoured glass windows away from the flow. With this design the flow guide windows could be made of untoughened soda-lime glass and regarded as sacrificial in the event of cavitation damage, since they would be inexpensive and simple to produce. The flow guide windows consist of four 10 mm thick soda lime untoughened glass windows ground on their upper and lower faces to engage with shoulders machined into the test section body. The manufacture of the metal and glass shoulders was closely controlled to give a step of less than 0.025 mm between metal and glass. The glass flow guide windows are backed up by a sliding seal which runs in a rectangular bore with radiused corners. A sliding seal was necessitated by the differential thermal expansions consequent upon the tunnel operating temperature range of 300° K to 475° K. The sliding seal contains a number of light compression springs located in holes in the top and bottom faces, their purpose being to ensure that the flow guide windows are always pressed firmly against their locating metal shoulders in the main test section body. The sliding seals contain a circumferential 'o' ring seal, and a face seal against the main pressure windows. These latter windows are made of

25 mm thick toughened boro-silicate glass, two of them being mounted back to back on each face of the test section. They are separated from each other by a p.t.f.e. joint, which acts as an elastic foundation to prevent the generation of excessive contact stresses at local high spots on the windows. Each of these glasses will safely withstand the full rated tunnel pressure on its own, so that the outer glass is there as a safety glass. The pressure windows are backed up by pressure plates, which have a series of counter bored holes containing Bellville washer stacks running on guide pins. The function of these Bellville washers is to provide a low stiffness means of tensioning the pressure windows, sliding seal and inner windows during temperature-induced dimensional changes. For the relatively small thermal expansions of the glass and metal constituting the expansion path, the Bellville washers are effectively a constant force device. The pressure plates are rigidly restrained by the window clamp plates, which are bolted directly onto the test section body. All the aforementioned components are so toleranced as to preclude lock-up of the glass windows at higher tunnel operating temperatures. The test section is joined to its mating components, the contraction and diffuser, via 16 mm thick packer plates. The function of these is to form a section in which profile blending can be performed and which can be easily replaced for drilling holes for model mountings and instrumentation access tappings. They are dowelled onto the test section and its mating components to ensure accuracy of location.

The test section body was manufactured from an 18/8 stainless steel forging, the inner profile and the stepped rectangular recesses in the sides being produced by spark

machining. The spark machining of the inner profile was done from the outlet end so that the taper arising from electrode wear would assist in ameliorating the effects of boundary layer growth on the test section walls. Fig. I.28 shows an oblique side elevation of the test section mounted between the contraction and diffuser.

I.4.7 DIFFUSER AND RECTANGULAR-TO-ROUND TRANSITION SECTION

The main constructional features of the diffuser and transition section are shown in Fig. I.29. The tapered portion of the diffuser consists of four suitably shaped 18/8 stainless steel plates welded at their abutting edges to form a box section. Full penetration welds with a large root gap were used to produce a fillet rather than a sharp corner on the inside, thus reducing the corner stress concentration factor. The diffuser outlet section was machined from a flame cut blank and circumferentially welded onto the box section. The weld joint was made well away from the outlet end so that the outlet dimensions would not be affected by weld shrinkage, which was substantial in the box section on account of the large corner welds. Weld shrinkage at the inlet end caused both a reduction in the dimensions and bowing of the main plates. This was rectified by milling out the inlet section walls parallel to the diffuser centre line and blending out the resulting joint line, as shown in the detail of Fig. I.29.

The rectangular-to-round transition section was produced by machining a tapered bore from the outlet end back to within 50 mm of the inlet end face, the bore diameter at this latter location corresponding to the

diagonal dimension of the diffuser outlet flow passage. A rectangular hole of the same dimensions as the diffuser outlet was machined in from the front face to break into the tapered bore. 16 s.w.g. sheet metal flow guides were tack welded into the bore to produce an approximately constant area transition. Since the inlet end of the conical bore constitutes a high point in the tunnel, provision was made to remove any gas tending to collect at this location. The de-gas point consists of a hole drilled through the joint flange to communicate with the conical bore, as shown in Fig. I.29. The uppermost flow guide has holes which allow gas to pass through to the de-gas point.

I.4.8 HEAT EXCHANGER

The main features of the heat exchanger are shown in Fig. I.30. The heat transfer element consists of 1027 tubes 4.75 mm outside diameter, 0.5 mm wall thickness which are brazed into two 25 mm thick circular tube plates to give an active tube length of 254 mm. The tubes were sized on the criteria of maintaining the maximum stress levels within the yield point of the tube material (18/8 hard drawn stainless steel), and maintaining the end load per tube below the buckling value. The point of maximum stress occurs at the tube outside diameter, where the tangential thermal stress adds to the tensile stress caused by the tube internal pressure load. The determination of the maximum thermal stress was done with the aid of the circuit performance simulation programs POWER, HEATU and HITER, since the wall temperature

gradients, thermal resistance and thermal stress level are functions of the tube dimensions.

The tube plates were designed with unrestrained ends, using the method of Miller^[34], since this produces a thinner tube plate for a given loading than a restrained tube plate configuration. Moreover, the free end design makes it easier to allow for the thermal expansion of the tubes. The tube plate at the inlet end of the heat exchanger usually operates at a higher temperature than the outlet end. Since the temperature gradients in the tube plates are complex, it is difficult to predict the radial expansion of a tube plate under operating conditions. To circumvent this problem a sliding seal was provided for the inlet tube plate (see detail, Fig. I.30), the design philosophy being that the sliding seal would tend to attain the primary fluid temperature and could expand and contract freely. The outlet tube plate is sealed by a conventional bore type 'o' ring seal backed up by anti-extrusion rings.

Cooling water enters through two pipes screwed into the main pressure shells, and flows across the tube bundle to the diametrically opposite outlet ports. To prevent channelling of the coolant flow, a split annular distributor casing completely surrounds the tube bundle. Six horizontal baffle plates are positioned through the tubes, parallel to the plane of the coolant entry ports, and the distributor casing is drilled with graded holes to promote uniform coolant distribution. The heat exchanger pressure containment shell is formed of two 18/8 stainless steel castings joined together on the longitudinal centre-line by a quick release type coupling, alignment being achieved by a spigoted joint which contains a low pressure

coolant 'o' ring seal. To ensure a reasonable inlet flow to the tube bundle, the 30 degree included angle inlet diffuser is fitted with a three element conical splitter vane assembly. The two main pressure shell castings are provided with 1.5 mm diameter de-gas passages at their uppermost points, since these locations constitute potential gas traps. The heat exchanger loop pipework was fabricated from 18/8 stainless steel proprietary pipe-fittings. The butterfly type bypass flow control valves are of the wafer type, being through-bolted between their mating pipe flanges. They are manually actuated through levers mounted on the butterfly shafts.

I.4.9 CIRCULATING PUMP

As noted in Section I.1.3 some difficulty was encountered in obtaining a commercial pump of the required characteristics, within the cost ceiling of the project. This difficulty was caused primarily by the high suction pressure requirement of 35 bar at a temperature of 475° K. The pump ultimately selected was a British Labour type 70 MHC mixed flow machine. In standard form this pump had a suction pressure capability of 30 bar at 475° K and some modifications had to be made to increase this. These consisted of producing the pump casing in 18/10 stainless steel rather than the 18/8 type used as standard, thus allowing a higher working stress level. The bolted connection between the pump casing and rear closure plate had to be strengthened by the provision of more bolts of a higher strength material.

Fig. I.31 shows the operating characteristics of the family of different diameter impellers that may be run in

the type 70 MHC pump. The required duty point was obtained by using a 12 inch (305 mm) diameter impeller. The somewhat low efficiency obtaining at the designated duty point (65%) is a consequence of the fact that the pump casing is hydrodynamically designed for a 14 inch (356 mm) diameter impeller and thus the 12 inch (305 mm) impeller is mis-matched with the pump volute. However, since the power absorption of the tunnel in the low speed test section configuration and with water as working fluid, is considerably less than that available from the hydrostatic drive, this low pump efficiency is of no immediate consequence. Fig. I.31 further indicates that if the tunnel is modified to incorporate a high speed test section then the corresponding pump duty point could be approached, although not attained, by fitting a 14 inch (356 mm) diameter impeller). In this case, the operational pressure of the tunnel would still, of course be limited to 37 bar. This limitation can only be circumvented by installing a different pump with the requisite suction pressure capability.

The type 70 MHC pump is fitted with a mechanical shaft seal which was specified to be chemically compatible with water and Freon 113.

I.4.10 MANUFACTURE, ERECTION AND TESTING OF TUNNEL STRUCTURE

Due to the size and weight of most of the tunnel components, the majority of the tunnel manufacture was performed by specialist organisations. In particular the heat exchanger shells, corner cascade housing, all the quick release type pipe couplings and the fabrication of

the tunnel pipework were performed by Destec Engineering (Lincoln) Ltd. The diffuser was manufactured completely by Norris Brothers (Reading) Ltd. The transitional portion of the contraction was produced by A. H. Austin (London) Ltd. and the test section spark machined by Holt and Mosedale (Warley) Ltd.

Smaller components such as flow straighteners, turning vanes, and wire gauze screens were produced in the workshops of the Engineering Science Department. All main structural welds on tunnel components were specified in accordance with British Standard B.S. 1515 Part 2, Class I, with 10% radiographic inspection. All castings were likewise specified to be radiographically inspected.

The designated location for the High Temperature Cavitation Tunnel (HTCT) was the Engineering Science Department Annexe, at 19 Parks Road, Oxford. This is also the location of the Oxford University Cold Water Cavitation Tunnel, parts of the hydrostatic power transmission of which are used to drive the HTCT circulating pump. The tunnel was mounted on a structural steel framework in the disposition shown in Fig. I.32. This framework was fixed rigidly to the Laboratory floor using a specialized adhesive (Adhesium). The only part of the tunnel which is fixed to the framework is the circulating pump. The remaining parts of the circuit rest on support stands rising from the structural steel framework. The circuit is supported thus since the tunnel structure was estimated to expand by 12 mm in length and 4.5 mm in breadth over the operational temperature range. This expansion is accommodated by the tunnel components sliding over their respective support stands with respect to the fixed location of the circulating pump. Figs. I.32 and I.33

show photographs taken during the erection of the tunnel. Fig. I.32 shows, from right to left, the corner cascade, turbine flowmeter, settling section, contraction, test section and diffuser located in their final positions. The left hand side of the photograph shows the heat exchanger loop under construction. The details of the tunnel support structure may also be seen in this picture. Fig. I.33 is a photograph taken from above the heat exchanger looking down along the tunnel. In the left foreground is the test section and contraction. To the right hand side of the test section is the pressurizer plenum with its associated liquid level gauge. On the right hand side of the picture is the circulating pump and its suction and discharge pipes.

When the tunnel had been completely assembled the circulating pump was removed from the circuit and the open ends of the suction and discharge pipes closed off with purpose made blank plugs. The circuit was then filled with water and hydrostatically pressurized to 56 bar to test the structural integrity. This value is 50% above the maximum tunnel operating pressure, as specified by British Standard B.S. 1515 Part 2. The circulating pump casing had previously been subjected to a witnessed hydrostatic test to the same pressure by the manufacturers, but was removed from the circuit to prevent possible damage to the mechanical seal unit.

CHAPTER I.5

TUNNEL ANCILIARY SYSTEMS

This chapter outlines the details of the hydrostatic drive and velocity control system, the pressure, temperature and volume regulation systems and the method used for adjusting the gas content of the working fluid.

I.5.1 HYDROSTATIC DRIVE AND VELOCITY CONTROL SYSTEM

As noted in Section I.1.2, it was decided at the outset of the project to utilize part of the 60 KW hydrostatic power transmission driving the circulating pump of the Oxford University Cold Water Cavitation Tunnel. A schematic of this hydrostatic transmission is shown in Fig. I.34. The input power to the system is derived from a 75 KW three-phase electric motor running at a nominally constant speed of 750 r.p.m. This is mechanically coupled to a Lucas Hydraulics Ltd. Type PM 1000 DB2 variable displacement swash plate pump. This unit supplies oil to a fixed swash plate Type PM 1000 DB2 hydraulic motor which is mechanically coupled to the driven element. The mechanical couplings are made using toothed pulleys and belts, the velocity ratios of the pulleys being determined by the electric motor and driven element speeds. The pump swash plate is actuated by a double acting swash servo piston. The actual control variable is motor output shaft rotational speed rather than tunnel flow velocity, the assumption being that for a constant tunnel hydrodynamic resistance, flow velocity is proportional to pump rotational speed. The control /

system is of the cascade type, as shown in Fig. I.34. Desired changes in circulating pump speed are initiated from a setpoint potentiometer. This signal, together with that from a tachogenerator mounted on the driven element shaft are fed into a controller giving proportional and integral action. The output from this unit forms the setpoint to a proportional controller, the other input to which is a signal generated by a transducer measuring the position of the swash servo piston. The output from this position controller energises a Moog type electro-hydraulic spool valve, which actuates the swash servo piston to produce the required swash plate deflection and hence change in oil flow. No further mention will be made of the engineering aspects of the hydrostatic power transmission in the form in which it drives the circulating pump of the Cold Water Cavitation Tunnel since these have been previously reported in detail by Bailey^[35], who originally installed it. Rather the discussion will outline the modifications which had to be made to the hydrostatic power supply to enable it to drive the circulating pump of the HTCT. To do this, isolating valves were installed in the pipes conveying the power and return oil flows and the suction boost and cooling and vent flows between the existing swash plate pump and motor. Additional pipework was installed so that the aforementioned flows could be ducted to a second fixed swash hydraulic motor coupled to the circulating pump of the HTCT. Fig. I.35 is a schematic of the modified hydrostatic drive. Since the HTCT is located on the floor above the site of the hydrostatic power supply, the additional pipework rises up to the first floor. The electronic control system serves both cavitation tunnels,

a single changeover switch being used to connect in the appropriate tachogenerator and velocity setpoint potentiometer. The maximum rotational speed of the Cold Water Cavitation Tunnel circulating pump is 750 r.p.m., whereas that of the HTCT is 1500 r.p.m. Since the same control system is used for both tunnels, the tachogenerator is connected to the HTCT circulating pump shaft through a two to one reduction gear. As shown in Fig. I.35 the power oil flow and return pipes are connected with the requisite hydraulic motors using the valves RH, RC, FH and FC.

Closure of the valves FH and RH isolate the flow and return power oil pipes from the motor driving the HTCT circulating pump. Likewise the valves FC and RC isolate the power oil flow and return pipes to the hydraulic motor driving the circulating pump of the Cold Water Cavitation Tunnel.

A secondary small bore pipe system connects the hydraulic pump and motors with a 7 bar gear pump and a heat exchanger. The gear pump draws oil from the low level reservoir and discharges it into the power oil return line and into the casings of the hydraulic motors. Return pipes connect the motor casings with the heat exchanger and reservoir. The purpose of the gear pump is to provide enough pressure at the pump suction and motor discharge ports to prevent cavitation and also to circulate oil through the low pressure parts of the pump and motors to abstract the power dissipation in these units. The flow from the pump also replenishes fluid lost in leakage between the high and low pressure oil paths in the pump and motors. The flow and return pipes of this secondary system may be diverted to the appropriate motor by using

the valves V1, V2, V3 and V4. The valves V1 and V3, in the low pressure return line, should never be totally closed, since this low pressure line also acts as a vent line for the internal motor leakage flows. Closure of these valves may result in the failure of the spring retaining bolts which locate the hydraulic motor main shafts.

Fig. I.36 is a photograph of the circulating pump drive mechanical details. The power from the hydraulic motor is transmitted through a toothed belt to a pulley mounted on a layshaft between self-centering bearings. The tachogenerator with its reduction gear drive is shown protruding from the layshaft on the left hand side of the picture. The power is transmitted from the layshaft to the circulating pump shaft through a flexible coupling which compensates for any misalignment of the shaft during assembly and for thermal expansion of the circulating pump when operating the HTCT at elevated temperatures.

I.5.2 WORKING FLUID TEMPERATURE REGULATION SYSTEM

Fig. I.37 is a schematic plan view of the HTCT and the 1st floor of the Cavitation Laboratory showing the disposition of the tunnel heating and cooling systems and the location of the concomitant controls. Heat is supplied to the working fluid by glass wool insulated Nicrome heater tapes wrapped around the pump suction, pump discharge and heat exchanger bypass pipes of the tunnel. These heater tapes are rated at 1 KW each and are connected to a three-phase electric power supply through a solenoid-actuated contactor situated on the east wall of the laboratory. The contactor unit contains lamps which

indicate whether the tapes are switched on. The contactor solenoid is energised by a variable setpoint thermostat located above the tunnel diffuser inlet. This thermostat consists of a mercury-in-steel thermometer, the indicating temperature gauge of which contains the appropriate contacts to make or break the three-phase contactor solenoid circuit. The thermometer bulb pad is located on the tunnel pipework at the flowmeter outlet, and hence measures the temperature of the outside of the tunnel structure rather than the working fluid inside it. Thus the temperature indicated by it is not accurate enough for experimental purposes but is quite adequate for monitoring the tunnel performance, which is its function. Using the thermostatically-controlled power contactor, the tunnel may be left unattended, for example to increase the temperature by a large amount, which takes considerable time. Alternatively, in the performance of experiments at elevated temperatures, the thermostat may be used to maintain the temperature overnight for resumption of experiments the following day. It is most important that this latter procedure only be performed with the appropriate vent valves open and closed (see Section I.5.3), and in any event the thermostat setpoint temperature must never be greater than 95° C when the tunnel is left unattended.

Heat is abstracted from the working fluid using the tunnel heat exchanger described in Sections I.2.3 and I.4.8. The cooling water of this unit passes through a fan-assisted falling film type cooling tower situated outside the east wall of the laboratory on the ground floor level. The cooling water is circulated through the tunnel heat exchanger by a centrifugal pump located at the base of the

cooling tower and the flow and return pipe connections are located on the laboratory west wall adjacent to the heat exchanger. Each pipe is surmounted by a tee-fitting providing two flow and return ports corresponding to those on the heat exchanger shell. The tee-fittings are terminated by Saunders valves from which the cooling water is ducted to the heat exchanger by flexible hoses running at floor level. The on-off switch for the cooling tower pump and fan is located on the laboratory east wall, adjacent to the heater tape controls. Indicator lamps show when the fan is running and whether the circulating pump is running or stalled. Heat abstraction is adjusted primarily by varying the flow passing through the heat exchanger. This is done using the two bypass flow control valves, BV1 and BV2, one of which is situated in the bypass leg and the other in the heat exchanger loop, as shown in Fig. I.37. The operational sequence of these valves is very important. When heating up the working fluid with the tunnel running, valve BV1 is maintained fully open and BV2 fully closed, the total flow then bypassing the heat exchanger. To abstract heat, valve BV2 should be used initially, valve BV1 being maintained fully open. If, and only if, the tunnel operating temperature is such that the required heat abstraction cannot be effected even with BV2 fully open should valve BV1 be moved from its fully-open position. With valve BV2 fully open and BV1 fully closed, the tunnel temperature will attain its lowest possible operating value at a given tunnel flowrate. On shutting down after a run in which the setting of valve BV1 has been moved from its fully-open position, the valve should be returned to its fully-open position before leaving the tunnel. It is most important that both BV1 and BV2 never

both be closed simultaneously since they are of the butterfly type and the resulting pressure drop across them could bend the butterfly spindles. This could cause failure of the spindle seals resulting in the escape of working fluid. This would be particularly hazardous when operating at elevated temperatures and pressures. For the same reason these valves should never be used to throttle the tunnel flow. Using the valve operation sequence outlined above ensures that one of the valves always remains fully open.

A secondary function of the heat exchanger is to cool the tunnel rapidly in the event of a hazardous situation arising. In these circumstances it is necessary to depressurize the tunnel as quickly as possible. This may not be done, however, before the working fluid has cooled to below its atmospheric boiling point. This is done (assuming that the cooling tower fan and pump are operating) by fully opening valve BV2 and fully closing valve BV1, in that order. This will cool the tunnel in the shortest possible time. The tunnel structure is thermally insulated with glass fibre matting both to reduce heat losses from the tunnel and to allow a tolerable working environment in the laboratory.

I.5.3 PRESSURE AND VOLUME CONTROL SYSTEMS

As outlined in the Introduction, pressure regulation is an important consideration in cavitation tunnel design and operation, since for a given flow velocity and temperature the degree of cavitation on a test body is directly proportional to the tunnel static pressure. With a variable temperature tunnel there is the further

requirement of accommodating temperature-induced volumetric changes in the working fluid. To satisfy this requirement, a gas on free liquid surface type pressurization system was adopted, as shown in Fig. I.38. The pressurizer plenum is located adjacent to the tunnel test section, with its vertical centre line (nominal free surface level) in the same plane as the test section axis (see photograph, Fig. I.33). Near to the plenum is a stainless steel sump tank of 0.3 m^3 capacity. The lower end of the plenum is connected to the sump tank, the tunnel test section outlet and the discharge port of a Madan pneumatically-actuated pump, through the pipes and valves shown schematically in Fig. I.38. The line connecting the plenum with the test section is 3 m long and has a bore of 2 mm, the objective being to provide a long diffusion path from the plenum gas space to the working fluid in the tunnel. The gas space of the plenum connects with the gas space of the sump tank, a vacuum pump and the outlet of the pressure regulator of a high pressure nitrogen cylinder. The high points of the tunnel circuit, which constitute potential gas traps are all connected by pipes to the sump tank. Referring to Fig. I.38, valves 1, 2 and 3 are associated with the volume control system, valves 4, 5, 6 and 7 with the de-gassing system and valves 8, 9, 10 and 11 with the pressure control system. Under steady running conditions, with the tunnel pressurized, only valves 2, 8, and 11 are open, all others being fully closed. If the tunnel temperature is lowered, the working fluid suffers a decrease in volume, which is manifested as a drop in the level of fluid in the plenum, which may be viewed at the plenum sight glass. The plenum fluid level is restored by opening valve 1, thereby connecting the Madan pneumatic pump with the plenum liquid

space. The discharge pressure of this pump is preset by means of an air pressure regulator mounted on the pump casing, the discharge fluid pressure being registered on a Bourdon gauge. This pressure should always be set to a value greater than the maximum tunnel static pressure to be used during any experiment, thus ensuring flow from the sump tank to the plenum when valve 1 is opened. Failure to preset the pressure at this valve is not dangerous, since the pump discharge line contains a non-return valve, but if this is not done, the pump will not provide the required flow and the plenum level will continue to fall. This situation can be remedied as it occurs by screwing in the pump air pressure regulator. If this action is not taken, pressurization gas will ultimately enter the tunnel circuit thereby necessitating complete de-gassing of the working fluid. When the plenum liquid level attains the required value, valve 1 is closed to cut off the flow from the Madan pump. Under increasing temperature conditions the plenum level rises. The level is restored to its correct value by opening valve 3 thus spilling fluid from the plenum into the sump tank, which must always be vented to atmosphere (valve 11 open) when running under pressurized conditions.

Valves 8, 9, 10 and 11 in Fig. I.38 are associated with the pressure and vacuum control systems. For operation under pressurized conditions it is imperative that valve 11 in the atmospheric vent line from the sump tank, always be fully open. This is so since the sump tank is of light construction, having a failure pressure of about 2 bar. Since the sump tank fluid temperature may often approach 100° C, and since it is situated close to the tunnel operating area, failure of the tank could prove dangerous to personnel. For pressurized tunnel operation,

valves 11, 8 and 2 in Fig. I.38 are open, all the other valves remaining closed, except during the aforementioned corrective action following a rise or fall in the plenum liquid level. The following pressure regulation technique was evolved from operating experience with the tunnel. From the initial phase of the experiments reported in Part II of this thesis, the main problem of pressure regulation was identified to be that of reducing the pressure in small enough decrements rather than that of setting it at some predetermined value. This was a consequence of the types of experiment performed. All the states of both limited and developed cavitation investigated were induced by lowering the tunnel static pressure. In the case of cavitation inception, it was found that small decrements in pressure were required to achieve the limiting incipient condition. In studying the development of cavitation, very small decrements in pressure were required to characterize adequately the changes in flow regime that were found. Detailed descriptions of these experiments are given in Part II, and for the present purpose it is sufficient to note that the principal requirement of the pressure regulation system is the ability to reduce pressure by very small amounts. To do this, in a given experiment, the tunnel static pressure is set coarsely to a value somewhat greater than that experimentally required using the pressure regulator on the nitrogen cylinder (valves 11, 8 and 2 being open, the remainder fully closed). The plenum is then isolated from the pressure regulator by closing valve 9. Tunnel static pressure may then be lowered in small decrements by bleeding fluid from the plenum to the sump tank using valve 3. The plenum liquid level then falls in proportion to the fluid bled off,

thereby allowing the isolated mass of gas above the liquid in the plenum to expand. In this manner, extremely small pressure decrements may be achieved. If the total pressure decrement required using this method is large, it is best to start the experiment with the plenum more than half full of liquid, since the plenum might otherwise be emptied before the experiment is completed. For any given experiment, trial runs should be performed to establish the appropriate starting level for the plenum fluid. This technique should, in principle, also be applicable when pressure increments are required. In this case the initial plenum fluid level would be low and fluid would be pumped into the plenum in small quantities by opening valve 1. However, as previously stated, all the experiments reported in Part II were performed under decreasing pressure conditions, so no further comments can be made as to the practicability of this latter method.

For operation under vacuum conditions, valve 11 is closed, isolating the sump tank from atmosphere. Valves 9 and 10 are opened to connect the sump tank and the plenum to the vacuum pump. The absolute pressure of the tunnel can then be increased, as required, by closing valve 9 and opening valve 8 to admit nitrogen to the pressurizer plenum. The technique of lowering the tunnel pressure by bleeding fluid from the plenum to the sump tank can be used as long as the plenum pressure is greater than that prevailing in the sump tank. When these two pressures are equal, flow between the plenum and sump tank ceases and the lowest tunnel operating pressure is attained. The safety valve SV1, which has a blow-off pressure of 24 bar, was installed as part of an early modification to the tunnel which was never completed, and is isolated by valve 12 in

Fig. I.38. This safety valve plays no part in the operation of the tunnel in its present configuration. All the aforementioned valves are labelled at their respective locations on the tunnel.

I.5.4 DE-GASSING

Since a design study had shown that it was not feasible to incorporate a resorber in the tunnel circuit to continuously control the free gas content of the working section flow, tunnel gas content is adjusted prior to a given experiment by boiling the working fluid. To do this, valves 4, 5, 6, 7, and 11 (Fig. I.38) are opened, all other valves being closed. The tunnel heater tapes (Fig. I.37) are then switched on and the tunnel temperature increased until boiling of the working fluid commences. Because the tunnel pipework is asymmetric in the vertical plane, once boiling commences the working fluid is circulated around the tunnel by density gradients, in the opposite direction to the normal tunnel flow. The increase in tunnel pressure which follows the onset of boiling causes flow of vapour, liquid and released gas from the tunnel to the sump tank through valves 4, 5, 6 and 7. The outflow pipe from these lines is drowned below the free surface of the sump tank liquid and thus some of the vapour inflow condenses in the sump fluid, the fraction depending on the temperature difference between them, and the remainder is vented to the atmosphere with the released gas flow. To make up for the vented flow, fluid is pumped back into the tunnel by actuating the Madan pump and opening valve 1. To ensure that the sump tank fluid is de-gassed, the tank is evacuated by actuating the vacuum pump, closing valve 11 and opening valve 10. During

de-gassing the sump tank should be continuously evacuated to an absolute pressure of 0.4 bar, which is the minimum operating suction pressure for the Madan pump. Below this pressure the pump suction cavitates and flow ceases completely. The gas content is measured by a Van Slyke Blood Gas Content Analyzer using a 50 cc working fluid sample extracted from the tunnel settling section through valve 13 in Fig. I.38. In practice the tunnel gas content, as measured with the Van Slyke apparatus, may be reduced to 10 parts per million (gravimetric) by boiling the tunnel contents for about 2 hours. By extending the boiling period, gas contents down to 5 parts per million were recorded (Van Slyke measurement). For a given investigation the appropriate working fluid gas content depends on the type of experiment to be performed. The investigations reported in Part II of this thesis were concerned primarily with viscous effects in limited and developed cavitating flows, and the influence of gas content was not systematically studied. Rather, since the investigations relied to a considerable extent on visual observations, the gas content was reduced to low values so that the viscous effects studied would not be obscured by the occurrence of pseudo-cavitation^[36].

I.5.5 VAPOUR VENTILATION SYSTEM

The tunnel is provided with a vapour ventilation system which allows new experiments to be carried out on the entrainment theory of thermodynamic effect. Being fully commissioned and operational, the system is properly regarded as one of the ancilliary systems of the tunnel. Nevertheless, its specification and design were necessarily

preceded by a careful review of previous work on entrainment in natural and ventilated cavities and the methods of correlating the thermodynamic effect, and an analysis of the proposed technique of vapour ventilation. In view of this, the system merits a chapter to itself, and this follows.

CHAPTER I.6

A VAPOUR VENTILATION SYSTEM FOR EXPERIMENTS ON
ENTRAINMENT FROM CAVITATING FLOWS

As the assembly of the cavitation tunnel neared completion, attention was given to formulating an experimental program. It was decided that the initial program would aim at studying vapour entrainment from cavitating flows with a view to obtaining a better physical insight into the entrainment theory of thermodynamic effect. It was recognised that some form of cavity ventilation system would be required in order to obtain quantitative entrainment data. Since the tunnel had no provision for the removal of any gas which might be used to ventilate cavities, it was clear that difficulties would be encountered with gas ventilation. A system was ultimately evolved based on ventilating cavities with vapour of the tunnel working fluid. A theoretical analysis of this system indicated that it might offer significant advantages over foreign gas ventilation techniques. As will be reported in Part II, the proposed entrainment study was not, in the event, performed, since other phenomena were observed which were deemed more important to investigate. Even so, the vapour ventilation system was designed and constructed, and engineering trials and preliminary experiments have shown that it performs its design functions. Since it may be used in conjunction with a variety of test bodies, it may now be regarded as part of the tunnel ancillary equipment but to deal with it adequately it is treated here as a separate project.

I.6.1 NATURAL AND VENTILATED CAVITIES

Cavitation may be produced in the flow field around a body by reducing the static pressure of the flow to some critical value at which cavitation inception occurs. If the static pressure is further reduced the extent of the cavitating region increases, resulting in what is termed the limited cavitation flow regime. Further static pressure reduction causes the cavity to extend in the downstream direction and when the streamwise length of the cavity is of the same order as the characteristic body dimension the cavitation is termed developed. With a confined flow, further reduction in static pressure beyond a certain value produces no change in the cavity geometry and the flow is then in the choked condition. In the developed cavitation flow regime, if the onset flow temperature is small compared to the critical temperature and if the fluid foreign gas content is low, the cavity pressure approximates to the vapour pressure of the fluid at the onset flow temperature. Cavitating flows produced in the aforementioned manner are termed natural cavities and, under the restrictions stated above, are characterised by the cavitation number based on free stream vapour pressure.

$$\sigma_v = (P_s - P_v) / \frac{1}{2} \rho u_\infty^2 \quad \text{I.46}$$

It is well known^[37,2] that cavitation may be induced or increased by introducing a supply of foreign gas into the cavitating or low pressure region of the flow about a body. In this case the relevant cavitation number is based on the total cavity pressure,

$$\sigma_c = (P_s - P_c) / \frac{1}{2} \rho u_\infty^2 \quad \text{I.47}$$

The fact that cavitation may be produced by ventilation demonstrates the existence of a corresponding entrainment process in natural cavities whereby vapour is removed from the cavity. It further indicates that entrainment flowrates from natural cavities may be inferred from ventilation experiments. It is clear that the validity of such a technique depends on geometric and dynamic similarity between the natural and ventilated cavity flows. The influence of some of these similarity parameters is outlined below.

I.6.2 VAPOUR ENTRAINMENT FROM NATURAL CAVITIES

In certain classes of natural cavity flow, vapour is entrained from the cavity by re-entrant jets which are generated in the cavity closure region and flow back through the cavity in the upstream direction. These re-entrant jets, an idealized form of which is shown in Fig. I.39a for a two-dimensional cavity terminating in a free stream, are formed at the juncture of the cavity free streamlines. If the fluid flowing around the cavity outside the free streamline is itself considered as a liquid jet, it is clear from the cavity geometry that the liquid jets bounding the upper and lower free streamlines of the cavity cannot be parallel to the onset flow velocity vector of the downstream end of the cavity. From a consideration of the continuity of mass, energy and momentum for the collision of the two mainstream jets, on the basis of steady two-dimensional flow, it may be seen that an upstream jet results from the collision. If the cavity terminates on a solid body, as shown in Fig. I.39b, it may be shown that a similar jet is formed from the collision of

the fluid outside the free streamline with the solid boundary. The causal mechanism of re-entrant jets as outlined above is an extreme simplification of these jets as observed experimentally. In the latter case, the re-entrant jet motion is of an inherently unsteady character in which the jet flows through the cavity in the upstream direction, thus partially filling the cavities with jet fluid. At a certain point the jet collides with the cavity free streamline, or solid boundary, whereupon the portion of the cavity downstream of the jet breaks away and is convected downstream with the liquid flow. Flows of this type were observed by Knapp et al.^[2] on an axisymmetric head form where the cavity terminated on the body. High speed motion pictures clearly showed the phases of cavity formation, filling and break-off, and revealed the cyclic nature of the re-entrant jet flow. Re-entrant jet flows have been observed on a two-dimensional Göttingen hydrofoil at incidence^[51]. Furness^[38,39] observed a similar sequence of events in a convergent-divergent nozzle, and recorded the cyclic behaviour of the flow with high speed films and a number of pressure transducers sited within the cavitating region. He found that large pressure fluctuations were generated at the point where the re-entrant jet collided with the test body at the upstream end of the cavity. Since the high speed films showed that cavity breakoff occurred at this point, Furness concluded that the aforementioned transient pressure rise suppressed cavitation at the upstream cavity separation point for a sufficient period for the main cavity to become detached and break away. Fig. I.40 shows a schematic of a typical re-entrant jet cycle observed in the above experiments. Furness devised a numerical simulation of the re-entrant jet flow

in his particular experimental geometry, in which the time dependent inviscid flow equations were reduced to a series of boundary value problems at small time increments and solved numerically to yield the jet trajectory, and thus the cavity shape, as a function of time^[40]. The initial values required to start the simulation were obtained from the high speed film records. Although the simulation predicted a sequence of events similar to that observed, the experimental data showed a relationship between onset flow velocity and cavity shedding frequency which became increasingly non-linear with increasing fluid temperature. Data obtained with Freon 11 indicated that below a certain threshold velocity the cavities became much steadier. Furness attributed this to the lower surface tension of Freon 11 producing a bubble filled rather than a vapour-filled cavity in which entrainment occurred by condensation at the trailing end of the cavity. These results illustrate the strong influence of real fluid effects on re-entrant jet motions, and some of the difficulties in predicting the form of entrainment which will occur in a given flow situation.

In the re-entrant jet flows observed by Furness and Knapp et al. virtually the complete cavity broke away at the end of each cycle, except for infrequent cases where only partial breakoff occurred. In other experimentally observed flows at low cavitation numbers the re-entrant jet motion is limited to the trailing end of the cavity, leaving the upstream portion of the cavity substantially steady in terms of its geometry and interior pressure. Examples of this type of flow were reported by Brennan in the axisymmetric flows around discs and spheres^[41] and Gadd and Grant on discs^[42]. One of the differences between

the flows observed by Knapp et al. and Furness on the one hand and Brennan and Gadd and Grant on the other is that in the former case, the re-entrant jet motion was adjacent to the test body, while in the latter the trailing end of the cavity was many body diameters downstream of the bodies in question. That the portion of the cavity influenced by the re-entrant jet motion decreases with decreasing cavitation number for this type of flow, was also found by Reichardt^[37] with ventilated cavities behind discs. It cannot, however be generally stated that the extent of the re-entrant jet motion and the consequent unsteadiness of a cavity decreases at lower cavitation numbers. In experiments on a plano-convex hydrofoil, Wade and Acosta^[43] reported the existence of a partially cavitating and a fully developed flow regime both of which exhibited steady forces and moments. Between the two regimes they found a region of highly unsteady flow with large fluctuations in the hydrodynamic forces and cavity length.

A peculiarity of fully wetted flows around two-dimensional bluff bodies is the generation of Karman vortex streets due to the instability of the separated free shear layers. The occurrence of cavitation within such a flow is known to modify both the vortex street geometry and the characteristic Strouhal shedding frequency. In experiments on two-dimensional wedges of various nose angles, Holl and Young^[44] found that, typically, as the cavitation number was reduced below about one half the inception value, the Strouhal shedding frequency started to increase above the fully wetted value for a given wedge. A further decrease in cavitation number produced a maximum value of Strouhal number, which occurred at a characteristic cavitation number for a given wedge nose angle. Further reduction in

cavitation number caused a sharp decrease in Strouhal number and shedding became spread over a wide band of frequencies. Fig. I.41 shows plots of Strouhal number against normalized cavitation number taken from Ref. 44. The authors concluded that the Strouhal number peak delineated two different flow regimes. At cavitation numbers greater than that corresponding to this peak, the flow was basically a vortex street modified by the presence of cavitation, while below the peak the coherent vortex structure degenerated into a fully developed cavity flow with entrainment occurring at the trailing end of the cavity.

Varga and Sebestyn^[45] investigated vortex shedding from cavitating two-dimensional circular cylinders for cavitation numbers from inception to approximately the choked condition. They found the Strouhal number to decrease smoothly with decreasing cavitation number and from dimensional considerations proposed the relationship

$$S_t = c/\sigma_v \quad \text{where } c \text{ is a constant} \quad \text{I.48}$$

A circular cylinder in a venturi was investigated by Syamala Rao et al.^[46,47], using high speed film records to determine cavity shedding frequency. Typical results presented by these authors, together with those of Varga and Sebestyn are shown in Fig. I.42. It may be seen that at low values of cavitation number the data of Syamala Rao et al. show a minimum in Strouhal number beyond which it increases rapidly with decreasing cavitation number. For cavitation numbers above this minimum value their Strouhal number data approximate reasonably well with the form of Equ. I.48. These authors were unable to propose a physical basis for the rapid increase in Strouhal number shown in Fig. I.42. Ewers^[48] performed experiments on a two-

dimensional circular cylinder in the Oxford University Cold Water Cavitation Tunnel and found a relationship between vortex shedding Strouhal number and cavitation number similar to that reported by Varga and Sebestyn^[45] when operating in the supercritical Reynolds number flow regime. He detected no tendency for the Strouhal number to increase at low cavitation numbers, as reported by Syamala Rao et al. The main deviation between his data and those of Varga and Sebestyn was a tendency for the Strouhal number (inferred from force balance measurements) to increase slightly prior to inception, which he attributed to pseudo-cavitation caused by the relatively high free air content of his tunnel flow. Ewers repeated the aforementioned experiments with a radial water jet flapped cylinder blowing moderately strongly at various jet angles. He reported little change in the Strouhal number versus cavitation number characteristics between all the blown and unblown cases, thus emphasizing the dominance of the vortex shedding mechanism in this cavitating flow regime.

The data of Varga and Sebestyn were performed at a nominally constant supercritical Reynolds number. For the experiments of Syamala Rao et al. the critical Reynolds number of their cylinder mounted in a venturi was not determined, but from the operating conditions quoted it appears that they were performed at Reynolds numbers above the critical value. Mussared^[49] has recently completed a study of vortex shedding behind a two-dimensional circular cylinder in the cavitation tunnel which is described in Part I of this thesis. Vortex shedding was detected using a strain gauge lift balance the output of which was analysed on a spectrum analyser. He reported a strong Reynolds number dependence of the form of the S_t versus σ_v

relationship for the cylinder used in his experiments. Fig. I.43 shows three sets of data obtained by Mussared at Reynolds numbers corresponding to the subcritical, supercritical and transcritical fully-wetted flow regimes. Fully-wetted vortex shedding could only be obtained at subcritical Reynolds numbers and it was concluded that at higher Reynolds numbers either coherent vortex shedding was not occurring or the fluctuating lift forces were too small to detect on the spectrum analyser. The subcritical Reynolds number plot in Fig. I.43 shows that the vortex shedding Strouhal number increased slightly as cavitation developed, thereafter decreasing as the choked condition was approached. For Reynolds numbers greater than the critical value, where no fully-wetted vortex shedding was detected, the inception of cavitation was accompanied by the appearance of narrow band vortex shedding. In the supercritical Reynolds number case, the spectral peak Strouhal number initially increased as the cavitation developed to reach a maximum value beyond which it decreases to converge with the subcritical Reynolds number data at low cavitation numbers. At a transcritical Reynolds number, a similar behaviour was found, with an initial increase in Strouhal number for cavitation numbers immediately below the inception value. Again, at low cavitation numbers the transcritical data merged with that of the other two plots. In all cases, as the cavitation number was reduced below about 1.5 the signal level from the strain gauge decreased and below $\sigma_v = 0.95$ no peak was discernable on the spectrum analyser. In the choked condition a re-entrant jet was clearly visible penetrating upstream through the cavity.

From the foregoing discussion, there appear to be at least three forms of entrainment in natural cavities. Two of these, the re-entrant jet and the cavitating vortex street are essentially dynamic in character. The third is the bubble type entrainment occurring below a certain threshold velocity, as reported by Furness and Hutton^[40] and Knapp et al.^[50]. The latter authors hypothesised that the steadiness of the cavity in this flow regime might be caused by the inability of the re-entrant jet to penetrate more than a short distance upstream into the cavity on account of the low jet momentum consequent upon the low free stream velocity. If this is so, this type of flow is similar to that observed around bodies trailing long cavities where the re-entrant jet effects are limited to the cavity closure region^[41,42], and thus does not constitute a different basic form of entrainment. If the contention of Furness and Hutton^[40], that entrainment in this flow regime is caused by bubble condensation at the cavity closure region, is true, this clearly constitutes a third basic form of vapour entrainment from natural cavity flows. It cannot be concluded that cavitating vortex street flows occur in situations in which the corresponding fully-wetted flow exhibits vortex shedding. The results of Young and Holl^[44] and Mussared^[49] (at subcritical Reynolds numbers) imply that this is so but in the experiments of Varga and Sebestyn^[45] and Syamala Rao et al.^[46] the presence of fully-wetted vortex shedding was not verified. Re-entrant jet flows appear to occur in flows which do not exhibit fully-wetted vortex shedding and may occur in cavitating vortex street flows below a critical cavitation number at which vortex shedding breaks down. In

this latter connection the results of both Holl and Young and Mussared indicate a transition from a cavitation modified vortex street flow to a fully developed cavity flow with re-entrant jet type closure. Since only two investigations of re-entrant jet flows have been reported, and with a similar paucity of data on cavitating vortex street flows it is not currently possible to predict with any certainty which form of entrainment will occur in a given cavitating flow, except in certain limiting cases such as axisymmetric bodies at low cavitation numbers where cavity closure occurs well downstream of the body.

I.6.3 ENTRAINMENT FROM VENTILATED CAVITIES

As stated in Section I.6.1, cavitation may be induced or its extent increased by the introduction of a foreign gas into the cavity or low pressure region downstream of the body. Three basic forms of entrainment have been reported for ventilated cavities, and these are outlined below.

Re-entrant jet entrainment occurs in ventilated as well as natural cavity flows. Reichardt^[37] demonstrated this similarity between natural and ventilated cavities by showing that the hydrodynamic forces and cavity geometry were the same for a given body at a given cavitation number (based on cavity pressure). With ventilation, longer cavities may be produced than could exist naturally (without ventilation) at a given flow velocity. In horizontal flow configurations, the force resulting from the density difference between the liquid flow and ventilation gas may cause the cavity to be inclined with respect to the onset flow. This asymmetry generates counter-rotating vortices

at the downstream end of the cavity, their axes being parallel to the onset flow. This type of entrainment mechanism has been observed on axisymmetric bodies^[52,53] and on three-dimensional hydrofoils^[54,55], and has been found to occur below a critical Froude number for the flow. With this form of entrainment, much larger ventilation rates are required to increase the cavity length by a given amount. A further gravitational effect was reported by Schiebe and Wetzel^[54] who studied cavities with re-entrant jet type closures in a horizontal towing tank and a vertical free jet water tunnel. They found that the vertically aligned cavities required twice the ventilation rate of similar cavities in the horizontal configuration.

If a cavity is ventilated such that the cavitation number is reduced to below about a fifth of its unventilated value a wave like oscillation of the cavity walls may occur resulting in the "pinching off" of the downstream portion of the cavity. This type of entrainment process has been investigated by Song^[56] for pulsating cavities behind cylinders and hydrofoils and considered theoretically by Woods^[57]. As with the spiral vortex type, this form of entrainment has not been observed in natural cavity flows.

In natural and ventilated cavities, gas contained in the free stream may diffuse into the cavity. With a natural cavity this mass transport increases the cavity pressure above the vapour pressure corresponding to the onset flow temperature, whereas for a ventilated cavity, the gas diffusion decreases the ventilation flow required to obtain a given cavitation number and hence cavity geometry. Such gas diffusion from the free stream into the cavity may thus be thought of as "natural" ventilation as opposed to the "forced" ventilation process by which gas is

supplied to the cavity from an external source. This gas diffusion arises if a gas content concentration gradient exists between the cavity and the free stream, the concentration gradient being proportional to the differences in the partial pressures of gas in the free stream and cavity. Brennen^[52] analysed the diffusion of air into developed cavities behind a sphere, assuming similarity of the turbulent boundary layer velocity and air concentration profiles and found good agreement with experimental data. Billet and Wier^[58,59] used Brennen's method to correct values of flow coefficients (non-dimensional volumetric ventilation flow rates) for the effects of gas diffusion in cavitating flows around ogival head forms. Their results indicated that if the cavity pressure is maintained at the same value as the partial pressure of the free stream gas, diffusion effects can be eliminated.

From the foregoing discussion it is possible to list some of the important similarity criteria for the valid simulation of entrainment from natural cavities using forced ventilation. For a given test body in pure drag cavity flows these are,

1. The cavitation numbers (based on cavity pressure) should be the same for both flows.

2. For natural cavities with re-entrant jet entrainment, the corresponding ventilated flow should also be of the re-entrant jet type. The Froude numbers of the two flows should be the same or at least, for the ventilated cavity, above the critical Froude number which delineates the change in entrainment mode from re-entrant jet to spiral vortex.

3. The orientation of the natural and ventilated cavities in relation to the gravitational force should be

the same^[54].

4. For re-entrant jet flows the main stream velocities of the natural and ventilated cavity flows should ideally be the same to give the same re-entrant jet momentum. At least the main stream velocities of both flows should be on the same side of the critical velocity at which transition may occur from a flow in which the re-entrant jet influences a large region of the cavity to the steady bubble type entrainment described by Furness and Hutton^[40] and Knapp et al.^[50].

5. The ventilated flow cavitation number should be greater than about one fifth of the unventilated value to preclude cavity pulsation effects^[56,57].

6. In respect of gas diffusion effects, the gas concentration gradients between the free stream and cavity should be the same for both flows (preferably zero) and, on the basis of Brennen's analysis^[52] the boundary layers on the cavity walls should be the same, implying similarity of the Reynolds number and free stream turbulence levels.

For two-dimensional bluff body flows there appear to be further, if vaguer, similarity requirements. Mussared's results^[49] indicate that the cavity shedding frequency and hence vapour entrainment rate depend on the corresponding fully-wetted flow regime, thus requiring the same Reynolds number for similarity of natural and ventilated cavitation on circular cylinders. The results of Young and Holl^[44], Mussared, Varga and Sebestyn^[45] and Ewers^[48] indicate a transition from a cavitating vortex type to a re-entrant jet type cavity closure below a certain cavitation number. This has yet to be verified. Ewers results also indicate that the ventilation flow itself might be expected to have

little influence on a cavitating vortex street flow.

With the present state of knowledge it is not possible to draw any further conclusions as to the validity of inferring vapour entrainment from natural cavities using forced ventilation and it is clear that many aspects of the similarity problems involved in the use of forced ventilation require further clarification.

I.6.4 METHODS OF CORRELATING THERMODYNAMIC EFFECTS IN CAVITATING FLOWS

On the assumption that the similarity requirements between natural and ventilated cavities, as outlined in the previous section, can be met, then entrainment data inferred from ventilation experiments may be used to investigate thermodynamic effects in cavitating flows. As noted in Section I.6.1, vapour is entrained from the closure region of natural cavities, resulting in a net flow of vapour out of the cavitating region. On a time-average basis, this entrainment flow must be balanced by a corresponding vapour flow from the free stream into the cavity. This vaporization of free stream liquid requires a heat flux from the free stream liquid into the cavity, which itself requires a temperature difference between the free stream and cavity. This temperature difference or "temperature depression" as it is sometimes called, constitutes the thermodynamic effect and is related to the corresponding "vapour pressure depression" through the slope of the vapour pressure versus temperature characteristic for a given fluid. Two methods have been previously used to analyse this thermodynamic effect, namely the β -factor theory and the Entrainment theory.

The former was first proposed by Stahl and Stepanoff [60]. It is based on a simple thermostatic model of an evaporative heat transfer process. Fig. I.44b shows a schematic of a developed cavity flow about an arbitrary body. The shaded area along the free stream lines bounding the cavity represents the volume of liquid V_L involved in the heat transfer process required to vaporize a volume V_v of vapour into the cavity. Using the notation of Fig. I.44b,

Heat lost by liquid of volume V_L = heat gained by vapour of volume V_v

$$\text{hence } \rho_f V_L C_{pf} \Delta T_c = \rho_g V_v h_{fg} = q \quad \text{I.49}$$

$$\text{from which, } \Delta T_c = \frac{\rho_g}{\rho_f} \frac{V_v}{V_L} \frac{h_{fg}}{C_{pf}} = \frac{\rho_g}{\rho_f} \cdot \frac{h_{fg}}{C_{pf}} \cdot \beta \quad \text{I.50}$$

The difficulty with using Equ. I.50 as a scaling relationship lies in the term β , the vapour to liquid volume ratio. Whereas all the other terms on the right hand side of Equ. I.50 are thermodynamic properties of the working fluid, it is not clear what physical factors influence β . To make Equ. I.50 useful as a scaling relationship, β must be given as a function of the appropriate variables governing the heat transfer process. Gelder et al.^[16] and Moore and Ruggeri^[65] used dimensional arguments to derive such a relationship in the form:

$$\beta = \beta_r \left\{ \frac{U_\infty}{u_{\infty r}} \right\}^x \left\{ \frac{D}{D_r} \right\}^y \left\{ \frac{L_c/D}{L_{c,r}/D_r} \right\}^z \left\{ \frac{k_f/\rho_f C_{pf}}{(k_f/\rho_f C_{pf})_r} \right\}^w \quad \text{I.51}$$

where the subscript r denotes a reference condition. The exponents of the dimensionless groups were determined by correlation of experimental data obtained from natural cavities in various fluids. Similar correlations, with

further dimensionless groups were proposed by Hord^[64] and applied to cryogenic fluids in a number of different flow devices.

A difficulty which invariably arises in the use of empirical correlations of the above form is that different values of the exponents are required for different flow geometries. Moreover, the factor β is a somewhat intangible quantity which gives little insight into the physical processes involved.

In the second method of analysing the thermodynamic effect the entrainment and vaporization process is treated on a steady flow basis as shown in Fig. I.44a. With this model, the entrainment flow $\dot{V}_e$ gives rise to a balancing evaporation flow $\dot{V}_w$ from the liquid at the cavity wall. For this situation a simple combined mass and energy equation may be written,

$$\dot{q} = \rho_g \dot{V}_e h_{fg} = \rho_g \dot{V}_w h_{fg} = h_c A_c \Delta T_c \quad \text{I.52}$$

where $\Delta T_c = (T_\infty - T_c)$

from which

$$\Delta T_c = \frac{\rho_g \dot{V}_e h_{fg}}{h_c A_c} \quad \text{I.53}$$

As shown by Holl et al.^[61] Equ. I.53 may be non-dimensionalized to give,

$$\Delta T_c = \frac{C_q}{C_A} \frac{P_e}{N_u} \cdot \frac{\rho_g}{\rho_f} \cdot \frac{h_{fg}}{C_{pf}} \quad \text{I.54}$$

$$\text{where } C_q = \frac{\dot{V}_e}{A_m} u_\infty \quad \text{I.55}$$

$$C_A = \frac{A_c}{A_m} \quad \text{I.56}$$

$$N_u = \frac{h_c D_m}{k_f} \quad I.57$$

$$P_e = \frac{u_\infty D_m \rho_f C_{pf}}{k_f} \quad I.58$$

Comparing Equ. I.53 with Equ. I.50, it may be seen that,

$$\beta = \frac{V_v}{V_L} = \frac{C_q}{C_A} \cdot \frac{P_e}{N_u} \quad I.59$$

which indicates that β depends on the cavity surface area, vapour entrainment rate and the characteristics of the vaporization heat transfer process. Of the four dimensionless groups given in Equ. I.55 to I.58, the Peclet number P_e is completely defined at a given onset flow velocity, test body size and onset flow temperature. On the basis that the cavity geometry is a unique function of cavitation number, the area coefficient C_A may be stated to be a function only of σ_c or cavity dimensionless length L_c/D_m . In Equ. I.54, this leaves the entrainment coefficient C_q and the Nusselt number N_u to be determined by the correlation of experimental data for a given flow geometry. Holl et al.^[61] assumed the entrainment coefficient could be represented in the form

$$C_q = C_2 R_e^b F_r^c W_e^d (L_c/D)^e \quad I.60$$

They used foreign gas ventilation, corrected for gas diffusion effects^[58,59] to determine the values of C_2 , b , c , d and e in Equ. I.60. Assuming that the Nusselt number could be expressed as,

$$N_u = C_3 R_e^f F_r^g W_e^h P_r^i (L_c/D)^j \quad I.61$$

they used measured values of temperature depression ΔT_c data from natural cavities in Equ. I.54 to determine the

constant and exponents of Equ. I.61. As might be expected, the values of both the constants and exponents in Eqs. I.60 and I.61 were found to be different for the two nose geometries of the ogives tested, but the signs of the exponents were the same for both geometries. Further correlations were presented by the same authors^[62] for developed cavitation in a venturi, and compared with the ogive data of Ref. 61. Two sets of correlating equations were used, one excluding Weber number ($d=0$, $h=0$ in Eqs. I.60 and I.61) and the other excluding Froude number ($c=0$, $g=0$ in Eqs. I.60 and I.61). The comparisons showed that the use of Froude number rather than Weber number had little effect on the C_q correlation (Equ. I.60) whereas it produced a considerable change in the values of the constant and exponents of the Nusselt number correlating equation (Equ. I.61).

It is apparent from the work reviewed above that foreign gas ventilation, when appropriately corrected for gas diffusion effects, is a very useful technique for inferring vapour entrainment in corresponding natural cavities, and represents a considerable improvement on the β -factor theory in that it gives a much better physical insight into the heat and mass transport processes involved. A shortcoming of the method is that it requires the correlation of data from natural and ventilated cavitation experiments in order to infer the characteristics of the heat transfer between free stream and cavity. This imposes considerable experimental difficulties in ensuring that the natural and ventilated cavities are completely similar while at the same time minimising gas diffusion effects. Clearly, if a ventilation technique could be designed which avoided the use of foreign gas, the problem

of gas diffusion effects could be greatly reduced, if not completely eliminated. Further, if the technique enabled the simultaneous investigation of natural and ventilated cavity flows the similarity problem would be greatly eased. One such technique is that of using vapour of the working fluid to ventilate natural cavities. An analysis of the additional information that a vapour ventilation system might be expected to give on the nature of the heat and mass transfer processes in developed cavity flows is given in the next section.

I.6.5 ANALYSIS OF VAPOUR VENTILATION OF DEVELOPED CAVITIES

Fig. I.45 shows a schematic of a vapour ventilated cavity, with the ventilation flow supplied from an external source. It is assumed that stable conditions prevail throughout the region of flow and that all the mass and energy fluxes are steady. For a given cavity (i.e. σ_c , T_∞ , u_∞ , P_s all constant) the mass continuity equation for the vapour phase is,

$$\dot{V}_v + \dot{V}_w = \dot{V}_e \quad \text{I.62}$$

This equation simply states that the entrainment from the cavity closure region is the sum of the externally-supplied ventilation flow and the vaporization rate of fluid from the cavity-liquid boundary. If $\dot{Q}_c$ is the heat flux required to vaporize a volumetric flow $\dot{V}_w$ from the cavity walls, then,

$$\dot{Q}_c = \rho_g h_{fg} \dot{V}_w \quad \text{I.63}$$

where ρ_g and h_{fg} are the saturated vapour density and latent heat of evaporation corresponding to the cavity pressure P_c .

From Eqs. I.62 and I.63,

$$\dot{Q}_c = \rho_g h_{fg} (\dot{V}_e - \dot{V}_v) \quad \text{I.64}$$

Following Holl et al.^[61], if the heat transfer process is assumed to be some form of forced convection, then Equ. I.64 may be expressed as,

$$\dot{Q}_c = h_c A_c (T_\infty - T_c) = \rho_g h_{fg} (\dot{V}_e - \dot{V}_v) \quad \text{I.65}$$

Considering Equ. I.62,

$$\dot{V}_v + \dot{V}_w = \dot{V}_e,$$

three states of ventilation may be identified:

(a) Cavity in thermal equilibrium with the ventilation flow.

This implies that $(T_\infty - T_c) = 0$

in which condition, from Equ. I.65

$$\dot{V}_e = \dot{V}_v = \dot{V}_* \quad \text{I.66}$$

In this state the ventilation system supplies all the vapour entrained from the cavity, the vaporization from the cavity wall thereby being suppressed for cavities having geometric and dynamic similarity, the equilibrium ventilation flow V_* represents the entrainment flow that would exist in the corresponding natural cavity.

(b) Partially-ventilated cavity.

If the equilibrium ventilation flow V_* is substituted into Equ. I.65

$$\dot{Q}_c = h_c A_c (T_\infty - T_c) = \rho_g h_{fg} (\dot{V}_* - \dot{V}_v) \quad \text{I.67}$$

In the above equation, if,

$$\dot{V}_v < \dot{V}_*$$

then $\dot{Q}_c$ and $(T_\infty - T_c)$ are positive and heat flows from the free stream to the cavity, with corresponding evaporation of fluid from the cavity walls.

(c) Over-ventilation.

In Equ. I.67 if $V_v > V_*$, then $\dot{Q}_c$ and $(T_\infty - T_c)$ are both negative which implies that heat is transferred from the cavity to the free stream with accompanying condensation of vapour on the cavity walls.

On the basis of the physical model postulated above it appears that the following effects may be investigated by ventilating cavities with vapour rather than a foreign gas.

(1) The thermal equilibrium ventilation flow may be determined from the condition of zero temperature difference between the free stream and cavity. As implied by Equ. I.67 an increase in the ventilation rate $\dot{V}_v$ beyond the equilibrium value, for a given state of cavitation, should cause vapour to condense onto rather than evaporate from the cavity walls. If the average cavity heat conductance is different for evaporation and condensation, over-ventilation would produce a discontinuity in the heat conductance versus temperature difference relationship. This discontinuity could be used as an alternative method for detecting the equilibrium condition.

(2) With foreign gas ventilation the cavity heat transfer coefficient h_c is estimated using temperature difference data obtained from natural cavities and entrainment rates inferred from ventilated cavities. The values of h_c thus estimated pertain to the maximum temperature difference between the cavity and free stream. With vapour ventilated cavities, from Equ. I.67, h_c is given by,

$$h_c = \frac{\rho_g h_{fg} (\dot{V}_* - \dot{V}_v)}{A_c (T_\infty - T_c)} \quad \text{I.68}$$

If $\dot{V}_*$ is established for a given cavity as described above then by recording the variation of $\dot{V}_v$ and $(T_\infty - T_c)$ as $\dot{V}_v$ is increased from 0 to $\dot{V}_*$, the behaviour of h_c as a function of $(T_\infty - T_c)$ may be determined for a given cavity.

To obtain the above information, the following measurements are required.

- (1) Onset flow temperature T_∞
- (2) Cavity to free stream differential temperature $(T_\infty - T_c)$
- (3) Heat supplied to the ventilation flow $\dot{Q}_v$
- (4) Volumetric ventilation flowrate $\dot{V}_v$
- (5) Static pressure of the onset flow P_s to determine the cavitation number based on vapour pressure σ_v
- (6) Differential static pressure between onset flow and cavity $(P_s - P_c)$ to determine the cavitation number based on cavity pressure σ_c
- (7) Onset flow dynamic pressure
- (8) Onset flow velocity U_∞

I.6.6 VENTILATION VAPOUR SUPPLY, FLOW DUCTING AND TEST BODY

The vapour ventilation system was designed and manufactured as the tunnel and ancilliary systems were being commissioned during 1975. In the course of the design of the ducting required to convey the ventilation flow from the vapour generator to a cavity, the form of test body had to be decided upon in order to match it to the supply duct. A two-dimensional circular cylinder was ultimately chosen, the intention being that it would be used as an initial test body to verify the operation of the system. A

cylindrical test body was chosen for the following reasons.

(1) A cylinder was very easy to make from standard steel tubes. In particular the tube bore provided the internal duct for the ventilation flow.

(2) A preliminary design study showed that a cylindrical body would be the simplest to connect with the ventilation supply duct from the evaporator, which had to pass through one of the test section walls.

(3) An investigation has just commenced in the Oxford University Cold Water Cavitation Tunnel to study fully wetted and natural and ventilated cavity flows around a two-dimensional circular cylinder (Ewers,^[48]). It was thought that if the initial test body used in the High Temperature Cavitation Tunnel were also a circular cylinder then some aspects of the two investigations might prove complementary. This was indeed the case as will be apparent in Part II.

(4) The behaviour of two-dimensional circular cylinders in single phase flow is well-documented. This was an important consideration, since the cylinder would be the first test body to be used in an otherwise unproven test facility. In this context it was of particular interest that the onset flow turbulence level may be inferred from cylinder drag or base pressure measurements.

In respect of (4) above it should be borne in mind that much of the cavitating wake results reviewed in Section I.6 had not come to light when the test body was chosen, indeed some of it had not been published at that date. In retrospect a more logical choice of test body would have been one for which the cavitation separation points are geometrically constrained rather than being influenced by viscous effects, for example, a two-dimensional wedge.

However, had a wedge been used, the interesting viscous effects on limited and developed cavities on a circular cylinder to be reported in Part II would not have been found.

The choice of cylinder diameter was a compromise between maintaining test body blockage at a reasonable level while allowing the straightforward manufacture of the test body and its associated vapour supply system. Again, with the objective of making the present investigation complementary with that of Ewers in the Cold Water Cavitation Tunnel^[48] the test body blockages were made approximately the same. Ewers used a 25.5 mm diameter cylinder with a test section depth of 229 mm, giving a blockage ratio of 11.1%. The present cylinder was made from the nearest standard tube size to correspond with this, which was 9.5 mm diameter giving a blockage ratio of 12.5% and an aspect ratio of 5.33.

The design parameters of the ventilation system were established from the relations,

$$C_q = \dot{V}_v / A_m U_\infty = \dot{Q}_v / (\rho_g h_{fg} A_m U_\infty) \quad 1.69$$

where C_q is the ventilation flow coefficient, or non-dimensional ventilation flow.

A_m is the model frontal area (diameter x span)

U_∞ is the onset flow temperature

ρ_g, h_{fg} the vapour density and latent heat of vaporization respectively.

A survey of foreign gas ventilation data given in Refs. 42, 52, 54, 55, 58, 59 and 62, indicated ventilation flow coefficients in the range $0.05 < C_q < 0.3$ for developed cavities. These limits were thus taken to design the evaporator and the ventilation flow ducting and flow

measurement system.

Fig. I.46 shows a schematic of the vapour ventilation system. The evaporator vessel is formed from a stainless steel pipe 2 m high and 125 mm in bore. A flanged connection at the lower end of the vessel houses the mounting flange of an array of six nickel alloy sheathed resistance heaters with a total maximum power rating of 30 KW. The maximum required evaporator power was determined from Equ. I.69 using a maximum value of $C_q = 0.3$ and with the relevant dry saturated water vapour properties evaluated at a temperature of 475° K, which corresponds to the maximum tunnel operating value; at evaporator temperatures lower than this, the power required to produce a C_q of 0.3 is less than 30 KW, since the thermodynamic property ($\rho_g h_{fg}$) decreases with decreasing temperature.

The heater elements are energised from a 40 KW direct current power supply which gives a continuously-variable output in the range 0 - 40 KW. The power output is controlled by a saturable reactor which is an integral part of the power supply. The control input is a direct current signal in the range 0 to 7.5 volts which produces an output of 0 to 40 KW. The maximum power output of the unit was limited to 30 KW by limiting the control signal. The evaporator operates in the pool boiling mode and vapour passes from the top of the vessel into a 25 mm bore transmission tube. The make-up flow required to maintain the evaporator liquid level is abstracted from the circulating pump discharge and ducted to an inlet port at the lower end of the evaporator vessel. The make-up flow is controlled by a solenoid valve activated by photo-electric cells on the evaporator level gauge. The solenoid operates in the

on-off mode and thus the evaporator level cycles between the limits of upper and lower photo-electric cells. The ventilation flow is measured by a turbine type flowmeter (Pottermeter) which is situated in the ventilation transmission tube between the evaporator and the test section. The flowmeter was sized to handle the ventilation flow required to produce a C_q of 0.3 at a tunnel onset flow velocity of 20 m/sec. Between the flowmeter and the test section is a manually operated globe valve which may be used either to throttle the ventilation flow or to isolate the evaporator completely from the tunnel. The ventilation flow transmission tube passes into the test section through a metal blank wall provided with suitable seals. After passing axially down the inside of the 9.5 mm diameter cylindrical test body, the ventilation vapour flows into the cavity through four slots machined into the downstream face of the cylinder. Fig. I.47 is a part sectional view of the test section in the plane of the cylindrical test body. To house the cylinder and ventilation supply tube the inner glass flow guide window and the two outer toughened glass windows were replaced by a pair of stainless steel blank windows. The inner one, adjacent to the flow surface acts as a flow guide and the 50 mm bore pressure tube is rebated and welded into its back face. This tube contains the pressurized working fluid which leaks around the unsealed flow guide windows. The cylindrical test body is welded into a shouldered bush with an 'o' ring seal on its outer diameter which mates and seals with a corresponding bore in another bush on the end of the ventilation tube. Fig. I.48 is a photograph of the cylindrical test body and its housing bush. The stepped flat on the housing bush face mates with a corresponding

step in the blank flow guide window to automatically ensure correct orientation of the ventilation slots during assembly. The cylinder passes through a mounting bush which can be changed to accommodate different types and sizes of test body, thereby minimising the number of components to be manufactured when the test body is changed. The assembly is designed so that the ventilation supply tube and test body can be removed or replaced without removing the inner blank window. This is done by unscrewing the locking tube (conventional right hand thread), and withdrawing the ventilation tube and then the test body. The assembly sequence is demonstrated in Fig. I.49 which shows an exploded view of (from right to left) the blank flow guide window and pressure tube, the ventilation tube with its associated seal housing spacer, the locking tube and a 90 degree bend tube-fitting for connecting the ventilation tube to the pipeline to the evaporator. In the foreground is the cylinder and its housing bush. The 3 mm diameter tube which can be seen emerging from the test body housing bush in Fig. I.49 is for communicating the cylinder cavity or base pressure to a transducer. This tube passes concentrically up the model tube to the mid span location where it turns through 90 degrees and penetrates through the tube wall. Upon assembly, the tube passes through a gland with an 'o' ring seal which is located on the 90 degree bend on the extreme left of Fig. I.49. Fig. I.50 shows an assembled view of the aforementioned components. It should be emphasized that for actual assembly on the tunnel, the metal flow guide window must be firmly in position and clamped up before the test body is inserted through it. This is important, since the glass window opposite the blank flow guide window is countersunk, the

end of the cylinder fitting into this hole to circumvent the problem of tip-clearance flows. Since this inner flow guide glass window is untoughened, it is quite fragile and any cylinder misalignment on assembly could easily damage or break it. Fig. I.51 is a photograph of the vapour ventilation supply system showing the evaporator, liquid level gauge, turbine flowmeter and the vapour supply pipework. The photograph also shows pipework added by Mussared^[49] at a later date for investigating the effects of vapour extraction from natural cavities. Full details of the modified pipework may be found in Ref. 49.

CHAPTER I.7

TUNNEL AND VAPOUR VENTILATION SYSTEM PROVING TRIALS

When the assembly of the tunnel and its ancilliary systems had been completed (late 1975) and the cylindrical test body installed, proving trials were performed to see if the tunnel behaved in accordance with its design specification. These trials were performed before the installation of the synchronised data acquisition system described in Section I.8, since it was thought best to ascertain the actual tunnel operating characteristics before finalizing the design of the data handling arrangements. The outputs of the requisite tunnel transducers were thus initially registered on analogue displays, no provision being made for data recording.

I.7.1 LOCATION OF TUNNEL TRANSDUCERS AND PROVING TRIAL INSTRUMENTATION

Fig. I.52 is a schematic of the measured and control variables of the instrumentation used in the initial proving trials of the tunnel. Only a brief description of the transducers themselves will be given, since they were subsequently incorporated into the synchronised data acquisition system and will be fully detailed in Section I.8. The physical locations of the transducers on the tunnel was the same for both the initial and final instrumentation systems, although in the latter case more transducers were added to the tunnel. The required 5 volt excitation for the two Statham differential pressure transducers was provided using a standard laboratory power supply, the corresponding

output being measured on two Phillips type 6020 voltmeters. The fluid temperature was measured by a nickel chrome/nickel aluminium thermocouple at the contraction inlet location which was differentially connected to a similar ice point reference thermocouple. The e.m.f. from the thermocouples was measured by another Phillips 6020 voltmeter. The ventilation and tunnel volumetric flows were registered on the analogue meters of two frequency to voltage converters. The evaporator input power was determined from current and voltage measurements. Fig. I.53 is a part sectional plan view of the test section and vapour ventilation supply showing the manner in which the cavity temperature and pressure were measured. The 2.5 mm bore cavity or base pressure sampling tube was used as a housing for a nickel chrome/nickel aluminium thermocouple, 0.5 mm in diameter. Since the thermocouple was so much smaller than the pressure sampling tube bore, the latter was not unduly blocked by the former. The thermocouple entered the tube through a P.T.F.E. packed screwed gland located on a elbow of the ventilation supply tube, and emerged to protrude from the tube end by 5 mm into the cavity or wake region downstream of the cylindrical test body. The thermocouple was of the sheathed insulated junction type, and was differentially connected to a second identical thermocouple at the contraction inlet. The e.m.f. generated in this circuit was thus proportional to the free stream to cavity differential temperature. Since it was expected that experimental values of this temperature difference would be small, some care was taken both with calibrating the thermocouples and with providing an accurate instrument to measure the e.m.f. in the thermocouple circuit. All the thermocouples were calibrated in a silicon oil bath. The

$(T_{\infty} - T_c)$ thermocouple circuit was first tested by placing the hot junctions in contact and binding them together with thin pure copper wire to ensure good thermal contact. They were then placed in the oil bath and their output measured on a digital voltmeter. As might be expected, the thermocouple outputs were not the same and zero errors up to 60 microvolts (1.5° centigrade) were recorded. Various thermocouples were tested in this manner to no avail, all of the circuits tested showing appreciable outputs with the hot junctions at the same temperature. This difficulty was circumvented by using a Keithley Type 149 high input impedance analogue micro voltmeter incorporating an adjustable zero offset bias facility. Using this facility the output from the thermocouple circuit was set to zero at the fully wetted flow condition immediately prior to establishing a ventilated cavity.

The control signal required to vary the output of the evaporator's power supply was derived from a standard laboratory variable voltage source. It should be emphasized that all the aforementioned instruments used to measure the outputs of the tunnel transducers were standard laboratory equipment and as such not ideally suited to the task. However, since the purpose was to verify the performance of the tunnel and its ancilliary systems rather than obtain scientific data, it was felt that lack of accuracy could be tolerated. Tunnel static pressure was adjusted using the pressure regulator on the nitrogen pressurization cylinder. Since the tunnel instrumentation console had not been built at this stage, the velocity setpoint potentiometer of the hydrostatic drive was located on the control console of the Oxford University Cold Water Cavitation Tunnel, which will be discussed in Chapter I.8.

Fig. I.54 is a general view of the tunnel working area showing the instrumentation used in the proving trials.

I.7.2 COMMENTS ON TUNNEL PERFORMANCE DURING PROVING TRIALS

The initial tunnel runs went reasonably smoothly, without any major problems occurring. It was discovered almost at once that the tunnel gradually filled with air when running under vacuum conditions. This leak was traced to the static fluid seal between the rotating portion of the circulating pump mechanical seal and the pump shaft. For some reason this seal was a C-section type, tensioned internally with a garter spring, the mouth of the C-section being on the tunnel side of the seal. This arrangement would clearly work well under pressurized tunnel conditions, since the fluid pressure would tend to open the C-section and press it against its mating joint faces. It was equally clear that under vacuum conditions the C-section was collapsing under atmospheric pressure and allowing the ingress of air into the tunnel. The seal housing was machined to accept a standard size Viton 'o' ring and no leakage problems were subsequently encountered.

The maximum attainable test section velocity corresponded closely to the design value of 30 m/sec. This is not surprising, since the estimated tunnel power absorption with the low speed test section was much less than that which the hydrostatic drive is capable of providing. The actual tunnel power absorption could have been estimated by measuring the pressure of the power flow oil at the maximum tunnel speed, and using the manufacturer's operating characteristics of the hydraulic pump and motor and the circulating pump. This was not done, since as the tunnel

performed to its design specification, it would have served no purpose. It would, however, be an essential exercise if a high speed test section were to be installed in the tunnel, since the tunnel pressure loss would be required to specify a higher duty impeller for the circulating pump, as mentioned earlier in Section I.4.9.

With a high working fluid free gas content it was found that the whole tunnel structure suffered severe low frequency vibrations at speeds above about 25 m/sec. These vibrations only occurred when the test body was cavitating, and could be arrested immediately by increasing tunnel pressure and suppressing cavitation in the test section. By contrast, with low working fluid gas contents, no change in vibration level could be detected in passing from the fully wetted to the cavitating flow regimes. In this situation the vibration was very low, indeed it was possible to stand a pencil on end on the top of the test section for considerable periods. When these aforementioned vibrations were first encountered they caused considerable alarm since the amplitudes appeared to be of the order of 10 mm, at frequencies of several Hertz. The origins of these vibrations were not investigated, rather it was resolved to avoid the operating conditions under which they occurred, on grounds of operator safety. Since they could be suppressed by increasing tunnel pressure, it is reasonable to assume that they were associated with cavitation, or pseudo-cavitation caused by the high free gas content, in some part of the tunnel. It is possible that they were related to the cavity compliance phenomena considered by Brennan and Acosta^[63].

The maximum tunnel temperature attained during the proving trials was 430° K. The tunnel took a considerable

time to reach this temperature and it became obvious that either more heater tapes must be provided to the tunnel pipework, or the tunnel structure must be more effectively thermally insulated. Neither of these measures were implemented, partly because of the time it would have involved and also because the experimental program reported in Part II only required tunnel temperatures in the range of ambient to 375° K. A further tunnel shortcoming which revealed itself at high temperature was the overheating of the fluid in the sump tank and its consequent loss to atmosphere by evaporation. This was caused by the thermal expansion of the working fluid and the consequent venting of fluid from the tunnel to the sump tank required to maintain a constant liquid level in the pressurizer plenum. At operating temperatures above the atmospheric boiling point, the fluid vented from the tunnel enters the sump tank as superheated vapour since the sump tank is always vented to the atmosphere during tunnel heat-up, as outlined in Section I.5.3. Since the vent tube discharge was drowned below the free surface of the sump tank liquid, this heated up to boiling point very quickly, resulting in loss of vapour through the atmospheric vent line. Also, as the sump tank fluid approaches the boiling point the Madan make-up pump ceases to function as it starts to cavitate on its suction stroke. This is not normally a problem during increasing tunnel temperature conditions, since it is never required to supply make-up flow while the working fluid is expanding. Because of its large thermal capacity, the sump tank water, once heated up, maintains its temperature for a considerable period. Thus if the tunnel is to be cooled after operating at a high temperature, the sump tank will remain at an elevated temperature and make-up flow cannot

be supplied since the Madan pump would cavitate. The situation outlined above only became a problem at tunnel temperatures greater than about 400° K, since below this temperature the amount of tunnel fluid required to be vented is insufficient to raise the sump tank temperature unduly. A further important point in connection with the sump tank is that since it is situated adjacent to the tunnel operating area, high temperatures thereof make the working environment uncomfortable. In the light of the foregoing it is clear that the sump tank requires a cooling coil to maintain it at a reasonable temperature. This topic is developed further in Part III. A cooler was not installed because of the time involved and because it was subsequently found that values of the experimental parameters required could be attained at temperatures below that at which sump tank overheating became a problem.

I.7.3 PERFORMANCE OF THE VAPOUR VENTILATION SYSTEM

The initial trials with the vapour ventilation system showed that there were strong interactions between the tunnel and the evaporator, which rendered the latter somewhat more difficult to operate than had been envisaged. In particular, once the evaporator was connected to the tunnel by opening the throttle valve in Fig. I.52, caution had to be exercised when reducing tunnel static pressure, since this was immediately transmitted to the evaporator. Under these conditions the evaporator liquid became superheated and some of it flashed to vapour causing a large increase in the ventilation flow and accompanying oscillations in the evaporator liquid level. This rapid increase in the ventilation flow inevitably caused over-

speeding of the turbine flowmeter used to measure it. To circumvent this problem, the following operating sequence was used.

The tunnel was heated up in the same manner as that used for de-gassing the working fluid, with the tunnel vented to the sump tank. At the same time the evaporator was heated up, with the ventilation flow valve wide open. When boiling of the tunnel and evaporator fluid commenced the hydrostatic drive was started and the tunnel velocity set to 5 m/sec. The static pressure was then set at the appropriate value to give the required cavitation number at the desired experimental test section velocity. Heat was then supplied to the evaporator and tunnel until the temperature reached the required value for a given experiment. The evaporator power was then switched off and the ventilation flow throttle valve closed. A developed natural cavity was then produced behind the cylindrical test body and the ventilation flow throttle valve opened to allow the cavity and evaporator pressures to equalize. The evaporator power was then switched on again and the resulting ventilation vapour flow decreased the cavitation number. This was adjusted back to the required constant experimental value by increasing the tunnel static pressure and observing the output of the differential pressure transducer measuring $(P_s - P_c)$, this being proportional to cavitation number at a given onset flow velocity and temperature. When reasonably steady conditions had been established following the static pressure adjustment, the values of $(P_s - P_c)$, $(T_\infty - T_c)$, $\dot{V}_v$, T_∞ and Q_v displayed on the analogue meters was noted. The ventilation flow was increased and the procedure repeated. During these experiments considerable difficulties were encountered in

trying to hold the experiment at a given operating point. Both T_{∞} and $\dot{V}_v$ were both subject to drift which rendered the achievement of a steady state virtually impossible while trying to record the required data and adjust the appropriate controls. The data taken during these proving trials showed that the vapour ventilation technique worked broadly in accordance with the analysis outlined in Section I.6.5, but indicated that further development effort is required to produce a system from which data of scientific value can be obtained. Fig. I.55 shows the cavity to free stream temperature differences induced by variation of the ventilation flow about its equilibrium value at a given velocity, temperature and cavitation number. The paucity of data points is a consequence of the operational difficulties outlined above. The data indicate the predicted reversal of the cavity to free stream temperature gradient at ventilation rates greater than the equilibrium value $\dot{V}_*$. A better demonstration of the existence of the three states of cavitation outlined in Section I.6.5 (i.e. thermal equilibrium, partially ventilated and over-ventilated cavities) was obtained under transient ventilation conditions. This was done by over-ventilating a developed cavity and then switching off the evaporator power completely. The decay time of the ventilation flow was found to be about 15 to 20 seconds, and during this period the $(T_{\infty} - T_c)$ and $\dot{V}_v$ displays were observed to move in unison, with the temperature difference starting positive and decreasing through zero to assume a negative value as the ventilation flow was reduced to zero. This transient mode of operation also had the advantage that during the ventilation flow decay time, negligible tunnel temperature drift occurred. Towards the end of

these ventilation system proving trials, the end of the thermocouple measuring the cavity temperature broke off. This appeared to be caused by vibration of the thermocouple in the fluctuating flow of the vortex shedding region immediately downstream of the cylinder. Support will have to be provided for a thermocouple at this location if further vapour ventilation experiments are to be performed.

From the operating experience gained during the proving trials, it became apparent that an automatic, centrally actuated data handling system would be the best way of handling the transducer measurements, thereby giving the tunnel operator maximum freedom to observe and adjust the experimental conditions. The tunnel data acquisition system is described in the next chapter.

CHAPTER I.8

INSTRUMENTATION AND DATA ACQUISITION SYSTEM

Although the ventilation experiments were not pursued beyond the stage of verifying the efficacy of the technique, the data acquisition system was designed to incorporate the basic measurements required to perform ventilation experiments, since the vapour ventilator was classed as being a tunnel ancilliary system.

I.8.1 TRANSDUCERS AND DATA LOGGER

The instrumentation system was based on a Hewlett Packard 2010 D data logger, which sequentially scans up to 600 input channels and measures them on a Hewlett Packard 2401 C Integrating Digital Volt Meter. Table I.5 shows the transducers used to measure the required tunnel experimental variables, and Fig. I.56 shows their physical locations on the tunnel. The locations of the transducers used in the proving trials were not changed and provision was made for the measurement of test section wall pressure distributions, since it was thought that this would be necessary to determine the choked cavity flow cavitation number of the cylindrical test body. Since the outputs of the transducers shown in Table I.5 were of different forms and ranges, an interface had to be provided to make the transducer outputs compatible with the 2401 C Integrating Digital Voltmeter. This interface was designed and built by the O.U.E.L. Electronic Workshop to specifications provided by the author. Since, in the chosen mode of operation, the full scale deflection of the digital

Table I.5 Tunnel Transducer Table

Transducer/Instrument	Manufacturer	Type	Accuracy	Sensitivity/Output Type
Tunnel flow turbine flowmeter	George Kent	5"-254-6-15-D/C	0.2%	frequency output
Ventilation flow turbine flowmeter	Potter Aero. Corp.	3/4 - 50345	0.2%	frequency output
Absolute pressure transducers	Bell and Howell	4-326-L514 4-326-L515	0.25% (non-linearity and hysteresis). Temperature sensitivity 0.018% /° C	40 mV F.S.D.
Differential pressure transducers	Statham	P.M. 280 TC	0.2% (non-linearity and hysteresis). Temperature sensitivity 0.018% /° C	20 mV F.S.D.
Thermocouples	Thermocouple Instruments	N _i C/N _i Al	0.2%	40 μ V/° C
Integrating Digital voltmeter	Hewlett Packard (part of 2010 D data logger)	2401 C	$\pm$ 0.5 μ V (5 digit display)	200 mV
Silicon oil bath (thermocouple calibrations)	P. M. Tamson N.V.	T. 45	0.02° C	
Hydraulic dead weight tester (pressure transducer calibrations)	Budenburg	240 D	$\pm$ 0.03%	

voltmeter was 200 m.v., the interface contains amplifiers to scale up the thermocouples to the required range. The pressure transducer outputs are large enough to be read directly by the digital voltmeter and thus these signals are not amplified. To assist in obtaining accurate pressure measurements, the excitation voltage as well as the output voltage of the pressure transducers is measured. This interface also contains frequency to voltage converters to process the frequency output signals from the tunnel and ventilation flow turbine flowmeters. A second interface provides a synchronous drive to the Type 48J7 Scanivalve, the first, or home channel of which is used to measure $(P_s - P_c)$ and the following 18 channels the 18 pressure tappings used to measure the test section wall pressure distribution. The outlet port of the Scanivalve connects the requisite input channel with the low pressure side of Statham type P.M. 280 TC differential pressure transducers of various ranges. This latter interface also incorporates a remote control facility for the data logger so that it can be actuated from the instrumentation console adjacent to the tunnel working area. Data output from the data logger is on punched paper tape, which is then digitally processed to obtain correctly scaled values of the tunnel state variables.

In order to provide information for experimental control purposes the values of the transducer measurements are registered on analogue and digital displays on the tunnel instrumentation console. This console is adjacent to the tunnel test section viewing windows, and is shown in Fig. I.57. It houses all the tunnel performance monitoring displays and control inputs required to run the tunnel, with the exception of the pressure regulator, which

is located on top of the nitrogen pressurizing gas cylinder on the opposite side of the tunnel working area. The set-point potentiometers of the tunnel velocity and evaporator power controllers are located on the top rack of the console, their functions being clearly labelled, as are those of all the other displays. The pressure transducers often have to be changed between experiments in order to obtain reasonable outputs for the wide ranges of velocity and static pressure attainable in the test section. For this reason, their monitoring displays are scaled as percentages of full scale output which is approximately the same for a given type of transducer. The main function of these pressure displays is to prevent overloading of the transducers, which are expensive items, particularly the differential ones. They are not normally used for monitoring the tunnel pressure since this is more conveniently done using the pressure gauge mounted on top of the pressurizer plenum, as shown in Fig. I.58, which is a general view of the tunnel working area. The hydrostatic drive starting controls are located on the instrumentation console of the Cold Water Cavitation Tunnel, which is adjacent to its test section on the laboratory east wall. The changeover switch which connects the appropriate hydraulic motor tach-generator and velocity setpoint potentiometer into the hydrostatic drive electronic control unit is also on this console, as noted in Section I.5.1. The electronic control unit is also located in the console, but access is only required to it for maintenance purposes. All the controls on the Cold Water Cavitation Tunnel instrumentation console are clearly labelled, and a general view of it is shown in Fig. I.59. Once the hydrostatic

drive is running, the High Temperature Cavitation Tunnel can be completely controlled from its instrumentation console.

I.8.2 TRANSDUCER CALIBRATION AND ACCURACY OF MEASURED DATA

All the tunnel transducers were calibrated prior to experiments in which they were used, where this was feasible. In the case of the turbine flowmeters, this was not possible, particularly with the 125 mm bore unit which measures the main tunnel flow. A certificated calibration of this flowmeter was performed by the C.E.G.B. Hydraulic Calibration Centre (Hams Hall, Warwickshire). The ventilation flowmeter was calibrated by the manufacturers (Potter Aeronautical Corporation, New Jersey) in a variable pressure air calibration facility. All pressure transducers were calibrated in the laboratory using a Budenburg Type 240 D dead-weight tester with a certificated accuracy of $\pm 0.03\%$ of the pressure being tested. These calibrations were performed by removing a given transducer from the tunnel and placing it on the dead-weight tester, the output at a given loading being read from the 2401 C Integrating Digital Voltmeter. Thus the complete system was calibrated, from transducer diaphragm through to the D.V.M. The thermocouples were calibrated in a Tamson thermostatically controlled silicon oil bath which, according to the manufacturer's data, holds a given temperature to within 0.02°C . The calibration was based on the oil temperature, as measured on an N.P.L. certificated resistance thermometer, with the thermocouple output read from the Integrating Digital Voltmeter. From the transducer and digital voltmeter accuracies quoted in Table I.5, it may be seen that a

high overall accuracy should have been achieved with the measurement system used. Unfortunately, after the data acquisition system had been built and the experimental program started, it was found that the data logger was subject to thermal drift. Subsequent observations of the magnitude of this drift indicated that it served to limit the achievable accuracy of a given channel to about 1% of its full scale value, which, in view of the level of hydrodynamic noise usually present in cavitating flows, was deemed satisfactory. Also because it could be actuated by a single switch, the data acquisition system allowed the tunnel operator to spend most of the time observing the flow through the test section viewing windows and ensuring that the required experimental conditions were obtained. It was felt that this advantage outweighed the limitation on overall accuracy imposed by the thermal drift of the data logger.

I.8.3 DATA ACQUISITION AND PROCESSING

The 45 data channels used by the data logger are defined in Table I.6. The paper tape output from the data logger is in an eleven character integer format, a typical example of which is shown in Table I.7, together with the data sequence of a typical experimental tape. Fig. I.60 shows a simplified block diagram of the Fortran program "Cavitace" used to process the output tape through a digital computer. The program initially reads a starting tape containing the experimental run number and an integer number which designates the range of the P_s pressure transducer used in a given experiment. This is required for the program to select the correct scale factor for the

Table I.6 Definition of Data Channels

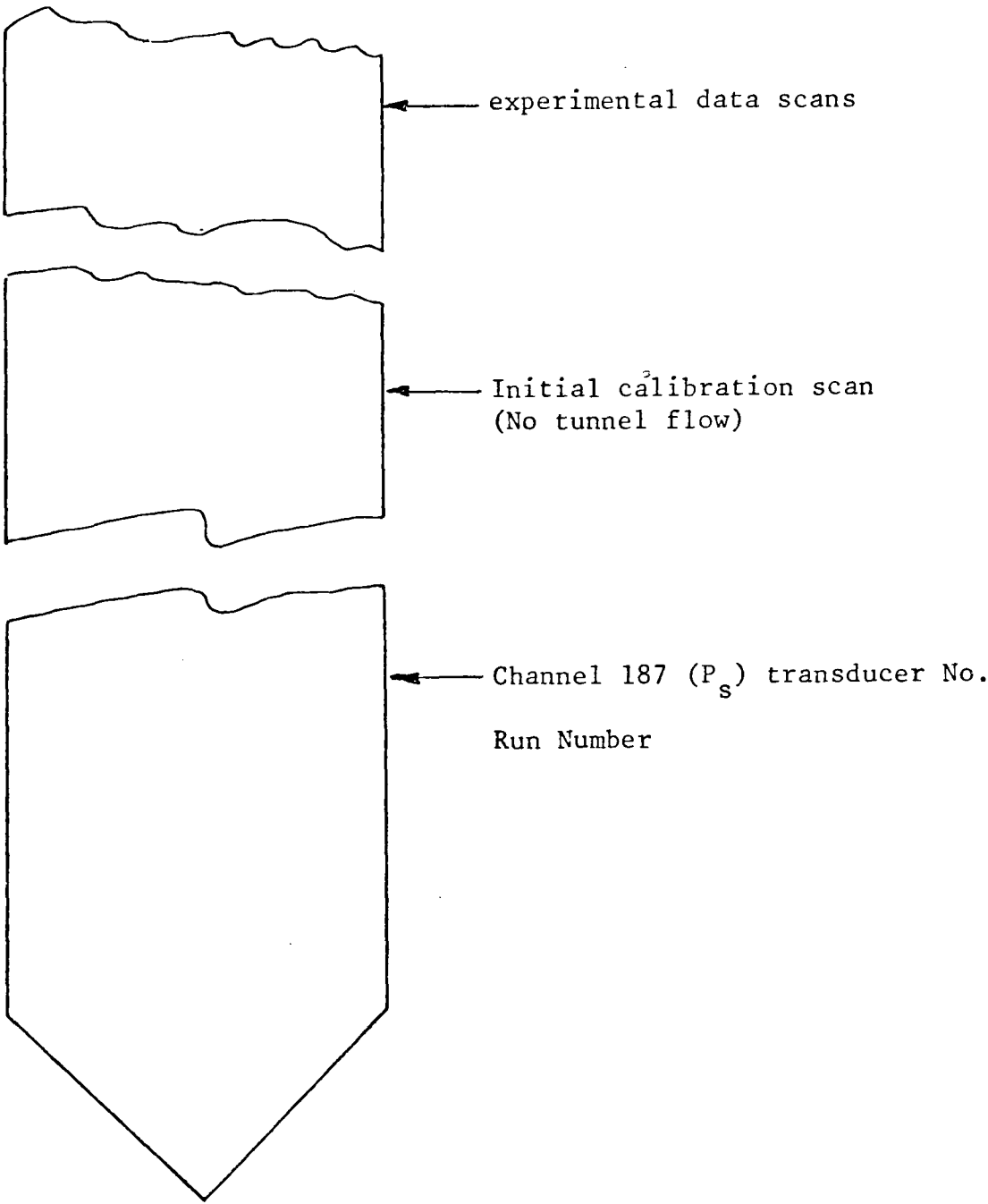
Channel number	Variable	
175	$T_{\infty} - T_c$	
176	Differential temperature	*
177	T_{∞}	
178	T_v	
179	Absolute temperature	*
180	Absolute temperature	*
181	Absolute temperature	*
182	Absolute temperature	*
183	Excitation voltage of $(P_s - P_c)$ and $(P_u - P_s)$ transducers	
184	$P_s - P_c$	
185	$P_u - P_c$	
186	Excitation voltage of P_s transducer	
187	P_s	
188	Ventilation flow rate range	
189	Ventilation flow rate output	
190	Circulation flow rate range	
191	Circulation flow rate output	
192	Evaporator input volts	
193	Evaporator input current	
194		*
195		*
196	currently unused	*
197		*
198		*
199		*
200	Excitation voltage of $P_w(x)$ transducer	
201		
202		
203		
204	$P_w(x)$	
205	(test section wall pressure distribution)	
206		
207		
208		
209		
210		
211		
212		
213		
214		
215		
216		
217		
218		

NOTE: Channels marked * are not currently used.

Table I.7 Typical paper tape output record

±	184	249718	6
Sign of channel output	Channel No.	channel output (microvolts) F.S.D. = 250000 uV	exponent of channel output scale factor (varies with D.V.M. range setting)

In the above example, the output of channel 184 is $\pm 249718 \times 10^{-6}$ VOLTS



transducer used. The program then reads the first scan of the main data tape, which contains initial values of the outputs of all the transducers at zero tunnel circulation flow. These initial values are required for some of the pressure transducers since, being of the strain gauge type, they have residual bridge unbalance. Further, as noted in the previous section, the Integrating Digital Voltmeter suffers from thermal drift. This problem is at least partially circumvented by reading the output of the D.V.M. for each channel, under no load conditions, and subtracting these values from those obtained in subsequent scans. In this way the measurements become differential to the initial values. Having read the initial values tape, the program decomposes the eleven integer character into the sign, microvolt value and exponent, as indicated in Table I.7, and computes the output value of each channel in the appropriate units. Further ancilliary data which depend on the tunnel state variables are computed for each scan. These include working fluid density, viscosity and vapour pressure, which are obtained from polynomial functions of the onset flow temperature, and the Reynolds number and cavitation number (or cylinder base pressure coefficient) based on cavity pressure and vapour pressure. Program output may be printed or graphical. The above computer programs were written by M. Mussared, and detailed program descriptions and running instructions may be found in Ref. 49. Fig. I.61 shows the Hewlett Packard 2010 D data logger and paper tape output.

CHAPTER I.9

TUNNEL AND DATA ACQUISITION SYSTEM OPERATING PROCEDURES

The operation of many of the tunnel systems has already been covered in their general descriptions in Chapters I.5 and I.6. Rather than describe the operating procedure of the various tunnel systems and the data acquisition system separately, they are described together in Appendices A.I.1 and A.I.2.

CHAPTER I.10

CLOSURE TO PART I

In Part I of this thesis the tunnel design has only been briefly outlined, since it involved the author in the production of about 170 engineering drawings and their concomitant design calculations. These drawings, together with manufacturers' data relating to tunnel components supplied by outside organisations, form an essential part of the tunnel documentation system. The engineering drawings are located in the office of the O.U.E.L. Departmental Design Engineer. Outside suppliers' data such as transducer specifications, operating characteristics of pumps and motors, and a wide variety of maintenance documentation are contained in the filing systems of the Cavitation Laboratory. Detailed drawings, specifications and maintenance data relating to the interface and data logging portions of the data acquisition system are held by the O.U.E.L. Electronics Workshop. In connection with the engineering drawings of the tunnel pressure structure, it is very important that these be referred to in the event of any modifications being made to it. Although the tunnel containment structure (with the exception of the circulating pump casing) was designed to operate at a pressure of 69 bar with a fluid temperature of 475° K, certain modifications might have a disproportionate effect on the local strength of the circuit, for example making a relatively large opening through a tunnel wall without providing reinforcement, or introducing a stress concentration factor which would produce a local stress

exceeding the strength of the material. One part of the tunnel where modifications must be effected with caution is the test section, the components of which are carefully toleranced to preclude lock-up and consequent crushing of the pressure windows under the action of differential thermal expansion. These tolerances are all clearly marked on the relevant engineering drawings.

P A R T I I

VISCOUS EFFECTS ON THE INCEPTION AND DEVELOPMENT
OF CAVITATION ON A TWO-DIMENSIONAL CIRCULAR CYLINDER

CHAPTER II.1

INTRODUCTION

In the course of establishing natural cavities behind the cylindrical test body during the tunnel proving trials, it was necessary to pass through the incipient cavitation state. It was observed that both the physical appearance and the location of these incipient states varied with the test section onset flow velocity and temperature. Some further specific cavitation inception experiments revealed three basic forms of inception with secondary types associated with the transition between these basic forms. Of these three basic types of inception, two were attached, or very close to the cylinder surface and one completely detached in nature, occurring well downstream in the wake region of the cylinder. Of the former two types, one was markedly two-dimensional in character, occurring as a thin band of cavitation across the back face of the cylinder at about 110 degrees from the frontal stagnation point. The other was highly three-dimensional, appearing as wedges of cavitation at discrete locations across the cylinder span at approximately 90 degrees from the frontal stagnation point. Further, the initial phase of the development of cavitation from these three basic incipient states was quite different, the cavities only appearing to be geometrically similar as the choked test section condition was approached, and even then differences in the angular location of cavitation separation from the cylinder were observed. These initial experiments also indicated that the transitions between the observed incipient cavitation

flow regimes could be correlated with the Reynolds number based on cylinder diameter and onset flow velocity (the kinematic viscosity ν being evaluated at the appropriate free stream temperature).

In the light of the current world wide interest in the problem of cavitation inception prediction and scaling, it was resolved to investigate these different forms of limited and developed natural cavitation, rather than pursue the ventilation experiments, particularly as this latter course would have involved considerably more engineering development work to overcome the operational problems encountered with the vapour ventilation system during its proving trials.

Little attention seems to have been given in the past to the cavitation inception performance of two-dimensional circular cylinders. Previous research has chiefly been concerned with the developed cavitation flow regime, in particular with drag coefficient measurements^[66,67], cavitating wake dynamics^[45,46], cavitation damage^[68], viscous effects on cavitation separation^[69] and the influence of Polymer additives^[70]. The only recent reported investigation of inception on circular cylinders that has come to the author's attention is that of Palanichamy et al.^[71], who observed inception on circular cylinders of different diameters and surface roughnesses in a venturi-type test section. These authors reported that at high tunnel velocities, inception occurred on the cylinder surface, while at lower velocities, it occurred downstream in the wake. No attempt was apparently made to determine the critical Reynolds number R_{CR} for the cylinders, the data obtained being presented as plots of

inception and choked flow cavitation numbers versus test section velocity. Some further interpretation will be made of their observations, in the light of the present investigation, in Section II.5.2. Since one of the aims of the investigation reported herein is to relate the viscous behaviour of the fully wetted flow to the modes and locations of the observed incipient cavitation states, it is deemed worthy to review the salient features of the single phase flow around a circular cylinder in the region of the critical Reynolds number. This is given in the following chapter.

CHAPTER II.2

SOME RELEVANT VISCOUS PHENOMENA IN SINGLE PHASE FLOWS

II.2.1 SINGLE PHASE FLOW ROUND A CIRCULAR CYLINDER FOR REYNOLDS NUMBERS GREATER THAN 10^5

The following outline pertains to a cylinder of small surface roughness in an incompressible plane stream of moderate onset flow turbulence level. Fig. II.1 shows a typical relationship between drag coefficient C_D , or its approximate equivalent, the absolute value of cylinder base pressure coefficient, and Reynolds number R_d for $10^5 < R_d < 10^6$. In the subcritical regime, for R_d greater than about 10^5 , the laminar boundary layer separates from the cylinder at an angle in the range 70 to 80 degrees from the frontal stagnation point^[72], with turbulent transition occurring in the free shear layers downstream of the body^[73]. This early laminar separation produces a broad wake with consequent low base pressure and high drag. In this regime, narrow band vortex shedding occurs at a Strouhal number of about 0.2^[74]. The critical regime may be said to commence with the fall in drag coefficient below the subcritical value (see Fig. II.1). The laminar boundary layer separation point moves downstream with increasing Reynolds number until it attains a value somewhat greater than 90 degrees. At the critical Reynolds number R_{CR} the proximity of the previously detached turbulent transition to the cylinder surface enables the separated turbulent free shear layers to reattach to the cylinder surface to form laminar separation bubbles. The turbulent boundary layers finally separate at about 140

degrees from the frontal stagnation point^[72,75]. The formation of laminar separation bubbles may be taken to mark the end of the critical regime. Increasing onset flow turbulence level^[76] and surface roughness^[77] both reduce the Reynolds number range within which laminar separation bubbles occur. In the supercritical regime the formation of separation bubbles and late turbulent boundary layer separation produce a narrow wake and consequent high C_{pb} and low C_D (both of absolute value 0.2 to 0.3 referred to infinite stream conditions). In low turbulence streams ($T_u < 0.002$), narrow band vortex shedding has been reported at a Strouhal number of 0.45^[76]. The supercritical regime may be said to terminate when the location of boundary layer turbulent transition moves far enough upstream to disrupt and finally eliminate laminar separation^[78,79]. At higher Reynolds numbers, in the transcritical regime, the turbulent boundary layer separation point moves in the upstream direction, producing a decrease in C_{pb} and a corresponding increase in C_D . At very high Reynolds numbers, C_D appears to reach a limiting value of about 0.7^[79].

II.2.2 FLOW IN THE REGION OF LAMINAR SEPARATION BUBBLES

A comprehensive wind tunnel investigation of attached and separated boundary layer turbulent transition on axisymmetric bodies was reported by Huang and Hannan^[80]. They investigated transition on two large scale axisymmetric headforms in a wind tunnel, one of which exhibited attached transition and the other a laminar separation bubble detached transition. In the latter case they found very large pressure fluctuations where the separated tur-

bulent flow reattached to the body, these fluctuations being an order of magnitude greater than those typically found under the fully developed turbulent boundary layers further downstream on the same body. In the case of attached transition, the corresponding factor was 2 to 3. The authors used the linear stability method of Smith^[81] to predict the growth of Tollmein-Schlichting wave-like disturbances in a laminar boundary layer. This is basically a linearized solution of the Orr-Sommerfeld equation governing the behaviour of two-dimensional disturbances in laminar boundary layers. They reported close correspondence between the frequency content of their hot-wire signals and those predicted using the above method, just upstream of the transition point. With attached transition they found that the start of the transition process occurred at a disturbance amplification ratio (ratio of disturbance amplitude to that at the point of neutral stability) of e^{11} , as computed using Smith's method. For separation bubble triggered transition, the location of maximum pressure fluctuations correlated well with a predicted amplification ratio of e^{16} . Three fluctuating pressures were measured at the transition location during these experiments, the r.m.s. value, the largest negative value occurring during a one second sampling interval and the largest value in a 40 second interval. Expressed as percentages of the onset flow dynamic pressure, these fluctuating components, for the detached transition, were 14.5%, 35% and 51.7%. The corresponding values quoted for attached transition were 3.8%, 10% and 20.8%. The laminar separation bubbles occurred at Reynolds numbers (based on stream velocity and boundary layer momentum

thickness at laminar separation) in the range $400 < R_{\theta_s} < 800$ at corresponding ratios of dead fluid region length S to boundary layer momentum thickness θ_s of about $200 > S/\theta_s > 125$. From fully wetted experiments on an ogival headform in a water tunnel, Arakeri^[82] reported large pressure fluctuations in the reattachment region downstream of laminar separation bubbles. The frequency content of the measured fluctuating wall pressures just prior to transition agreed well with those predicted by the linear stability method of Smith^[81]. The r.m.s. and negative peak values of the measured pressure fluctuations were approximately 10% and 26% of the onset flow dynamic pressure. Arakeri concluded that these pressure fluctuations were the probable cause of cavitation inception in flows exhibiting laminar separation bubbles. In further experiments to determine the extent of the separated flow region and the Reynolds number range for its occurrence, Arakeri^[69] reported that with the ogival headform, laminar separation bubbles occurred in the Reynolds number range $250 < R_{\theta_s} < 500$, with streamwise separated region lengths of $200 > S/\theta_s > 80$. For a hemispherical nosed body the corresponding ranges were $200 < R_{\theta_s} < 400$, at $200 > S/\theta_s > 150$. Using a hot wire, Gaster^[83] probed the flow field within laminar separation bubbles on a flat plate with an auxiliary aerofoil in a wind tunnel, reporting large velocity fluctuations in the region of turbulent flow reattachment. The range of dimensionless bubble lengths and Reynolds numbers reported were $170 > S/\theta_s > 120$ at $170 < R_{\theta_s} < 450$.

The author is not aware of any similar reported data pertaining to flow fluctuations in the region of boundary layer transition on circular cylinders. Tani^[84] gives

some experimental data for the occurrence of laminar separation bubbles on two-dimensional circular cylinders in a low turbulence water tunnel. The streamwise lengths of the laminar separation bubbles were determined using a dye injection technique. The stream velocity at laminar separation was computed from the measured pressure distribution, and the laminar boundary layer momentum thickness at separation computed by Thwaites' method^[128] using measured values of the streamwise location of laminar separation. Figs. II.2, II.3 and II.4 show the cylinder data reported by Tani. As shown in Appendix II it may be deduced from Tani's data that in his experiments, laminar separation bubbles of dimensionless length $270 > S/\theta_s > 220$ occurred in the Reynolds number range $380 < R_{\theta s} < 500$. As may be seen from Fig. II.3, Tani reported that with laminar separation bubbles present, turbulent transition, as evidenced by the dye injection technique, occurred at an angular location of approximately 110 degrees, increasing slightly with Reynolds number. Fig. II.5 shows a resume of the dimensionless separation bubble length and Reynolds number data for the various bodies discussed above, adapted from Huang and Hannan^[80] and with the extreme values of Tani for circular cylinders appended. Fig. II.5 indicates that the separation bubbles reported by Tani were somewhat longer and occurred broadly within the same Reynolds number range as those on the other bodies in question. This indicates that separation bubbles on circular cylinders are not markedly different from those reported on other bodies. Further, since the pressure and velocity fluctuations which occur at transition and result from the growth of upstream disturbances in laminar boundary layers are a general viscous phenomena^[91], it is

reasonable to assume that similar pressure fluctuations occur at the locations of attached and detached turbulent transition on circular cylinders as those reported on other bodies^[80,82,83].

CHAPTER II.3

SOME INFLUENCES OF VISCOUS EFFECTS ON CAVITATION INCEPTION

From a comparison of the stall characteristics and cavitation inception behaviour of certain aerofoils, Alexander^[85] proposed a relationship between flow separation and the occurrence of cavitation. He postulated that laminar separation bubbles could provide stream nuclei with a long residence, low pressure environment in which to grow to their critical size. Bailey^[86] examined previously reported single phase and cavitating flow data on NACA 4412 and 661-012 aerofoil sections and emphasized the similarities between the behaviour of laminar separation bubbles and cavitation inception at various values of aerofoil incidence and Reynolds number. Casey^[87,88] used oil film flow visualization to determine the locations of laminar separation bubbles on a NACA 0015 section hydrofoil in a water tunnel and showed that limited cavitation occurred at the same chordwise location as the separation bubbles. He also found good correlation between the pressure coefficients at laminar separation and the desinent and incipient cavitation numbers. Arakeri and Acosta^[89] performed a classic series of experiments on axisymmetric headforms in a recirculating type water tunnel and a free surface tunnel. They used a Schlieren system to visualize both the fully wetted and corresponding cavitating flows in the region of laminar separation bubbles. High speed motion pictures were taken of the inception process on a hemispherical headform and a Swedish headform. From the experiments on the hemi-

spherical headform, the following conclusions were reached.

1. Visible macroscopic cavitation inception occurred in the turbulent reattachment region at the downstream end of the laminar separation bubbles.

2. Microscopic bubbles formed in the reattachment region were entrained by the reverse flow in the separated region and recirculated in the upstream direction at low velocities. Even with a relatively long residence time in the separated region (of order 0.5×10^{-3} secs.) these bubbles experienced little growth.

3. Growth times of macroscopic bubbles in the reattachment region were estimated to be less than 10^{-4} secs.

For the corresponding experiments with the Swedish headform, inception also occurred within the separated region downstream of the minimum pressure point on the headform. With the experimental technique used, the details of the inception process were not resolvable to the same degree as with the hemispherical headform, since the height of the separated region was much less.

The same authors investigated cavitation inception on a 1.5 calibre ogival headform^[90] in the supercritical Reynolds number flow regime (defined by these authors as when laminar separation is precluded by turbulent transition). In this case the inception cavitation numbers were found to correlate well with the pressure coefficient at transition. The authors surmized that under these conditions cavitation inception might be associated with the pressure and velocity fluctuations which are known to occur at the location of attached transition^[80]. At supercritical Reynolds numbers, inception occurred at discrete locations around the head-

forms as opposed to the circumferentially continuous 'sheet' type cavities observed at subcritical Reynolds numbers with laminar separation bubbles present. Further confirmation of a relationship between attached turbulent transition and cavitation inception at supercritical Reynolds numbers was found by Arakeri and Acosta during experiments on axisymmetric headforms fitted with boundary layer trips on the upstream high pressure regions of the forebodies^[92]. They reported that with a boundary layer trip fitted, cavitation inception could not be induced on a hemispherical headform even at cavitation numbers of less than half the corresponding inception value without the trip fitted. They hypothesized that with a boundary layer trip fitted, turbulent transition could occur well upstream of the minimum pressure point on a body, as opposed to natural transition which, even at very high Reynolds numbers would be unlikely to move much upstream of the minimum pressure point on account of the favourable pressure gradient in this region of the flow. On the basis that cavitation inception at supercritical Reynolds numbers occurs at the location of transition, then causing the latter to occur in the upstream high pressure region could lower the inception cavitation number below that attainable in a given facility. This hypothesis received some verification from experiments on a hemispherical headform fitted with a boundary layer trip which enabled laminar separation bubbles to occur downstream of the minimum pressure point of the headform below a critical tunnel velocity. Attached cavitation was induced at a tunnel velocity below this critical value and then the velocity increased at constant static pressure. As the velocity was increased through the critical value the

cavitation disappeared completely. This apparently paradoxical behaviour, wherein cavitation was eliminated by decreasing the cavitation number, was taken by the authors to indicate an upstream movement of the transition location at tunnel velocities greater than the aforementioned critical value at which laminar separation bubbles downstream of the minimum pressure point disappeared.

Attached cavitation inception in the region of laminar separation bubbles on a two-dimensional hydrofoil was reported by Blake, Wolpert and Geib^[93] who took high speed motion pictures of the inception process. They found inception to occur at the reattachment region of the separation bubbles in a similar manner to that reported by Arakeri and Acosta^[89]. With a boundary layer trip fitted on the forward region of the hydrofoil, Blake et al. found inception to be in the form of travelling cavitation bubbles. Those far from the hydrofoil surface grew in size as they passed downstream through the minimum pressure region, thereafter decreasing in size. A large number of those bubbles which were formed close to the hydrofoil took on nearly hemispherical shapes that persisted until after they passed the minimum pressure point, whereafter the liquid flow over these bubbles separated. The portion of the bubble adjacent to the hydrofoil surface became stationary, the outermost part continuing to move with the free stream. This stretching ultimately caused the bubbles to split and be entrained by the three dimensional vortex generated by the separated liquid flow. It is clear from the investigation of Blake et al. that the observed interaction between the cavitation bubbles and their bounding liquid flow is a complex dynamic

phenomena.

Acosta and Parkin^[94] classified three basic types of cavitation inception on smooth bodies, any or all of which may exist separately or together.

1. Travelling or growing cavitation bubbles originating from a position near the minimum pressure point on a body.

2. Fixed "spot" nucleation sources near the position of minimum pressure behind which are regions of attached cavitation.

3. Attached "band" or "sheet" cavities consisting of many transient bubbles or appearing as a smooth well defined glossy surface.

The same authors emphasized the difficulties in reconciling the cavitation inception behaviour of similar test bodies in different test facilities. They cited the International Towing Tank Conference round-robin tests, wherein the same test body (an ellipsoidal headform) was tested in different facilities, exhibiting widely differing inception characteristics^[95]. Huang and Peterson^[96] classified the factors influencing cavitation inception into those associated with the single phase liquid flow around a given body and those concerned with the gas and vapour phase components of the flow, this latter group being termed the cavitation 'resistance' or 'susceptibility' of a particular flow. The former group includes such effects as non-uniformities in a flow which might distort the static pressure distribution around a body, and the influence of free stream turbulence and acoustic and mechanical vibration on the development of the boundary layers on a body. The cavitation 'resistance' of a flow depends on such factors as the number, size, spatial

distribution and surface tension of free stream nuclei and the number of active nucleation sites on a given test body^[97,98,99]. Recent reviews of the influence of free stream factors and bubble dynamics on cavitation inception have been presented by Gates and Acosta^[100] and Plesset and Prosperetti^[101] respectively. The experiments that will presently be described, on cavitation inception on a circular cylinder in the Oxford University High Temperature Cavitation Tunnel, were aimed at investigating viscous effects on cavitation inception, rather than the influences of the cavitation 'resistance' of the tunnel flow. These experiments were performed at nominally constant low values of working fluid gas content (typically 10 parts per million, gravimetric) in order to minimise the effects of streamborne nuclei and gas bubbles. Hence no further mention will be made of the influence of the cavitation 'resistance' or 'susceptibility' of the tunnel flow, except to note that the results to be presented pertaining to cavitation inception on a circular cylinder must, to some extent, reflect the characteristics of the flow in the High Temperature Cavitation Tunnel. Hence completely similar inception characteristics may not be found in other test facilities, even using the same cylindrical test body.

Most of the cavitation inception investigations discussed above were performed on axisymmetric headforms equipped with cylindrical afterbodies or two-dimensional streamlined hydrofoils at moderate incidences. Inception has also been observed to occur in the detached free shear layers downstream of bluff axisymmetric and two-dimensional bodies. Kermeen and Parkin^[102] observed inception to occur in the free shear layers behind sharp

edged disks and reported a correlation between the incipient cavitation number and Reynolds number. Arndt^[103] derived an expression for the minimum pressure in the vortex system downstream of a disk by equating the vorticity in the boundary layer leaving the disk to that contained in the ring-vortex system. A Reynolds number dependence of the minimum pressure in the vortex rings was predicted by this analysis due to variations in the relevant boundary layer thickness at separation. This Reynolds number dependence showed good agreement with experimental data. Young and Holl^[44] observed inception in the vortex streets downstream of two-dimensional wedges, but the Reynolds number range used in the experiments was too narrow to detect any viscous effects on the incipient cavitation number. Reports of inception on bluff bodies which are well rounded on both their frontal and downstream profiles, appear to be very scarce indeed. This is unfortunate, since bodies of this form exhibit wide Reynolds number dependent variations in flow pattern since free streamline separation is determined by viscous effects rather than by geometric constraints as is the case with bodies exhibiting profile discontinuities. Thus these bodies might be expected to exhibit a corresponding variety of viscous effects on cavitation inception. Knapp et al.^[104] presented photographs of detached inception occurring downstream of a sphere at a Reynolds number of about 1.5×10^5 (estimated from quoted flow velocity and sphere diameter). This is of some interest, since it is well known^[84,105,106] that in the region of the critical Reynolds number the behaviour of boundary layer transition and separation is similar for spheres and cylinders, both of which exhibit a laminar separation bubble flow regime.

On the basis of Achenbach's data^[105], the photographs of Knapp et al. would correspond to a subcritical Reynolds number, assuming a smooth sphere and a moderate free stream turbulence level, thereby indicating detached type cavitation inception on this class of body at subcritical Reynolds numbers (that is, Reynolds numbers below which turbulent reattachment of the separated laminar boundary layer occurs). Unfortunately there appear to be no reported observations of cavitation inception on spheres at supercritical Reynolds numbers which might corroborate the relationship between laminar separation bubbles and cavitation inception for this class of body. As will be reported in Section II.5.1, with a circular cylinder both the 'band' and 'spot' types of inception reported by Acosta and Parkin^[94] were observed, and also a detached form of inception occurring well downstream of the cylinder in the wake at subcritical Reynolds numbers. This latter form may be similar to that observed by Knapp et al. on a sphere in the same flow regime, as discussed above. In the absence of any reported observations of cavitation inception on spheres at different Reynolds numbers, this latter point must await further clarification.

CHAPTER II.4

EXPERIMENTAL SET-UP AND FULLY WETTED FLOW EXPERIMENTS

This chapter describes the fully wetted flow experiments which were performed to determine the pertinent characteristics of the viscous flow around the cylindrical test body. No direct observations or measurements of the test body boundary layers were made, since the test section and cylinder were not provided with the requisite instrumentation. Rather the characteristics of the cylinder boundary layers were inferred from their influence on the free-streamline separation points as manifested by the variation of cylinder base pressure and test section wall pressures with Reynolds number. In this context it was fortuitous indeed that a cylindrical test body was used since, as outlined in Chapter II.2, the large body of single phase flow data on circular cylinders renders this aforementioned inference possible. The fact that test section wall pressures are a measure of the disposition of the free-streamlines of a confined flow has been utilized previously by various investigators to determine blockage corrections to the force coefficients of both lifting and pure drag wake and cavity flows^[107,108].

The cylindrical test body was the same one used in the vapour ventilation experiments reported in Chapter I.6, and the instrumentation similar to that described in Chapter I.8, except that the thermocouple measuring the cavity temperature was removed from the cylindrical test body since it was not required for the experiments to be reported presently. Some further details of the

experimental set-up are given below.

II.4.1 EXPERIMENTAL SET-UP

During all the fully wetted and cavitating flow experiments which will be reported in Part II of this thesis, the vapour ventilation system was isolated from the cylindrical test body by completely closing the vapour ventilation flow throttle valve (see Fig. I.46). Fig. II.6 is a schematic of the tunnel working section showing the tunnel state variables measured during these experiments. A Hasselblad camera was used to take still photographs during the cavitation experiments and was positioned on a mounting bracket which could be clamped to the top or side of the test section to take overhead or end-view photographs of the flow field in the region of the test body. The cylinder base pressure P_b sampling tube was connected to the first, or home position inlet port of the type 48J Scanivalve. The 18 wall pressure P_w sampling tubes were sequentially connected to the next 18 inlet ports of the Scanivalve such that when it was actuated, the wall pressures were sampled starting at the upstream location. The wall pressure tappings were uniformly spaced along the test section wall in the stream-wise direction.

The cylindrical test body had a series of 3.175 mm deep ventilation flow admission slots machined through its back face. As shown in Fig. II.7, these slots subtended a semi-angle of 19.5 degrees on either side of the cylinder centreline. Since the maximum value of the turbulent boundary layer separation angle is known to be about 145

degrees for supercritical Reynolds numbers^[72,75,84], the slots could be expected to remain well within the cylinder wake for all the flow regimes covered in the experiments. It was thus assumed that they would have no effect upon the attached region of flow around the cylindrical test body.

II.4.2 FULLY-WETTED FLOW EXPERIMENTS

Since the experiments spanned the Reynolds number range $10^5 < R_d < 0.85 \times 10^6$, it was important to determine the critical Reynolds number R_{CR} at which turbulent reattachment of the separated laminar boundary layers occurred. This was done in two ways, first by examining the variation of C_{pb} over the required range of R_d , and then comparing this to the corresponding behaviour of the wall pressure distributions, these reflecting the disposition of the free-streamlines in confined flow.

The operating range of the cylinder-in-test section is shown in Fig. II.8, which shows curves of constant Reynolds number plotted against axes of test section velocity and onset flow temperature. The fully wetted experiments were performed by adjusting T_∞ and U_∞ to give the required value of R_d and scanning the tunnel transducers twice at each operating state. The cavitation number σ_v was maintained above a minimum value of 3.5 throughout these experiments. Fig. II.9 shows the variation of the cylinder base pressure coefficient C_{pb} and the minimum test section wall pressure coefficient C_{wMIN} with Reynolds number over the range $10^5 < R_d < 8.5 \times 10^5$. At the lowest Reynolds number attained during the experiments ($R_d = 10^5$), C_{pb} was about -0.95. With

increasing Reynolds number, C_{pb} increased rapidly to a maximum value of -0.25 at $R_d = 3 \times 10^5$, thereafter decreasing again to attain a value of approximately $C_{pb} = -0.5$ at $R_d = 8.5 \times 10^5$. The general shape of the C_{pb} versus R_d plot shown in Fig. II.9 is similar to that obtained by many other investigators for single phase flow around a circular cylinder, as outlined in Chapter II.2. The corresponding variation of C_{wMIN} over the same Reynolds number range illustrates well the abruptness of the change from a broad subcritical Reynolds number wake to a narrow supercritical one, as noted by Tani^[84] and Achenbach^[72]. Fig. II.10 shows the variation of cylinder base pressure coefficient C_{pb} with Reynolds number, the latter variable being plotted on a more extended scale than as in Fig. II.9. From this plot it may be seen that, with increasing Reynolds number, C_{pb} decreases from its subcritical value of -0.95 , to attain a value of -0.55 at $R_d \approx 2.3 \times 10^5$, at which Reynolds number it decreases abruptly to a value of $C_{pb} \approx -0.4$. The observed behaviour of the cylinder base pressure in this region was thus very similar to that observed by Bearman^[76,78] during wind tunnel experiments on a circular cylinder in the same Reynolds number regime. In particular, he reported the establishment of laminar separation bubbles, verified by oil film flow visualization, at a base pressure coefficient of $C_{pb} \approx -0.55$. The corresponding Reynolds number was higher in his experiments, but this can be attributed to the fact that he used a highly polished cylinder in a very low turbulence wind tunnel ($T_u < 0.002$). Comparing the behaviour of the cylinder base pressure in the present experiments it may be deduced that the critical Reynolds number at which laminar separation bubbles occurred on the

cylinder was 2.3×10^5 . It should be emphasised that in the present experiments the presence of laminar separation bubbles was not verified by flow visualization, but rather inferred from the observed mode of cavitation inception at Reynolds number greater than 2.3×10^5 . A further feature of Fig. II.9 worthy of comment is the difference in behaviour of C_{wMIN} and C_{pb} at Reynolds numbers lower than the critical value. Whereas C_{pb} increased uniformly in the critical Reynolds number flow regime, the abruptness of the increase in C_{wMIN} at the critical Reynolds number is much more marked. This is probably a consequence of the fact that the behaviour of the minimum wall pressure coefficient is a better reflection of the free streamline behaviour since, being remote from the cylinder, it is not so influenced by the unsteady flow in the wake downstream of the cylinder.

None of the aforementioned data are corrected for the effects of tunnel wall constraint, primarily because of the lack of measured drag coefficients and also because there appears to be no correction rule which is valid throughout the range of Reynolds number considered. This difficulty was noted by Achenbach^[72], whose experiments on a circular cylinder in a wind tunnel covered a similar Reynolds number range to those reported herein, at a somewhat higher cylinder blockage. He thought it best to present uncorrected data. The data shown in Figs. II.9 and II.10 could have been corrected using the method of Maskell^[109]. This correction rule, however, requires the invariance of $C_D/(1-C_{pb})$, and since the cylinder drag coefficient was not measured the validity of this assumption could not be checked. In this context, Ewers^[40] measured the drag and base pressure coefficients for a

circular cylinder in the Reynolds number range $10^5 < R_d < 1.7 \times 10^5$ and reported considerable variation in the parameter $C_D/(1-C_{pb})$. A further reason for presenting uncorrected data is that the purpose of the experimental program was a comparison of fully wetted and cavitation inception behaviour, absolute values of the flow parameters being of secondary importance.

The value of the cylinder critical Reynolds number deduced from the behaviour of the cylinder base pressure coefficient ($R_{CR} = 2.3 \times 10^5$) is what would be expected for a smooth cylinder in a stream of moderate turbulence level. Fig. II.11, taken from Ref. 111 shows the influence of free stream turbulence level T_u on the critical Reynolds number of a smooth cylinder in unbounded plane flow. Bearing in mind that in the present fully wetted flow experiments the velocity at the cylinder was uncorrected for the effects of wall constraint, Fig. II.11 indicates a free stream turbulence level in the region of $T_u = 0.01$. This compares favourably with the cylinder experiments of Achenbach^[72], in which the critical Reynolds number (velocity uncorrected for constraint) was found to be approximately 2.6×10^5 , with a cylinder blockage ratio of 0.166 and a measured free stream turbulence level of $T_u \approx 0.007$.

CHAPTER II.5

CAVITATION INCEPTION ON THE CYLINDRICAL TEST BODY

This chapter describes the experiments performed to determine the types of cavitation inception occurring on the cylindrical test body in the Reynolds number range $10^5 < R_d < 0.85 \times 10^6$. The observed incipient states are then discussed and their appearance and location explained in terms of the fully wetted flow corresponding to each of them.

II.5.1 EXPERIMENTAL TECHNIQUE AND OBSERVED FORMS OF INCIPIENT CAVITATION

Cavitation inception experiments were performed over the same range of Reynolds numbers as depicted in Fig. II.9. These experiments were done by setting the test section velocity and temperature to produce a given Reynolds number and then reducing the static pressure in the test section until visible cavitation occurred within the flow. The pressure reduction technique used was that described in Section I.5.3, wherein fluid was bled from the pressurizer plenum to the sump tank (see Fig. I.38). Inception was detected visually under continuous quartz-iodine illumination. The tunnel transducers were scanned to obtain the requisite state variables for both the fully-wetted flow conditions immediately prior to inception and for the corresponding limited cavitation flow. Three modes of cavitation inception were found to occur, depending on the Reynolds number R_d .

At Reynolds numbers less than about $R_d = 2 \times 10^5$,

no attached cavitation was observed in any of the experiments. In all cases, inception occurred downstream of the cylinder within the wake. Fig. II.12 is a photograph of typical wake type inception taken from directly over the cylinder, at a Reynolds number of 1.2×10^5 and an incipient cavitation number of $\sigma_{iw} = 2.2$. The dark spot just behind the trailing edge of the cylinder on the right hand side was caused by foreign matter trapped between the inner and outer test section windows. It can be seen from the photograph that the front of the cavitated region was approximately one diameter downstream of the trailing edge of the cylinder, with cavitation extending some distance downstream. This represented the minimum state of cavitation attainable, any increase in static pressure eliminating the cavitation completely. The location and extend of this form of inception appeared to remain constant over the limited range of Reynolds numbers investigated ($10^5 < R_d < 2 \times 10^5$).

Above a certain Reynolds number, the mode of inception changed from the detached wake type to an attached form, occurring with marked spanwise uniformity across the back faces of the cylinder at about 110 degrees from the frontal stagnation point. Following Acosta and Parkin^[94], this is classified as band type inception, since it appears to be the two-dimensional equivalent of the attached band or sheet type limited cavitation reported by them on axisymmetric headforms. Fig. II.13 is a photograph of typical band inception from which it can be seen that the cavitation band extends across the central portion of the cylinder span, being missing only from the end-span locations next to the test section walls. The Reynolds number range over which the band type inception

was found is depicted in Fig. II.14. It proved difficult to obtain steady enough flow conditions to study the transition from the wake to the band type of inception. Preliminary experiments had been performed before the installation of the scanning data acquisition system, using a manually switched digital voltmeter to measure the output of the tunnel transducers^[110]. These showed considerable variation in the Reynolds number at which transition from wake to band type inception occurred. These variations were attributed mainly to lack of data synchronization, but may have been caused by effects other than purely viscous ones, since the same nominal Reynolds numbers were generated by wide variations in tunnel velocity and temperature. In the present experiments, the transition range shown in Fig. II.14 was determined using a narrow range of tunnel velocity, tunnel temperature being used to vary the Reynolds number. The unsteadiness of the flow in the transition region between the wake and band types of inception was undoubtedly caused by the high sensitivity of cylinder base pressure and hence drag to small changes in Reynolds number. Bearman^[76] encountered similar difficulties when investigating laminar separation bubbles on a cylinder in a wind tunnel. From what could be observed, it appeared that the stream-wise length of the minimum sustainable cavitation band increased with decreasing Reynolds number. At a certain Reynolds number, the attached band shed sheets of cavitation into the wake before becoming detached. It should be emphasized that the above observations were made under unsteady flow conditions and should thus be treated with reservation.

Above a certain Reynolds number, the spanwise uniformity of the band inception was disrupted by the disappearance of the bands from sections of the span and the coincident appearance of wedges of cavitation at certain spanwise locations. These wedges were aligned in the direction of flow, with their apexes pointing in the upstream direction. Fig. II.15 is a photograph of discontinuous band and spot type incipient cavitation existing together at a Reynolds number of 3.9×10^5 and a cavitation number of $\sigma_v = 2.55$. The spot on the right hand side of the cylinder is situated within a section of the band cavitation, with its apex just upstream of it. The spot on the left hand side is closer to the 90 degree cylinder generator. The transition range of Reynolds numbers for band and spot type cavitation existing together is shown in Fig. II.14. Further increase in Reynolds number beyond the aforementioned transition range resulted in the disappearance of the band type cavitation completely, leaving only the spot type. Fig. II.16 shows a photograph of typical spot type inception, with one spot just to the left of the mid-span location and the other nearer the right hand end of the cylinder. The former spot may be seen to be nearer the 90 degree cylinder generator than the latter, on either side of which may be seen the last remnants of a cavitation band. The spot type inception persisted up to the highest Reynolds number attained in the inception experiments ($R_d \approx 10^6$). The number and location of incipient cavitation spots changed between experiments, even at the same Reynolds number, one, two or three incipient spots appearing at varying spanwise locations. Fig. II.17 shows incipient cavitation spots at $R_d = 7.1 \times 10^5$ at a corresponding inception cavitation

number of $\sigma_{is} = 2.23$.

During the experiments performed in the band and spot incipient cavitating flow regimes the photographs shown in Figs. II.13 to II.17 were all taken from the overhead position. The cavitation inception behaviour of the lower surface of the cylinder could not be photographed since the bottom wall of the test section consisted of a blank metal window to accommodate wall pressure tapings. The lower surface of the cylinder could be viewed, albeit somewhat obliquely, from the test section side window, and this was done at each incipient cavitation state in order to detect any asymmetries in behaviour. None were found for the band type inception which appeared to occur coincidentally on both the upper and lower surfaces of the cylinder. This is of some interest since Bearman [76,78] reported the establishment of a single laminar separation bubble on the lower surface of a cylinder within a narrow Reynolds number range, which manifested itself in a measured lift coefficient of 1.3. Beyond this narrow Reynolds number range, a separation bubble was established on the upper cylinder surface with consequent disappearance of the aforementioned mean lift coefficient. No such asymmetries were observed with the band type inception found in the present experiments. Bearman's experiments were performed in a wind tunnel with a very low free stream turbulence level ($T_u = 0.002$) and in this situation it is possible that the laminar boundary layers in the downstream regions of adverse pressure gradient on the upper and lower cylinder surfaces would be sensitive to small differences in surface roughness between the upper and lower surfaces, resulting in the earlier turbulent transition and subsequent reattachment of the

boundary layer on the cylinder lower surface. In the present experiments, the estimated free stream turbulence level was considerably higher than that obtaining in Bearman's experiments ($T_u \approx 0.01$), in which case the turbulent transition and reattachment of the boundary layers might be more influenced by free stream turbulence than by variations in surface roughness. In the incipient spot cavitating flow regime, however, it was observed that inception always occurred initially on the lower surface of the cylinder in all the present experiments. The smallest achievable decrement in tunnel static pressure then served to produce inception on the upper surface of the cylinder, as well as the lower. It was not possible to estimate the spanwise location of the incipient cavitation on the lower surface, due to the obliquity of the view through the test section side window. It appeared that the location of the incipient spots on the lower surface varied randomly between experiments, in a similar manner to that exhibited by the cavitation spots on the cylinder upper surface.

II.5.2 DISCUSSION OF OBSERVED TYPES OF CAVITATION INCEPTION

Of the three forms of cavitation inception observed in these experiments, the band type can most confidently be related to the viscous behaviour of the fully-wetted flow. Considering the fully wetted and cavitation inception data of Arakeri and Acosta^[89], and the known location and Reynolds number range for the occurrence of laminar separation bubbles on circular cylinders (Tani^[84], Achenbach^[72] and Bearman^[76,78]), it can be concluded that the observed band type inception is associated with

the turbulent reattachment region downstream of laminar separation bubbles. The incipient cavitation band in the present experiments was observed to occur at an angular location of 110 degrees from the frontal stagnation point. No measurable change in its location was observed over the Reynolds number range in which it occurred. Tani^[84] observed turbulent transition to occur at about 110 degrees from the frontal stagnation point of his cylinders, with subsequent reattachment of the turbulent boundary layer occurring at about 117 degrees, this angle decreasing slightly with increasing Reynolds number, as shown in Fig. II.3. No further conclusions can be drawn from a comparison of the present band inception with the data of Tani shown in Fig. II.3, since he did not measure fluctuating wall pressures in the region of the observed laminar separation bubbles. In any event, differences in the location of laminar separation and turbulent reattachment might be expected between the present experiments and those of Tani, on account of differences in cylinder blockage ratio and free stream turbulence levels.

The only firm conclusion that can be drawn from the wake type inception experiments is that no attached cavitation was observed below a certain Reynolds number. As can be seen in Fig. II.12, the streamwise distance of the leading edge of the cavitated region from the cylinder varied between about half and one diameter from the downstream face of the cylinder, the cavitation extending a further cylinder diameter in the downstream direction. This form of wake inception may be similar to that reported by Knapp et al.^[104] for subcritical Reynolds number flow around a sphere as previously noted in

Chapter II.3. It is possible that at subcritical Reynolds numbers the observed detached cavitation inception is associated with the detached nature of turbulent transition in this flow regime. Tani^[84] reported that in the range of high subcritical Reynolds number covered in his experiments, laminar separation occurred at 75 degrees from the frontal stagnation point, with transition taking place at about 90 degrees. Even at a very much lower Reynolds number ($R_d = 8.5 \times 10^3$), transition has been found to occur 0.7 cylinder diameters downstream of laminar separation^[115]. It thus appears unlikely that the wake type inception is directly associated with turbulent transition, since this could reasonably be expected to occur much further upstream than the leading edge of the cavitation shown in Fig. II.12. Moreover, Gaster^[113] has suggested that it is unlikely that the turbulent pressure fluctuations in the wake at the observed location of cavitation would be great enough to cause cavitation inception. A further possibility is that inception in this flow regime may occur within the cores of the vortex street downstream of the cylinder in a similar manner to that reported by Young and Holl with two-dimensional wedges^[44]. Fig. II.18 shows the distribution of static pressure on the centreline of a wake behind a circular cylinder in a wind tunnel at a high subcritical Reynolds number, presented by Roshko^[114]. Of interest in the present context is the occurrence of a low static pressure region about one diameter downstream of the cylindrical test body, that is, at about the same streamwise location at which wake type inception was observed in the present experiments. This low pressure region results from the vortex shedding dynamics. The minimum pressure coefficient C_p measured by

Roshko was -1.3, as shown in Fig. II.18, as opposed to a typical incipient cavitation number of 2.2 found in the present experiments. This indicates that wake inception is not simply related to this low static pressure region. It is clear from the foregoing discussion that further experiments are required to resolve the causes of wake inception on circular cylinders. Suggestions for further experiments will be offered in Part III.

The spot type inception found at higher Reynolds numbers appears to be the two-dimensional equivalent of that found by Arakeri and Acosta^[92] on axisymmetric bodies at supercritical Reynolds numbers. Although no fully wetted measurements were made of the variation in the location of turbulent transition in the present experiments, the appearance of the cavitation spots, together with the partial disruption of the band cavitation indicates that at certain spanwise positions the location of turbulent transition has moved far enough upstream to disrupt laminar separation. This is corroborated by the hot-wire vortex shedding measurements of Bearman^[78] behind a cylinder in air over a similar Reynolds number range to that covered in the present experiments. Bearman found that with laminar separation bubbles present on the cylinder, narrow band vortex shedding occurred at a Strouhal number of about 0.45. At higher Reynolds numbers the vortex shedding became spread over a wide band of frequencies as "turbulent wedges" disrupted the laminar separation bubbles. Thus in the transcritical flow regime, cavitation inception appears to visualize the turbulence wedges which occur at transition, and which invariably start from some isolated surface-roughness^[113].

It is possible that the discrete nature of the spot type inception is related to three-dimensionalities of the flow in the region of the critical Reynolds number. Humphreys^[116] performed flow visualization experiments on a circular cylinder in a wind tunnel and observed regular spanwise variations in the flow in the region of boundary layer separation. Using a hot-wire probe he found alternate regions of separated laminar and attached turbulent flow, the spanwise length of these regions being of the same order as the cylinder diameter. Korotkin^[117] measured spanwise static pressure distributions on circular cylinders at supercritical and transcritical Reynolds numbers, and reported large and regular static pressure variations across the cylinder span in the region of turbulent boundary layer separation. The peak values of these pressure variations was one unit of pressure coefficient. The fact that Korotkin found large spanwise pressure variations at an angular location and within a Reynolds number range where laminar separation bubbles would be expected to occur is markedly at variance with the two-dimensionality of the band type inception found in the present experiments. This cannot presently be taken further, since the cylindrical test body used in the present experiments was not instrumented to measure spanwise pressure distributions.

As previously mentioned in Chapter II.1, the only recently reported observations of cavitation inception on circular cylinders that came to light during a literature survey were those of Palanichamy et al.^[71]. Their experiments were conducted with cylinders of moderate and large surface roughnesses in a cold water cavitation tunnel. With the cylinder of large surface roughness,

cavitation inception always occurred on the roughness elements on the cylinder surface, and since this form of cavitation is not pertinent to the present investigation it will not be discussed further. The smooth cylinder studied by these authors was 20 mm in diameter and it was mounted in a rectangular venturi the dimensions of which gave the cylinder a solid blockage ratio of 0.2 and an aspect ratio of 1.95. No fully wetted experiments were performed to determine the critical Reynolds number of the cylinder. From the quoted ranges of test section velocity and tunnel water temperature it appears that the Reynolds number range covered in the experiments was $1.55 \times 10^5 < R_d < 3.1 \times 10^5$. The authors reported that above a tunnel velocity of about 9 m/sec. the incipient cavitation was attached to the cylinder while below this velocity, inception occurred within the wake downstream of the cylinder. This velocity, on the basis of the quoted tunnel water temperature, corresponded to a Reynolds number of 1.9×10^5 , which would be a reasonable value of the critical Reynolds number for the cylinder in the test section which they used since their tunnel had no turbulence damping screens and had a contraction ratio of 4.6. These authors presented a still photograph of incipient cavitation on the smooth cylinder (Fig. 4 in Ref. 71), taken from the cylinder end-span location at an oblique angle. This photograph showed two discrete regions of cavitation which were similar in appearance to the spot type inception found in the present experiments. The authors stated that this form of inception occurred at velocities greater than 9 m/sec. which, on the basis of the previous estimate of the critical Reynolds number for their cylinder ($R_{CR} = 1.9 \times 10^5$), corresponded to the supercritical Reynolds number

flow regime. The authors did not report the occurrence of a band type cavitation inception flow regime. It is possible that the effects of free stream turbulence and the small aspect ratio of the cylinder produced a very narrow Reynolds number range in which laminar separation bubbles occurred, or even precluded their formation altogether. Palanichamy et al. attributed the wake type inception found in their experiments to the growth of gas bubbles by diffusion in the cylinder wake, at tunnel velocities low enough to provide the necessary time for bubble growth. In the light of the present experiments, it is much more likely that the wake inception observed by them was caused by viscous effects.

Some further details of the cavitation inception experiments reported in this chapter may be found in Ref. 126.

CHAPTER II.6

THE DEVELOPMENT OF CAVITATION ON THE CYLINDRICAL TEST BODY

Two-dimensional circular cylinders in single phase subsonic flow exhibit a large variety of flow patterns. This is a consequence of the cylinder being the ultimate example of a two-dimensional well-rounded bluff body. On account of this bluntness the flow must separate at all but the lowest Reynolds number, and the constant curvature of a cylinder results in boundary layer separation being determined solely by viscous effects.

For developed cavitating flow, the cavitation separation points on a circular cylinder are also subject to viscous effects. Brennen^[70] reported a strong Reynolds number dependence of the cavitation separation points of a cylinder at subcritical Reynolds numbers. For well-rounded bodies exhibiting fully wetted laminar separations, Arakeri^[69] proposed a correlation for the streamwise displacement between the boundary layer and cavitation separation points in terms of the Taylor-Saffman parameter (the ratio of Weber to Reynolds numbers). This correlation showed good agreement with observations on axisymmetric headforms, spheres and a cylinder at subcritical Reynolds numbers. Syamala Rao and Chandrasekhara^[118] measured cavitation separation angles on cylinders of various diameters in a venturi. Although the critical Reynolds numbers were not determined for these cylinders, from the Reynolds numbers quoted ($2.97 \times 10^5 < R_d < 7.32 \times 10^5$), it may be concluded that all the experiments were performed in the supercritical

flow regime. The authors reported that the developed cavitation separation angle decreased with decreasing cavitation number and increased with decreasing Reynolds number. At high Reynolds numbers, the cavitation separation angles approached those given by Rosko's notched hodograph^[119]. These previously reported results on cavitating flows around circular cylinders all pertain to the developed cavitation flow regime. With limited cavitation, there are strong interactions between the fully wetted flow and the initial phase of cavitation development. Martyrer^[120] investigated the influence of Reynolds number on the drag coefficient versus cavitation number behaviour of two-dimensional circular cylinders. Using cylinders of two different diameters, corresponding to subcritical and supercritical Reynolds numbers for his experimental configuration, he obtained the drag coefficient data shown in Fig. II.19. At supercritical Reynolds numbers, as σ_v was reduced below the inception value, C_D rose very rapidly to a peak value of about 1.35, thereafter falling away to approximate to the relationship,

$$C_D/(1+\sigma_v) = \text{constant} \quad \text{II.1}$$

For a cylinder at a subcritical Reynolds number, Martyrer reported no such increase in C_D just below inception. As cavitation commenced, C_D decreased smoothly from the fully wetted value of 1.1 to coincide with the supercritical Reynolds number data at lower values of σ_v . Fig. II.19 also shows drag coefficient data obtained by Silberman in a free-jet tunnel, for a circular cylinder at a subcritical Reynolds number^[67]. Silberman's data is very similar to that of Martyrer for $R_d < R_{CR}$, the main difference being

the fully wetted drag coefficient values. These differences was probably due to variations in cylinder surface-roughness and free stream turbulence between the two experimental configurations and the influence of constraint on Martyrer's data. Varga and Sebestyn^[45] presented drag coefficient versus cavitation number data for a cylinder at a supercritical Reynolds number, the form of which was similar to that of Martyrer for $R_d > R_{CR}$. Varga further stated that the "critical" cavitation number at which the drag coefficient increased rapidly, with the development of cavitation at supercritical Reynolds numbers, is constant for a given cylinder in a given test section, and independent of Reynolds number.

The purpose of the investigation to be reported in the next section was to study the development of cavitation from the wake, band and spot incipient states described in the previous chapter. One of the motivations for these experiments was to attempt to explain differences between the band type inception found in the High Temperature Cavitation Tunnel and that found by Ewers^[48] on a circular cylinder in the Oxford University Cold Water Cavitation Tunnel when operating at a supercritical Reynolds number. Ewers found that the apparent incipient cavitation state consisted of a band of cavitation occurring uniformly across the span at an angle of about 90 degrees from the frontal stagnation point. The streamwise extent of this cavitation band was considerably greater than the band type inception found in the present experiments. As will be described in Section II.6.1, on the basis of the present studies of the development of cavitation from the incipient band state, it was concluded that the minimum sustainable state of cavitation attained in Ewer's

experiments was not in fact the true incipient state. Rather it was limited cavitation which was similar to that observed in the present experiments at cavitation numbers below the inception value.

II.6.1 EXPERIMENTAL TECHNIQUE AND RESULTS

The various flow regimes investigated were characterized by the cylinder base pressure coefficient C_{pb} . This is a useful parameter in experiments covering the fully wetted to the fully cavitating flow regimes, at various Reynolds numbers spanning the critical regime, since it gives an indication of the disposition of the free-streamlines of the flow through the inviscid steady flow relationship,

$$U_s = U_\infty (1 - C_{pb})^{\frac{1}{2}} = U_\infty (1 + \sigma_c)^{\frac{1}{2}} \quad \text{II.2}$$

Further, in the present context, C_{pb} and σ_c are obtained from the same pressure transducer, thus providing continuity of measurement throughout the flow regimes considered. The cavitation number σ_v was likewise taken to represent the degree of cavitation on the cylinder.

Before commencing each experiment the tunnel gas content was measured with the Van Slyke apparatus. If required, the tunnel working fluid was de-gassed to attain a measured gas content of approximately 10 parts per million (gravimetric), using the de-gassing method described in Chapter I.5.4. The pressure transducers were bled before each experiment by pressurizing the tunnel to about 2 bar and opening the transducer bleed screws. The required incipient cavitation states were established by adjusting the tunnel temperature and velocity to give the requisite Reynolds number, as defined in Fig. 11.14, and

lowering the tunnel static pressure to induce cavitation. Inception was detected visually under continuous quartz-iodine illumination. It was deemed desirable to obtain as many data points as possible for each type of cavitation development and thus the tunnel pressure was lowered by very small decrements between data scans. This was done using the techniques described in Chapter I.5.3. The tunnel pressure was set to a value just greater than that required to produce inception at a given operating state, using the pressure regulator (see Fig. I.38). The pressure regulator was then isolated from the pressurizer plenum, and the pressure lowered by bleeding small quantities of liquid from the plenum back to the sump tank. At each state of cavitation the data logger was actuated to scan the tunnel transducers and record the required state variables defined in Fig. II.6.

Fig. II.20 shows plots of the cylinder base pressure coefficient, or cavitation number based on cavity pressure versus cavitation number based on vapour pressure, from the fully wetted to the fully cavitating flow conditions. Each symbol denotes a data point at which the tunnel transducers were scanned to obtain the required state variables. For values of σ_v greater than σ_i corresponding to the given inception type, the base pressure coefficients had the fully wetted values at the appropriate Reynolds numbers, as may be seen from Fig. II.20 and Fig. II.10. With wake type inception ($R_d = 10^5$), visible cavitation occurred within the flow field at $\sigma_v = \sigma_{iw} = 2.2$, at a corresponding value of C_{pb} of -0.95. As σ_v was decreased, C_{pb} decreased slightly to a value of -1.0.

Further reduction of σ_v caused C_{pb} to increase smoothly towards the choked condition ($\sigma_c = -C_{pb} \approx 0.7$). For the band and spot type incipient states, inception occurred at values of σ_v of 2.75 and 2.3 respectively, at corresponding values of C_{pb} of -0.27 and -0.45. A reduction in σ_v below the incipient values caused an initial decrease in C_{pb} in both cases. At a critical value of σ_v ($\sigma_v = 2$ in Fig. II.20), the base pressure coefficient C_{pb} decreased abruptly to a value of -1.65, thereafter increasing smoothly towards the choked condition. The critical value of σ_v at which C_{pb} increased abruptly, σ_{CR} , was the same for the development of cavitation from both the band and spot type incipient states.

Figures II.22, II.23 and II.24 show photographs of the development of cavitation from the wake, band and spot type incipient states respectively. Fig. II.21 is a key to these aforementioned photographic data and relates the visible appearance of the cavitation to the behaviour of C_{pb} and σ_v . Frames 1 to 6 in Fig. II.22 show the development of cavitation from wake type inception, with σ_v decreasing sequentially through the frames. Frame 1 shows an overhead view of the detached wake type inception. At reduced σ_v , the extent of the detached cavitated region increased in an upstream direction, as shown in Frames 2 and 3. At a value of σ_v of about 1.5, the cavitation became attached to the cylinder, as shown in Frame 4, the cavitation separation being located at approximately 90 degrees from the frontal stagnation point of the cylinder. Further increase in σ_v caused the cavitation to extend in the downstream direction and the cavitation separation point to move upstream to attain a value of 80 degrees at the choked condition, as shown in Frames 5 and 6. The

end view photographs show that once an attached cavity was established, the cavitating wake was broader than the cylinder diameter, the apparent free-streamlines of the flow being convex to the stream direction. Frames 8 to 11 in Fig. II.23 show cavitation development from the incipient band state. Frame 7 shows band type inception with the cavitation band located at 110 degrees from the frontal stagnation point. In contrast to the wake type development, the cavitation was always attached to, or very close to the cylinder surface for the band type development. The initial decrease in C_{pb} , for values of σ_v between σ_{ib} and σ_{CR} , were observed to correspond to an upstream displacement of the cavitation separation location, from 110 degrees at inception to a value somewhat greater than 90 degrees at σ_v just greater than σ_{CR} . Frame 8 shows an end view, and Frame 8a an overhead view of the cavitating flow found in this phase of the development. It may be seen in Frame 8, that the cavitating wake is narrower than the cylinder diameter, the apparent free-streamlines of the flow being concave to the stream direction in the vicinity of cavitation separation. Figure 8a shows that at a value of σ_v of 2.1 the cavitation separation angle had moved upstream to an angular position somewhat greater than 90 degrees, from the incipient band location of 110 degrees. Upon attainment of the critical cavitation number σ_{CR} , the flow became unstable, changing randomly between two quasi-static states. One of these was the aforementioned concave free-streamline type cavity shown in Frame 8, and the other a flow with the cavity broader than the cylinder diameter and the apparent free-streamlines convex to the stream direction. The form of this latter flow is shown in

Frame 10. This instability was attributed to the abrupt decrease in C_{pb} , and hence increase in cylinder drag, in the region of σ_{CR} . Frame 9 shows a typical cavity found in the unstable flow regime. At values of σ_v less than σ_{CR} , the convex free-streamline cavities grew longer and broader as σ_v approached σ_{CH} , the cavitation separation angle moving upstream to attain a value of 82.5 degrees in the choked condition. Frame 11 shows an end view of a cavity at a value of σ_v just above σ_{CH} .

With the development from spot type inception, the shape of the cavities in the developed cavitation region ($\sigma_{CH} < \sigma_v < \sigma_{CR}$) were indistinguishable from those of the band type development. Frames 12 to 16 in Fig. II.24 show overhead views of the spot type development at a Reynolds number of 5.1×10^5 . Frame 12 shows the incipient condition with one cavitation spot at the mid-span location and two adjacent spots near the span-end location. At values of σ_v just below inception, these spots grew in the downstream direction and cavitation appeared at the other end of the span as shown in Frame 13. Further reduction in σ_v caused the cavitation to extend in both the spanwise and streamwise direction. Frame 14 shows this condition for σ_v slightly greater than σ_{CR} , with fully wetted areas remaining between the cavitated regions on the cylinder surface. At the critical cavitation number, the flow was again unstable, alternating between a concave free-streamline flow at σ_v greater than σ_{CR} and a convex free-streamline flow for σ_v less than σ_{CR} . In the developed cavitation flow regime, cavitation separation occurred continuously across the cylinder span, and moved upstream with decreasing σ_v to attain a value of 80 degrees at the choked condition.

In the developed cavitation flow regime, all the cavitation separations observed in the experiments, though occurring continuously across the cylinder span, exhibited small scale spanwise variations in separation angle. These variations can be seen in Fig. II.24, Frame 16, and they persisted down to the choked condition. These variations, which were not investigated in the present experiments, imply small scale three-dimensionalities in the flow in the region of cavitation separation. One source of such small scale effects is the generation of Görtler vortices in the regions of streamline concavity near the frontal stagnation point of a cylinder. Kestin and Wood^[129] analysed the stability of the linearised two-dimensional boundary layer equations to three-dimensional upstream disturbances, verifying the formation of Görtler vortices in the upstream stagnation region and their subsequent growth in the downstream direction. Their analysis showed the vortex pairs to have a characteristic spanwise wavelength Z , and predicted a relationship between Z , the cylinder diameter D and Reynolds number R_d which was,

$$Z = 1.79(\pi.D)/(R_d)^{\frac{1}{2}} \quad \text{II.3}$$

The existence of the predicted flow pattern was verified by wind tunnel experiments using an oil film visualization technique (Fig. 6, Reference 129). At the Reynolds number corresponding to Fig. II.24 ($R_d = 5.1 \times 10^5$), Equ. II.3 predicts a wavelength Z of 0.8 mm which is of the same order as that observed in the present experiments. This indicates that the spanwise variations observed in the developed cavitation flow regime may be caused by Görtler vortices, but this cannot be taken fur-

ther without additional experimental work.

The choked condition was designated as

$\sigma_{CH} = -C_{pb} = -C_{wMIN} = 0.7$. In theory, in the choked condition, the free-streamlines bounding the cavity should become parallel to the test section walls confining the flow. In this condition the cavity pressure should be the same as the static pressure of the liquid flow outside the free-streamlines, or in non-dimensional form $\sigma_c = -C_{wMIN}$.

It was not possible to achieve this condition experimentally.

Fig. II.25 shows a plot of minimum wall pressure coefficient

C_{wMIN} versus cavitation number σ_v for Reynolds numbers corresponding to the incipient wake, band and spot states.

The data form two distinct groups, band and spot in one and wake type development in the other. At low values of

σ_v both curves tend towards the value $\sigma_v = -C_{wMIN} = 0.7$,

but none of the data attain this value. This failure to

achieve the theoretical choked flow condition may be

caused by the tunnel diffuser, the entrance of which is

located some 20 cylinder diameters downstream of the

cylinder axis. The adverse pressure gradient in the

diffuser would tend to inhibit cavity growth beyond its

inlet. A further constraint on the low cavitation number

performance of the tunnel was the main circulating pump,

which itself started to cavitate near the choked test

section condition. On the basis of the asymptotic

behaviour of Figs. II.20 and II.25, the nominal choked

condition for the 9.5 mm diameter cylinder was taken to be

$$\sigma_c \approx -C_{wMIN} \approx 0.7.$$

II.6.2 DISCUSSION OF OBSERVED FORMS OF CAVITATION DEVELOPMENT

The form of the C_{pb} versus σ_v plots shown in Fig.

II.20 is similar to the C_D versus σ_v data of Martyrer

[120], Silberman^[67] and Varga and Sebestyn^[45] in the limited cavitation flow regime. This is a reflection of the fact that for fully wetted flow about a circular cylinder, the drag is largely determined by the base pressure^[79]. From the results of the present experiments the difference between the behaviour of the base pressure coefficients during the development of cavitation at subcritical and supercritical Reynolds numbers may be explained on a simplified basis by the relative movement of the free-streamlines of the flows in question. Fig. II.26 shows a schematic of the free-streamlines of a two-dimensional flow about a circular cylinder for fully wetted and cavitating flows at subcritical and supercritical Reynolds numbers. In the fully wetted flow condition, the free-streamline detachment points correspond to the boundary layer separation points. For subcritical Reynolds numbers, laminar boundary layer separation occurs on the frontal portion of the cylinder, at supercritical Reynolds numbers, turbulent boundary layer separation occurs well back on the rear face of the cylinder. At transcritical Reynolds numbers the turbulent boundary layer separation moves in the upstream direction with increasing R_d , but for the range of R_d covered in the present experiments it may be assumed that it always occurs on the rear face of the cylinder, as evidenced by the fully wetted behaviour of the base pressure coefficient in Figs. II.9 and II.10. In the developed cavitation flow regime, the free-streamline (cavity detachment) angles were found to be in the range 80 to 90 degrees. Thus for subcritical R_d , on the basis of Fig. II.26, the movement of the free-streamlines between the fully wetted and fully developed cavitating states is minimal. As previously noted, the detached

nature of the wake type cavitation persisted for values of σ_v considerably below inception and it appeared that this phase of development consisted of the wake gradually filling with vapour as σ_v was reduced. This interpretation is compatible with the behaviour of C_{pb} in this region since it remained substantially constant as σ_v was reduced from inception to a value of about 1.5. It is interesting to note that Silberman^[67], in experiments on a cylinder in a free-jet at subcritical Reynolds numbers, termed this phase of development as "bubble cavitation" as opposed to the developed cavities with well defined separations found at lower cavitation numbers.

The band type inception occurred at 110 degrees from the frontal stagnation point of the cylinder. In the limited cavitation regime, the point of cavitation separation was observed to move in the upstream direction, this displacement corresponding to the initial decrease in C_{pb} as σ_v was reduced below the inception value (see Fig. II.20). In this condition the apparent free-streamlines of the flow, as visualized by the cavitation, were concave to the stream direction, whereas in the developed cavitation regime they were convex. It thus appears that the abrupt decrease in base pressure coefficient which occurs at the critical cavitation number σ_{CR} is associated with the free streamlines of the flow changing from a convex to a concave configuration. This is a very simplified physical model of the flow behaviour since there is clearly a complex interaction between the turbulent boundary layer and the limited cavitation. In fully wetted and developed cavity flows, the free-streamline separation points are the boundary layer and cavity

detachment points respectively. It should be emphasized that in the limited cavitation regime, the classical concept of a free-streamline may not be applicable. The term is presently used to denote a flow pattern inferred from the cylinder base pressure.

The development from spot type inception was similar to the band type in respect of the behaviour of C_{pb} , except in the limited cavitation regime, where the variations in C_{pb} were much smaller. Whereas the band type inception was highly two-dimensional and occurred at an angle of 110 degrees, the spot type was three-dimensional and occurred nearer the 90 degree location. The initial development of the spot type cavitation consisted of achieving spanwise continuity of the cavitation separation without appreciable change in the separation angle. In the developed cavitation region the C_{pb} versus σ_v plots of the spot and band type development were virtually coincident and both diverged from the line $\sigma_v = \sigma_c$ at increasing values of σ_v above the choked condition, as shown in Fig. II.20. The plots are quite different from the wake type development, where the C_{pb} versus σ_v curve showed a much greater divergence from the line $\sigma_v = \sigma_c = -C_{pb}$. In all cases this divergence indicates that the cavity pressure P_c exceeded the vapour pressure P_v , for σ_v greater than the choked value. It should be noted, however that the measured values of cavity pressure used in the computation of C_{pb} were some time-mean average of the fluctuating cavity pressure. As previously mentioned in Chapter I.6, it is well known that vortex shedding may persist in the cavitating wakes behind circular cylinders down to very low cavitation numbers. In a theoretical potential cavity flow, at low

values of reduced onset flow temperature (relative to the critical temperature) and foreign gas content, the flow would be characterized by $\sigma_v = \sigma_c$, since $P_c = P_v$. It is probable that the wake dynamics caused the divergence from the line $\sigma_v = \sigma_c$ in the developed cavitation regime. This is indicated by the fact that near the choked flow state, σ_v tended towards σ_c (or $-C_{pb}$), in which condition it is reasonable to assume that a cavitating vortex street at higher values of σ_v had changed to a fully developed cavity with a re-entrant jet type closure. This view is reinforced by a recent study by Mussared^[49] of the dynamics of cavitating wakes behind a circular cylinder in the same facility used in the present experiments. The influence of wake dynamics is also indicated by the drag coefficient data of Waid^[66] and Ewers^[48], both of whom measured significantly lower drag coefficients for circular cylinders at low cavitation numbers than were predicted by planar flow theory.

The observed flow instability and abrupt change in C_{pb} at the critical cavitation number may be further explained on a simplified basis. For a convex free-streamline cavity flow to change steadily to a concave free-streamline form would require the radius of curvature of the free-streamlines to become infinite at some point in the transition.

Theoretically this condition is only satisfied in the choked flow state and since σ_{CR} is much greater than σ_{CH} , no such smooth flow transition is possible. The critical cavitation number was found to be independent of Reynolds number in the present experiments, for Reynolds numbers greater than the critical value. It would undoubtedly be influenced by a change in cylinder

blockage ratio, which has a direct influence on the local flow velocity around a cylinder.

Some further details of the cavitation development experiments described in this chapter are given in Ref. 127.

CHAPTER II.7

CLOSURE TO PART II

II.7.1 SUMMARY OF OBSERVATIONS ON CAVITATION INCEPTION AND DEVELOPMENT ON A CIRCULAR CYLINDER

Three forms of cavitation inception were found on the two-dimensional circular cylinder in the Reynolds number range $10^5 < R_d < 8.5 \times 10^5$. The Reynolds number ranges in which these three forms occurred were found to correspond broadly to those delineating the changes from subcritical to transcritical fully wetted flow over the cylinder. The band and spot types of cavitation inception can with some confidence be attributed to pressure fluctuations in the region of boundary layer turbulent transition. Further experiments are required to determine the nature of wake type inception.

From the studies of the development of cavitation from the three incipient states, the following conclusions were reached.

1. The detached form of wake inception persisted down to cavitation numbers well below the inception value. The behaviour of the cylinder base pressure indicated minimal displacement of the free-streamlines between the fully wetted and fully cavitating states.

2. With development from band and spot type inception, a critical cavitation number (σ_{CR}) was found at which the apparent free-streamlines of the flow changed from a concave to a convex configuration, with an accompanying large decrease in cylinder base pressure.

3. The value of σ_{CR} was 2 for the experimental

configuration used and did not vary with Reynolds number.

4. For the initial phase of development from band type inception, the cavitation separation remained largely two-dimensional and moved upstream.

5. The initial phase of development from spot type inception consisted of the cavitation separation achieving spanwise continuity, with little displacement of the separation location.

P A R T III

SUGGESTIONS FOR FURTHER RESEARCH IN AND
MODIFICATIONS TO THE HIGH TEMPERATURE CAVITATION TUNNEL

CHAPTER III.1

SUGGESTIONS FOR FUTURE RESEARCH

The new observations on cavitation inception and development on a circular cylinder which have been reported in Part II demonstrate the strong, almost dominant influence of viscous effects on cavitation on well rounded two-dimensional bluff bodies. It is clear that there are many problem areas in cavitating cylinder flows which require attention. On the basis of the present experiments some of these may be identified, and they are given below.

III.1.1 BAND AND SPOT TYPE INCEPTION

The band and spot type incipient cavitation observed on the cylinder have been related to boundary layer turbulent transition, on the basis of their macroscopic appearance and the Reynolds number range and the location of their occurrence, their appearance being determined from still photographs. In order to gain further understanding of the interaction between the cavitation and its bounding liquid flow at inception, a high speed photographic study should be performed, in a similar manner to that of Blake et al. on a two-dimensional hydrofoil^[93]. This could yield further much-needed information on the bubble dynamics of cavitation inception. In this context, it might be worth investigating the possibility of designing an experiment capable of studying inception under a variety of boundary layer conditions. This might be done with some form of experiment similar to Gaster's flat plate and auxiliary aerofoil^[83], or alternatively using a

test body with provision for boundary layer suction, as proposed by Prasad^[121]. It would certainly be worth fitting the cylinder used in the present experiments with boundary layer trips to ascertain if these would suppress cavitation inception completely, as found by Arakeri and Acosta^[92], or cause the cavitation bubble type inception found by Blake et al.^[93]. If this were done, fully wetted experiments would have to be performed to ensure that the trips completely eliminated laminar separation, and that turbulent transition occurred well upstream of the low pressure region of the cylinder.

III.1.2 WAKE TYPE INCEPTION

As previously noted in Section II.5.2, further experiments are required to clarify the causes of wake inception. Gaster^[113] has suggested that fully wetted measurements be made of the fluctuating pressure in the region of the cylinder wake where inception was observed. This would be difficult to do using the High Temperature Cavitation Tunnel on account of the very small pressure transducer and an associated support system that would be required because of the small test section and model dimensions. Such measurements could be obtained more easily in a wind tunnel, in which a large cylinder could be used to minimise the influence of the pressure transducer and its support on the flow field in the cylinder wake. Such a cylinder could be enclosed in a rectangular section duct so dimensioned as to give the same blockage and aspect ratios as those used in the present experiments. As with the band and spot types of inception, a high-speed photographic study should be made of the wake type inception.

A further experiment, which would not require any elaborate modification of the cylinder or test section, could be the investigation of cavitation inception at subcritical Reynolds numbers with a splitter plate downstream of the cylinder. A splitter plate of the appropriate streamwise length^[122] would suppress fully wetted vortex shedding completely and it would be interesting to ascertain the type of cavitation inception thus found.

III.1.3 VISCOUS EFFECTS ON CAVITATION ON AXISYMMETRIC WELL-ROUNDED BLUFF BODIES

As previously noted in Chapter II.3 the fully wetted flow around a sphere is very similar to that around a cylinder in the region of the critical Reynolds number. Both these bodies exhibit a laminar separation bubble flow regime^[72,105] and vortex shedding also occurs from spheres at subcritical Reynolds numbers^[123]. It would be interesting to instal a sphere in the High Temperature Cavitation Tunnel to confirm that the strong viscous effects on cavitation inception found with a cylinder also occur on a sphere. A sphere 7 mm in diameter would occupy only 1% of the tunnel test section flow area.

According to the data for spheres in turbulent wind-streams presented by Dryden et al.^[124], the critical Reynolds number for a sphere in a stream with a turbulence level of 1% (the estimated value for the High Temperature Cavitation Tunnel) is about $R_{CR} = 2.15 \times 10^5$. Thus, subcritical, supercritical and transcritical Reynolds number flow conditions could be attained by adjusting the tunnel flow velocity and temperature over the range shown in Fig. II.8. A sphere could be sting-mounted from supports built into the packer plate at the test section

outlet location. If the sting were made hollow, it could be used to transmit the sphere base pressure to a pressure transducer. Because of the aforementioned similarities in the fully wetted flow around cylinders and spheres, the development of cavitation on the latter body should be similar to that on the former. Experiments with a sphere could thus confirm the strong viscous effects on both the inception and the development of cavitation found in the present experiments.

III.1.4 THE LIMITATION OF THE GAS CONTENT CONTROL PROCEDURE

Because of the wide range of Reynolds numbers which may be attained in its test section, the High Temperature Cavitation Tunnel should be suitable for studying a variety of both single and two-phase flow phenomena. Its main limitation is that it does not have a resorber to allow test section flows with low free gas content at variable total gas content. The experiments reported in Part II of this thesis were all performed at low values of gas content (typically 10 parts per million as measured with a Van Styke apparatus). Flows with a high free gas content may easily be generated in the tunnel by allowing nitrogen pressurization gas to enter the tunnel fluid. In principle, the free gas content of the tunnel could be varied by adding known quantities of gas to the working fluid. This was not attempted during the experimental program reported herein.

CHAPTER III.2

SUGGESTED MODIFICATIONS TO THE HIGH TEMPERATURE CAVITATION TUNNEL

This chapter suggests some modifications which would make the High Temperature Cavitation Tunnel safer and easier to use, and some of the data that should be obtained to define the quality of the test section flow.

III.2.1 TEST SECTION FLOW QUALITY

The quality of the test section flow has not, to date, been specifically investigated since the viscous effects on cavitation reported in Part II of this thesis were observed during the tunnel proving trials and it was thought best to investigate them rather than delay the research program in order to design and manufacture the equipment necessary to measure test section velocity distributions and turbulence levels.

The fully wetted experiments described in Section II.4.2 revealed a cylinder critical Reynolds number of $R_{CR} = 2.3 \times 10^5$, from which the r.m.s. free stream turbulence level was estimated to be about 1%. This is considerably higher than the value that the screens had been designed to produce, but is probably lower than that found in many cavitation tunnels. The turbulence level estimated from the present experiments is the same as that reported for the St. Anthony's Falls free-jet water tunnel [125] which, being of the open-circuit type without a circulating pump should have a less disturbed flow than a forced circulation type tunnel. It would clearly be more scientific

to directly measure the turbulence level in the test section, rather than infer it from base pressure measurements such as was done in the present case. This could be done using hot-wire anemometry or possibly with a laser anemometer.

The variation of the mean flow velocity in the empty test section was not measured during the present investigations. On the basis of the two-dimensionality of the band type cavitation inception and the cavitation separations of developed cavities, it appears that there were no gross irregularities in the test section flow. The measurement of test section mean velocity profiles would be a very useful exercise. If this were done using a pitot tube, two of the test section windows would have to be replaced with specially designed blank windows to accommodate traversing pitot tubes. This could be avoided if the velocity profiles could be obtained using laser anemometry through the existing test section glass windows.

An investigation should also be made of the acoustic characteristics of the empty test section. The tunnel environment is very noisy in the audible range, on account of the hydrostatic drive. Whether or not this would prevent the application of the tunnel to the study of acoustic problems would depend on the relative acoustic power spectra of the tunnel and the particular phenomena to be investigated. As with the test section turbulence level, the acoustic characteristics could be expected to vary with tunnel flow velocity.

III.2.2 PREVENTION OF SUMP TANK PRESSURIZATION

As previously mentioned in Section I.5.3 the sump tank must only be isolated from the atmosphere when running the tunnel under vacuum conditions. Valve 11 in Fig. I.38 must always be fully open except when running the tunnel under vacuum conditions, since the sump tank will not withstand pressures in excess of about 1 bar. If valve 11 is inadvertently left closed under pressurized tunnel conditions a potentially lethal accident could occur. It is thus suggested that valve 11 in Fig. I.38 be replaced with an electrically actuated solenoid valve of the spring-to-open type. This should be electrically connected in parallel with the vacuum pump motor so that the solenoid valve can only close when the vacuum pump starts. The vacuum pump is currently actuated by means of a 13 amp plug and socket adjacent to the tunnel working area. The vacuum pump and solenoid valve electrical connections should be wired into the tunnel control console and a single switch provided to energise the required circuits. As a precautionary measure the solenoid valve should be provided with a bypass pipe housing a spring-loaded safety valve set to lift at 0.25 bar.

III.2.3 PREVENTION OF SUMP TANK OVERHEATING

The problem of overheating of the sump tank during degassing operations could be solved by immersing a copper cooling coil in the sump tank liquid. Provision would have to be made for the inflow and outflow pipes of the cooling coil to pass through the sump tank walls. The cooling coil could be connected into the circuit which connects the external cooling tower to the tunnel heat

exchanger (see Fig. 1.37).

III.2.4 TUNNEL HEAT LOSS AND HEAT-UP RATE

The tunnel proving trials showed that the present heat input and tunnel thermal insulation are not sufficient to allow high tunnel temperatures to be attained in a reasonable time. Two approaches can be made to this problem. Further lagging material can be wrapped around the tunnel structure and the heat supplied to the tunnel can be increased. Doing both these things would decrease the energy required to heat up the tunnel and increase the rate at which tunnel temperature may be changed. The heat supply rate to the tunnel could be more than doubled by using the vapour ventilation system as a steam injector. In its present configuration the volumetric vapour flow-rate that can be supplied to the tunnel is limited by the maximum allowable rotational speed of the turbine flowmeter measuring the ventilation flow (approximately 600 Hz). If some additional pipes and valves were added to the ventilation supply line the flowmeter could be bypassed as required and up to 30 KW of heat supplied to the tunnel working fluid. The total heat supply rate to the tunnel, using the existing heating tapes together with steam injection would be about 50 KW. Steam injection would also be more effective than the heating tapes on account of the direct contact between steam and working fluid. If the tunnel is used for studying test bodies without provision for the admission of a ventilation flow, additional modifications to the ventilation system pipework would have to be made in order to inject steam at some other point in the tunnel circuit.

III.2.5 TUNNEL DRAINING AND FILLING

The present procedure for draining and filling the tunnel is not satisfactory. The tunnel fluid is presently drained into a number of 45 litre plastic drums, from the tunnel low-point at the bottom of the circulating pump volute. The tunnel is filled by connecting the discharge of a small centrifugal pump to the tunnel low point, the pump section being immersed in the fluid in the 45 litre plastic drums. Draining and filling is thus a tedious and time consuming process. A big improvement on these arrangements would be the provision of a plastic dump tank. Because of the lack of space on the 1st floor of the Cavitation Laboratory, this dump tank would be best located on the ground floor adjacent to the hydrostatic drive. The tunnel could then be drained by gravity. The existing Stuart Turner centrifugal pump could be permanently plumbed in to refill the tunnel from the dump tank, its electrical connections being wired in to the tunnel control console so that it could be started from the tunnel working area. If this modification is effected, it would certainly be worth considering installing a resin-exchange type de-ionizer in the filling line between the dump tank and the tunnel.

III.2.6 HYDRAULIC COOLING AND VENT FLOW CONTROL VALVES

In Section I.5.1 it was emphasized that the valves V1 and V3 shown in Fig. I.35, which are situated in the low pressure part of the hydraulic pump and motor cooling and leakage flow vent line, should never be completely closed. This is because one of their functions is to allow the hydraulic pump and motor internal leakage flows

to be vented back to the oil reservoirs. If valves V1 and V3 are closed, failure of motor and pump components may occur. It is recommended that the valve plugs of these valves have holes drilled through them to communicate the upstream and downstream faces of the valve plugs. This would ensure that even if they were fully closed, the casings of the hydraulic pump and motors would always be vented to the oil reservoirs.

III.2.7 SPEED CONTROLLER CHANGEOVER SWITCH

As described in Section I.5.1, the electronic control unit of the hydrostatic drive serves both the Cold Water Cavitation Tunnel and the High Temperature Cavitation Tunnel. It is connected to the appropriate tacho-generator and velocity setpoint potentiometer through a changeover switch located on the control console of the Cold Water Cavitation Tunnel. This switch is currently a manually operated two-position switch labelled "small" and "large" relating to the High Temperature and Cold Water cavitation tunnels respectively. This arrangement is not ideal since if the switch position is inadvertently changed while either of the tunnels is running the hydrostatic drive might go out of control. A better arrangement would be the replacement of the present switch with a key-operated one. The key could be removed when a changeover had been made. Further if a multi-pole switch was used, the electrical power supply to the tunnel control consoles could be connected through the switch such that the consoles would only be energized when the switch was in the appropriate position. It would then be difficult to accidentally change the switch position with one of the tunnels running.

APPENDIX A.I.1

TUNNEL OPERATING PROCEDURES

The following Appendix describes the operating procedures for the High Temperature Cavitation Tunnel and its ancilliary systems in its present configuration. Words written in capital letters which appear in the following instructions denote titles or descriptions which are labelled on items of equipment used in conjunction with the tunnel. This is done to clarify the operating instructions and facilitate the location of various items by a potential user not conversant with the tunnel, while reading these instructions. As much detail as possible is given in the following procedures in order to illustrate the effects on the tunnel of implementing each stage in the procedure. The procedures apply only with water as the working fluid and if other working fluids are used in the tunnel, modifications will have to be made both to the tunnel itself and its operation. For brevity, the High Temperature Cavitation Tunnel is referred to as the H.T.C.T. and the Cold Water Cavitation Tunnel as the C.W.C.T. The pressure transducers required for a given experiment should be installed in the tunnel when the working fluid is cool since transducer changeover and installation must be done with the tunnel slightly pressurized (to about 0.5 bar) to prevent the ingress of air into the working fluid.

A.I.1.1 DE-GASSING

If the working fluid gas content requires reducing, the following procedure is used

1. Close valves 2, 3, 1, 9 and 10. (Fig. I.38).
2. Open valves 4, 5, 6 and 7. (Fig. I.38).
3. Open valve 11. (Fig. I.38). Valve 11 must be fully open at all times during the de-gassing procedure to prevent pressurization of the sump tank.

4. Set the temperature setpoint thermostat to 110° C (Fig. I.37).

5. Adjust the air pressure regulator on the Madan pump to give a pump discharge pressure of 80 p.s.i.

6. Open valve 1 (Fig. I.38) to allow fluid to circulate through the sump tank.

7. Switch on the heating tapes at the control box labelled HEATING TAPES on the east wall of the 1st floor Cavitation Laboratory.

8. Allow tunnel fluid to boil for a sufficient period to reduce the gas content to the experimentally required value. The gas content can be estimated using the Van Slyke apparatus adjacent to the tunnel. A fluid sample can be extracted from the tunnel through valve 13 (Fig. I.38). A 50 cc sample is usually used. Boiling the tunnel contents for two hours reduces the gas content to about 10 parts per million gravimetric, as measured with the Van Slyke apparatus.

A.I.1.2 PRELIMINARY HEATING TO EXPERIMENTAL TEMPERATURE

The tunnel temperature can be increased while the tunnel is running as will be described in Section A.I.1.8.

If the working fluid does not require de-gassing prior to an experimental run at elevated temperature, it is best to heat up the tunnel from low temperatures without the hydrostatic drive running, since this can be done without operator attendance. To do this the same procedure is followed as with de-gassing the working fluid, except that valve 1 (Fig. I.38) is closed and the temperature setpoint thermostat (Fig. I.37) is set to the required experimental temperature instead of 110° as when de-gassing. If this procedure is used to heat up the tunnel from cold, the thermostat must never be set at a value greater than 95° C if the tunnel is to be left unattended.

A.I.1.3 ELECTRIC POWER SUPPLIES

The heavy-current electrical supply to the 75 KW electric motor of the hydrostatic drive is shared with other users and has to be connected to the Cavitation Laboratory system by means of changeover links in a control box situated outside the east wall of the Cavitation Laboratory. Access to the control box is gained through a gate in the wire-mesh fence adjacent to the external cooling tower. A key to the lock on this gate is on the keyboard in the Cavitation Laboratory. Underneath the control box containing the changeover links is the main power circuit breaker. This must be switched off before removing the front panel of the control box. The three link-bars are changed over by undoing their securing nuts and connecting them to the three bus-bars labelled CAVITATION TUNNEL. Ensure that the nuts holding the links in position are screwed tightly, otherwise arcing may occur and damage the links. Switch on the main circuit breaker after

replacing the front panel of the control box. On the ground floor of the Cavitation Laboratory, switch on the circuit breaker labelled NEW CAVITATION TUNNEL and also the circuit breaker mounted on the side of the MAIN PUMP MOTOR STARTER. If the ventilation system is required switch on the circuit breaker labelled BBC 5.

A.I.1.4 HYDROSTATIC DRIVE

The hydrostatic drive is located on the ground floor of the Cavitation Laboratory and the following actions must be taken at this location to prepare the hydrostatic drive for start-up.

1. Ensure that valves RH and FH are fully open and that valves RC and FC are fully closed (Fig. I.35).
2. Valves V1, V2, V3 and V4 must be fully open when oil is diverted to the hydraulic motor driving the circulating pump of the H.T.C.T.
3. Open valve VC (Fig. I.35) to allow cooling water to flow through the auxilliary hydraulics cooler.

A.I.1.5 START-UP PROCEDURE

1. Switch on the tunnel instrumentation at the MAINS switches on the H.T.C.T. control console.
2. Ensure that ice is present in the flask comprising the reference temperature for the ice-point thermocouple.
3. Set the velocity setpoint potentiometer labelled SET SPEED, on the H.T.C.T. control console, to zero.

The following actions are required at the control console of the C.W.C.T.

4. Set the SMALL-LARGE changeover switch to SMALL.

5. Turn on the key-operated MASTER SWITCH.

6. Press the AUX. HYDR. switch.

7. Press home the red sampling switch to read the BOOST PUMP DELIVERY PRESSURE gauge. This must indicate a pressure of at least 800 p.s.i. If it does not, check the auxilliary hydraulic filters. (See Appendix A.I.3).

8. Press home the red sampling switch to read the SERVO PUMP DELIVERY PRESSURE. This must be greater than 150 p.s.i. If it is not, check the auxilliary hydraulics filters (see Appendix A.I.3).

9. Turn on the MAINS switch at the base of the console.

10. Press the INST. PWR. switch.

11. Press the MAIN HYDR. switch. No further action is required at the control console of the C.W.C.T.

12. Check that valve 11 is fully open and valves 9 and 10 are fully closed (Fig. I.38).

13. Set the pressure regulator valve to give a pressure of 3 bar and open valve 8.

14. When the plenum pressure reaches 3 bar, as indicated on the Bourdon gauge on the top of the plenum, close valve 8 (Fig. I.38).

15. Close valves 1, 3, 4, 5, 6 and 7 and set the air pressure regulator on the Madan pump to give a pump discharge pressure of 80 p.s.i.

16. Open valve 2 (Fig. I.38) to pressurize the tunnel from the plenum. If this is done immediately after de-gassing, the liquid level in the plenum will fall as the steam voids in the tunnel collapse. Open valve 1 (Fig. I.38) to admit fluid from the Madan pump to restore the plenum level. The tunnel is now ready to run.

17. Adjust the tunnel speed to the required value by slowly turning the SET SPEED potentiometer. The tunnel speed can be monitored on the digital display of the main flowmeter frequency, labelled CIRCULATION FLOW.

A.I.1.6 TUNNEL PRESSURE ADJUSTMENT

A variety of methods can be used to adjust the tunnel static pressure, depending on the requirements of a given experiment. The following procedures have been found to give satisfactory pressure regulation. Coarse pressure adjustment is performed thus

1. Set the pressure regulator valve to the value required.
2. Open valve 8 (Fig. I.38) to connect the pressure regulator with the plenum.

For fine pressure adjustments

3. Close valve 8 (Fig. I.38).
4. To reduce pressure, bleed fluid from the plenum to the sump tank using valve 3 (Fig. I.38).
5. To increase pressure, set the pressure regulator to give a value greater than that required and admit small quantities of gas to the plenum using valve 8 (Fig. I.38). Alternatively, increase pressure by admitting small quantities of liquid to the plenum using valve 1 (Fig. I.38). In this case the discharge pressure of the Madan pump must be greater than the plenum pressure.

A.I.1.7 OPERATION UNDER VACUUM CONDITIONS

Tunnel operation under vacuum conditions is usually only required when the tunnel pressure has been reduced to atmospheric pressure using one of the methods given in

Section A.I.1.6 . Starting at this situation, vacuum conditions are obtained as follows.

1. Start the vacuum pump, open valve 10 and close valves 11 and 8, in that order (Fig. I.38). The plenum pressure can then be lowered by,
2. Withdrawing small amounts of gas from the plenum using valve 9 (Fig. I.38), or alternatively,
3. Withdrawing small amounts of liquid from the plenum using valve 3 (Fig. I.38). This has proved the most effective method in practice.

A.I.1.8 INCREASING TUNNEL TEMPERATURE

Tunnel temperature is increased by using the heater tapes, the controls for which are located on the laboratory east wall (Fig. I.37) and are labelled HEATING TAPES. If the required temperature is less than 100° C, the procedure used is that given in Section A.I.1.2. If the required temperature is greater than 100° C the tunnel pressure must be controlled from the pressure regulator valve on the nitrogen cylinder (Fig. I.38), and the circulating pump must be running to prevent leakage through its mechanical shaft seal. Operator attendance is required in order to bleed fluid from the tunnel as it expands and raises the plenum level. The plenum fluid level is maintained using valve 3 (Fig. I.38).

A.I.1.9 REDUCING TUNNEL TEMPERATURE

The tunnel is cooled using the heat exchanger shown in Figs. I.37 and I.38. When running the tunnel bypass valve BV1 is normally fully open and BV2 fully closed (Fig. I.38). To abstract heat from the tunnel

1. Switch on the cooling tower pump and fan at the switch on the laboratory east wall (Fig. I.37). Check the indicator lamps adjacent to the switch to ensure that both the pump and fan are running.

2. Open the heat exchanger coolant flow control valves C1 and C2 (Fig. I.38).

3. Divert working fluid to the heat exchanger by opening valve BV2 (Fig. I.38).

4. If, and only if, insufficient heat can be abstracted from the tunnel with valve BV2 fully open, start closing valve BV1. With valve BV2 fully open and BV1 fully closed the tunnel will attain its lowest possible working temperature. It is most important that one of the valves BV1 or BV2 remain fully open at all times.

A.I.1.10 VAPOUR VENTILATION SYSTEM

The vapour ventilation system requires further development to overcome the operational difficulties described in Section I.7.3. The method of operation used thus far is as follows.

1. At the H.T.C.T. control console, set the evaporator power setpoint potentiometer labelled SET EVAPORATOR POWER to zero.

2. Open the vapour ventilation throttle valve (Fig. I.46).

3. At the ground floor of the Cavitation Laboratory, switch on the circuit breaker labelled BBC 5 and switch on the CONTACTOR on the front panel of the 40 KW power supply.

4. At the H.T.C.T. control console set EVAPORATOR POWER to give a reading of 5 KW on the adjacent analogue

meter labelled EVAPORATOR POWER, and heat up the evaporator to the desired tunnel temperature. If this temperature is greater than 100° C the circulating pump must be running and the test section velocity should be set at 5 m/sec when heating up the evaporator.

5. When vapour generation commences at the prevailing tunnel pressure, set the EVAPORATOR POWER potentiometer to zero and completely close the vapour ventilation throttle valve (Fig. I.46).

6. Increase the test section velocity to the desired value.

7. Slowly open the ventilation throttle valve (Fig. I.46) ensuring that the ventilation flowmeter does not exceed a rotational speed of 600 HZ. This can be read on the digital display labelled VENTILATION FLOW on the H.T.C.T. control console.

8. When the ventilation flow throttle valve is fully open, establish the required ventilation flow by slowly increasing the evaporator power input using the SET EVAPORATOR POWER potentiometer.

A.I.1.11 TUNNEL SHUTDOWN

If the vapour ventilation system is being used, it cannot be cooled down using the tunnel heat exchanger and can only reject heat by boiling induced by pressure reduction with no heat input. The method used for cooling the vapour ventilation system down from temperatures greater than 100° C is included below.

1. Set evaporator power input to zero.

2. Cool down the tunnel to at least 95° C using the procedure given in Section A.I.1.9, maintaining the

tunnel pressure at the last experimental value used. As the tunnel is cooled the plenum level will fall as the working fluid contracts. Restore the level by activating the Madan pump and opening valve I (Fig. I.38) as required.

3. When the tunnel temperature reaches 95° C, reduce the tunnel speed to 5 m/sec.

4. Reduce the tunnel pressure slowly to atmospheric pressure using the pressure regulator (Fig. I.38).

Pressure reduction will cause vapour generation in the ventilation evaporator. Prevent overspeeding of the turbine flowmeter by using the ventilation throttle valve (Fig. I.46). When the tunnel pressure has been reduced to atmospheric, vapour generation from the evaporator will cease when its temperature drops to 100° C.

5. When the tunnel pressure drops to atmospheric, check that valve 11 is open, open valve 5 to connect the tunnel to the sump tank and close valve 8 to isolate the plenum from the pressure regulator (Fig. I.38).

6. Switch off the cooling tower pump and fan (Fig. I.37) and reduce tunnel speed to zero.

7. Fully open valve BV1 and fully close valve BV2 (Fig. I.38).

At the C.W.C.T. control console,

8. Switch off MAIN HYDR., and wait until the 75 KW electric motor has completely stopped.

9. Switch off AUX. HYDR., INST. PWR., and MAINS switches.

10. Turn off the key-operated MASTER SWITCH.

At the ground floor Cavitation Laboratory,

11. Switch off CONTRACTOR on the front panel of the 40 KW evaporator power supply.

12. Switch off circuit breaker BBC 5.
13. Close valve VC(Fig. I.35) to stop the cooling water flow through the auxilliary hydraulics cooler.
14. Switch off the circuit breakers labelled NEW CAVITATION TUNNEL and MAIN PUMP MOTOR STARTER.

APPENDIX A.I.2

OPERATION OF THE DATA ACQUISITION SYSTEM

As described in Chapter I.8, the tunnel transducers are interfaced to a Hewlett Packard 2010 D scanning data logging system. The interfaces contain signal-conditioning equipment to make the outputs of the transducers compatible with the integrating digital voltmeter which measures the transducer outputs. One of the interfaces provides the synchronized drive required to advance the scanivalve, and also allows the data logger to be actuated from its control panel. This Appendix describes the procedures required to operate the data acquisition system to obtain paper tape records of the transducer outputs. A computer program which may be used to process the paper tape data is described in detail by Mussared^[49], who wrote it to obtain the derived numerical data presented in Part II of this thesis. For the present purpose it is assumed that the data acquisition system terminates at the paper tape punch of the data logger. Words written in capital letters in the following procedures denote function descriptions labelled on the items of equipment in question.

The data logger is operated as follows

1. Turn on the main power switch located at the top right hand corner of the chassis.
2. Switch on the VOLTMETER, SCANNER CONTROL AND PUNCH COUPLER.
3. Select the RANGE and SAMPLE PERIOD on the VOLTMETER. The 0.1 volt RANGE and 1 second SAMPLE PERIOD are usually used.

4. Set the VOLTMETER SAMPLING RATE potentiometer.
Turn it fully clockwise for visual readings of the digital voltmeter in the LOCAL mode, and fully anti-clockwise when using automatic scanning with the paper tape punch.

5. Set the SCANNER mode

SINGLE CHANNEL:	Scans only the channel displayed
SINGLE SCAN:	Performs one scan through the range of channels set by LOWER LIMIT and UPPER LIMIT dials when START switch is pressed.
CONT. SCAN:	Allows continuous scanning through the channel range in either LOCAL + STEP or AUTO mode.
RESET:	RETURNS THE SCANNER TO THE CHANNEL DEFINED BY LOWER LIMIT dial.
START:	Initiates scanning in LOCAL or AUTO modes.
STEP:	Advances the scanner to the next channel when in the LOCAL mode.
LOCAL:	Allows manual control from the scanner front panel.
AUTO:	Gives automatic scanning through the selected channel range at the set SAMPLE PERIOD.
REMOTE:	Allows SINGLE SCAN, CONT. SCAN, RESET and START to be performed from the panel next to the Scanivalve drive unit.

6. Set the PUNCH COUPLER.

NON-PUNCH:	Stops the paper tape punch.
AUTO:	Allows automatic scanning and punching of the voltmeter readings.
MANUAL + TAPE FEED:	Feeds blank tape through the paper tape punch.
MANUAL + SINGLE WORD:	Punches only the voltmeter reading currently displayed. Only used in LOCAL mode.

7. Set the SCANIVALVE controls.

AUTOHOME:	Returns Scanivalve to channel 01
CONTROL + MAN:	Allows manual stepping through Scanivalve channels up to UPPER LIMIT.
CONTROL + D/L:	Scanivalve under the control of the data logger. STEP/SCAN. switch must be set to SCAN.
STEP/SCAN:	Allows MAN TRIG to advance the channel by one or to start the continuous scanning as far as UPPER LIMIT.
MAN TRIG:	Triggers above scanning action.
STEP TIME:	Adjusts scan rate of scanivalve.
UPPER LIMIT:	Sets channel at which Scanivalve returns to channel 01 automatically.

8. Usual control settings and operating sequence.

(i) For visual voltmeter readings.

a. Set voltmeter RANGE to 0.1 volts and SAMPLE

PERIOD to 1 sec.

b. Turn SAMPLING RATE potentiometer fully clockwise.

c. Set LOWER LIMIT to 175, UPPER LIMIT to 219.

d. At PUNCH COUPLER, press AUTO and NON-PUNCH switches.

e. At SCANIVALVE control panel set the switches to CONTROL + D/L, SCAN and UPPER LIMIT 19.

f. At the SCANNER, press the SINGLE SCAN, LOCAL and RESET switches and STEP to the required channel.

(ii) For automatic scanning and tape punch operation.

a. Set voltmeter RANGE to 0.1 volts and SAMPLE PERIOD to 1 sec.

b. Turn the SAMPLING RATE potentiometer fully anti-clockwise.

c. Set LOWER LIMIT to 175.

d. Set UPPER LIMIT to 191 if wall pressures are not required or to 219 if they are.

e. At SCANIVALVE control panel, set the switches to CONTROL + D/L, SCAN and UPPER LIMIT 19.

f. At the SCANNER, press the LOCAL, RESET switches and then REMOTE.

g. At the PUNCH COUPLER, press AUTO and then release NON-PUNCH.

h. At SCANIVALVE control panel, press START and data scanning and logging commences.

APPENDIX A.I.3

ROUTINE MAINTENANCE OF THE HIGH TEMPERATURE CAVITATION TUNNEL

The following maintenance procedure should be observed to ensure the safe and efficient operation of the tunnel. The period between maintenance for each item is given, where appropriate, in units of hour of running time for the hydrostatic drive. Cumulative running time registers for both the main and auxilliary hydraulic systems are located on the control console of the Cold Water Cavitation Tunnel.

A.I.3.1 HYDROSTATIC DRIVE

The sight glass of the upper level oil header tank (Fig. I.35) should be checked each time the hydrostatic drive is run to ensure that the oil level is still within sight. Add Shell Tellus 127 oil to the tank as required. The oil filters located in the discharge pipes from the low level oil reservoir (Fig. I.35) should be replaced every 100 hours. Their technical specifications and code numbers are shown on Keelavite Hydraulics Ltd. Drawing No. C 35406, which is a drawing of the hydrostatic drive showing all the type numbers of the components thereof. Copies of this drawing, together with further technical information relating to the hydrostatic drive is located in the filing system of the Cavitation Laboratory. Blockage of these filters causes the boost pump and servo pump delivery pressures to fall below their required values, as indicated on their respective pressure gauges

on the control consol of the C.W.C.T. (See Section A.I.1.5).

The holding-down bolts of the Lucas PM 1000 hydraulic pumps and motors should be checked for security every 50 hours running time. If any part of the auxilliary hydraulics system downstream of the filters is dismantled, care should be taken to prevent dirt getting into the system since the Moog-valve is very sensitive to contamination.

The reduction gearing between the tacho-generator and the circulating pump shaft (Fig. I.36) should be lubricated prior to each tunnel run with a few drops of light machine oil.

A.I.3.2 CIRCULATING PUMP

The oil level should be checked every 10 running hours at the level glass on the pump bearing housing. If the level is low, add Esso Teresso 47 oil as required. The bolts securing the self-centring bearing housing to their support framework should be checked for security every 50 running hours, as should the retaining bolts of the flexible coupling (Fig. I.36).

A.I.3.3 TUNNEL COMPONENTS SUBJECT TO CAVITATION

The occurrence of cavitation damage is crucial to tunnel components which constitute part of the pressure containment structure. The areas where cavitation damage is likely to occur are the diffuser inlet region and the circulating pump volute. These regions have not been inspected internally to date, but this should be done at some stage. The diffuser inlet could be inspected without removing it from the tunnel circuit by taking out the test

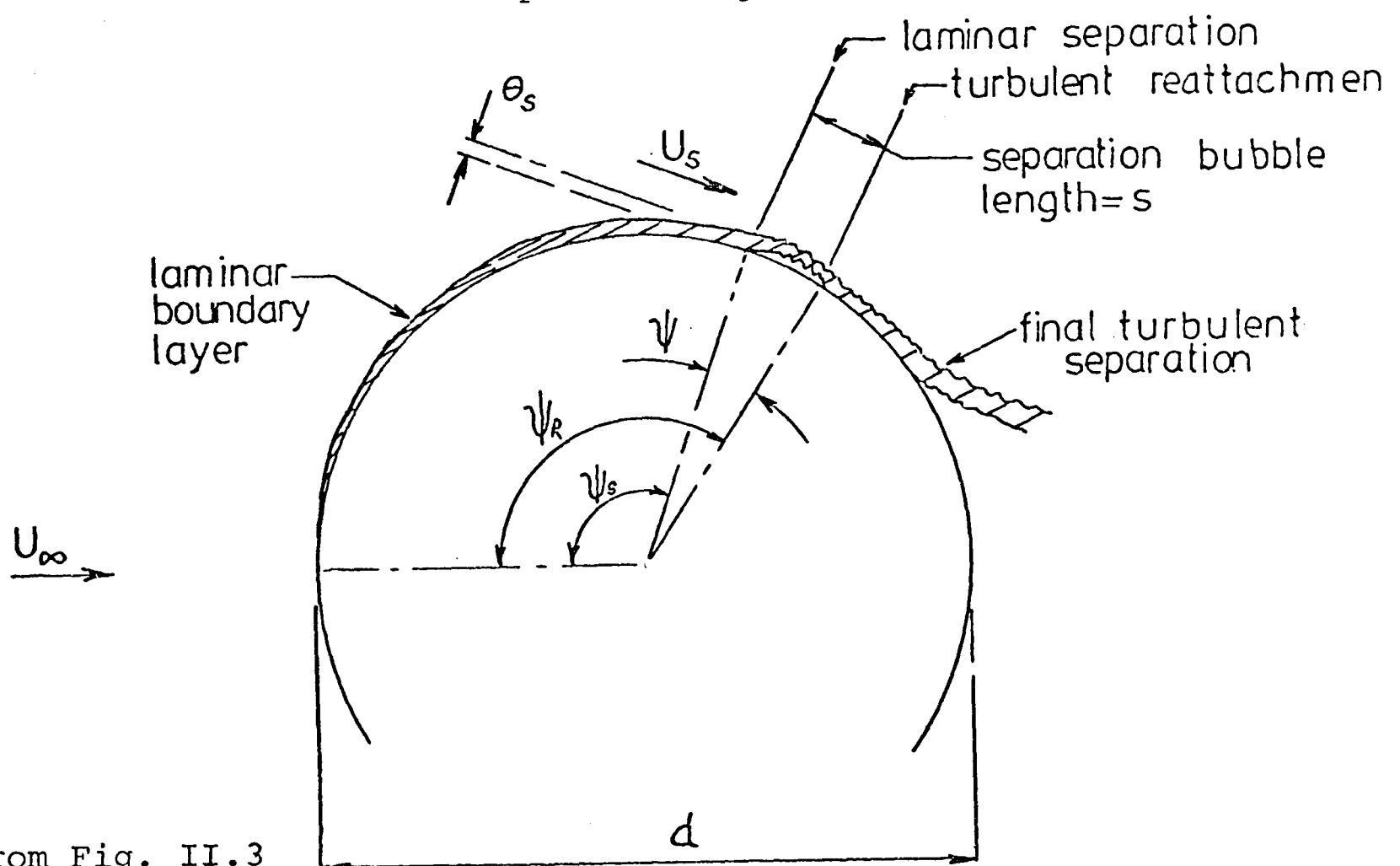
section windows and using a mirror with a suitable light source. The circulating pump volute could be likewise inspected since the pump back plate, bearing housing, main shaft and impeller can be removed as a single unit, thereby allowing access to the volute for inspection. The maintenance documentation, parts list and detailed engineering drawings of the circulating pump are contained in the filing system of the Cavitation Laboratory.

APPENDIX A.II

LAMINAR SEPARATION BUBBLES ON CIRCULAR CYLINDERS

Estimation of the non-dimensional bubble length and Reynolds number range for the occurrence of laminar separation bubbles on two-dimensional circular cylinders, from the data of Tani^[84].

Dimensions of the separated region



From Fig. II.3

at $R_d = 3.7 \times 10^5$, $\psi = \psi_R - \psi_s = 17.5^\circ = 0.305$ rads.

at $R_d = 9 \times 10^5$, $\psi = 8.5^\circ = 0.148$ rads.

From Fig. II.2

$(1-C_{ps}) = 3.34$ assumed constant with R_d

From Fig. II.4 (extreme values of data)

R_d	$R_{\delta s}$	$R_{\theta s} = R_{\delta s}/3.7$	ψ
3.7×10^5	1400	380	0.305
9×10^5	2000	550	0.408

$$\text{and } R_{\theta s} = \frac{\theta_s u_s}{\nu} ; R_d = \frac{d U_\infty}{\nu}$$

$$\text{hence } \frac{R_{\theta s}}{\theta_s u_s} = \frac{R_d}{d U_\infty} \quad \text{A.II.1}$$

$$\text{Separated region length } S = \frac{d}{2} \cdot \psi$$

$$\text{hence } \frac{S}{\theta_s} = \frac{d}{2} \cdot \frac{\psi}{\theta_s} \quad \text{A.II.2}$$

$$\text{From Equ. A.II.1, } \frac{d}{\theta_s} = \frac{R_d}{R_{\theta s}} \cdot \frac{U_s}{U_\infty} \quad \text{A.II.3}$$

Substitute Equ. A.II.3 into Equ. A.II.2,

$$\frac{S}{\theta_s} = \frac{\psi}{2} \cdot \frac{R_d}{R_{\theta s}} \cdot \frac{U_s}{U_\infty} \quad \text{A.II.4}$$

$$\text{and } \frac{U_s}{U_\infty} = (1-C_{ps})^{\frac{1}{2}} \quad \text{A.II.5}$$

Substitute Equ. A.II.5 into Equ. A.II.4,

$$\frac{S}{\theta_s} = \frac{\psi}{2} \cdot \frac{R_d}{R_{\theta s}} \cdot (1-C_{ps})^{\frac{1}{2}} \quad \text{A.II.6}$$

$$\text{Given } (1-C_{ps}) = 3.34, (1-C_{ps})^{\frac{1}{2}} = 1.825$$

Insertion of the above numerical tabulated data into Equ. A.I.6 yields the required bubble size and Reynolds number data.

$R_{\theta s}$	S/θ_s	R_d	ψ (rads)
380	272	3.7×10^5	0.305
550	221	9×10^5	0.148

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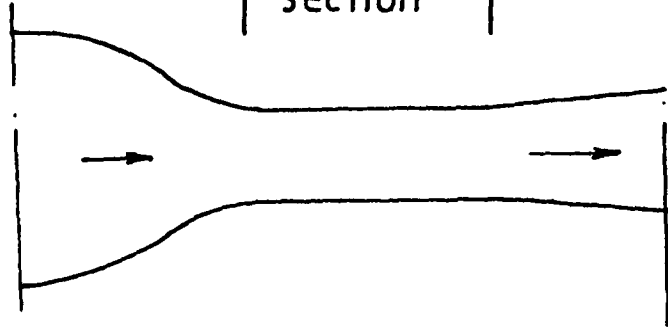
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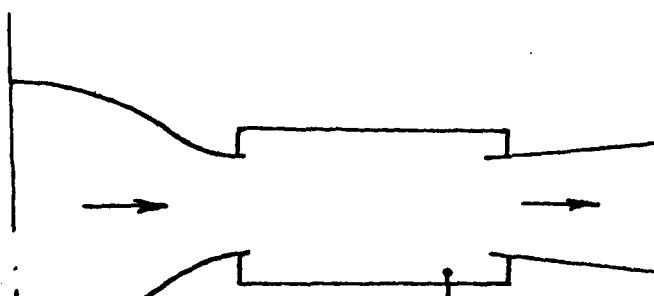
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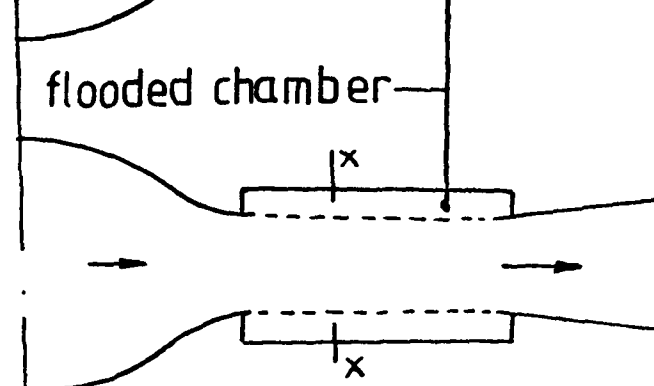
contraction test section diffuser



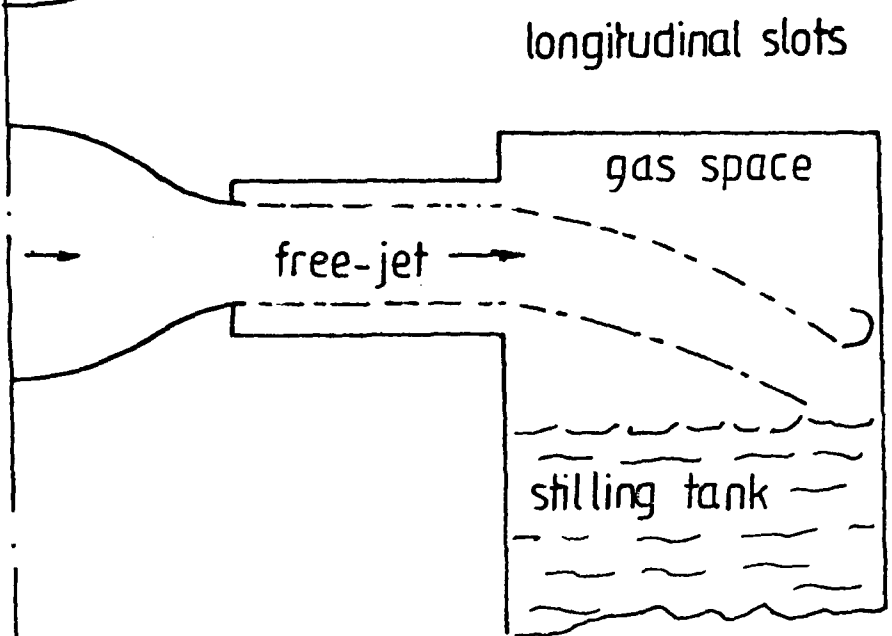
a. closed type



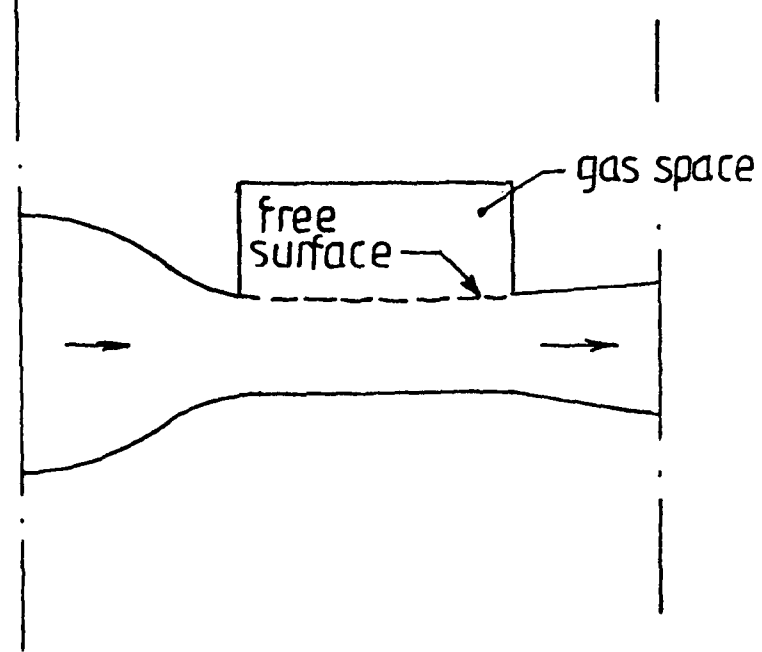
b. open type



c. slotted wall type



d. free — jet type



e. free-surface type

FIG. I.1 SCHEMATICS OF SOME BASIC TYPES OF WORKING SECTIONS FOR CAVITATION TUNNELS

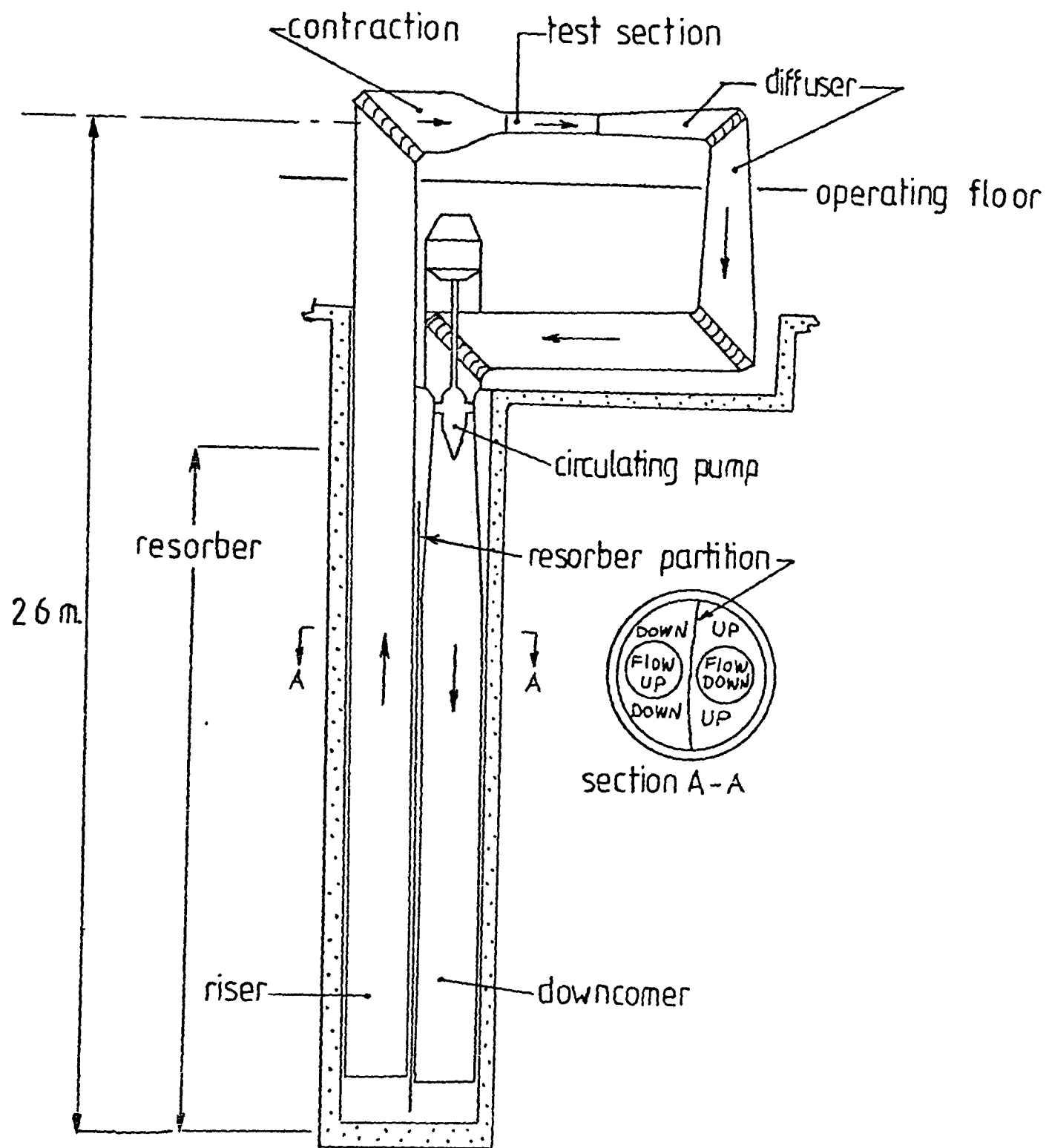
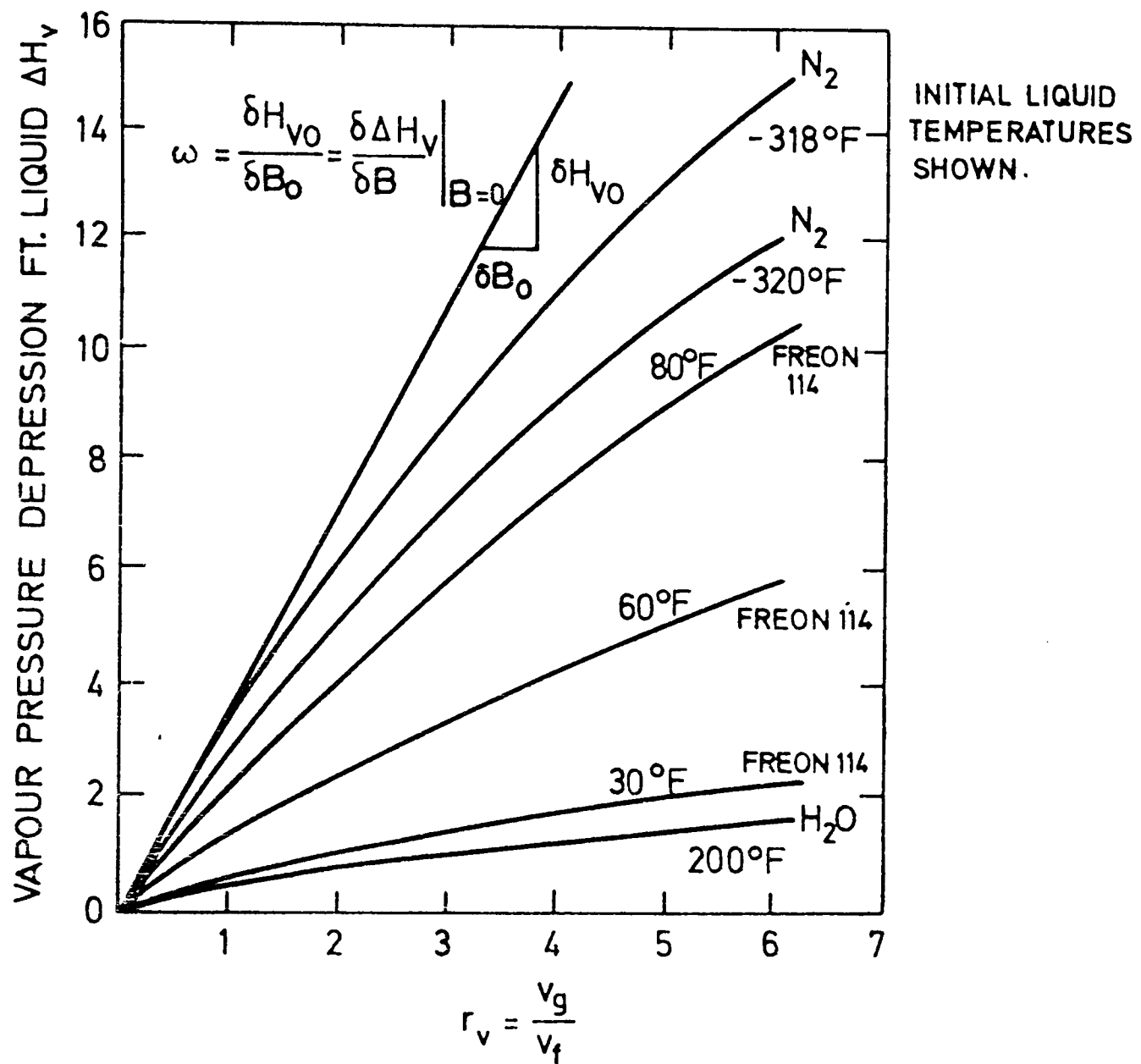


FIG. 1.2 CIRCUIT LAYOUT OF THE CALIFORNIA INSTITUTE OF TECHNOLOGY HIGH SPEED WATER TUNNEL SHOWING RESORBER



VAPOUR PRESSURE DEPRESSION AS FUNCTION OF LIQUID VAPOURIZED.

ADAPTED FROM GELDER, RUGGERI & MOORE (16)
N.A.S.A. TN D-3509

FIG. I. 3

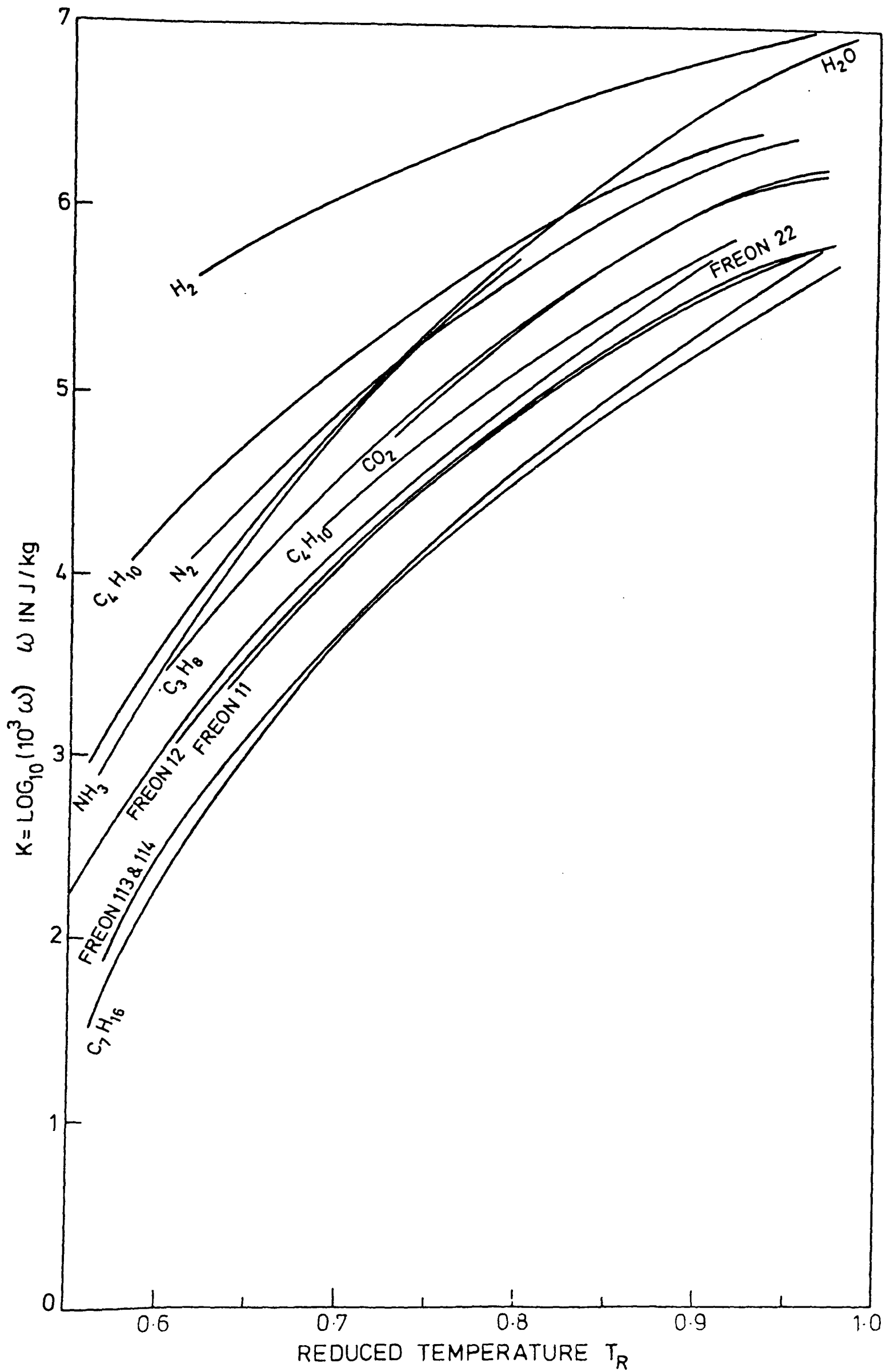


FIG.I.4 VARIATION OF K PARAMETER WITH REDUCED TEMPERATURE FOR VARIOUS LIQUIDS

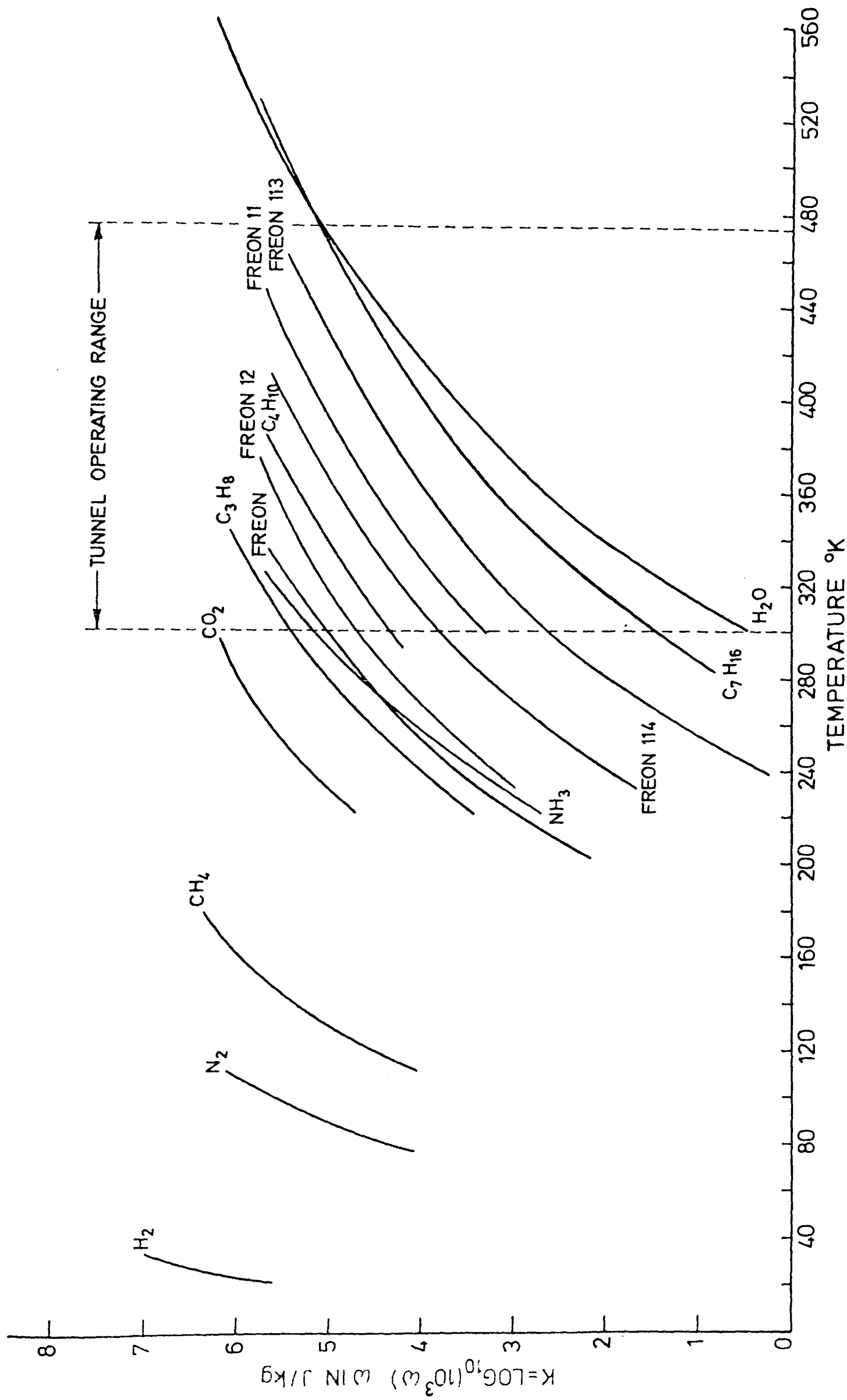


FIG.I.5 VARIATION OF K PARAMETER WITH ABSOLUTE TEMPERATURE FOR VARIOUS LIQUIDS.

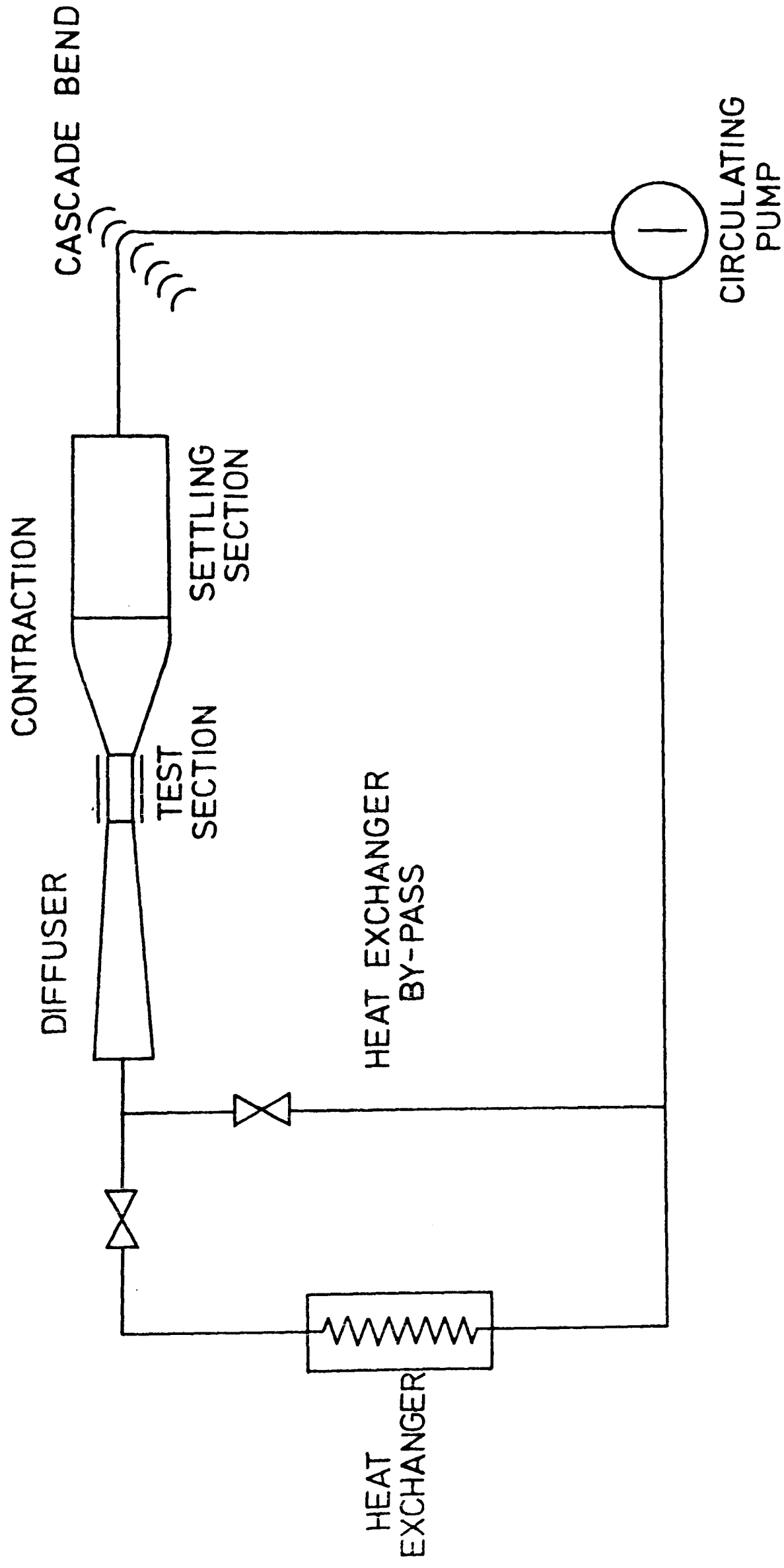


FIG.I.6 TUNNEL CONFIGURATION SCHEMATIC.

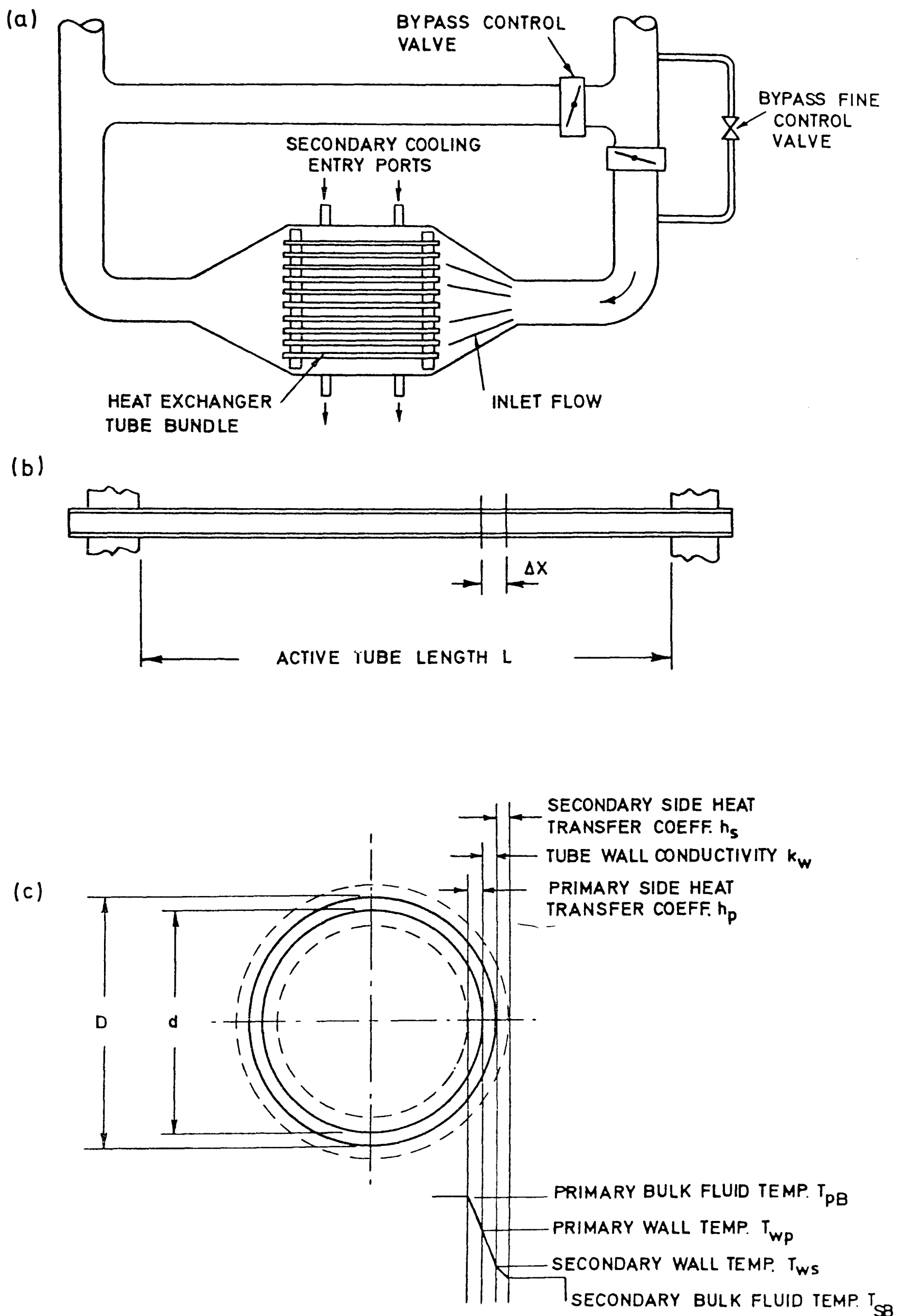


FIG.I.7 HEAT EXCHANGER AND BYPASS SYSTEM. TYPICAL TUBE AND TYPICAL CROSS SECTION THROUGH TUBE SHOWING SALIENT TEMPERATURES.

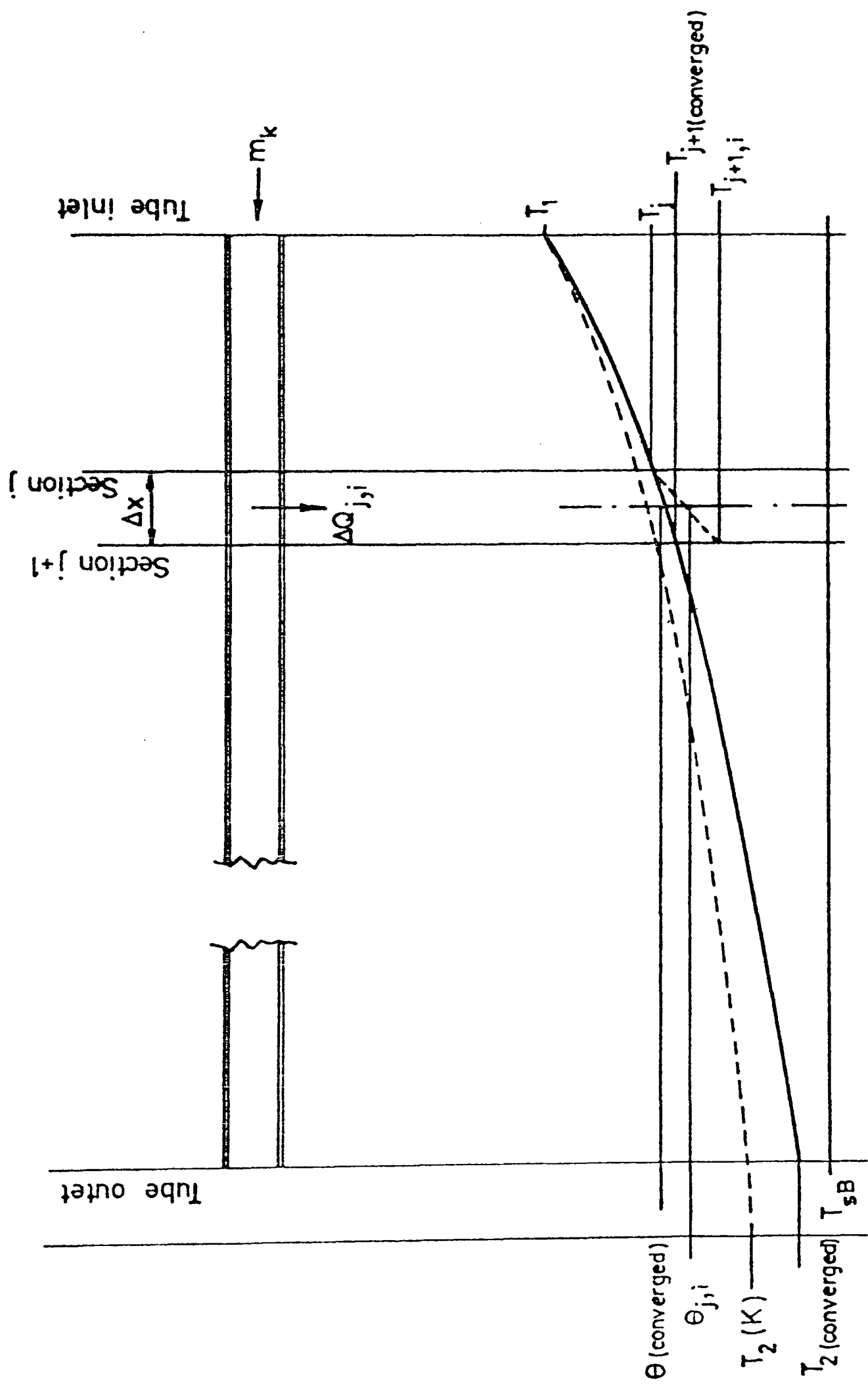


FIG.I.8 ITERATION SCHEME FOR COMPUTING THE TEMPERATURE DISTRIBUTION IN A TYPICAL HEAT EXCHANGER TUBE

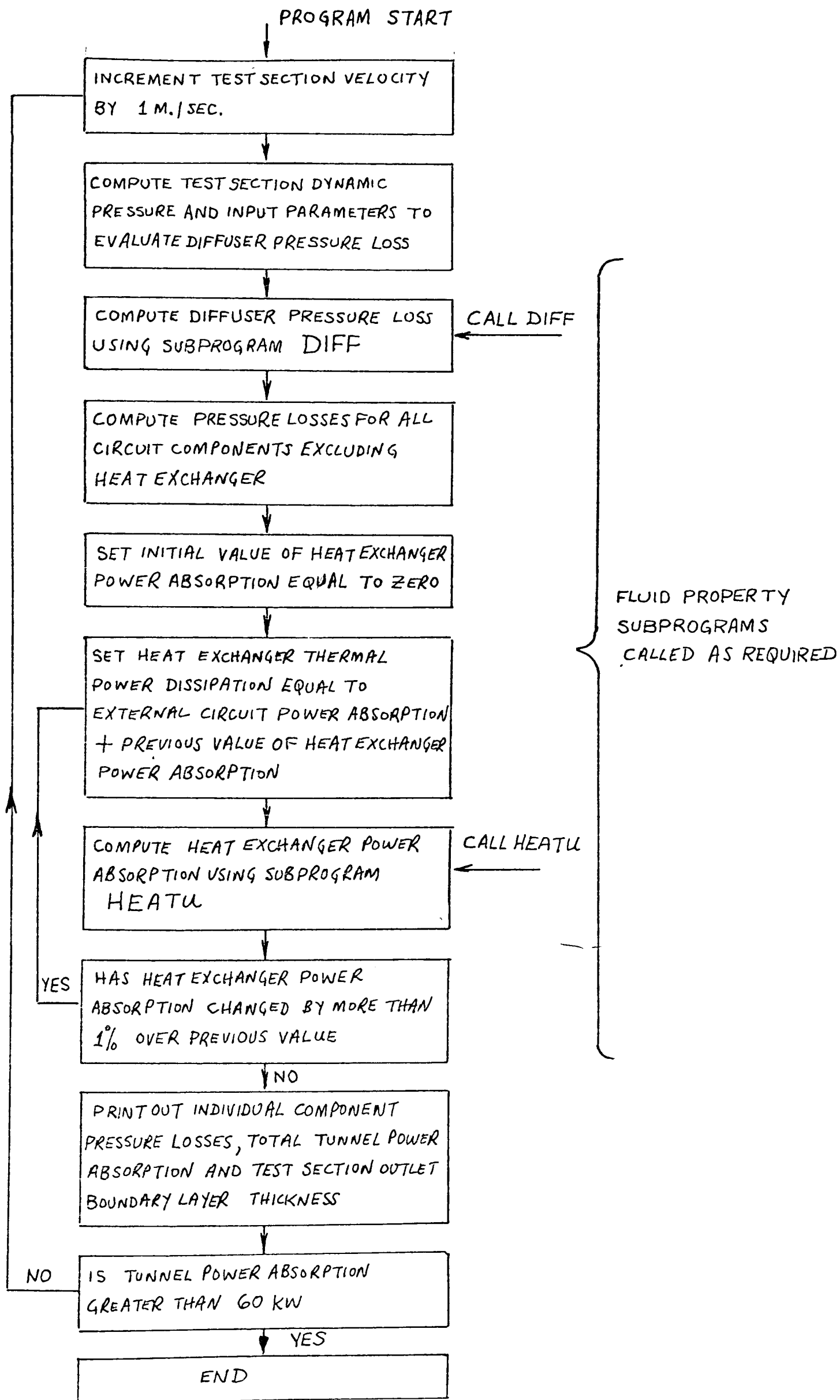


FIG. I.9 BLOCK DIAGRAM OF PROGRAM 'POWER'

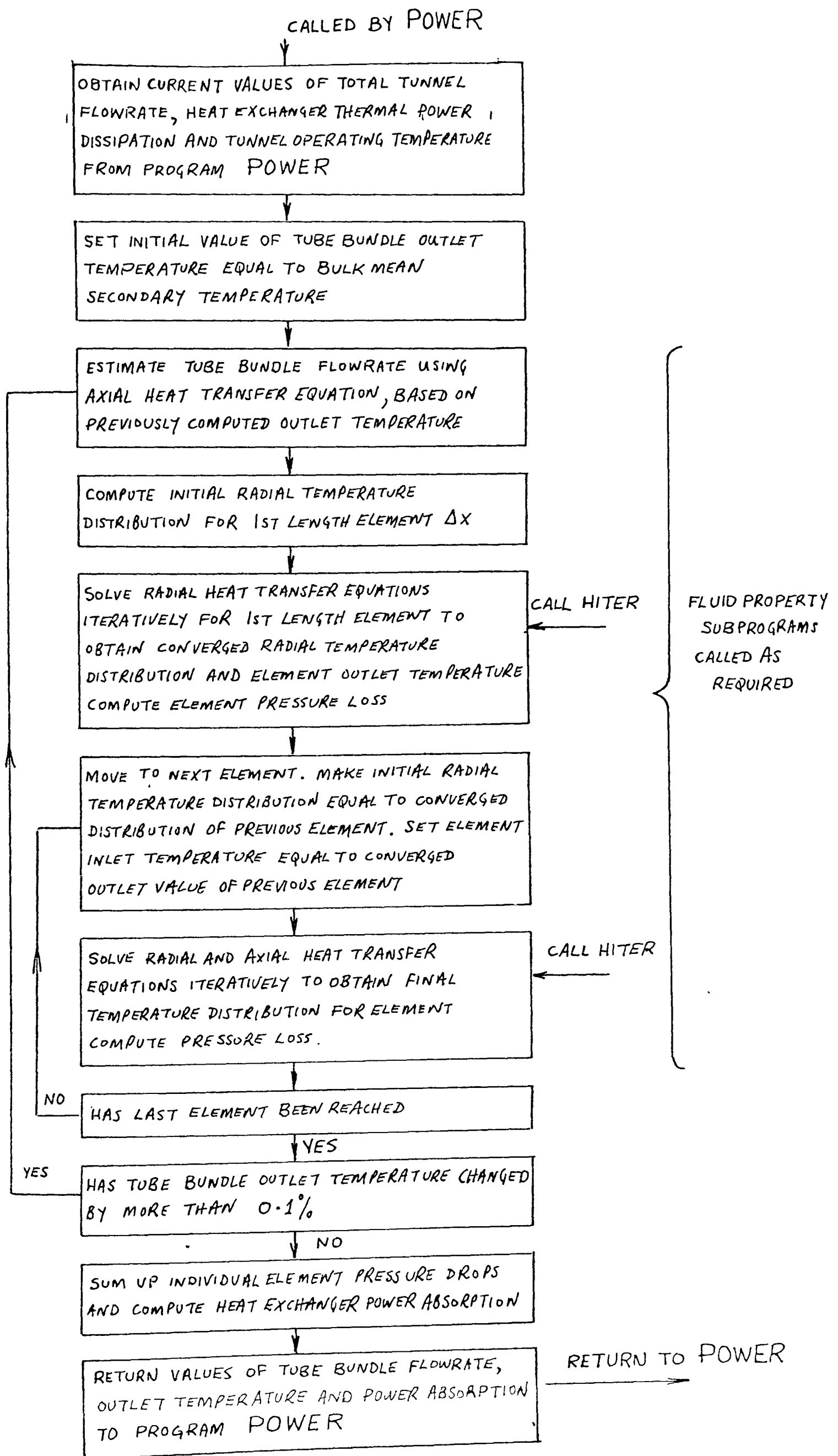


FIG. I.10 BLOCK DIAGRAM OF SUBPROGRAM 'HEATU'

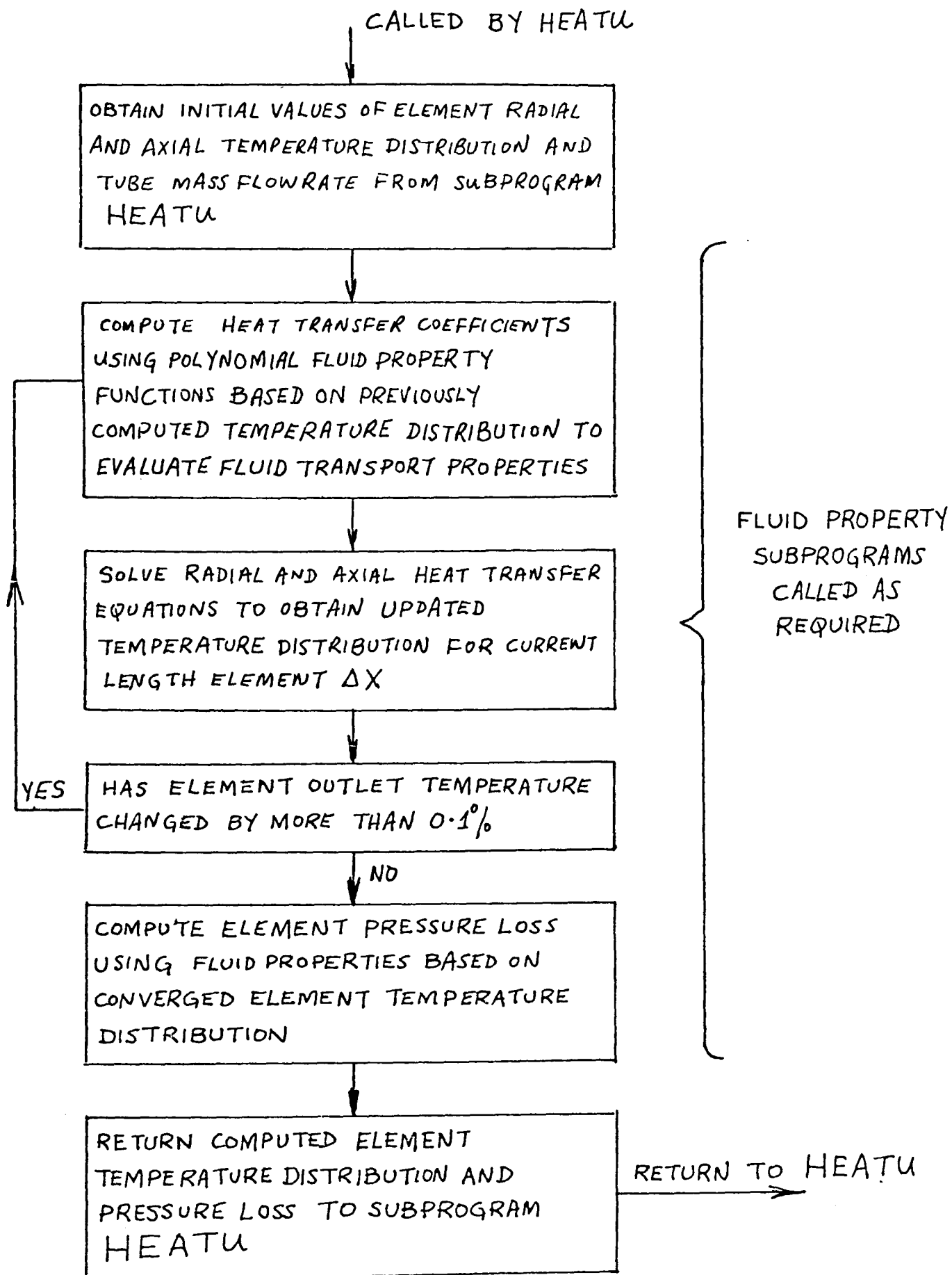


FIG.I.11 BLOCK DIAGRAM OF SUBPROGRAM 'HITER'

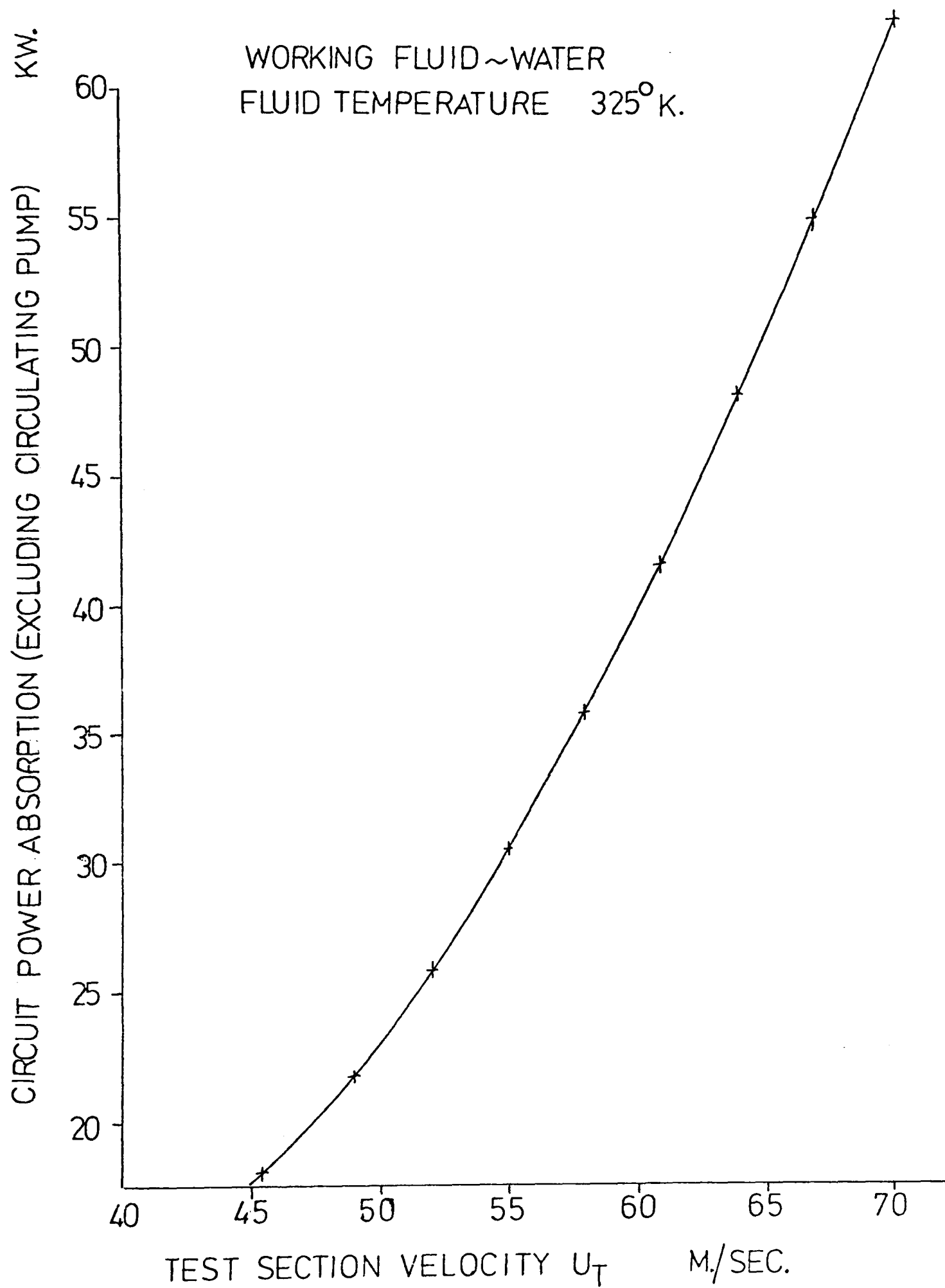


FIG. I.12 TYPICAL SIMULATED TUNNEL POWER ABSORPTION CURVE
FOR HIGH SPEED TEST SECTION CONFIGURATION

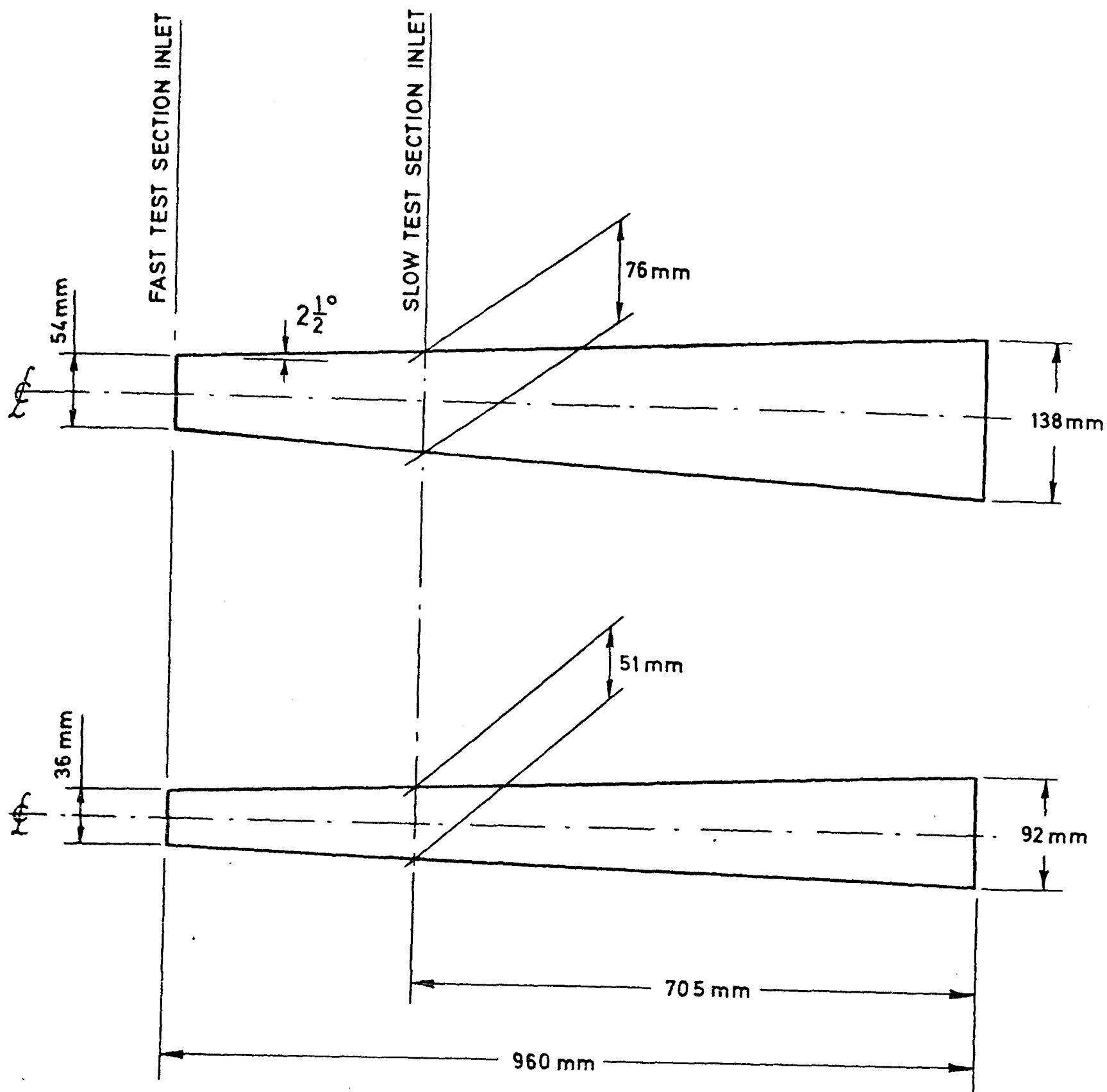


FIG.I.13 LINE DIAGRAM OF DIFFUSER - INTERNAL PROFILE.

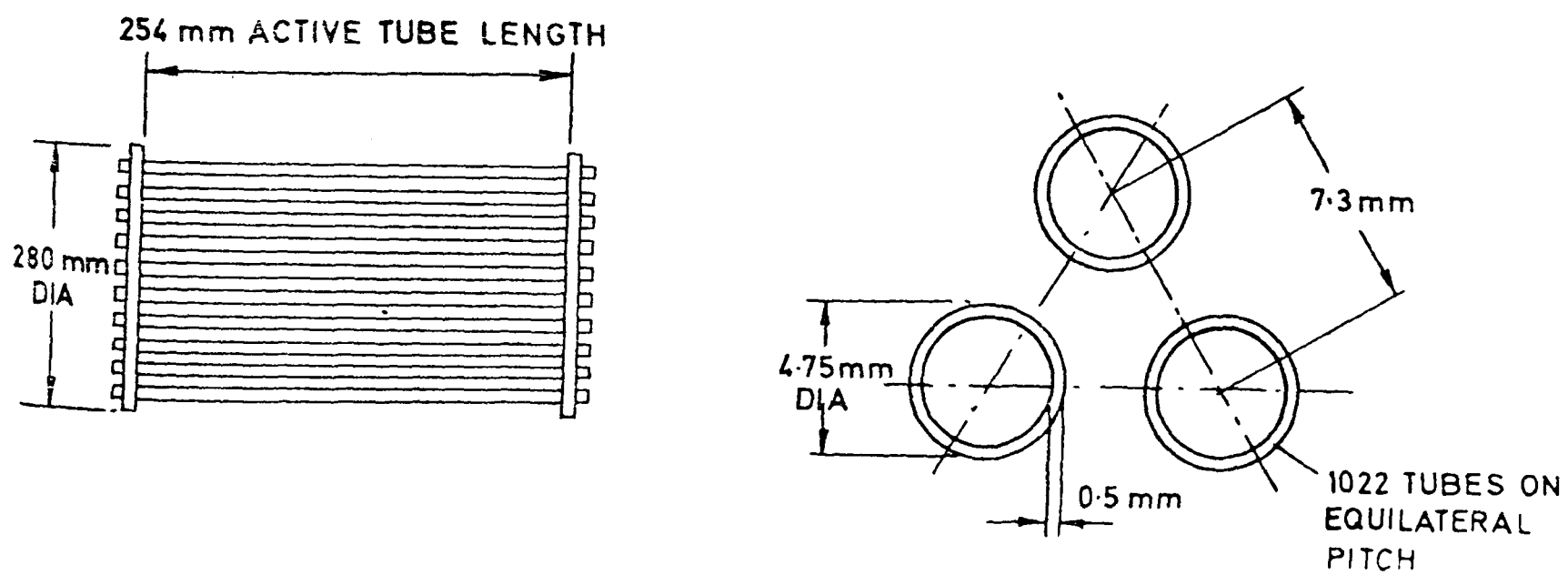


FIG.I.14 HEAT EXCHANGER TUBE BUNDLE DETAIL.

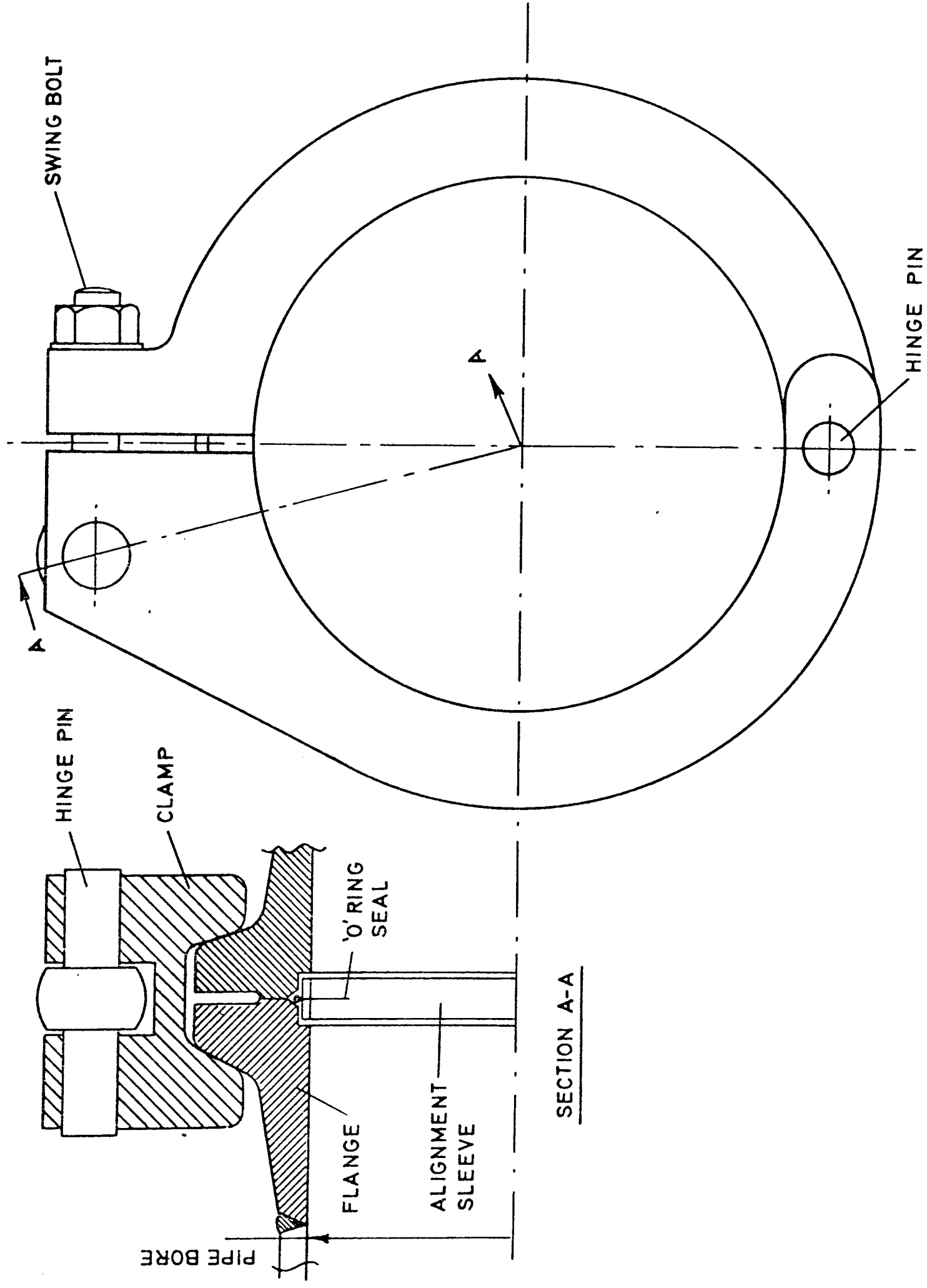


FIG.I.15 'GRAYLOCK' QUICK RELEASE TYPE COUPLING DETAIL

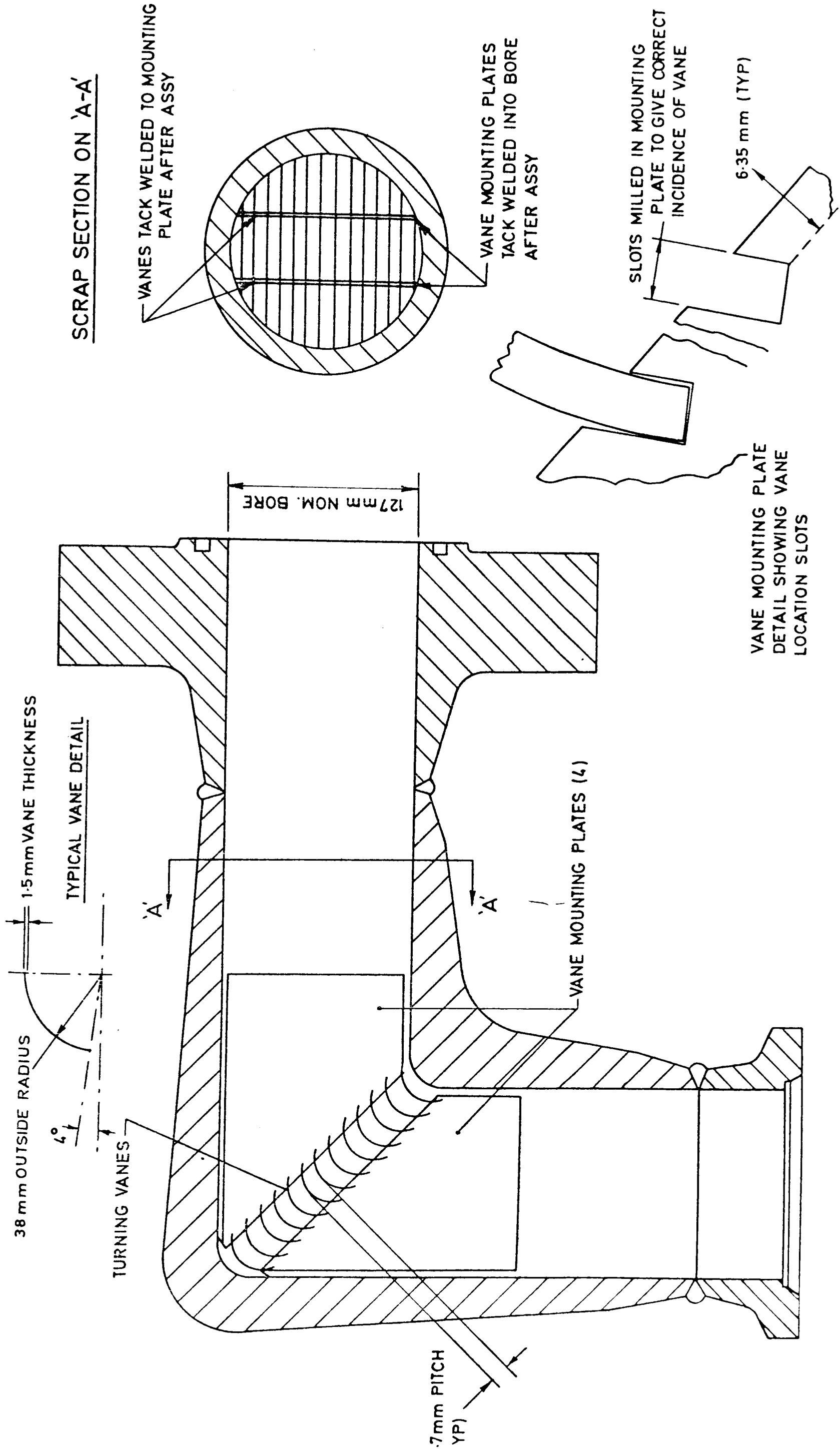


FIG.I.16 CORNER VANE ASSEMBLY SHOWING VANE FIXING DETAILS.

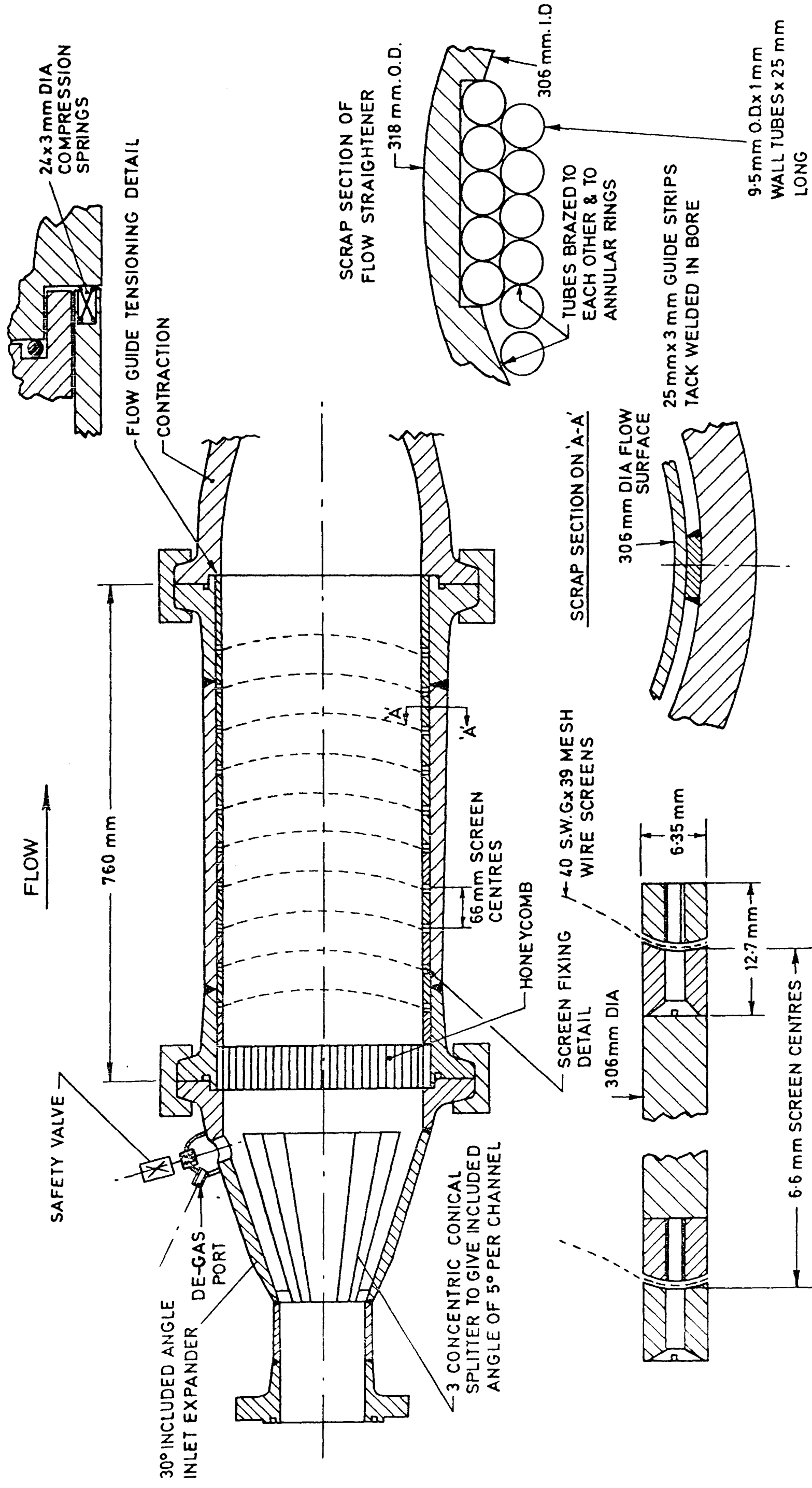


FIG.I.17 SETTLING SECTION ASSEMBLY

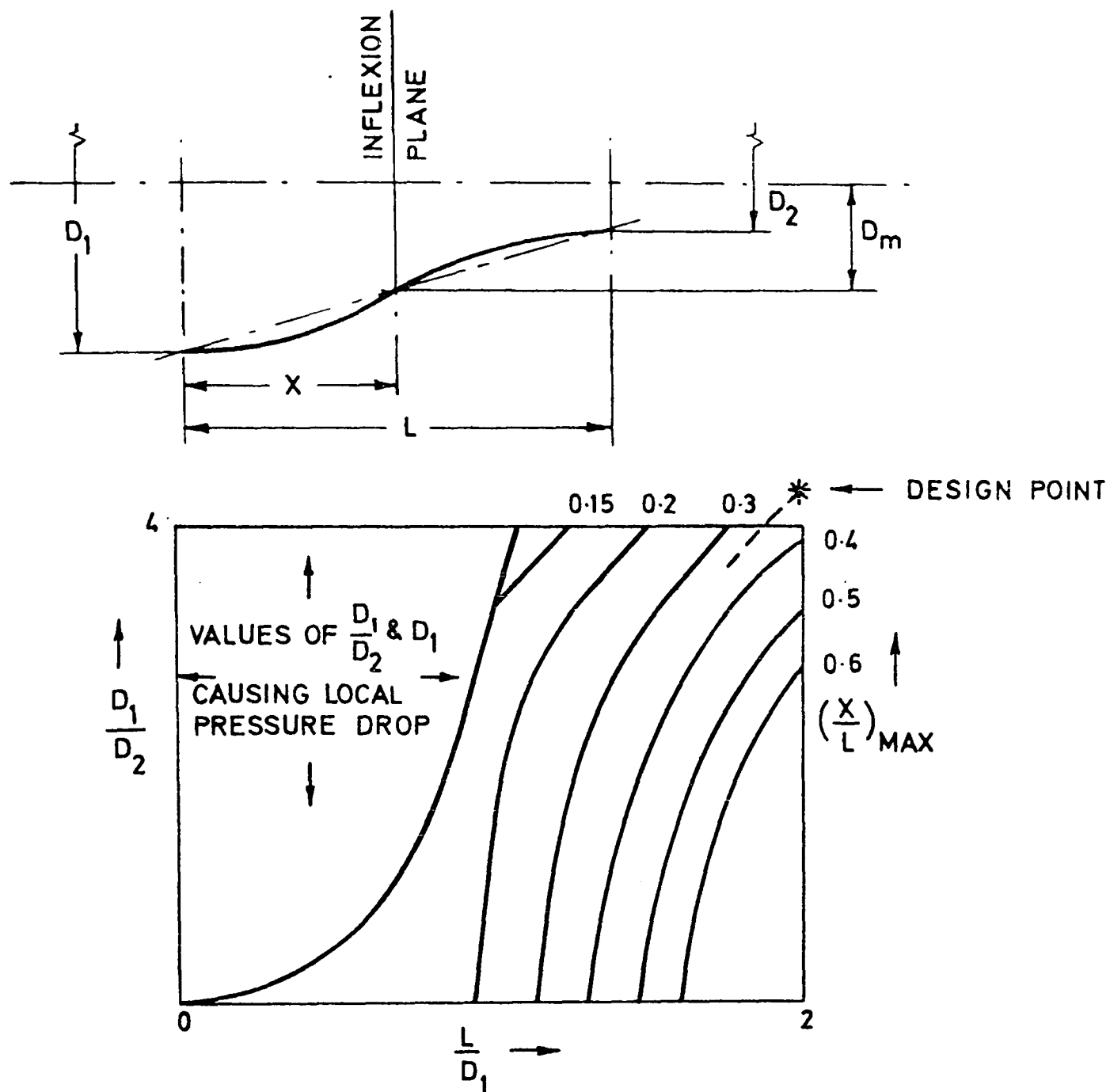


FIG.I.18 CONTRACTION DESIGN DATA OF ROUSE AND HASSAN

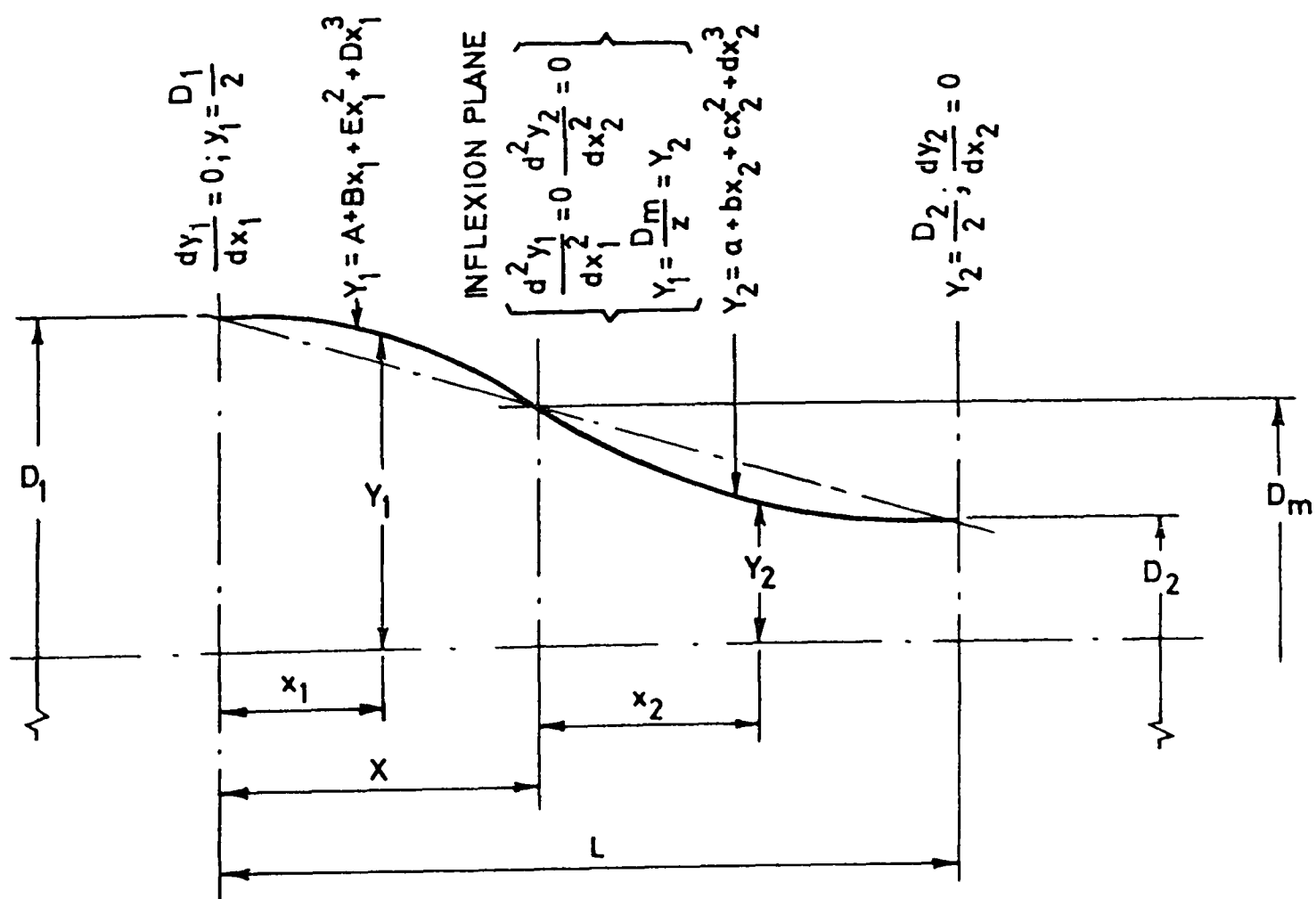


FIG.I.19 BOUNDARY CONDITIONS FOR COEFFICIENT OF CUBIC POLYNOMIALS DEFINING CONTRACTION WALLS.

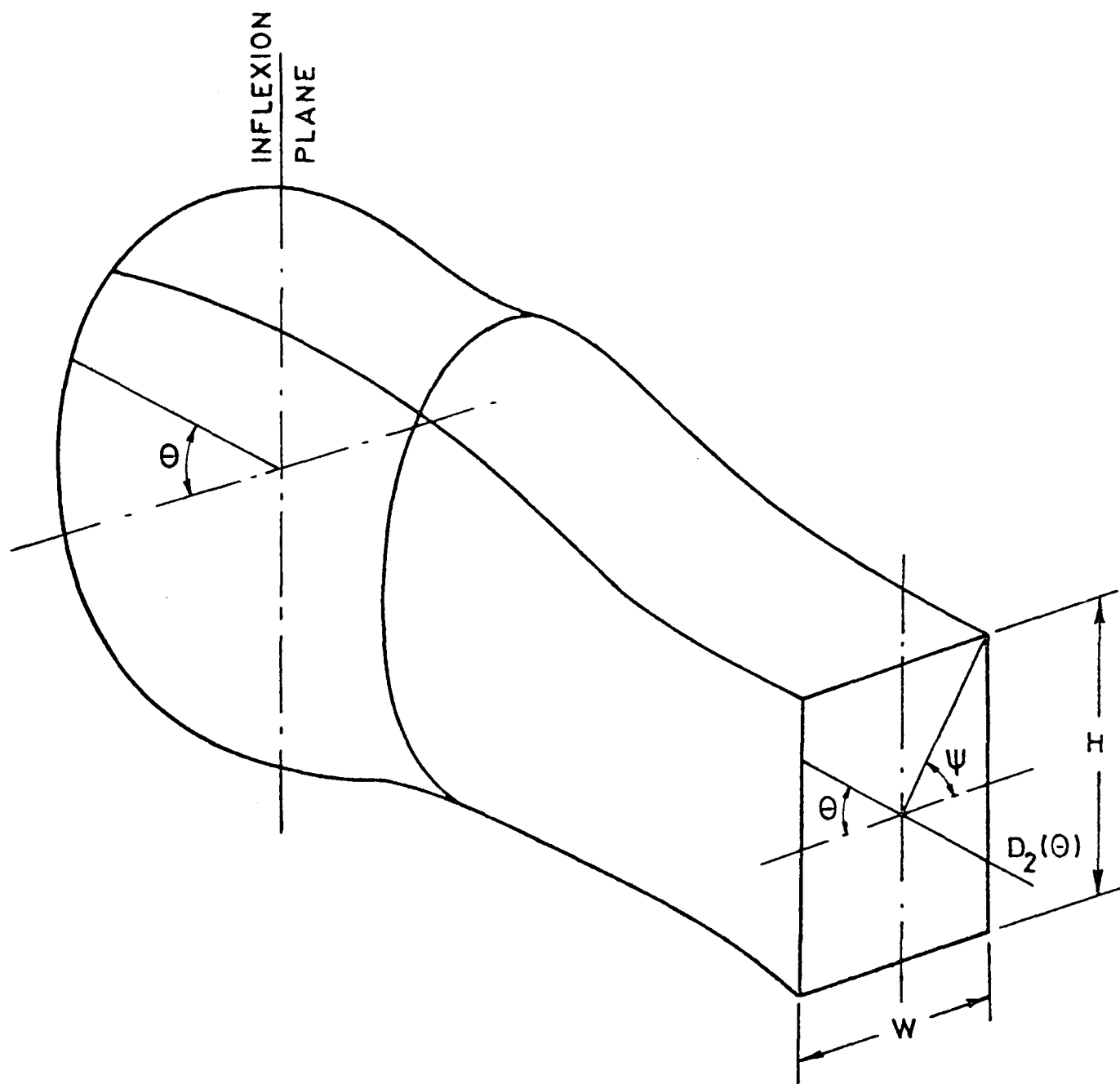


FIG.I.20 FITTING OF POLYNOMIAL CURVES TO TRANSITION CONTRACTION

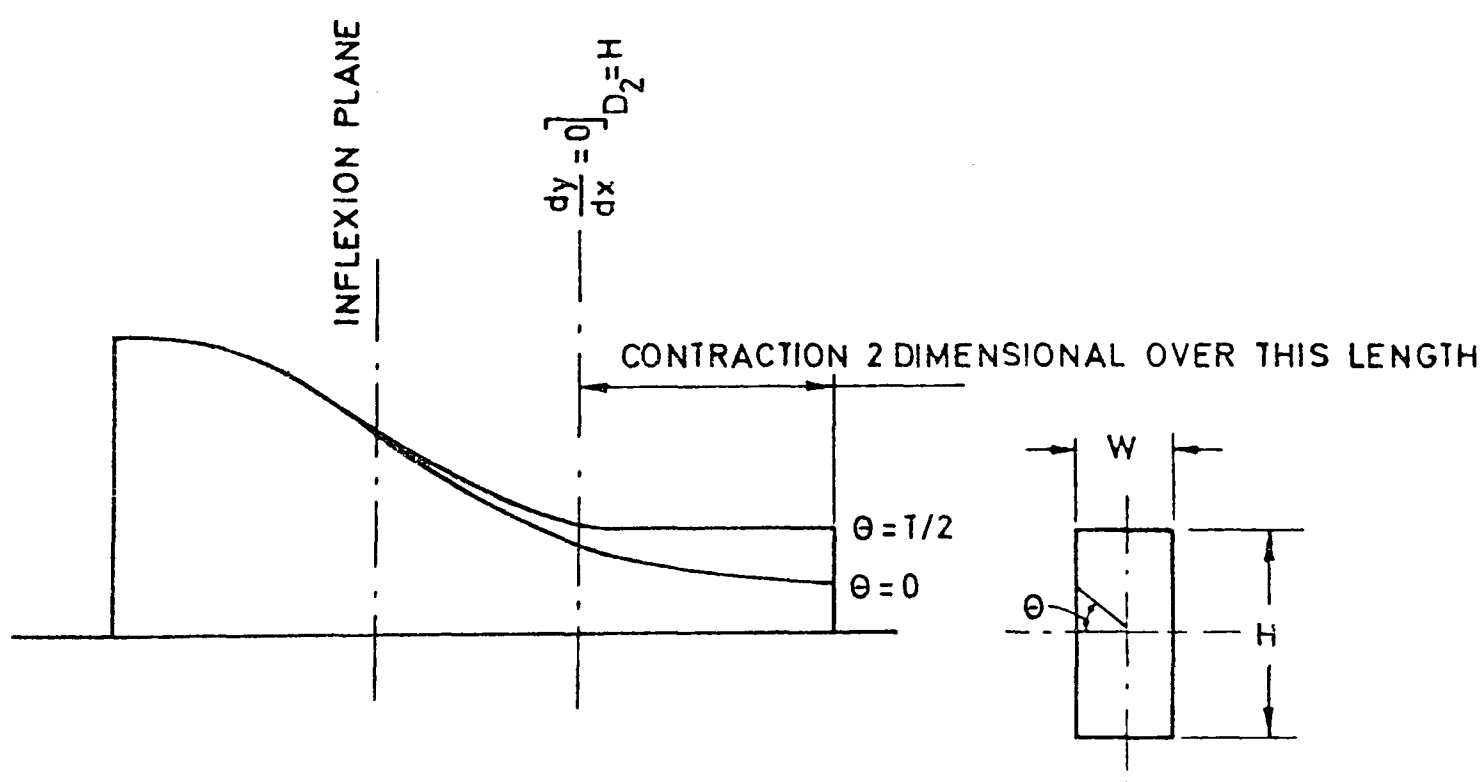


FIG.I.21 EFFECT OF VARIABLE TRANSITION LENGTH ON WALL PROFILE

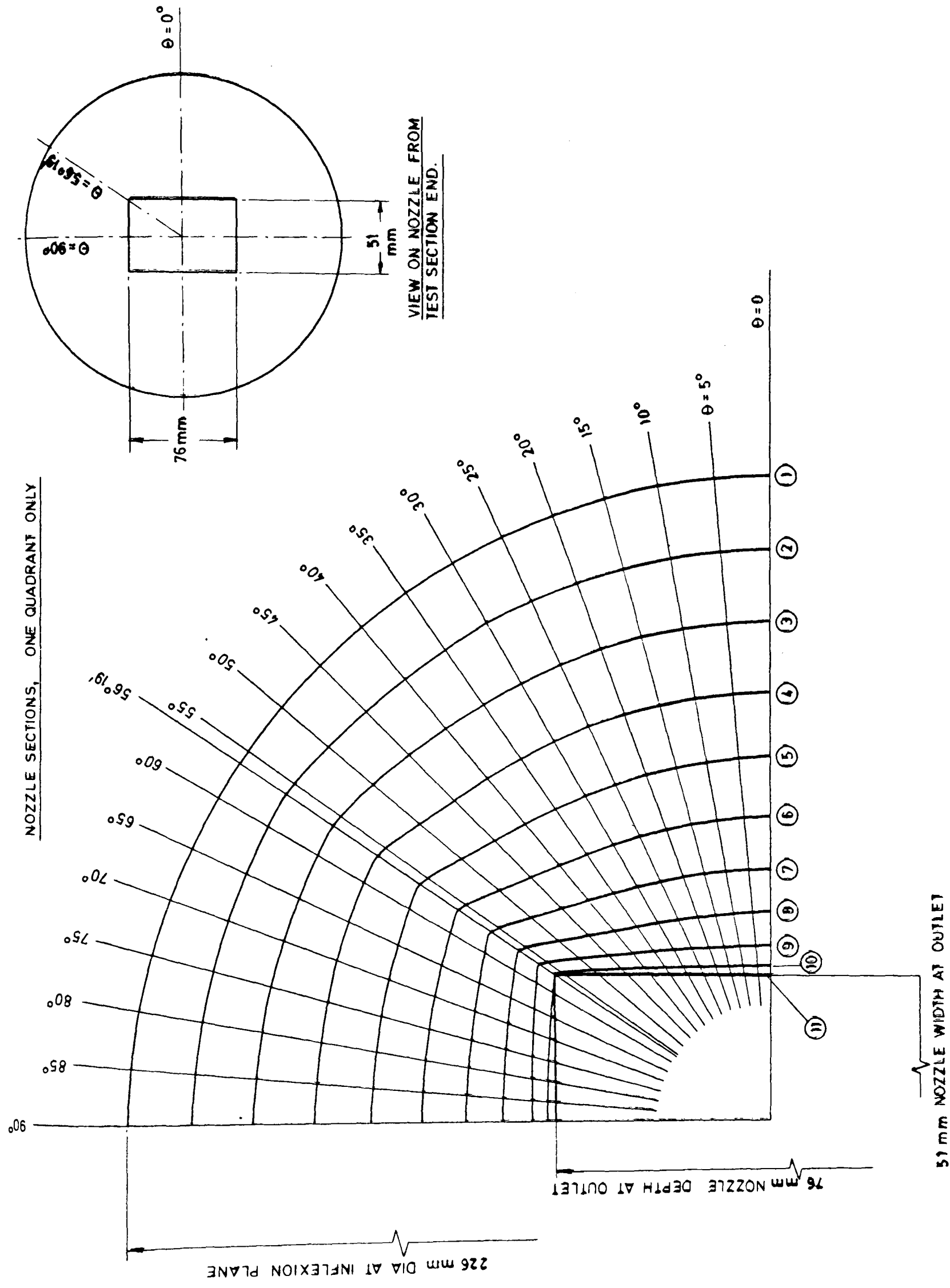


FIG.I.22 CONTRACTION CROSS SECTION AT VARIOUS AXIAL LOCATIONS.

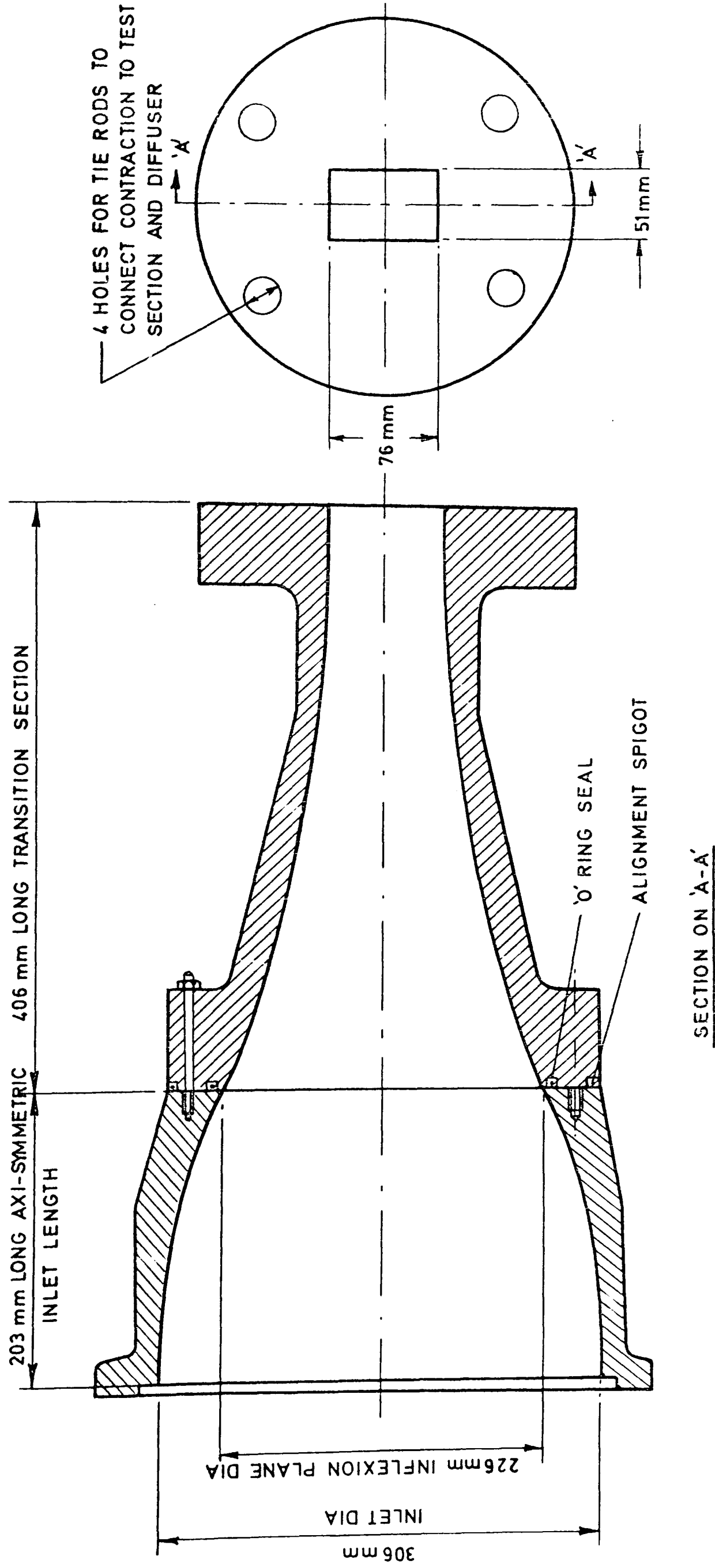


FIG.I.23 CONTRACTION ASSEMBLY

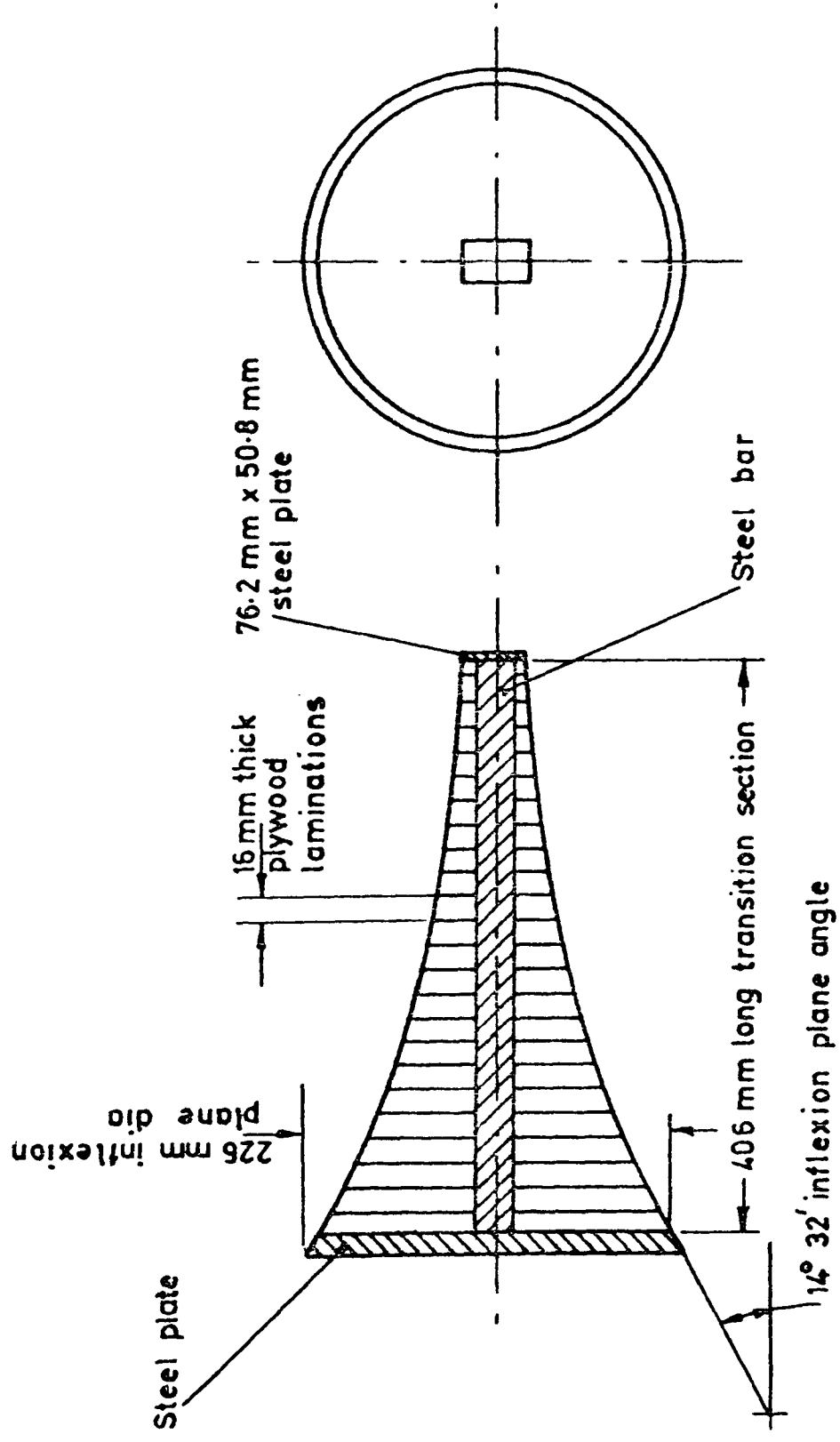


FIG. I.24 CONSTRUCTIONAL DETAILS OF TRANSITION SECTION PATTERN

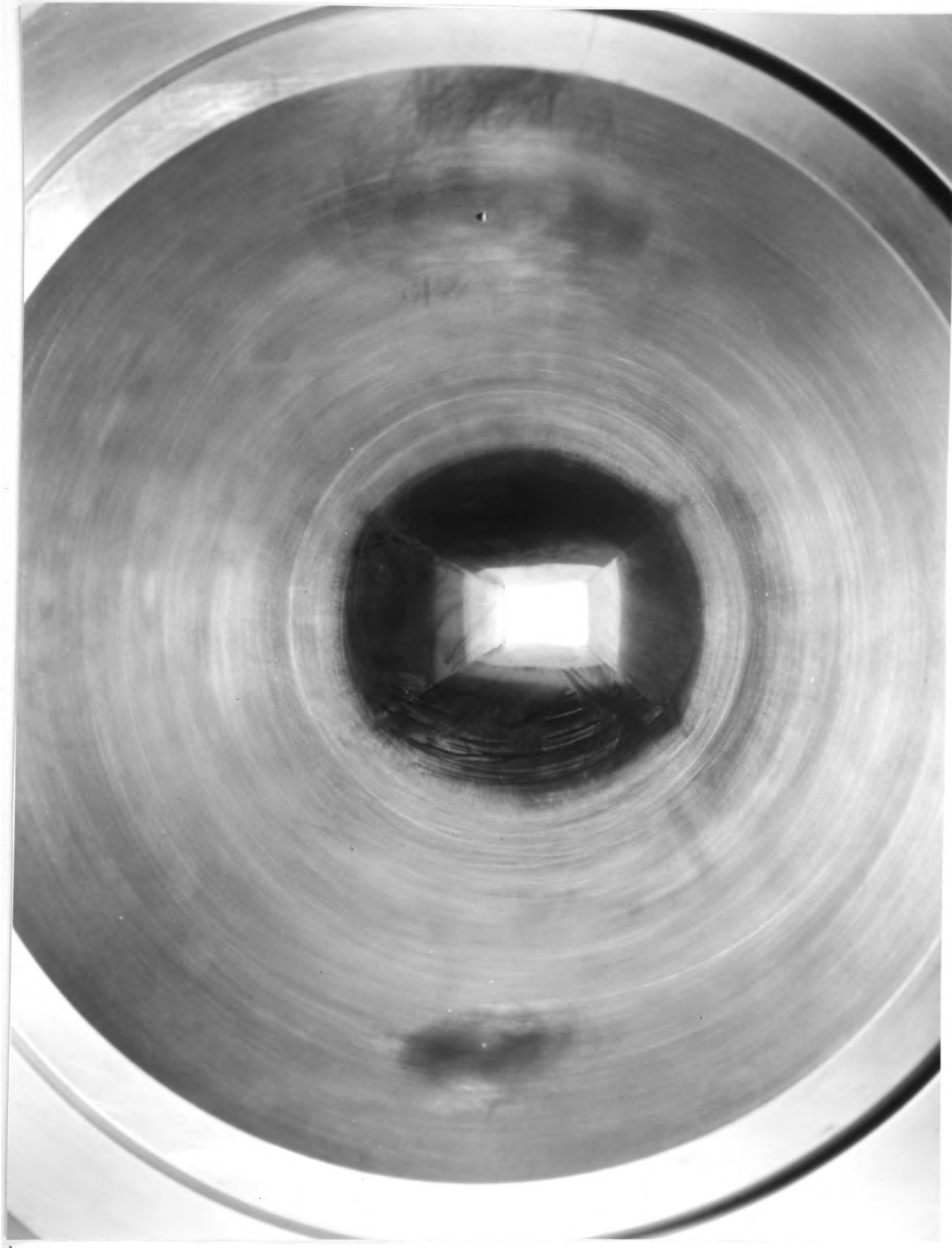


FIG. I. 25 INTERNAL VIEW OF CONTRACTION FROM INLET END

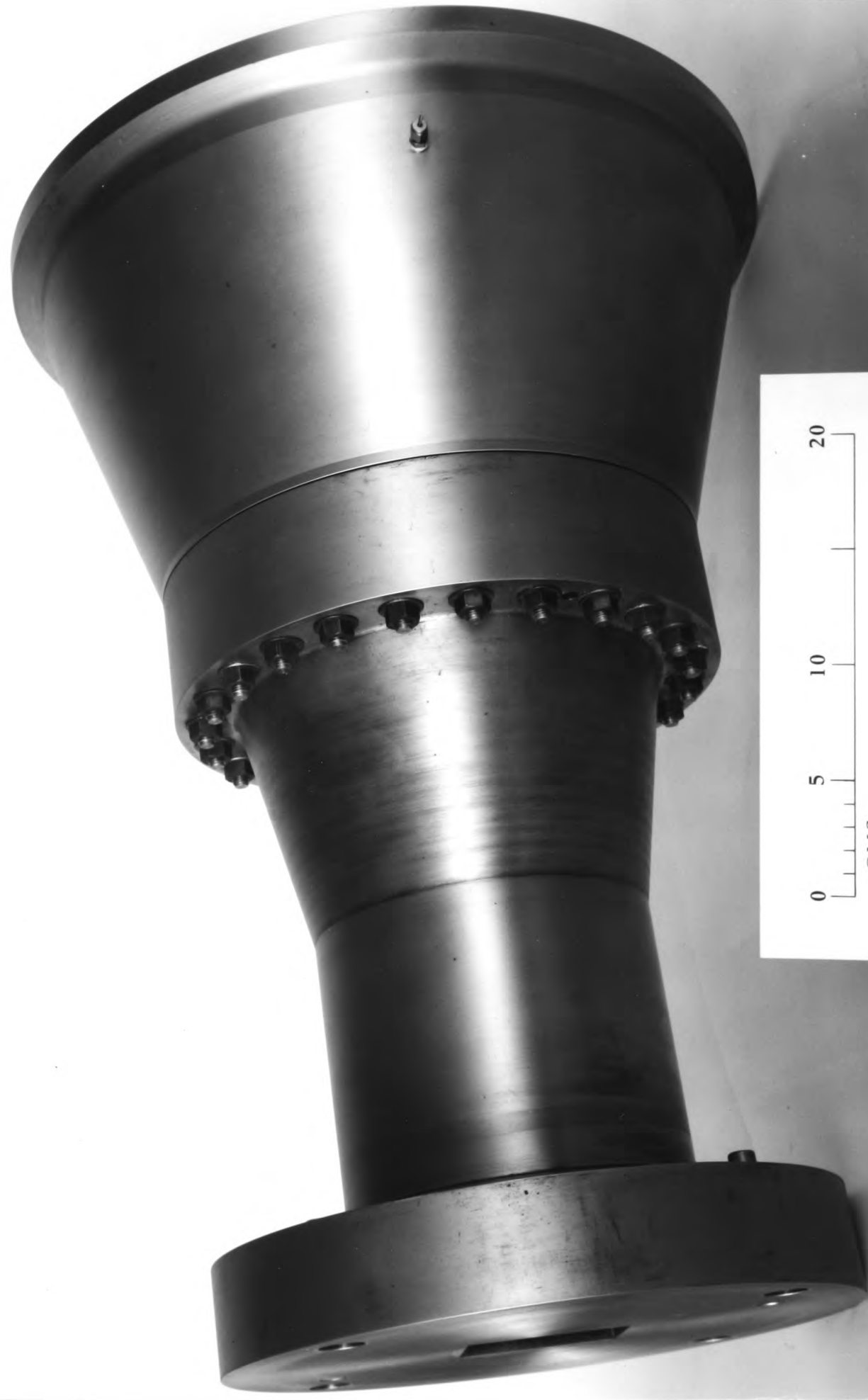


FIG. I. 26 OVERALL EXTERNAL VIEW OF ASSEMBLED CONTRACTION

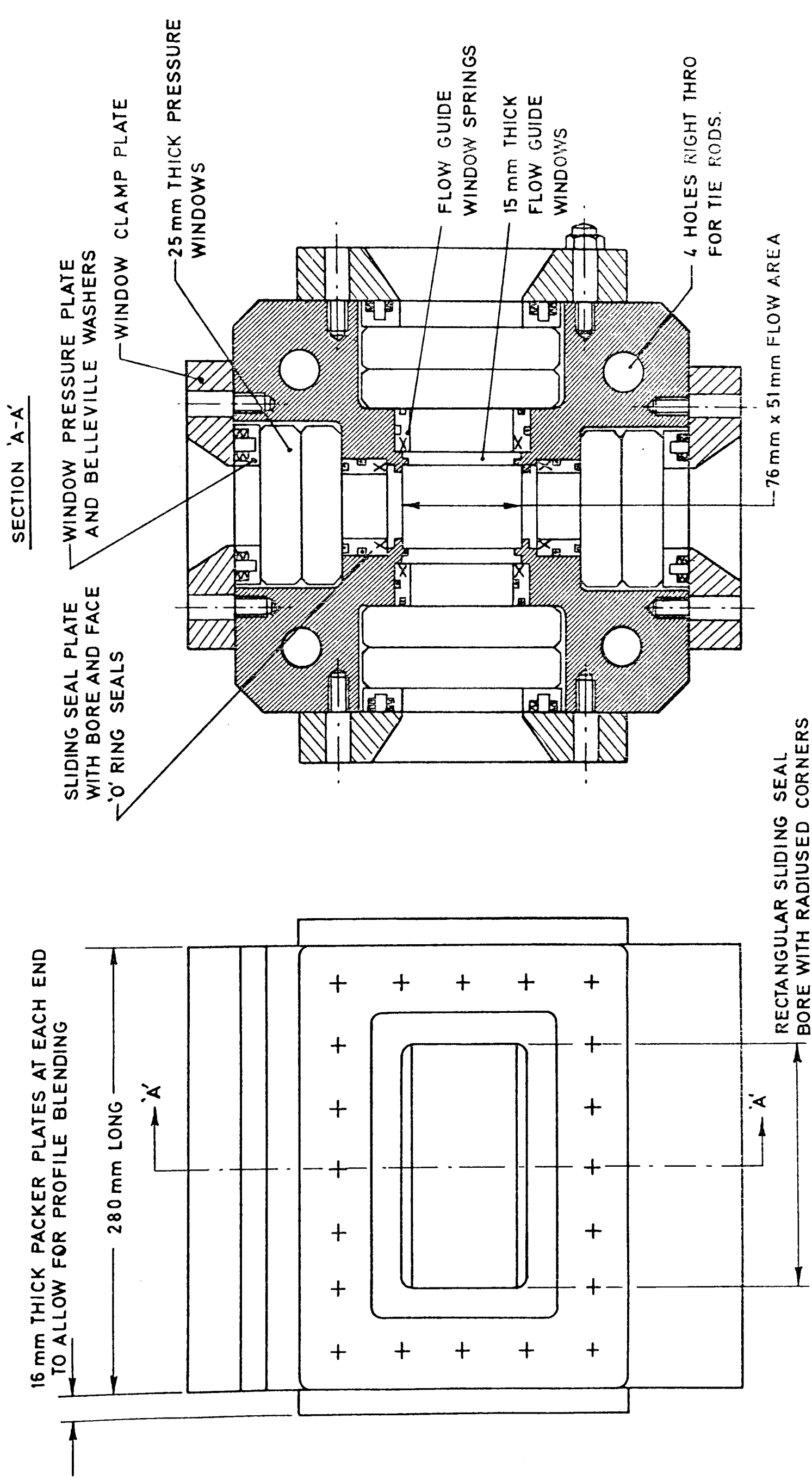


FIG.I.27 TEST SECTION ASSEMBLY

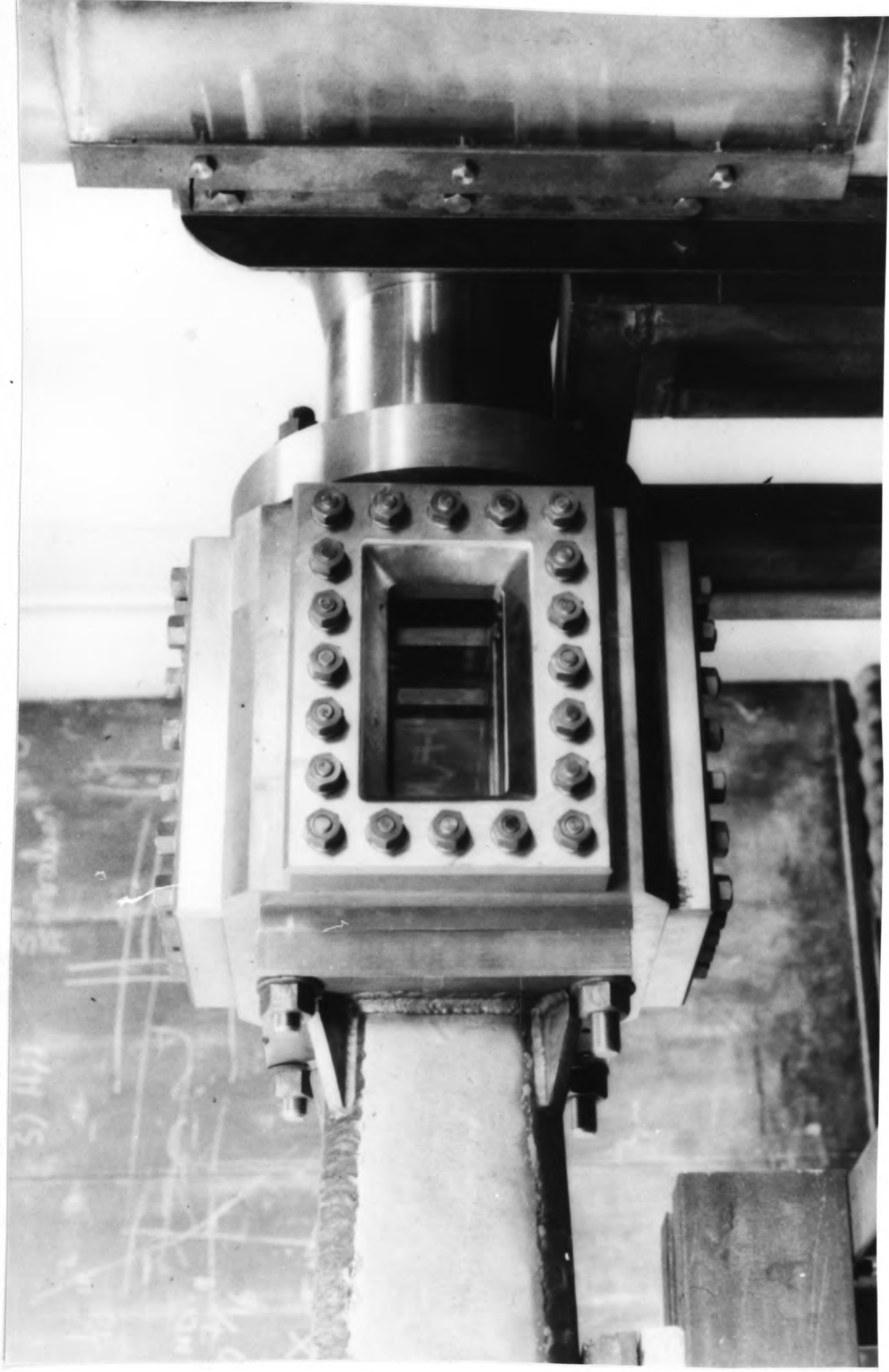


FIG.I.28 OBLIQUE SIDE ELEVATION OF TEST SECTION MOUNTED
BETWEEN CONTRACTION AND DIFFUSER

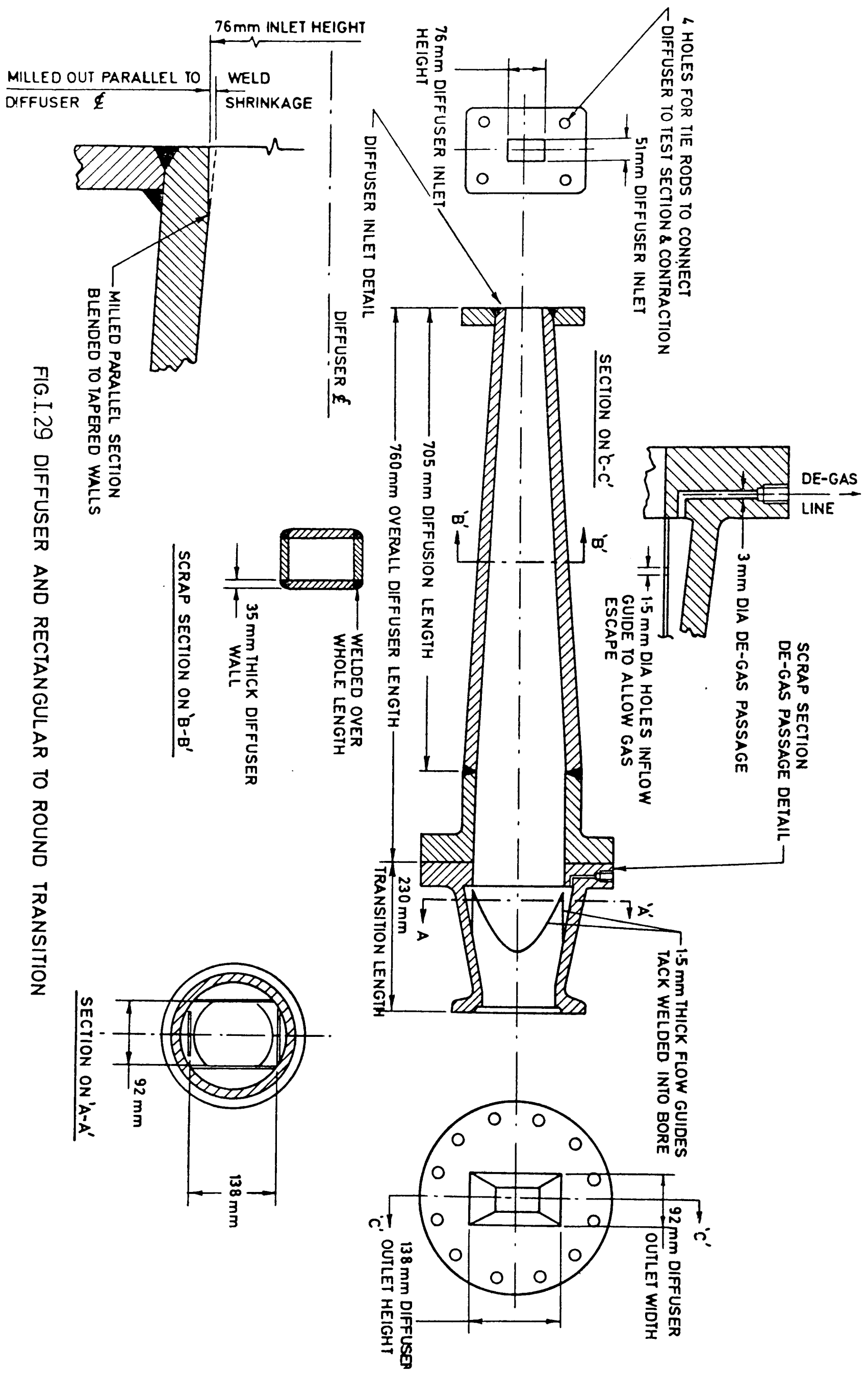


FIG.1.29 DIFFUSER AND RECTANGULAR TO ROUND TRANSITION

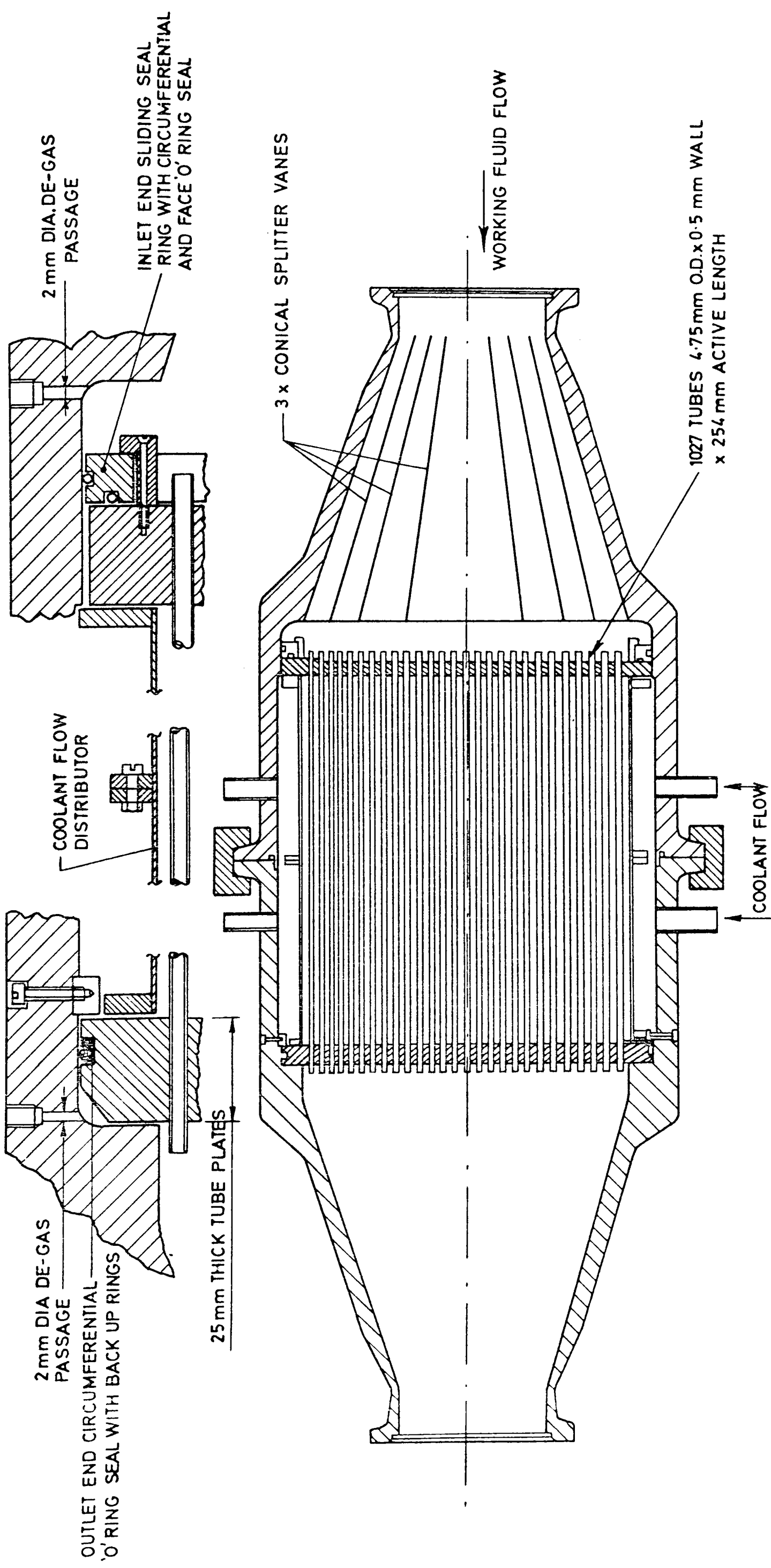


FIG.I.30 HEAT EXCHANGER ASSEMBLY AND DETAILS

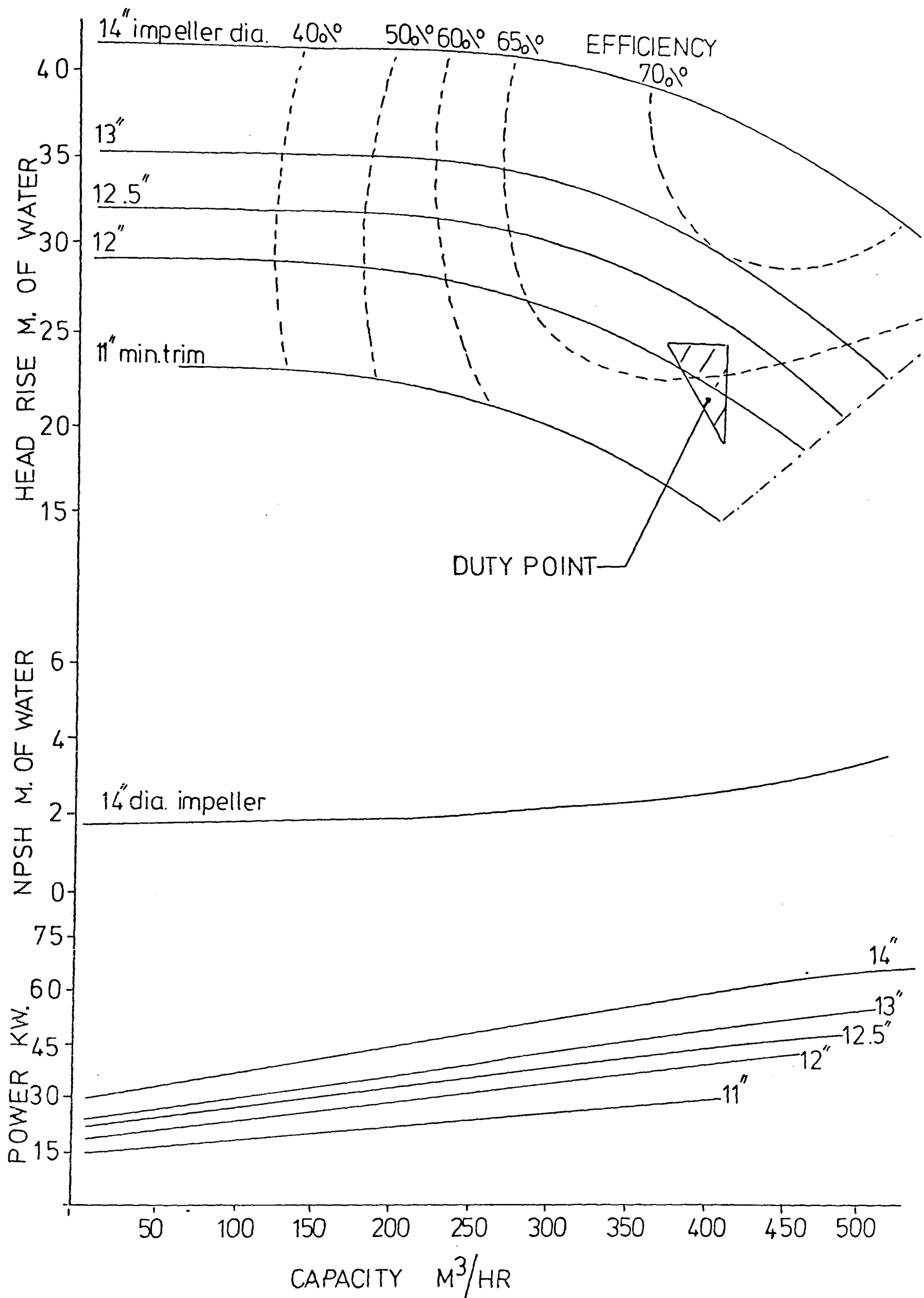


FIG.I.31 CIRCULATING PUMP CHARACTERISTIC
 AT 1500 RPM ROTATIONAL SPEED

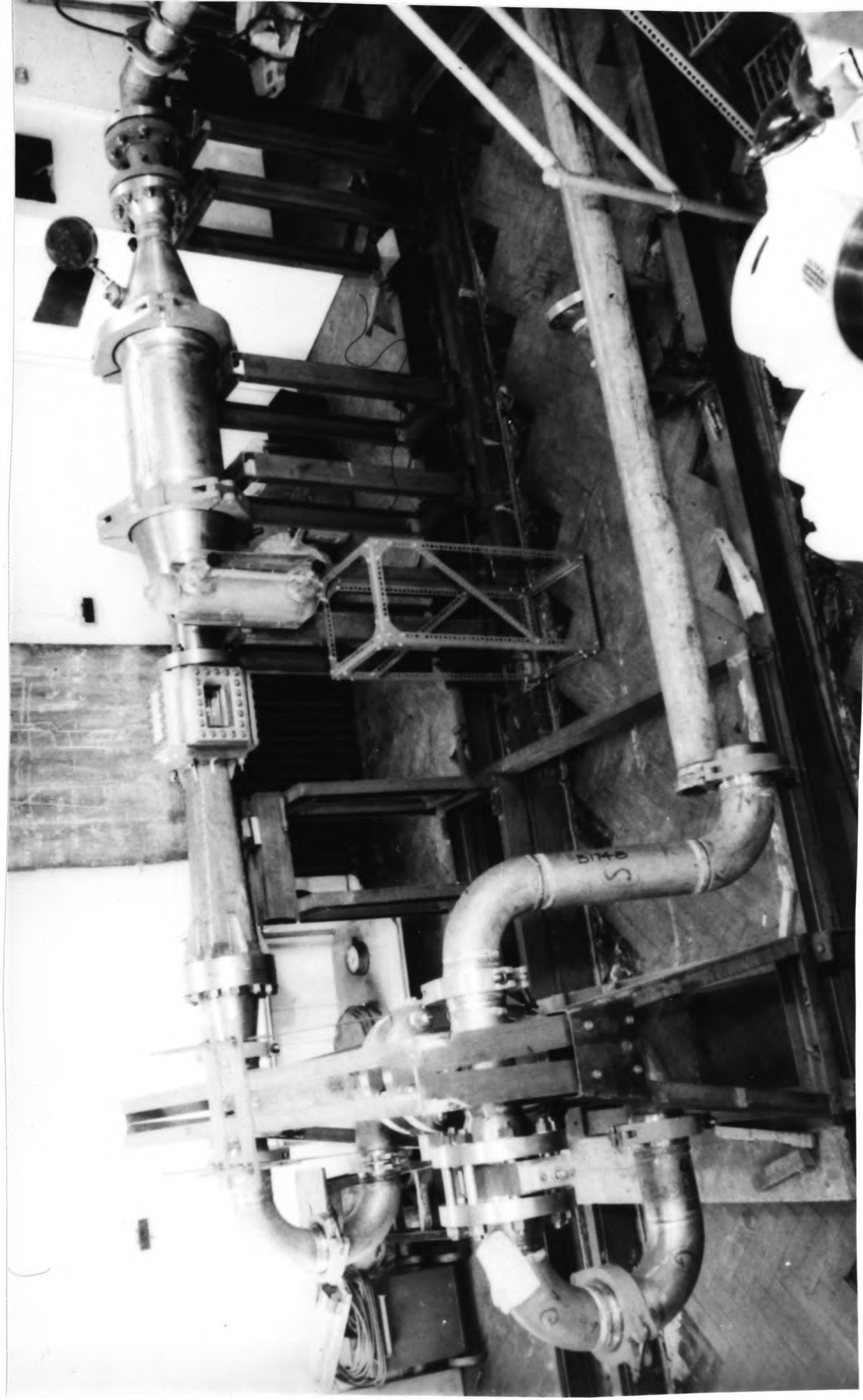


FIG. I. 32 OBLIQUE ELIVATION OF TUNNEL DURING ERECTION

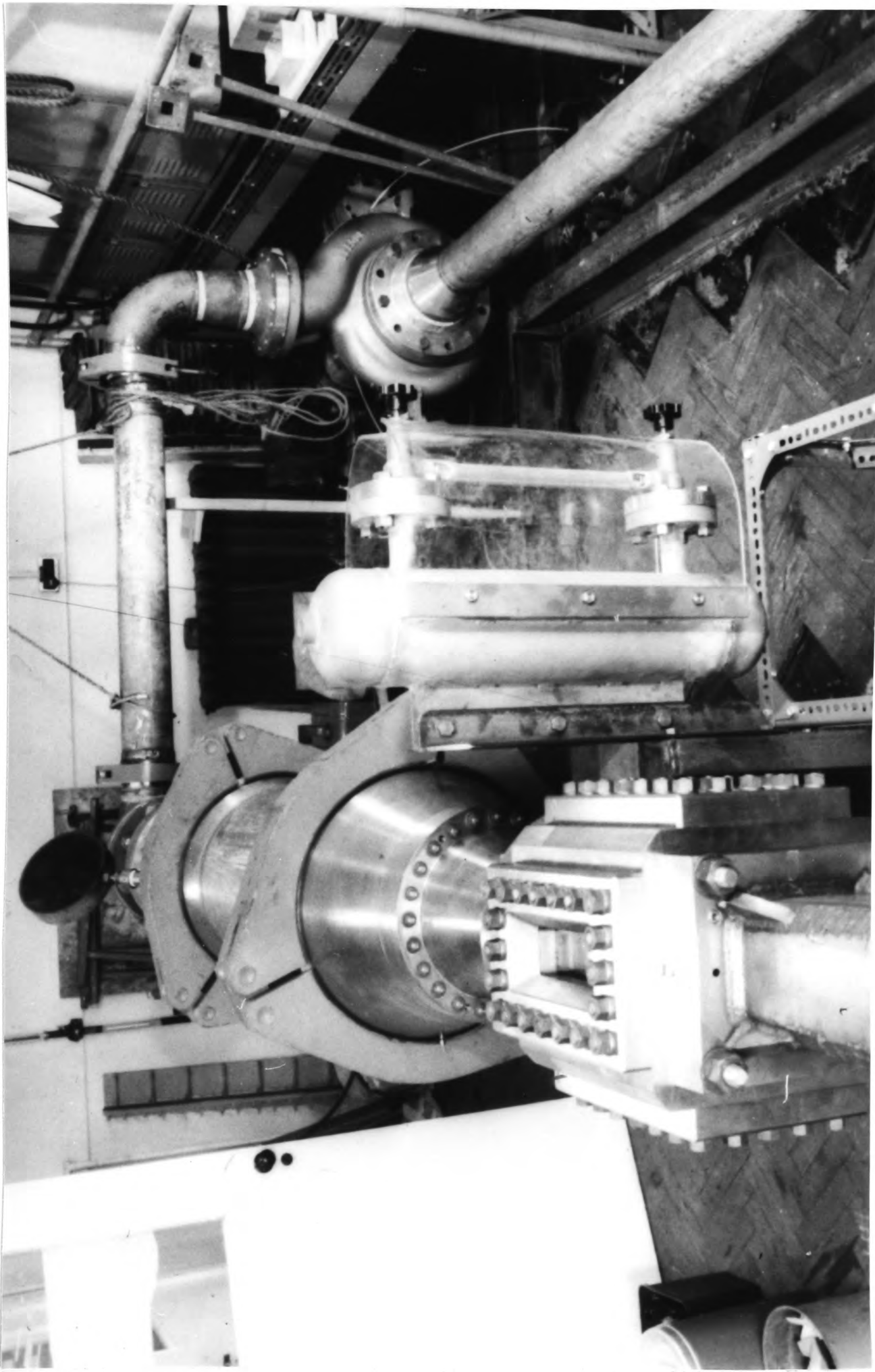


FIG. I. 33 PLAN VIEW OF TUNNEL DURING ERECTION

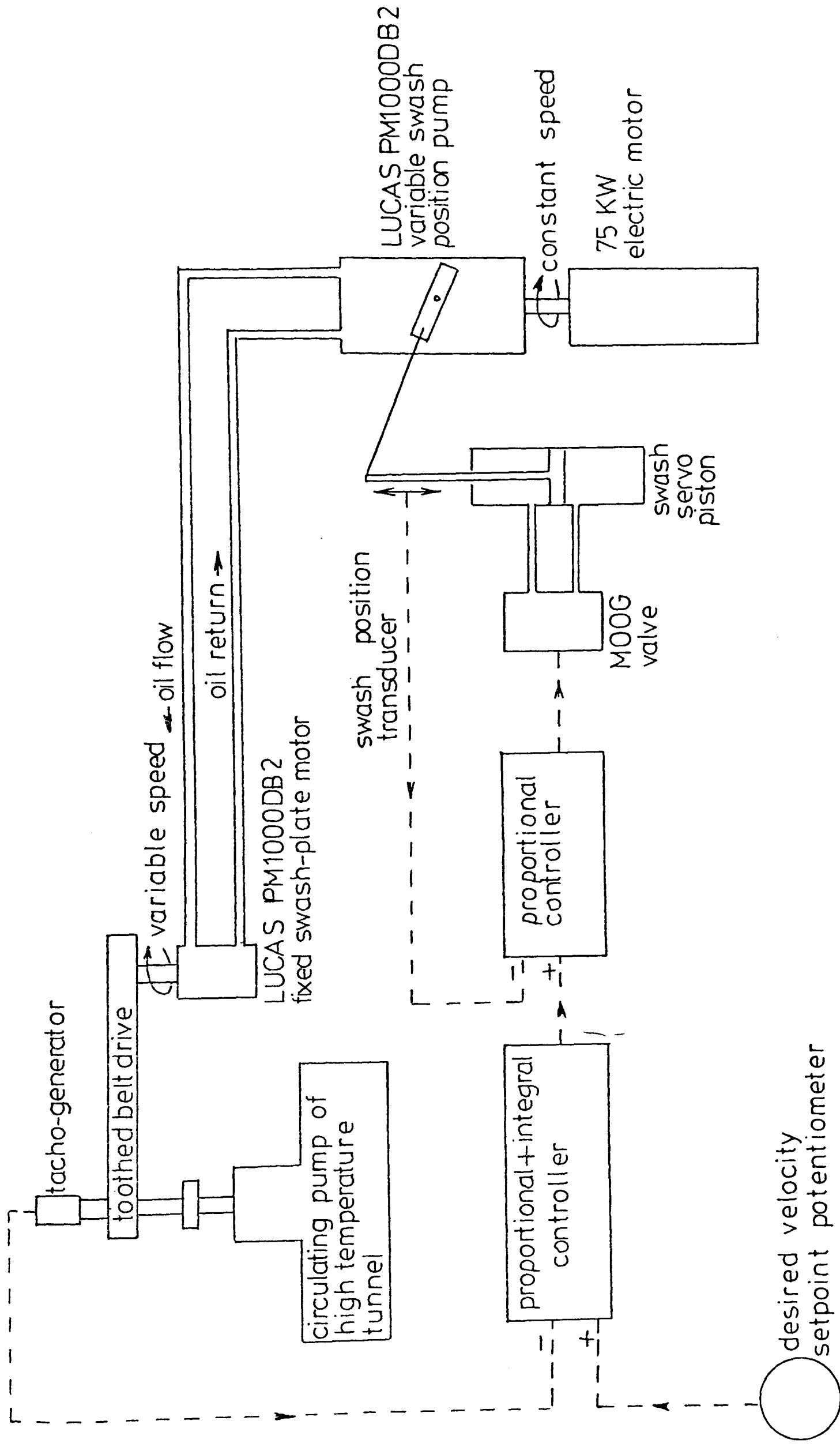


FIG I.34 SCHEMATIC OF HYDROSTATIC DRIVE

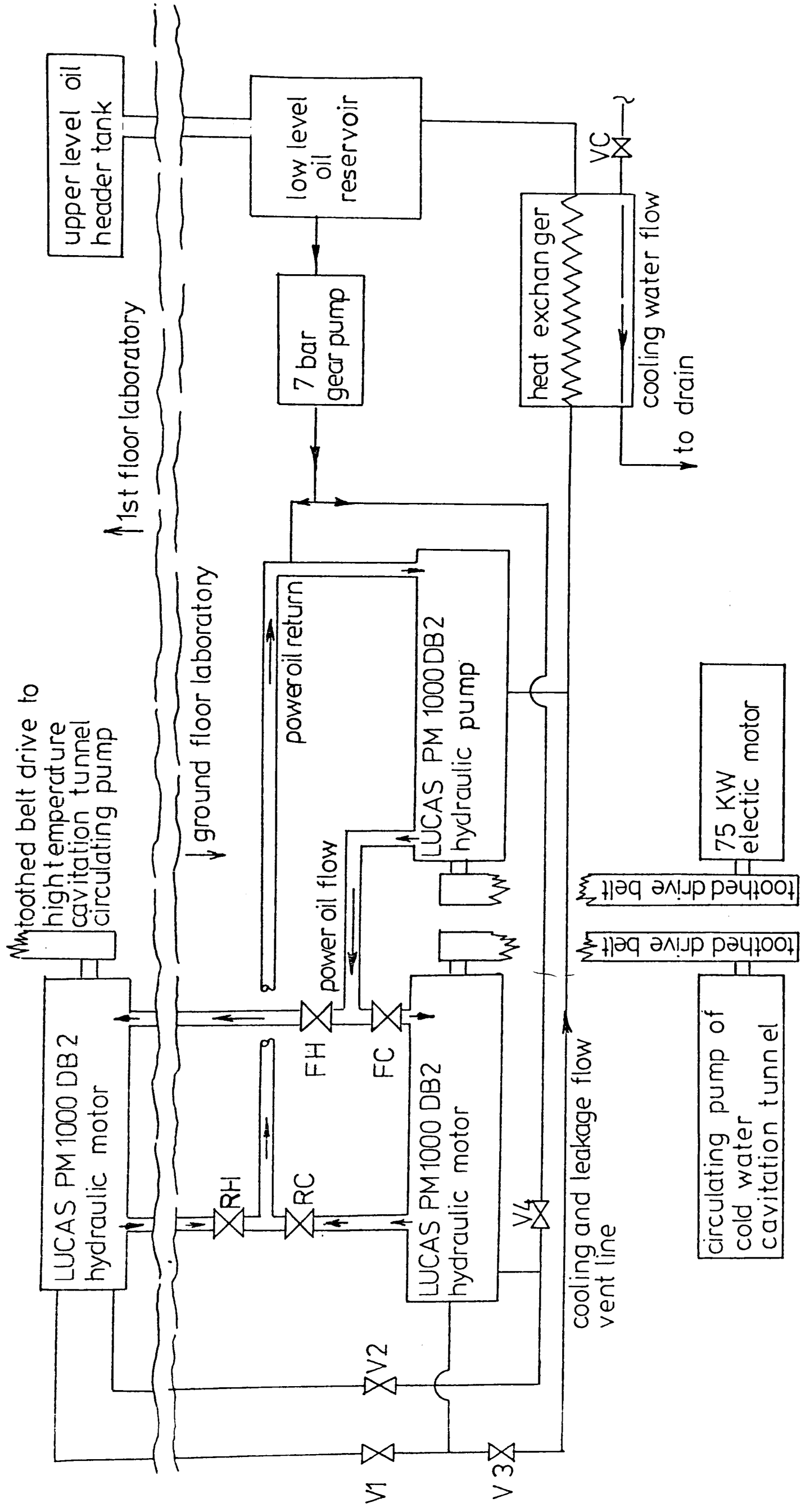


FIG. I.35 SCHEMATIC OF HYDROSTATIC DRIVE SHOWING OIL FLOW CHANGEOVER ARRANGEMENT

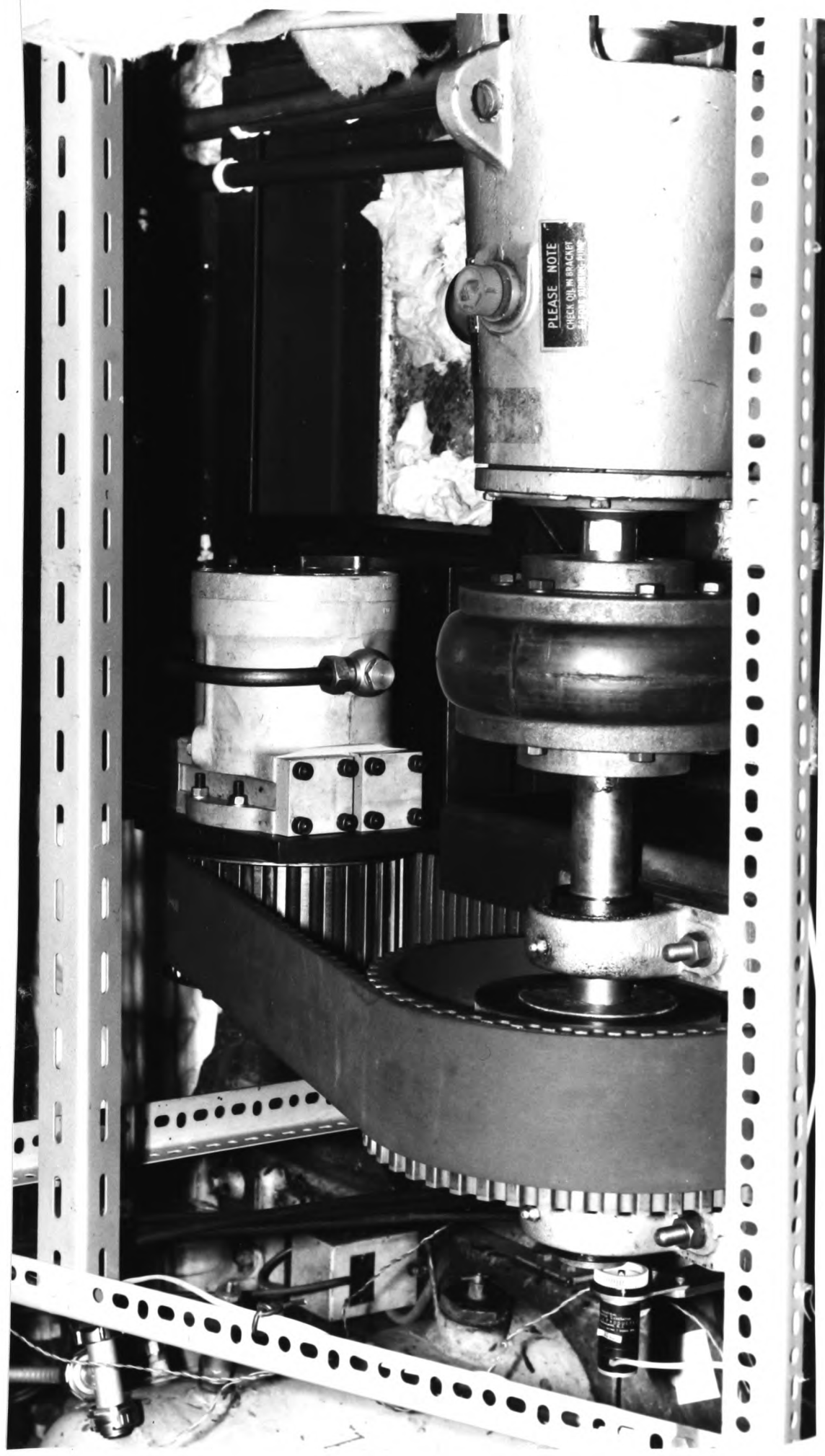


FIG. I. 36 CIRCULATING PUMP DRIVE DETAILS

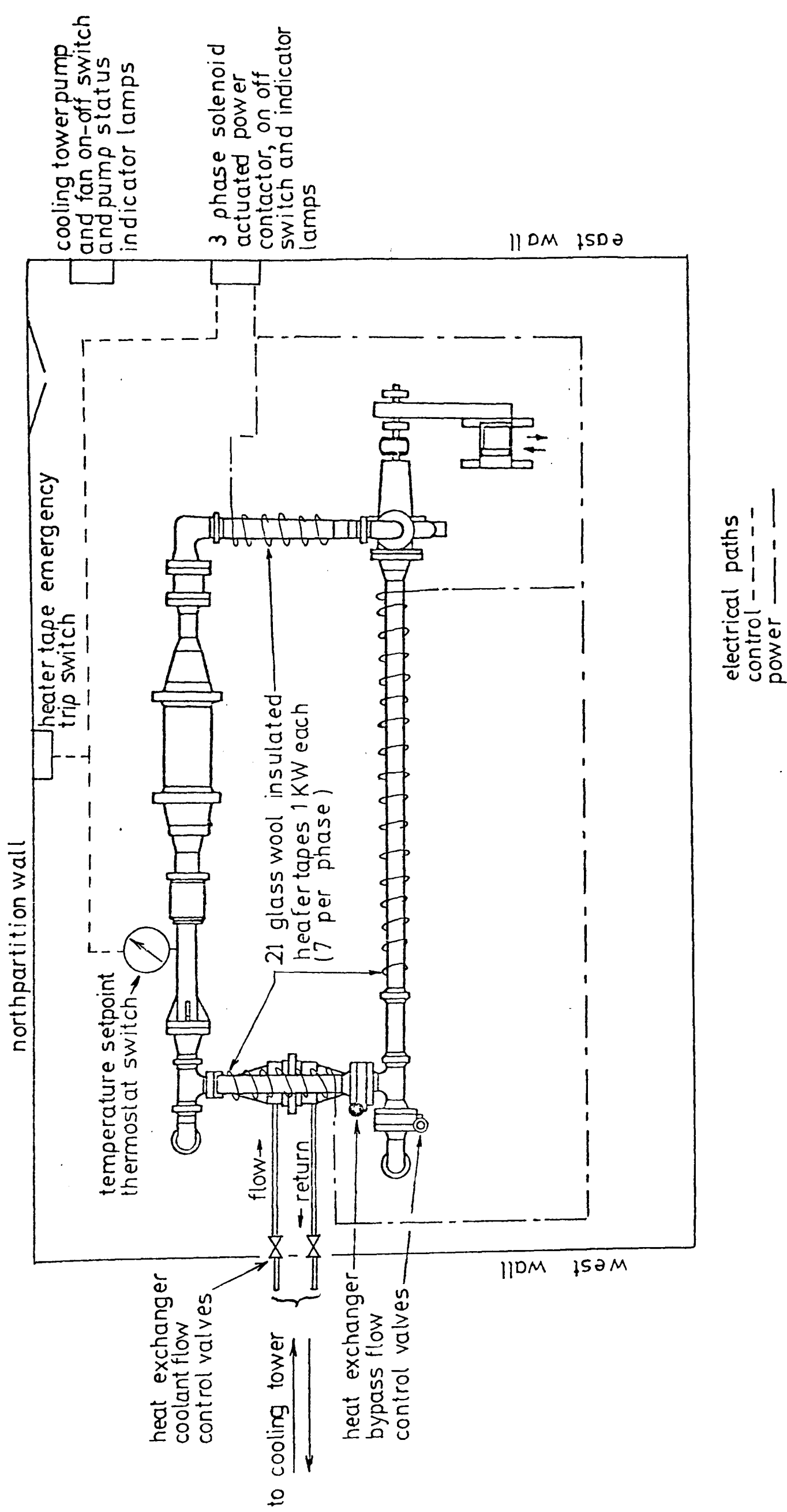


FIG.I.37 SCHEMATIC PLAN OF CAVITATION LABORATORY 1ST FLOOR LEVEL
SHOWING TEMPERATURE CONTROL ARRANGEMENTS

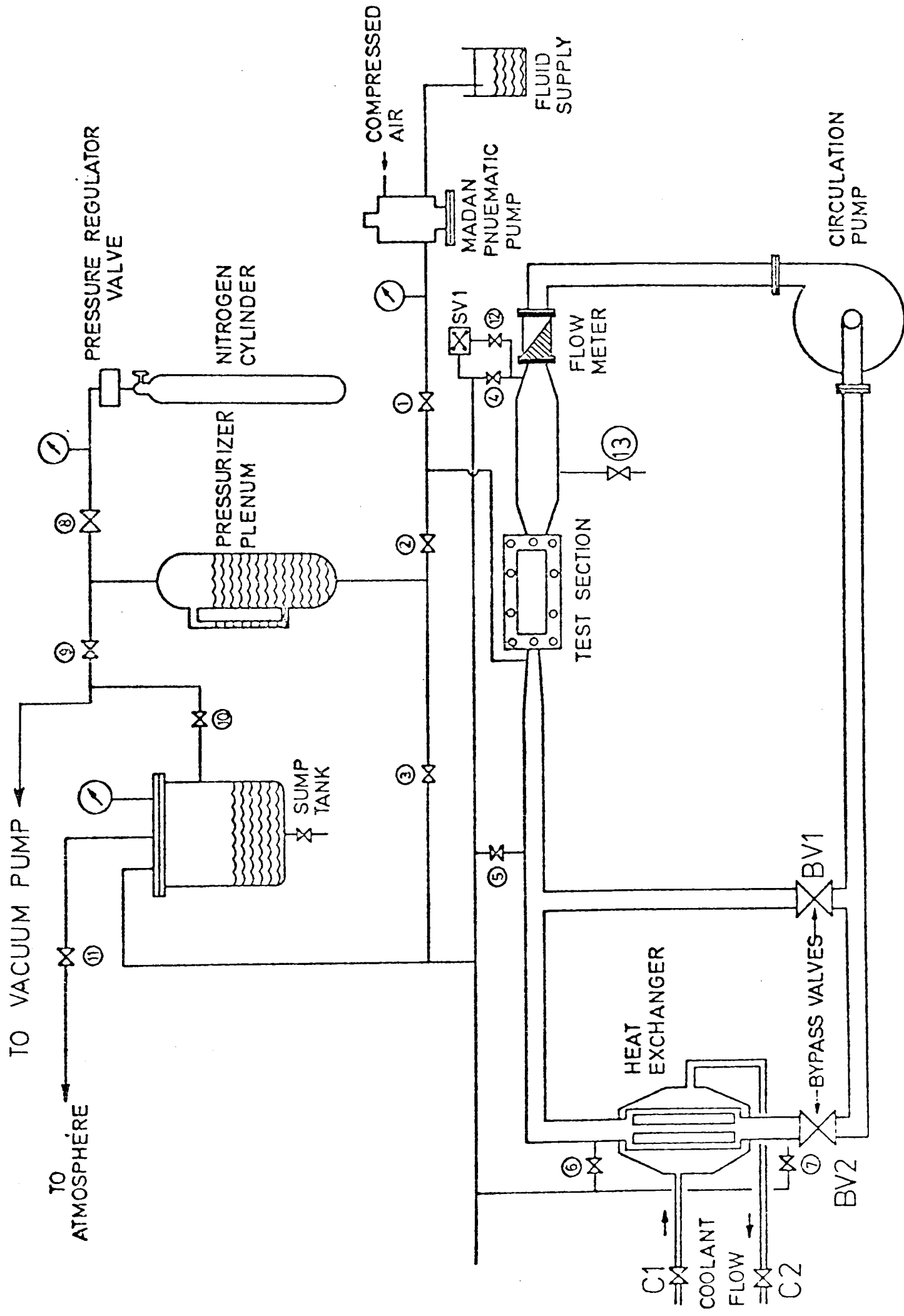


FIG.I.38 SCHEMATIC OF PRESSURE AND VOLUME CONTROL SYSTEM

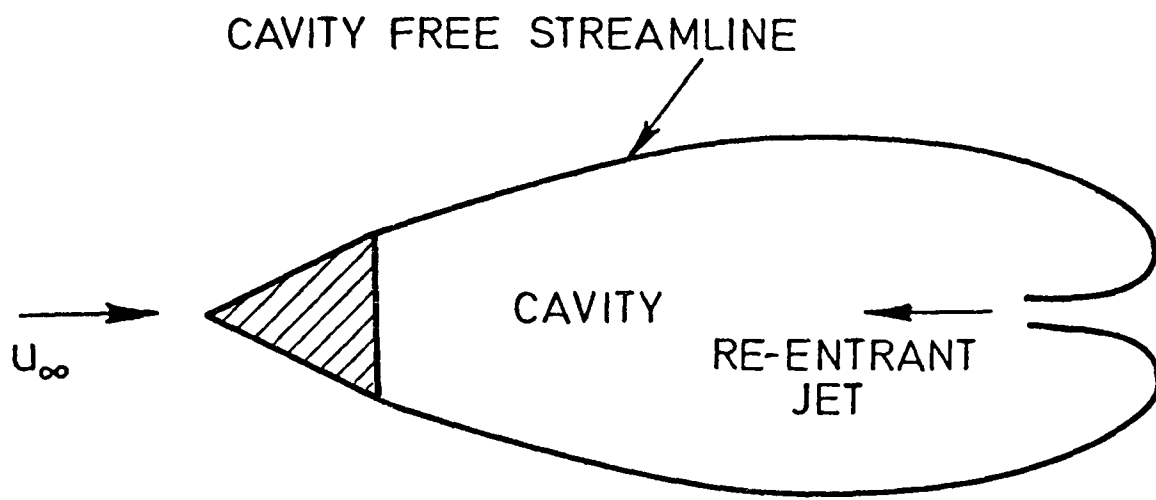


Fig. I.39(a)

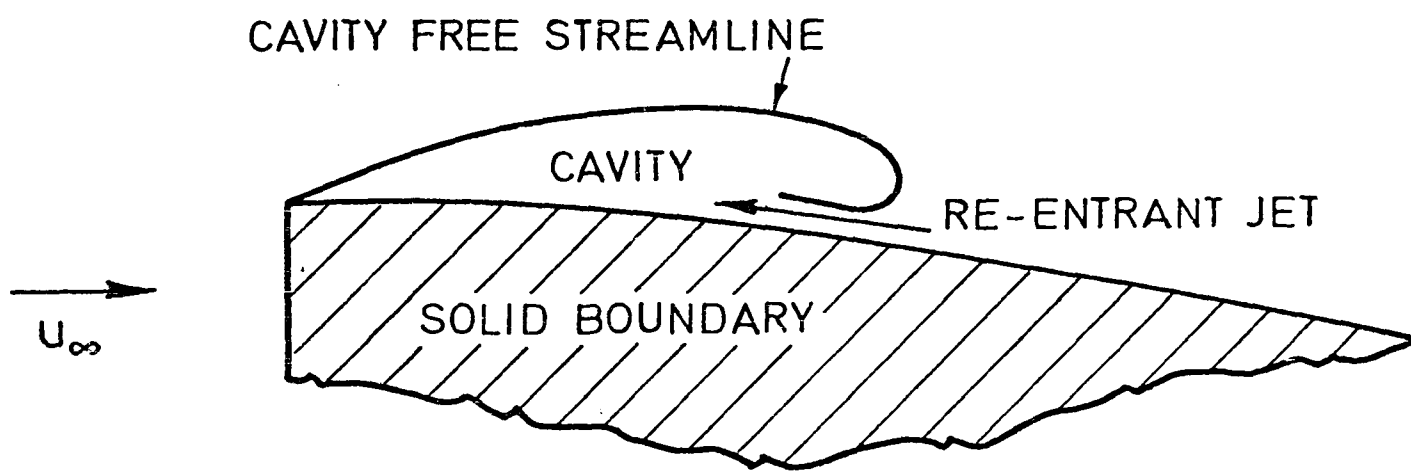
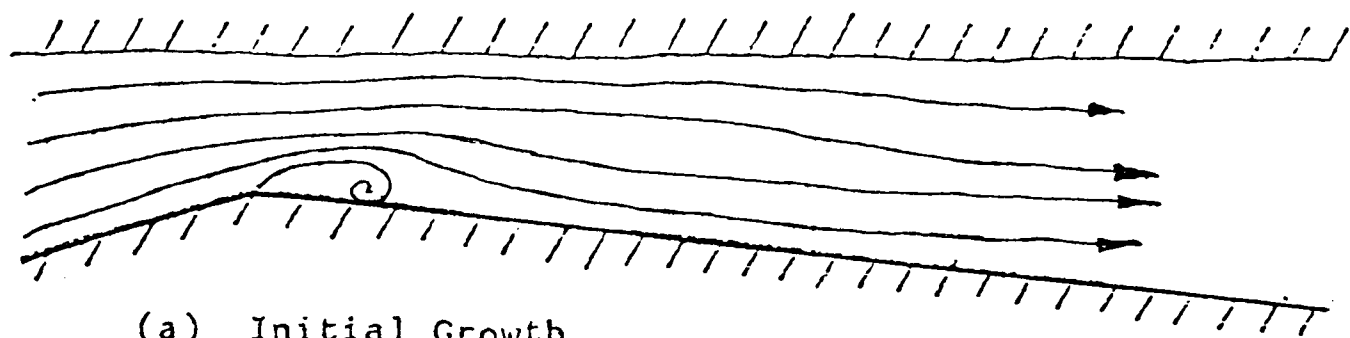
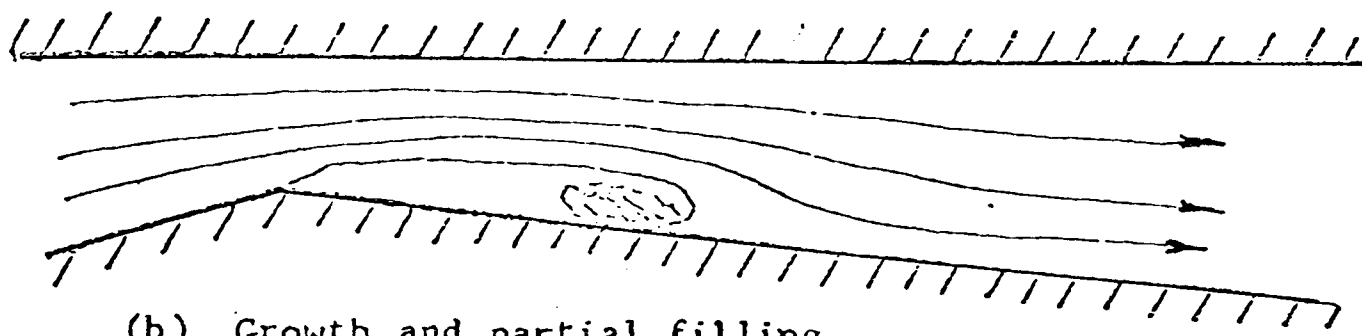


Fig. I. 39(b)

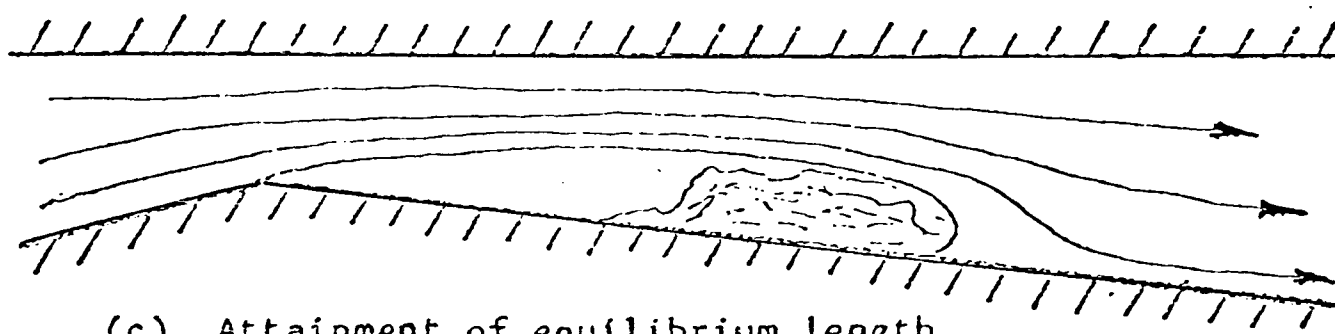
FIG. I. 39 SCHEMATIC OF RE-ENTRANT JET MOTION
FOR CAVITIES TERMINATING
a. IN THE FREE STREAM
b. ADJACENT TO A WALL



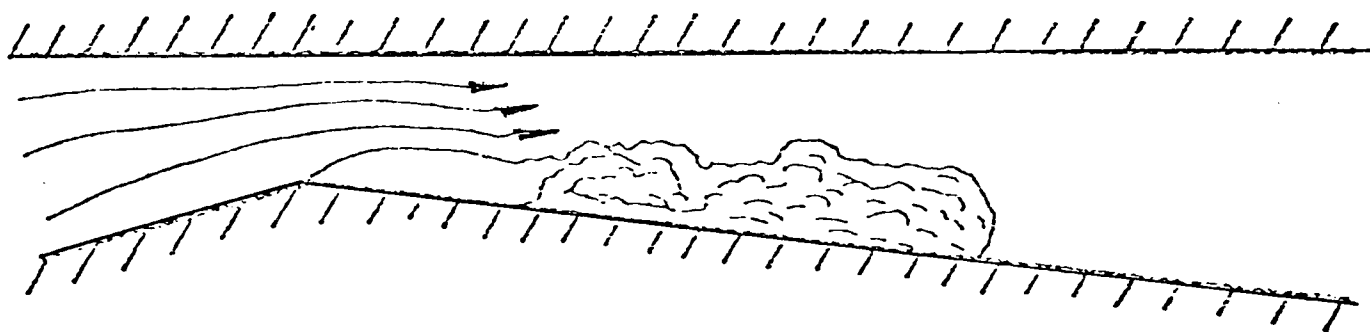
(a) Initial Growth



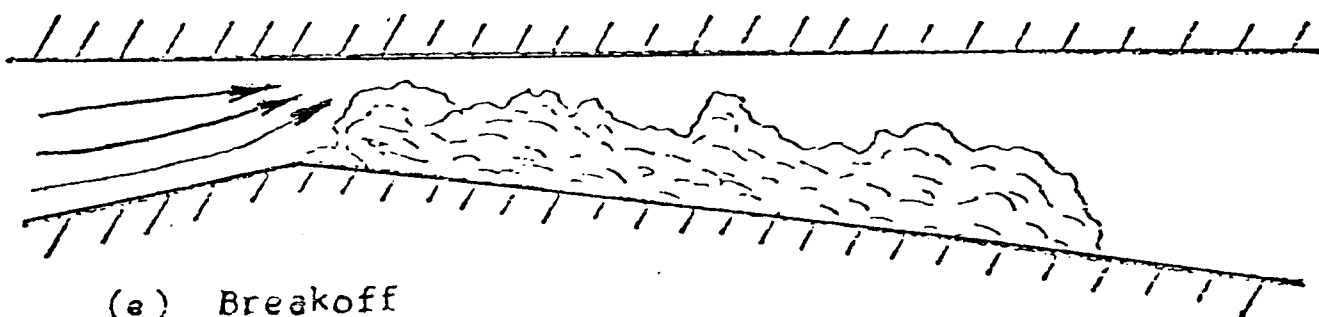
(b) Growth and partial filling



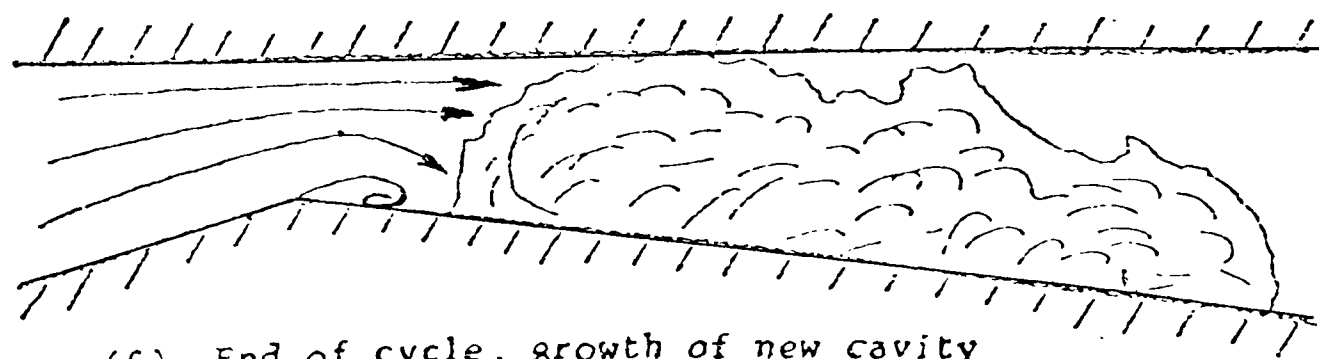
(c) Attainment of equilibrium length



(d) Filling



(e) Breakoff



(f) End of cycle, growth of new cavity

FIG. I.40 GROWTH, DEVELOPMENT AND BREAKOFF OF CYCLIC CAVITY. AFTER FURNESS (38)

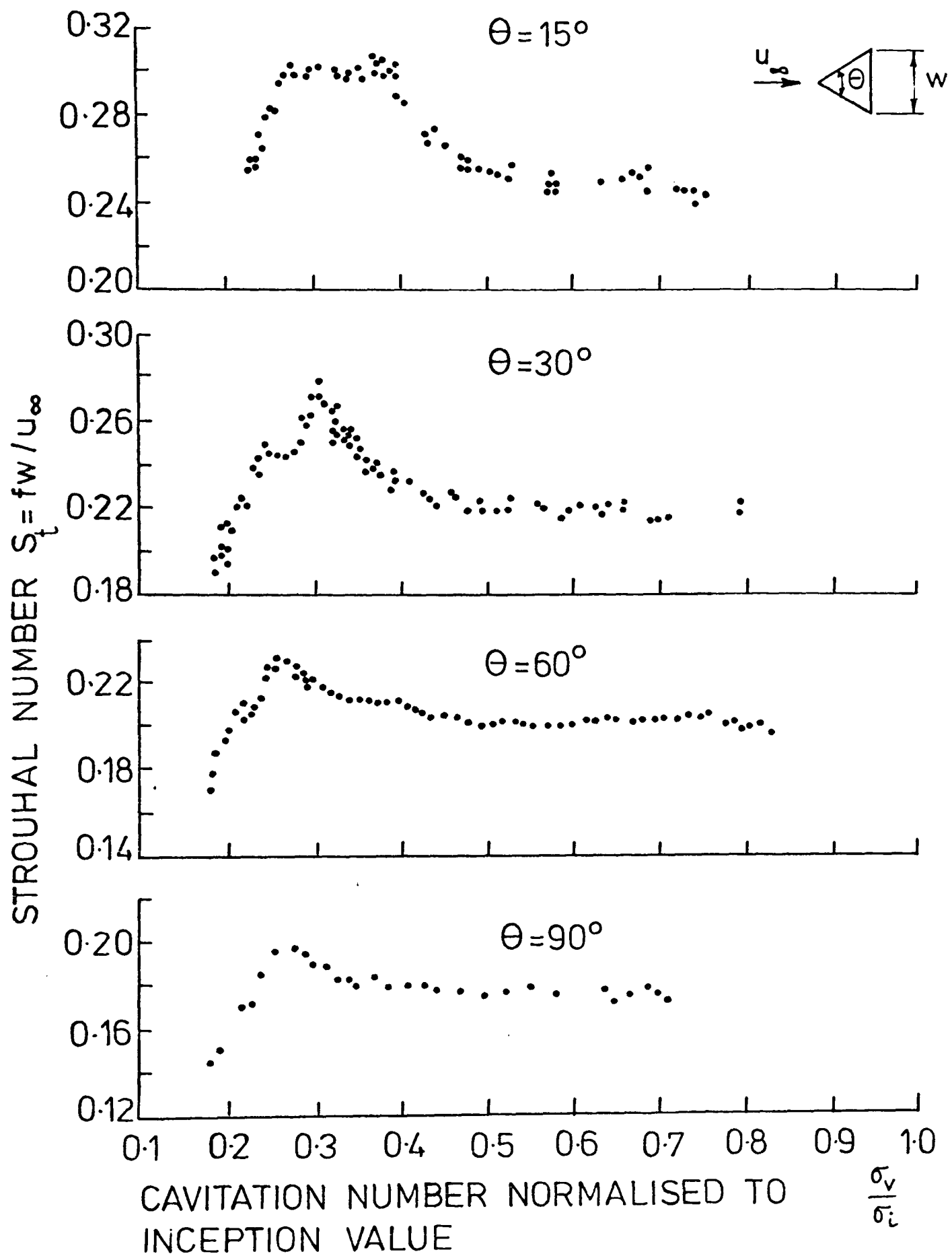


FIG. I. 41 VORTEX SHEDDING STROUHAL NUMBER
VERSUS NORMALIZED CAVITATION NUMBER
FOR TWO-DIMENSIONAL WEDGES OF VARIOUS
NOSE ANGLES. AFTER YOUNG AND HOLL (44)

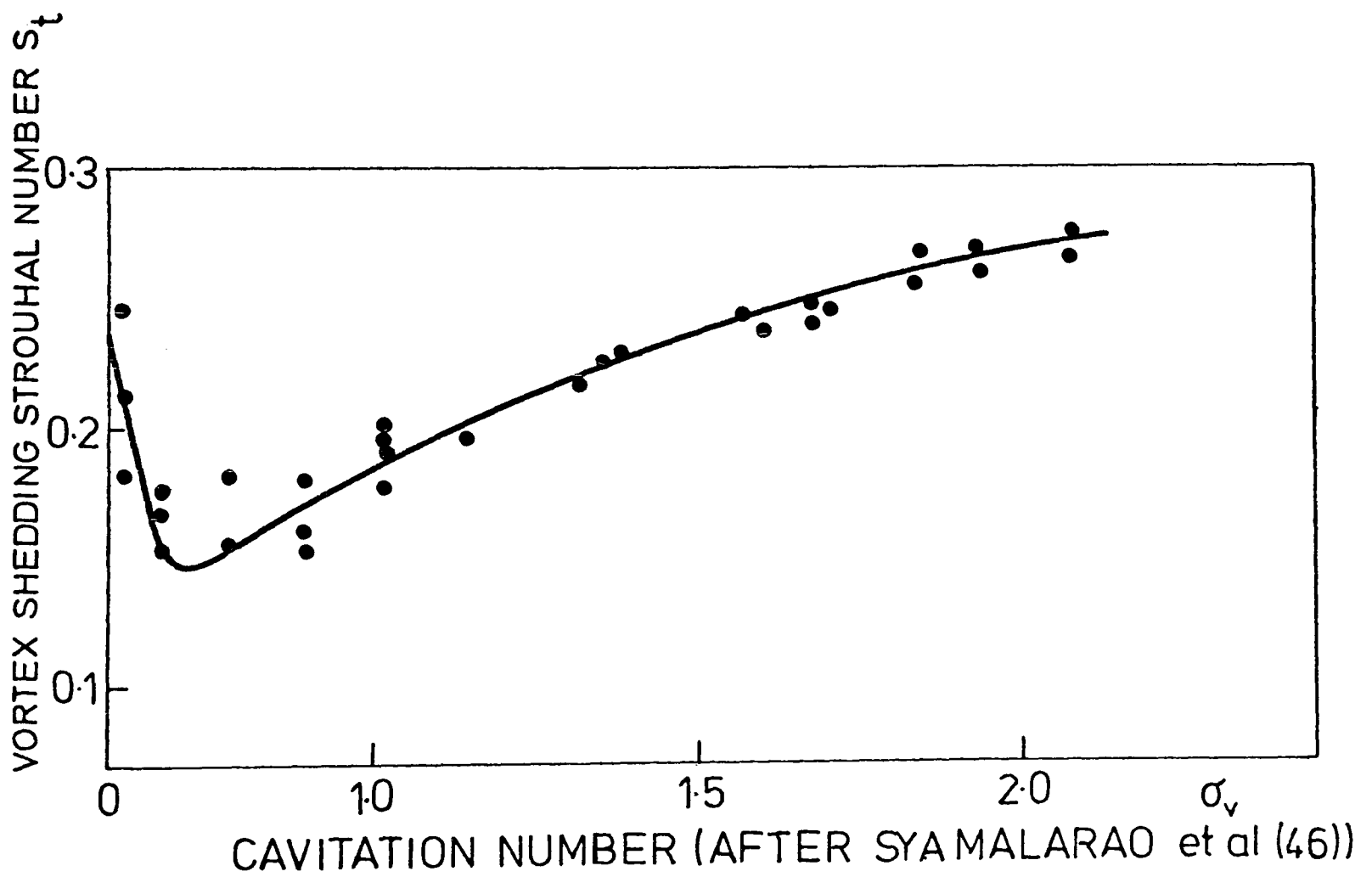
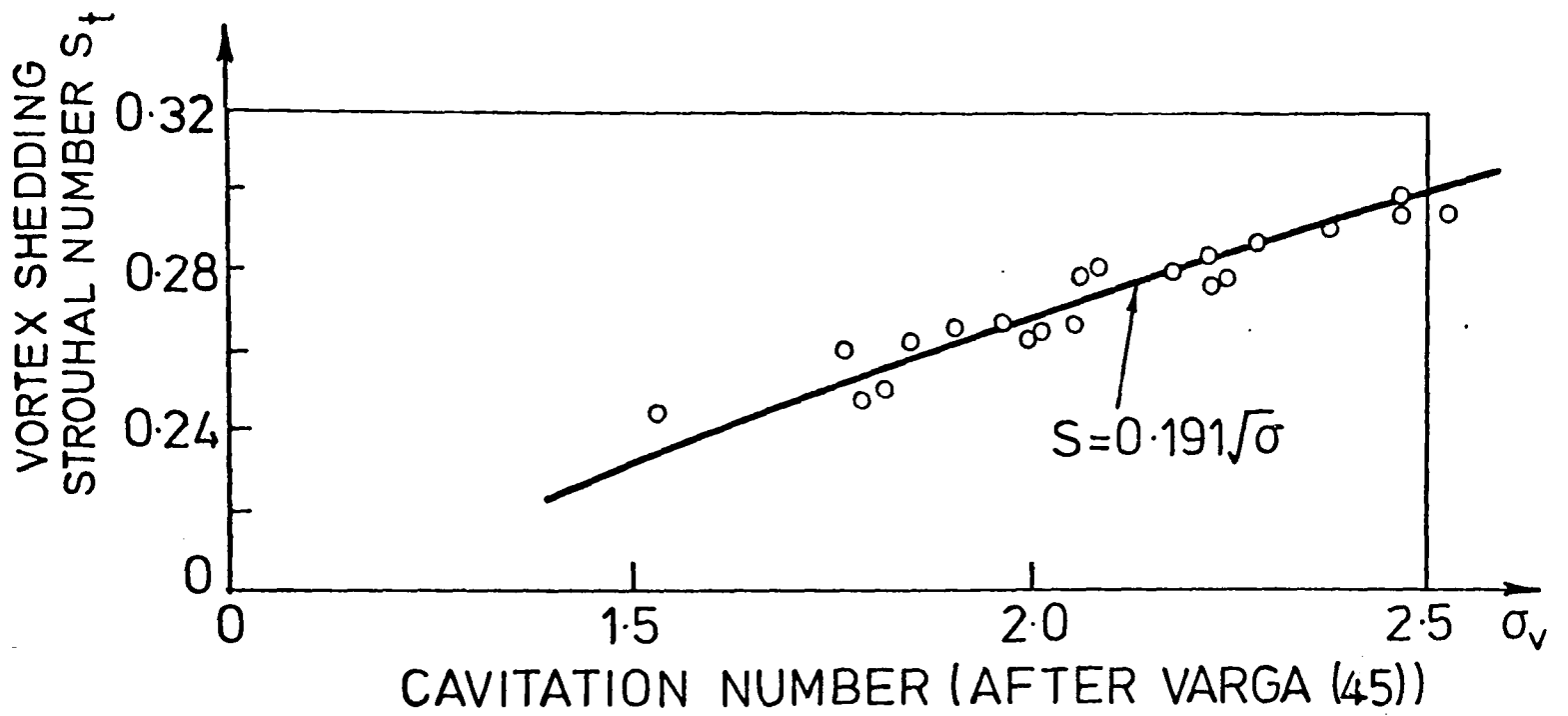


FIG. I. 42 VARIATION OF VORTEX SHEDDING STROUHAL NUMBER WITH CAVITATION NUMBER FOR A CIRCULAR CYLINDER. AFTER VARGA(45) AND SYAMALA RAO(46)

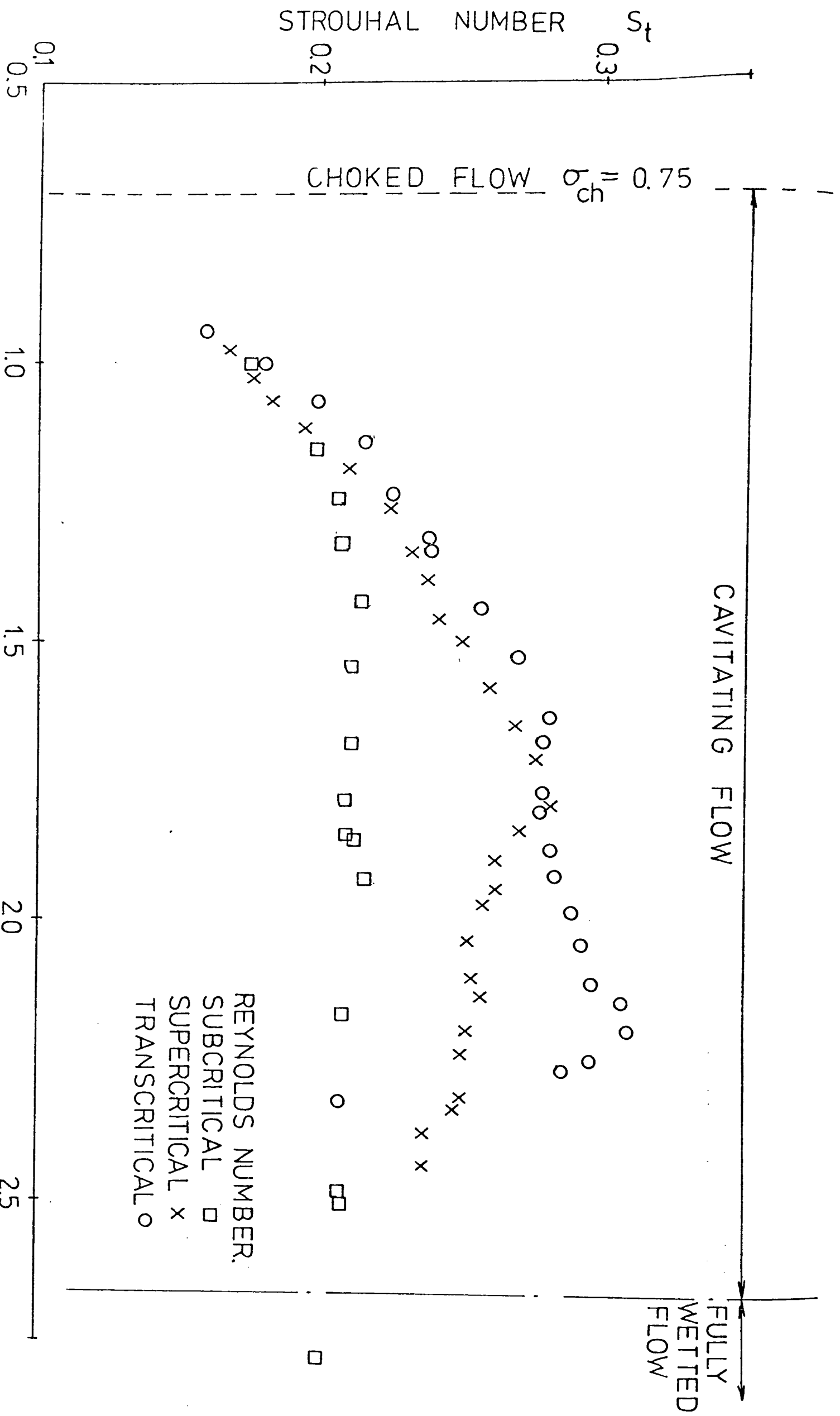
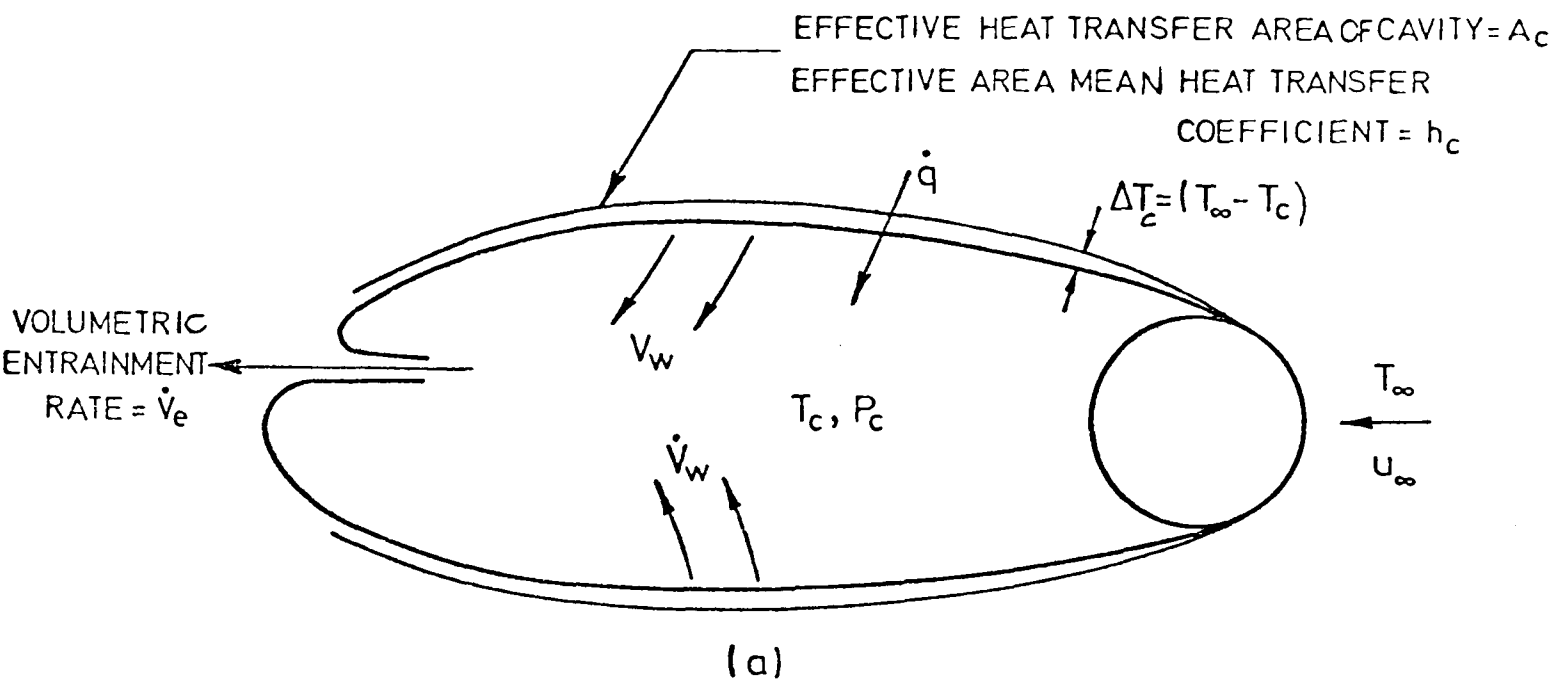


FIG. 1.43 CAVITATION NUMBER σ_v VERSUS CAVITATION NUMBER FOR A CAVITATING CYLINDER AT SUBCRITICAL AND SUPERCRITICAL REYNOLDS NUMBERS. AFTER MUSSAREDD(49)

ENTRAINMENT THEORY CAVITY SCHEMATIC.



β -FACTOR CAVITY SCHEMATIC.

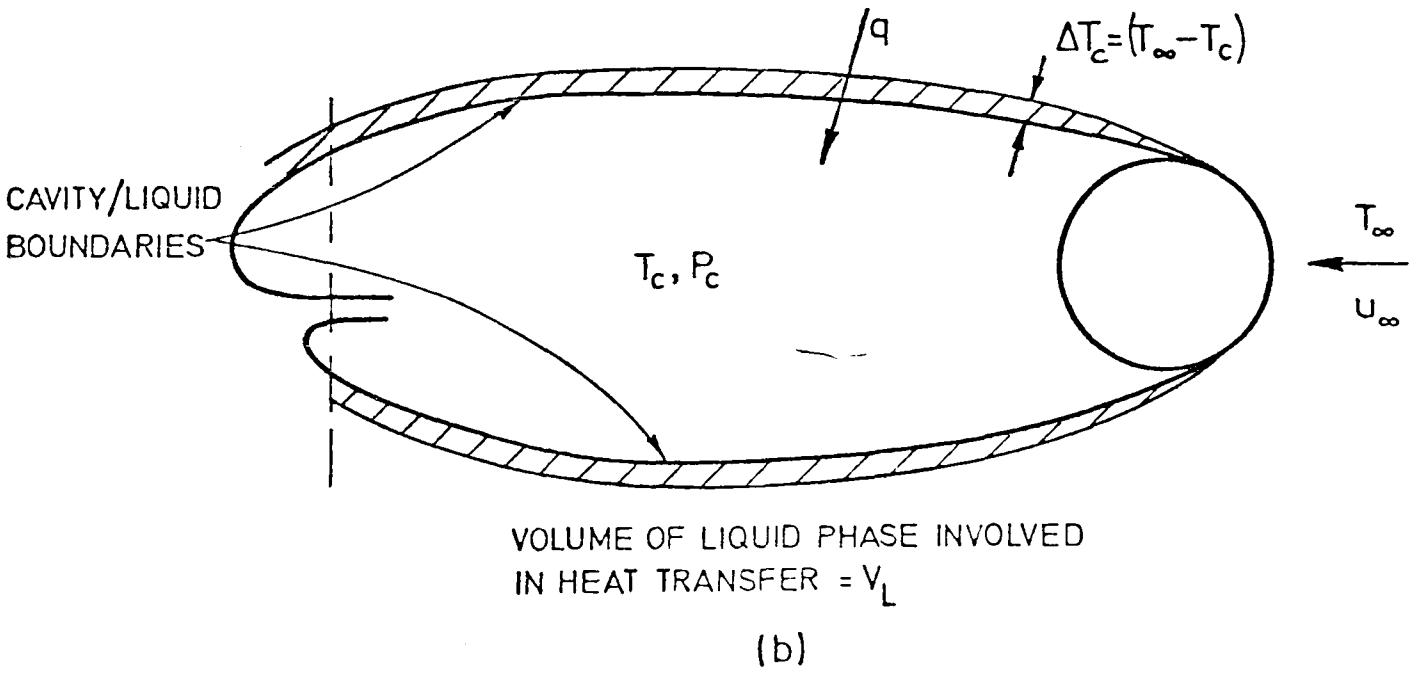
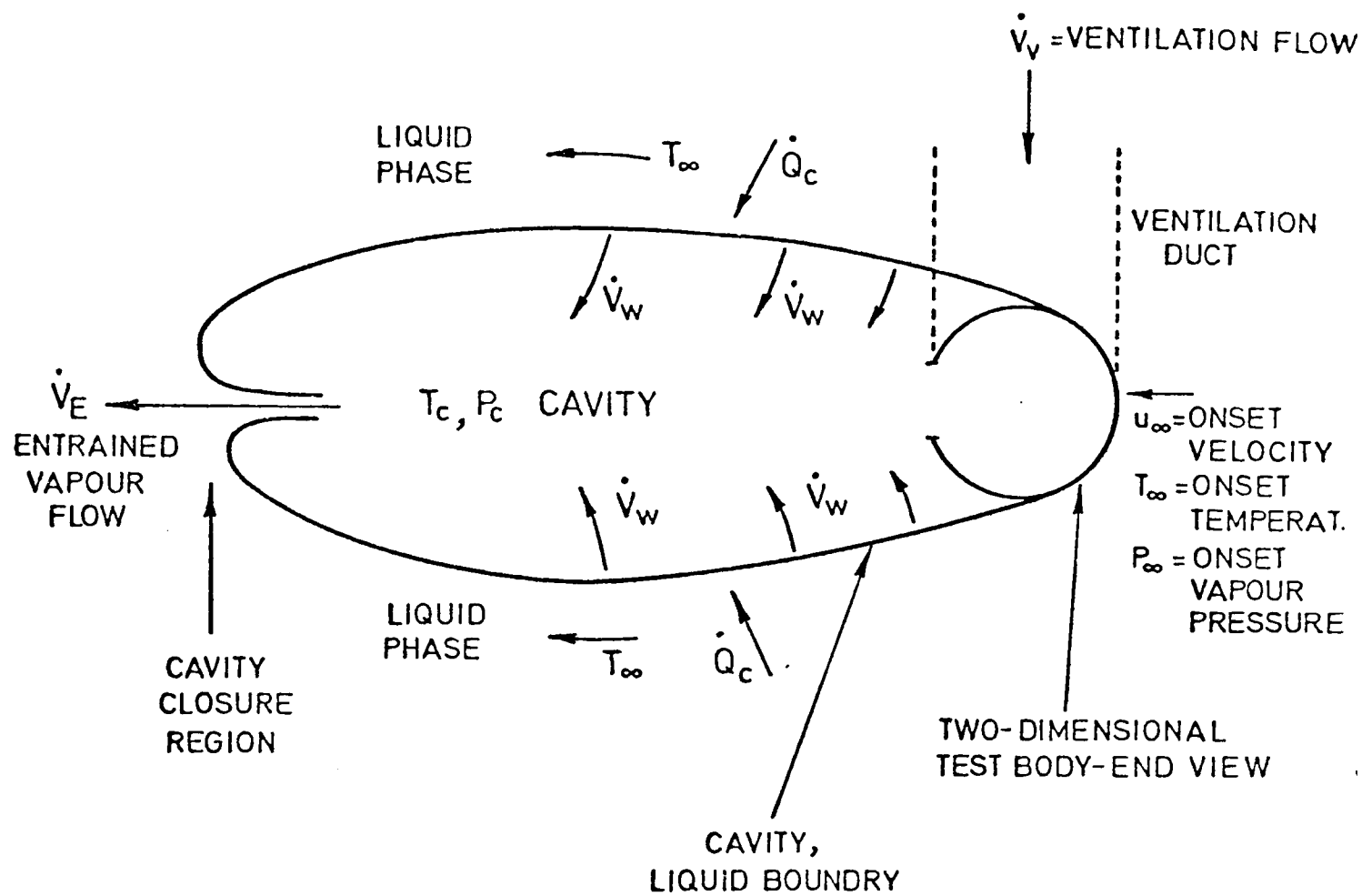


FIG. I.44 SHEMATIC OF PHYSICAL MODELS FOR THE β -FACTOR AND ENTRAINMENT THEORY OF THERMODYNAMIC EFFECT



$\dot{V}_w$ = EVAPORATION RATE FROM LIQUID PHASE INTO CAVITY OR RATE OF VAPOUR CONDENSATION ON LIQUID - CAVITY BOUNDARY.

$\dot{Q}_c$ = HEAT TRANSFER RATE BETWEEN CAVITY AND FREE STREAM LIQUID PHASE

FIG.I.45 CAVITY SCHEMATIC SHOWING HEAT AND MASS FLUXES IN A DEVELOPED VAPOUR VENTILATED CAVITY

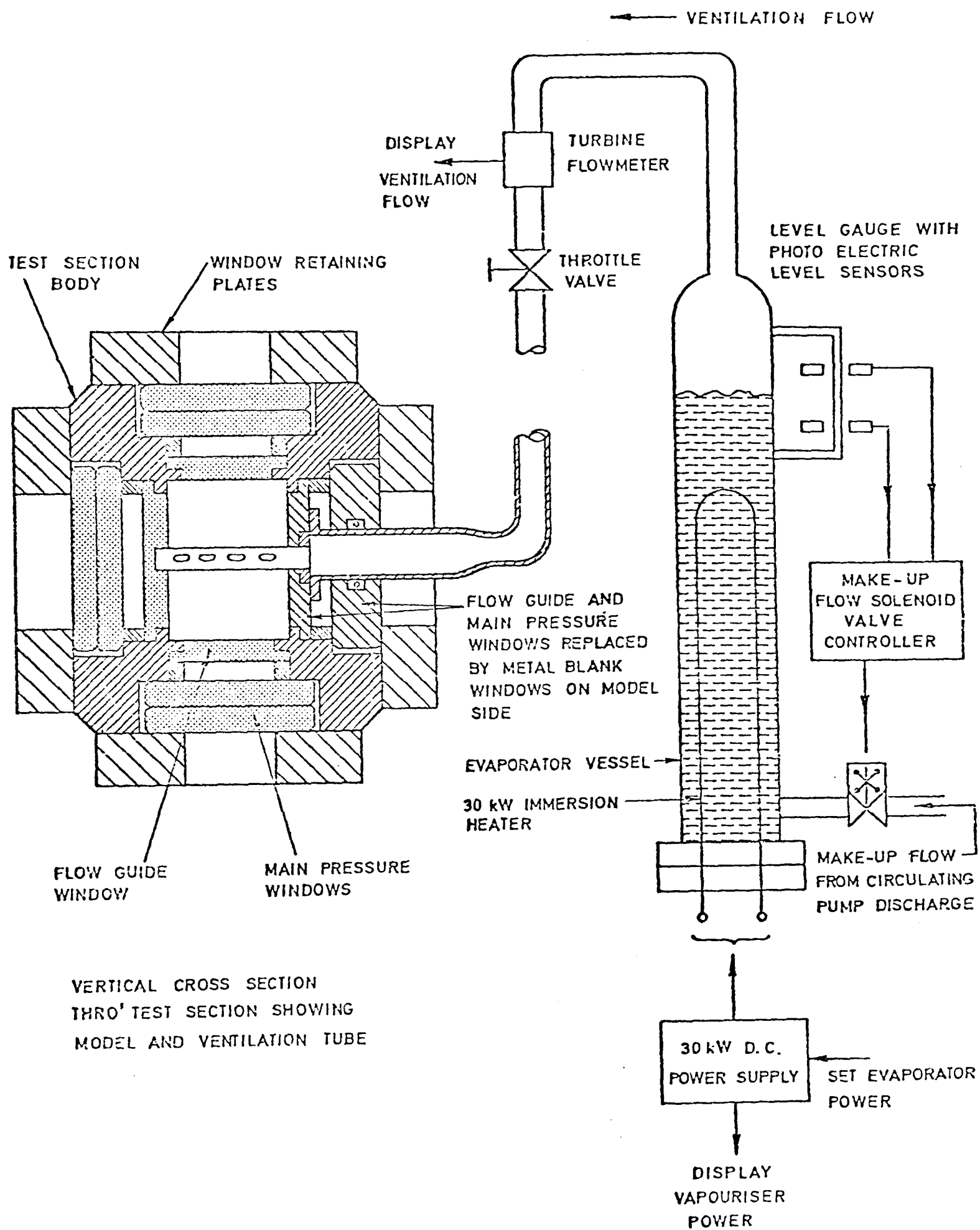


FIG. I.46 THE VAPOUR VENTILATION SYSTEM

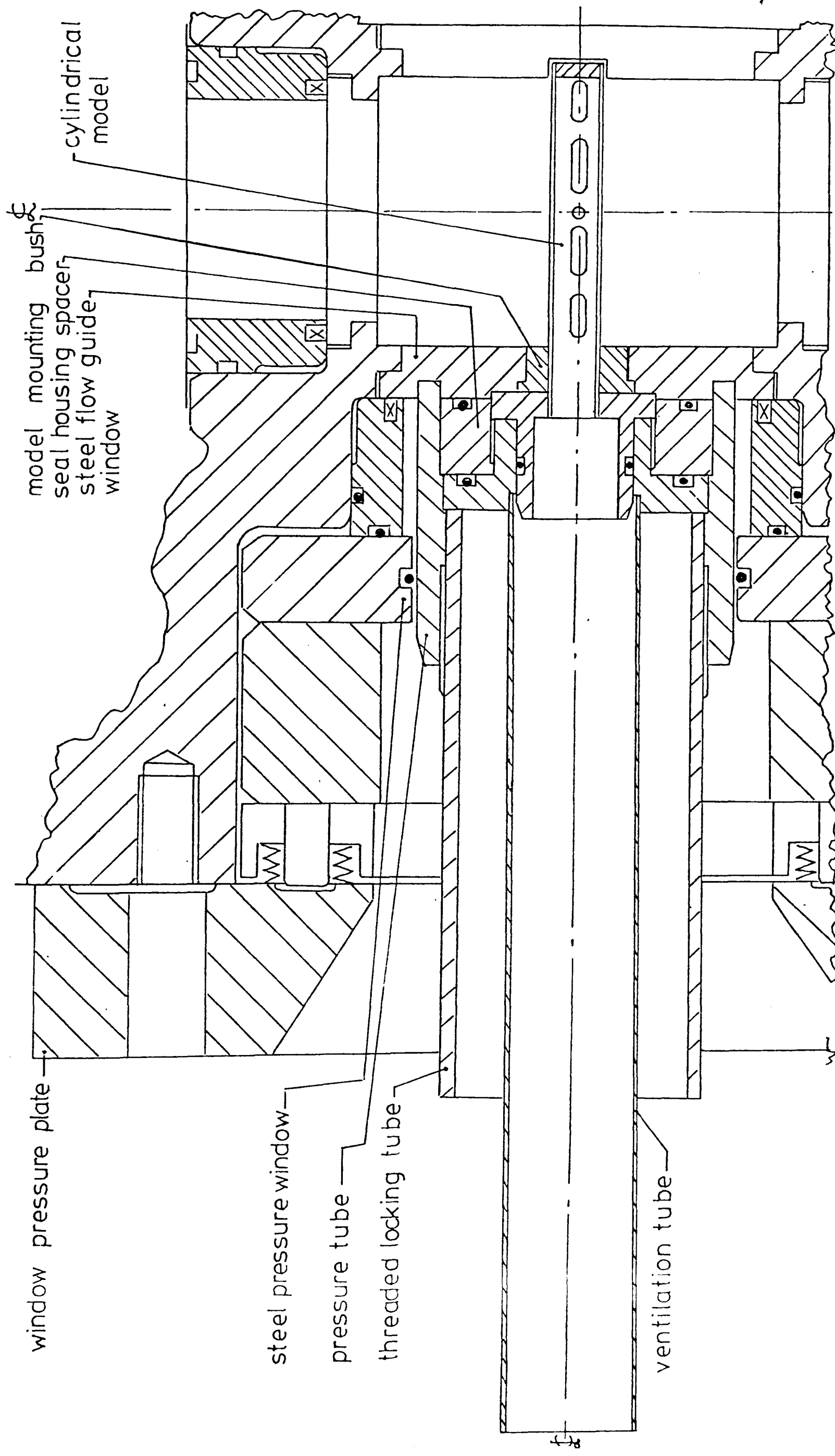


FIG. I.47 PART SECTIONAL VIEW OF TEST SECTION SHOWING DETAILS OF VENTILATION SUPPLY AND SEALING ARRANGEMENTS

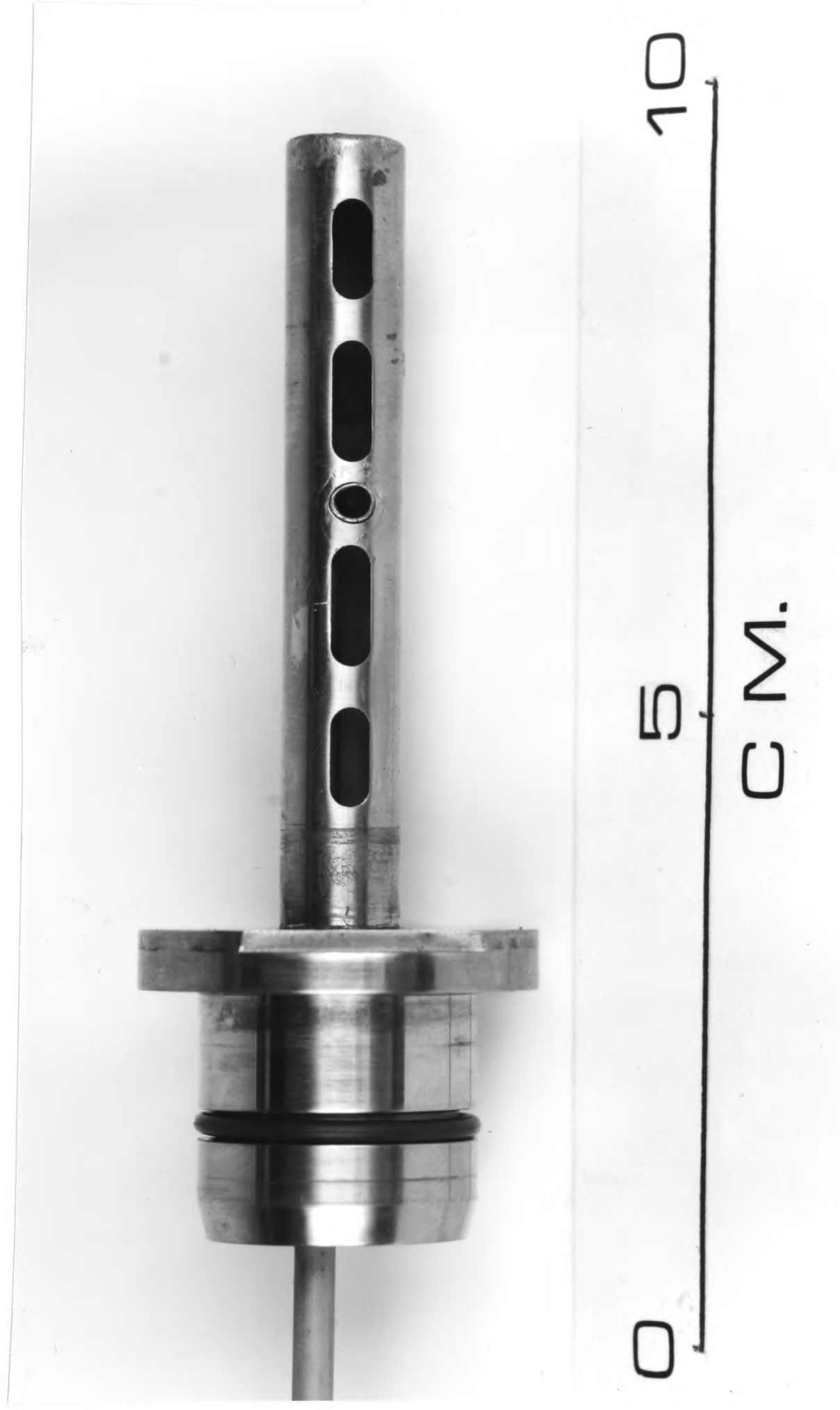


FIG. I.48 THE CYLINDRICAL TEST BODY AND MOUNTING BUSH



FIG. I. 49 EXPLODED VIEW OF TEST BODY AND VENTILATION
SUPPLY COMPONENTS



FIG. I. 50 ASSEMBLED VIEW OF TEST BODY AND VENTILATION COMPONENTS

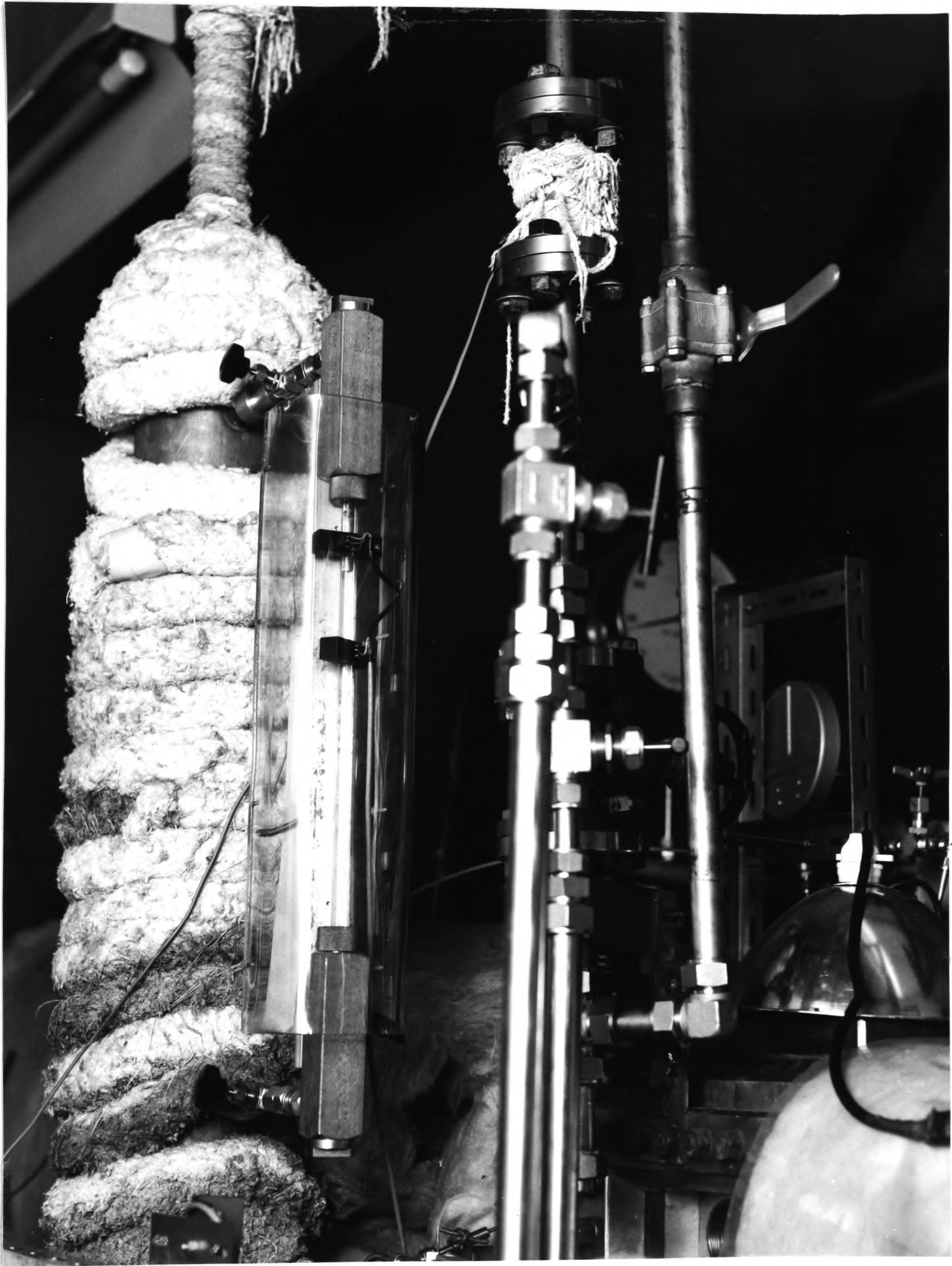


FIG. I. 51 GENERAL VIEW OF EVAPORATOR AND
 VAPOUR SUPPLY SYSTEM

VARIABLE TABLE	
VARIABLE	DESCRIPTION
u_{∞}	TEST SECTION FLOW VELOCITY
V_V	VOLUMETRIC VENTILATION FLOW RATE
P_u	STATIC PRESSURE AT CONTRACTION INLET
P_s	STATIC PRESSURE AT TEST BODY LOCATION
P_c	CAVITY PRESSURE
T_{∞}	FLUID TEMPERATURE AT CONTRACTION INLET
T_c	CAVITY TEMPERATURE
L_c	CAVITY LENGTH
Q_c	EVAPORATOR INPUT POWER

TRANSDUCER TABLE	
VARIABLE TYPE	TRANSDUCERS
DIFFERENTIAL PRESSURES	STATHAM D.P. TRANSDUCERS
TEMPERATURES	THERMOCOUPLES
FLOWS	TURBINE TYPE FLOWMETERS
ELECTRIC POWER	VOLTAGE AND CURRENT MEASUREMENTS
CAVITY LENGTH	VISUAL

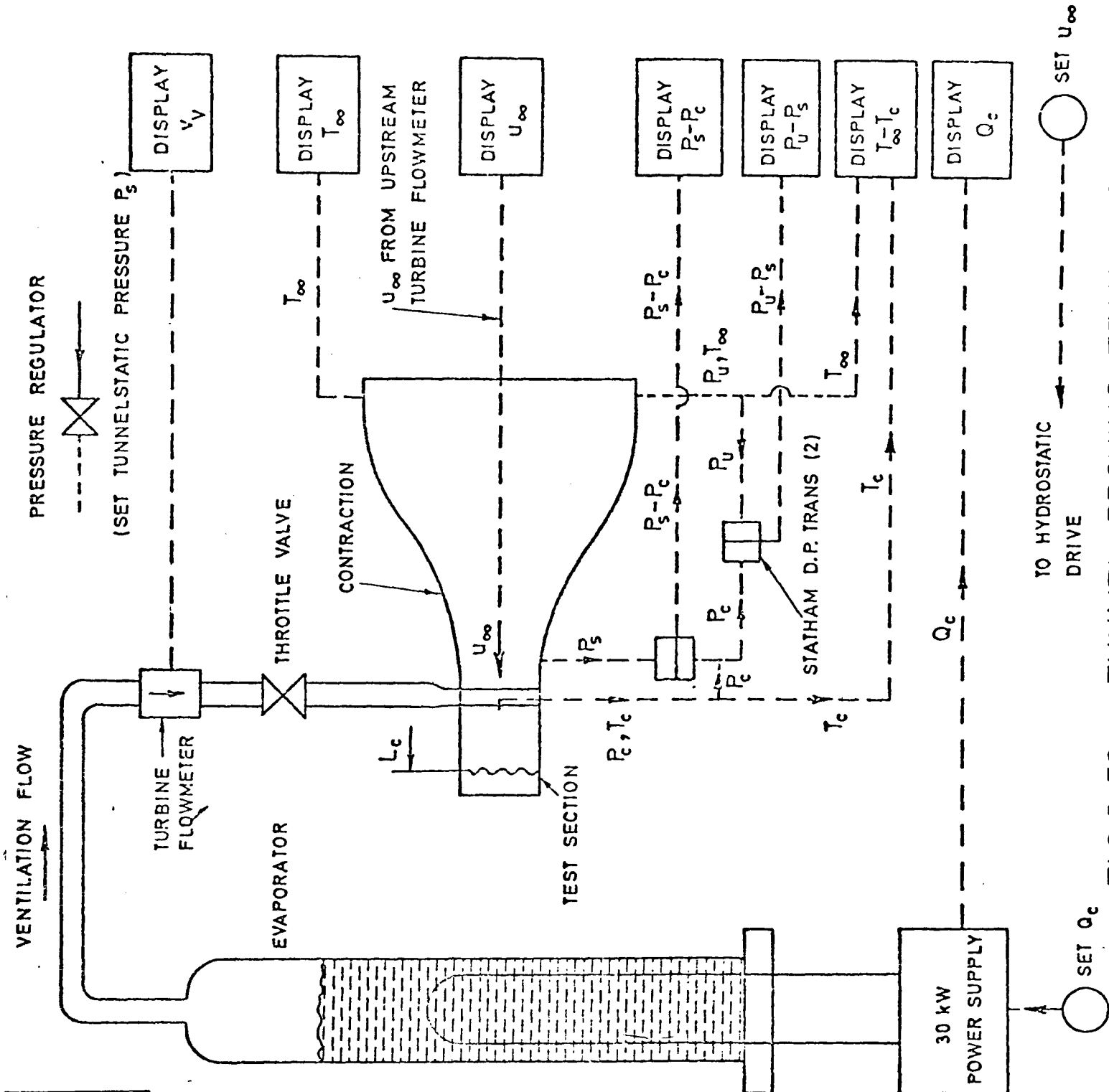


FIG.I. 52 TUNNEL PROVING TRIAL INSTRUMENTATION

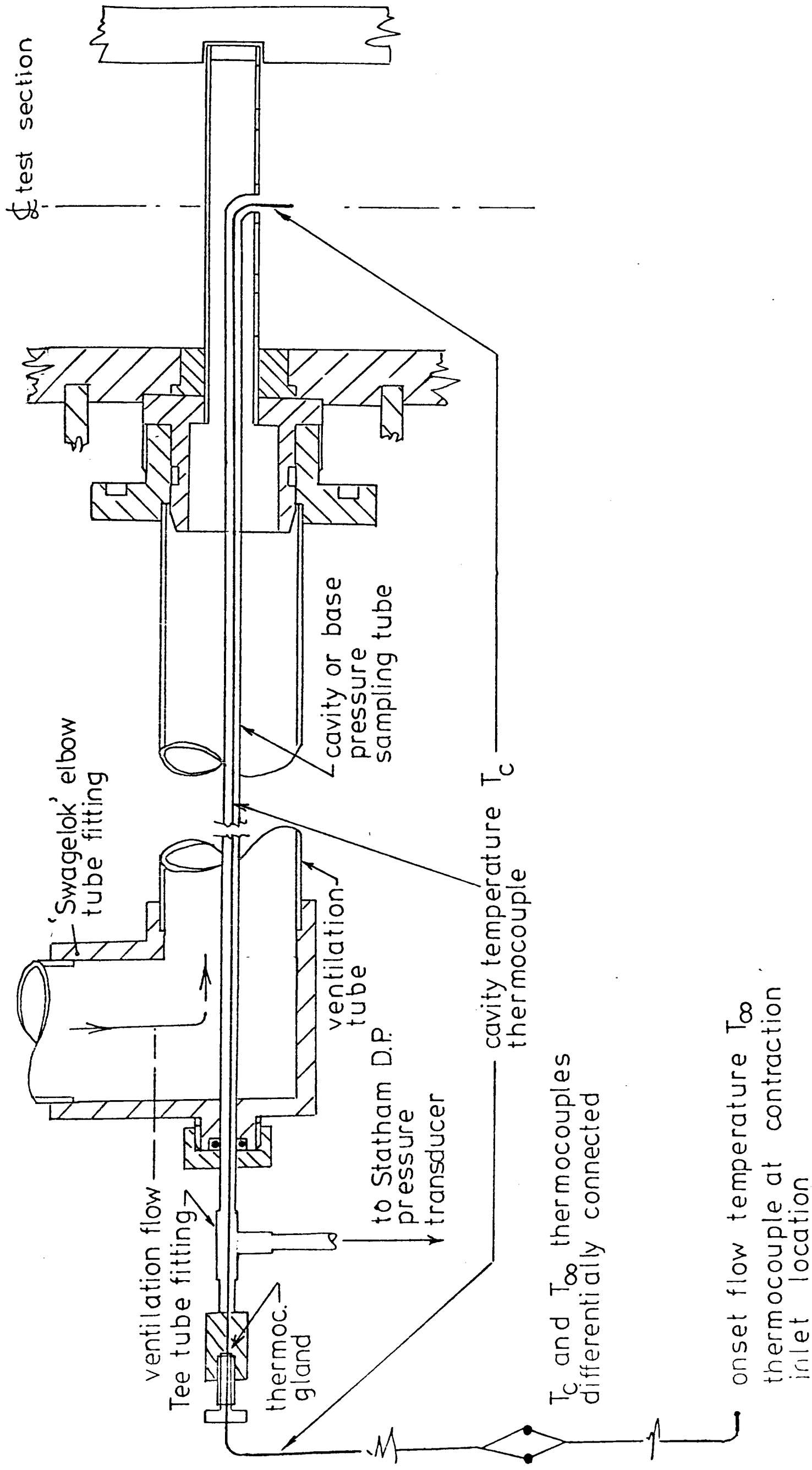


FIG. I.53 PART SECTIONAL PLAN VIEW OF TEST SECTION SHOWING CAVITY TEMPERATURE MEASUREMENT ARRANGEMENT

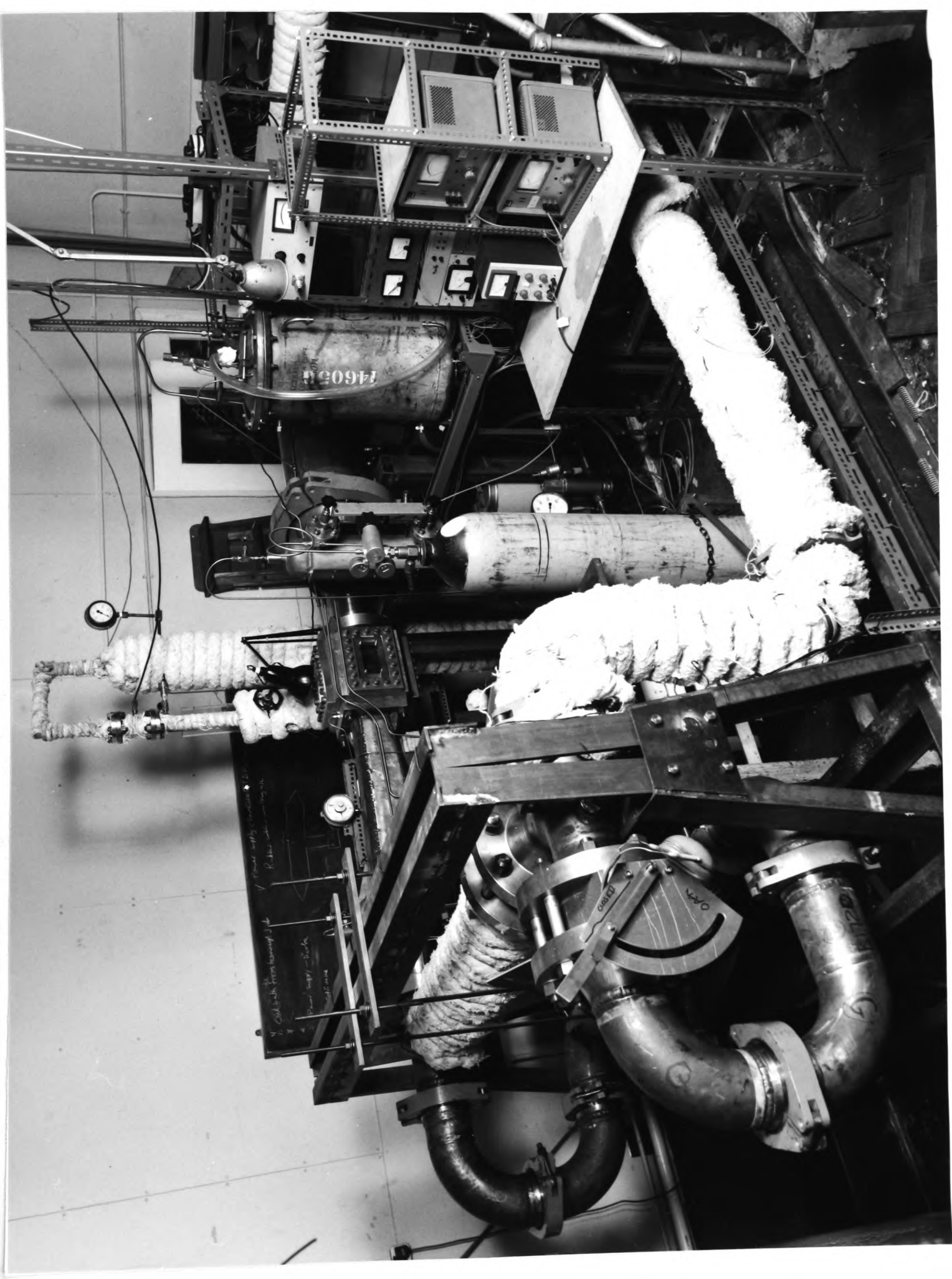


FIG.I. 54 GENERAL VIEW OF TUNNEL AND INSTRUMENTATION USED FOR
THE PROVING TRAILS

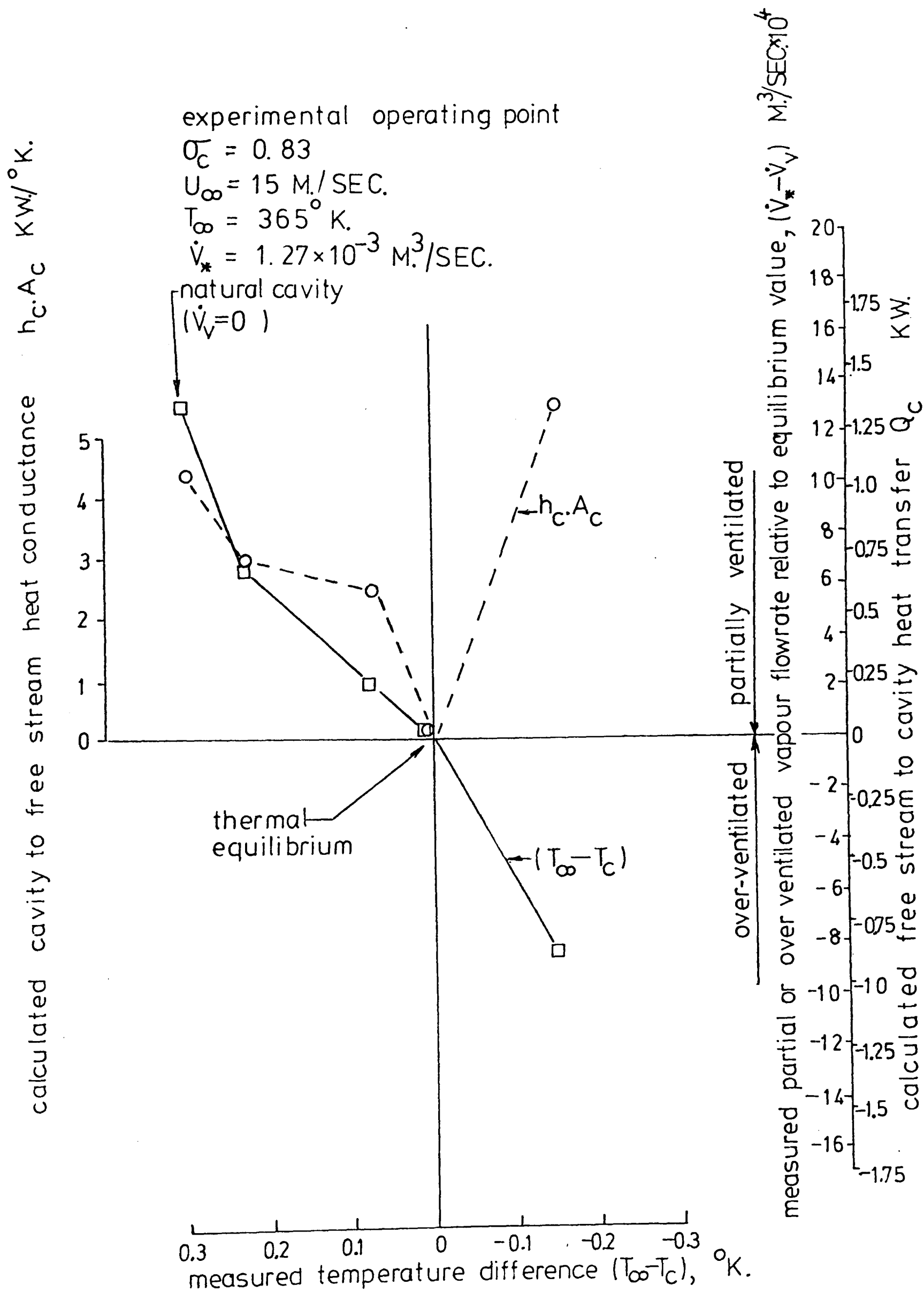


FIG. I.55 CAVITY TO FREE STREAM TEMPERATURE DIFFERENCE AND HEAT TRANSFER

VARIABLE TABLE	
symbol	variable
u_∞	test section velocity
$\dot{V}_V$	volumetric ventilation flowrate
P_U	static pressure at contraction inlet
P_S	static pressure at test section inlet
P_C	cavity or base pressure
T_∞	onset flow temperature
T_C	cavity temperature
Q_v	evaporator input power
$P_W(x)$	test section wall pressure distribution

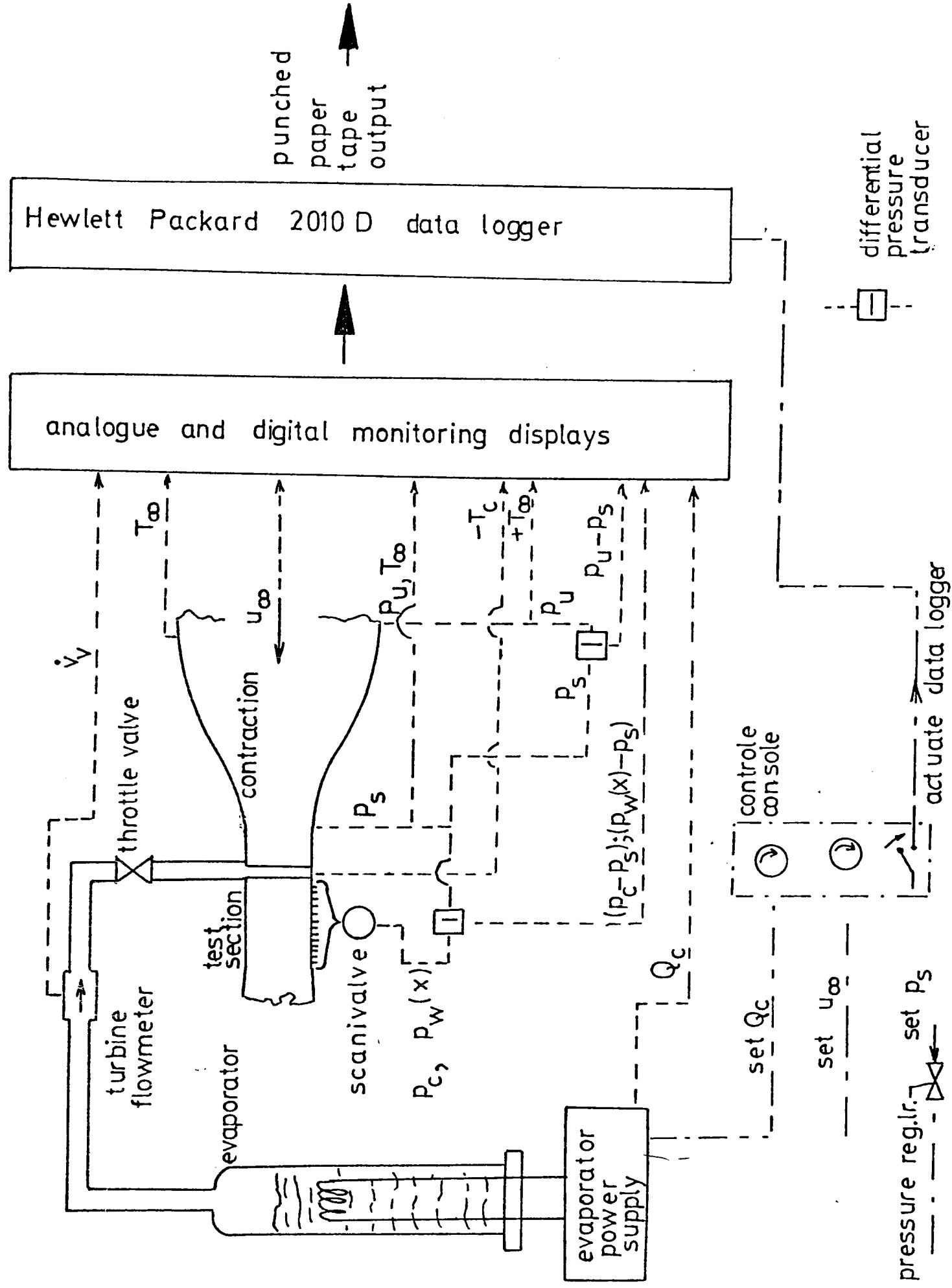


FIG. I.56 DATA ACQUISITION SYSTEM AND MEASURED AND CONTROL VARIABLES

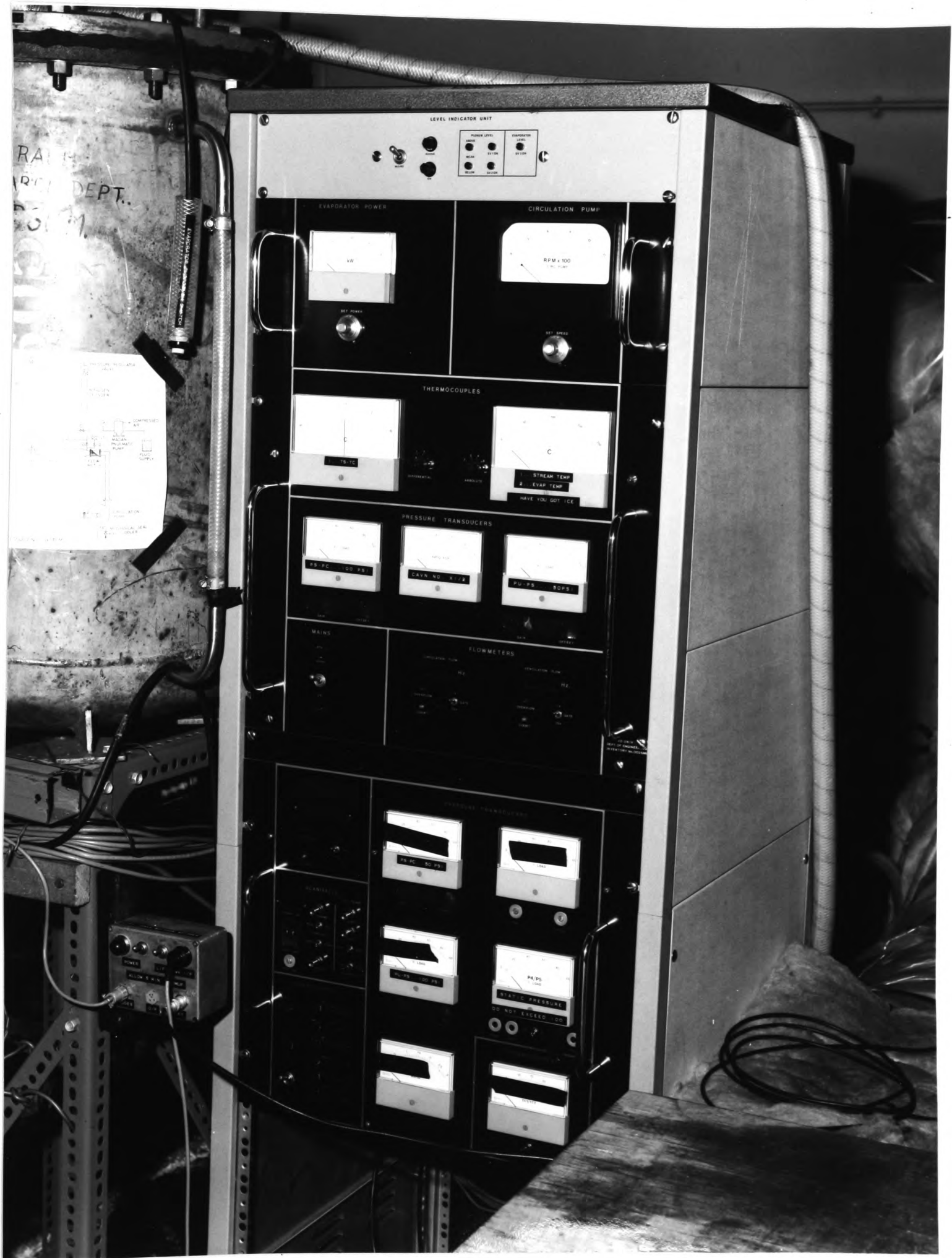


FIG. I. 57 THE TUNNEL INSTRUMENTATION CONSOLE

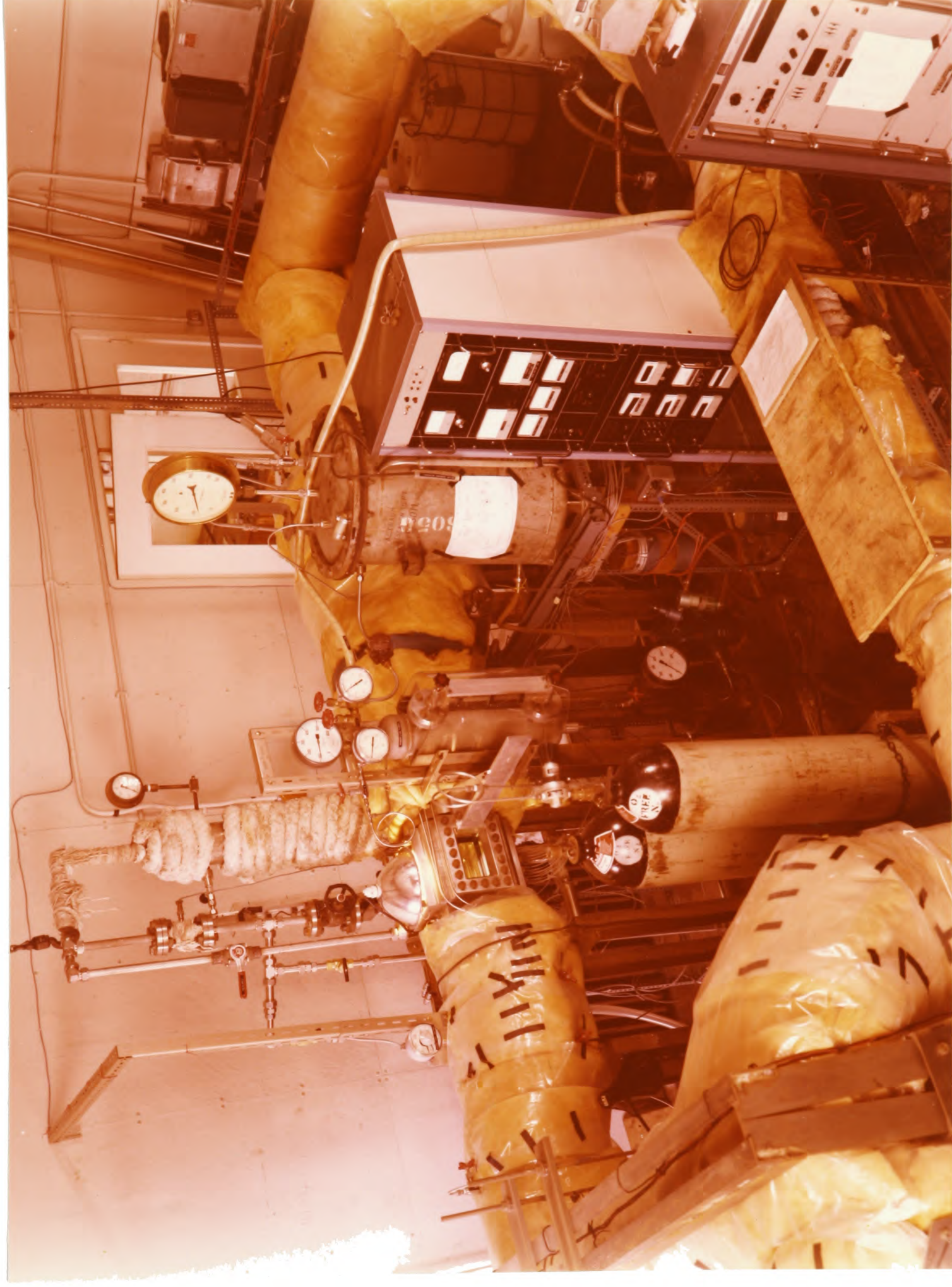


FIG. I.58 GENERAL VIEW OF TUNNEL WORKING AREA AND CONTROL CONSOLE



FIG. I. 59 CONTROL CONSOLE OF THE COLD WATER CAVITATION TUNNEL
SHOWING STARTING CONTROLS FOR HYDROSTATIC DRIVE

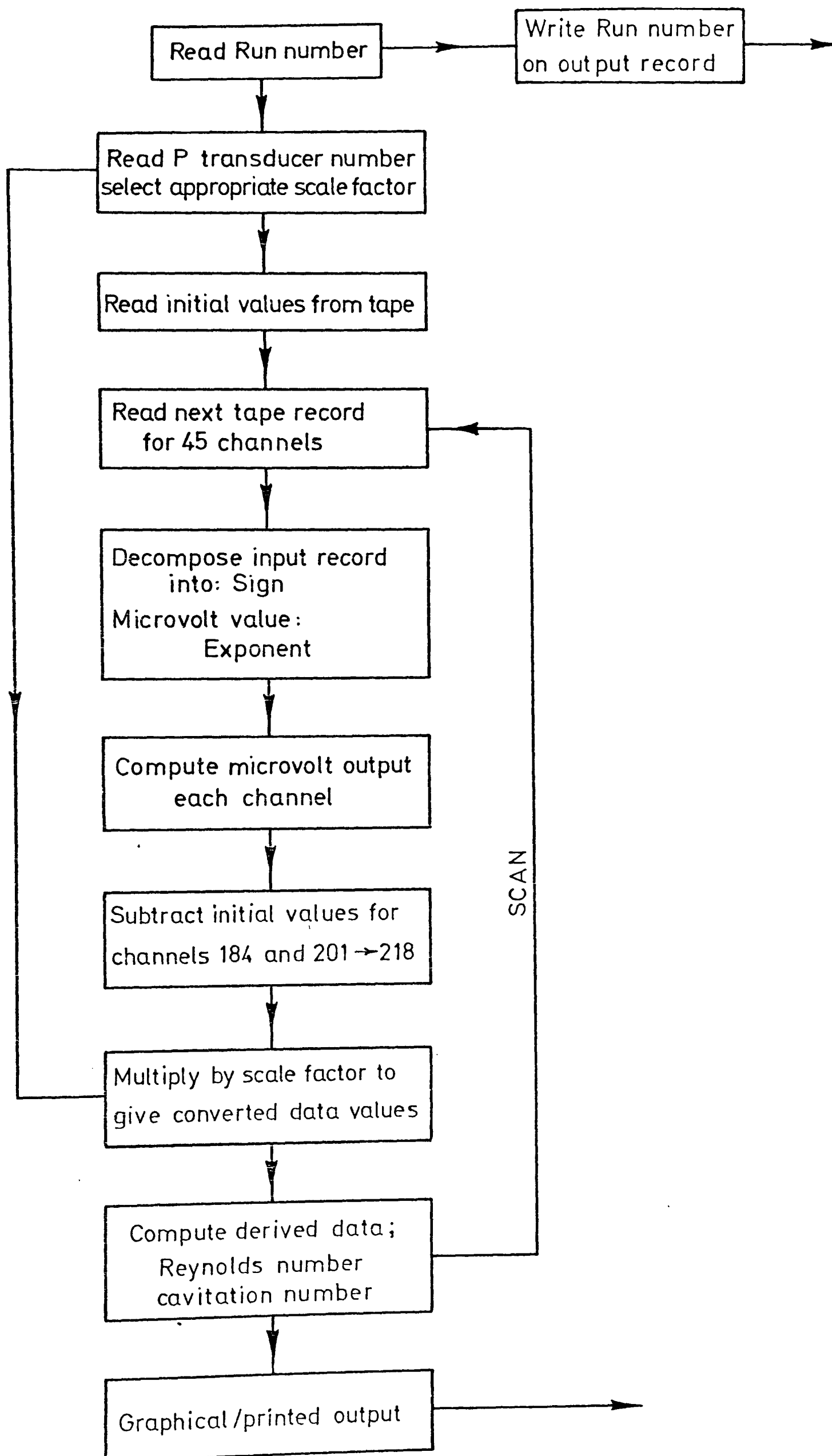


FIG. I.60 BLOCK DIAGRAM OF PROGRAM 'CAVITACE'

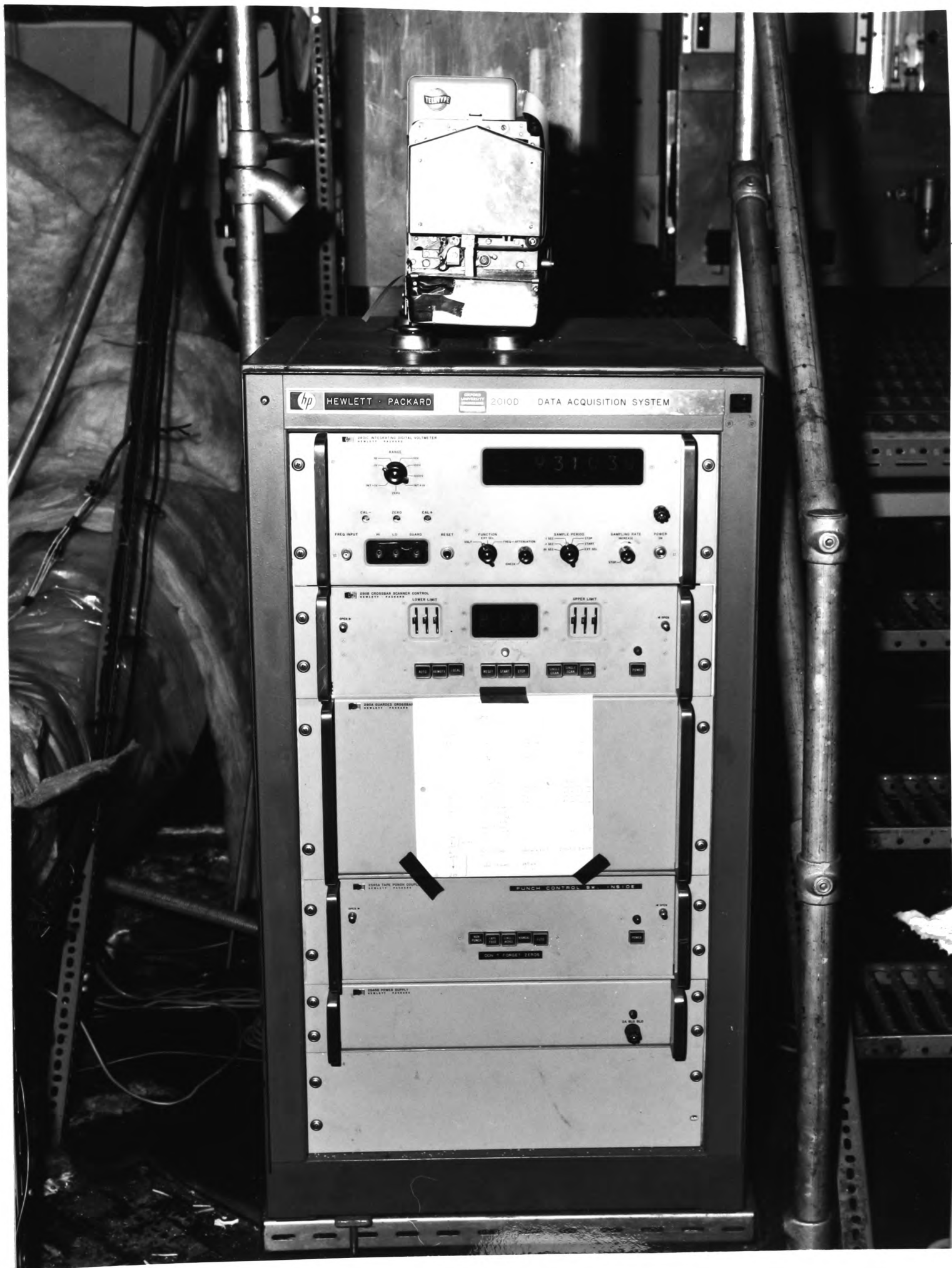


FIG. I. 61 THE DATA LOGGER AND PAPER TAPE PUNCH

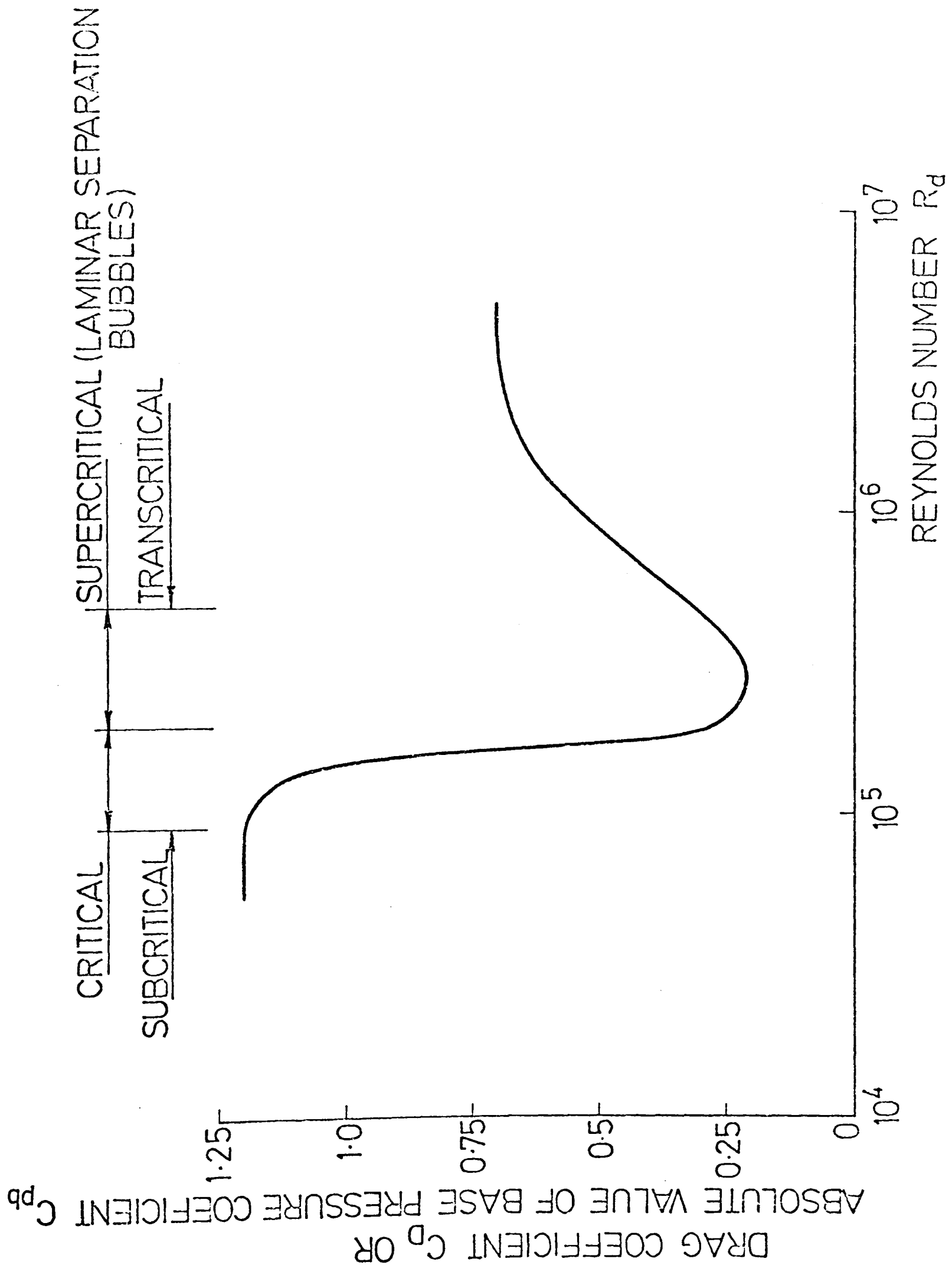


FIG. II. 1 TYPICAL PLOT OF DRAG COEFFICIENT OR ABSOLUTE VALUE OF BASE PRESSURE COEFFICIENT VERSUS REYNOLDS NUMBER FOR A CIRCULAR CYLINDER

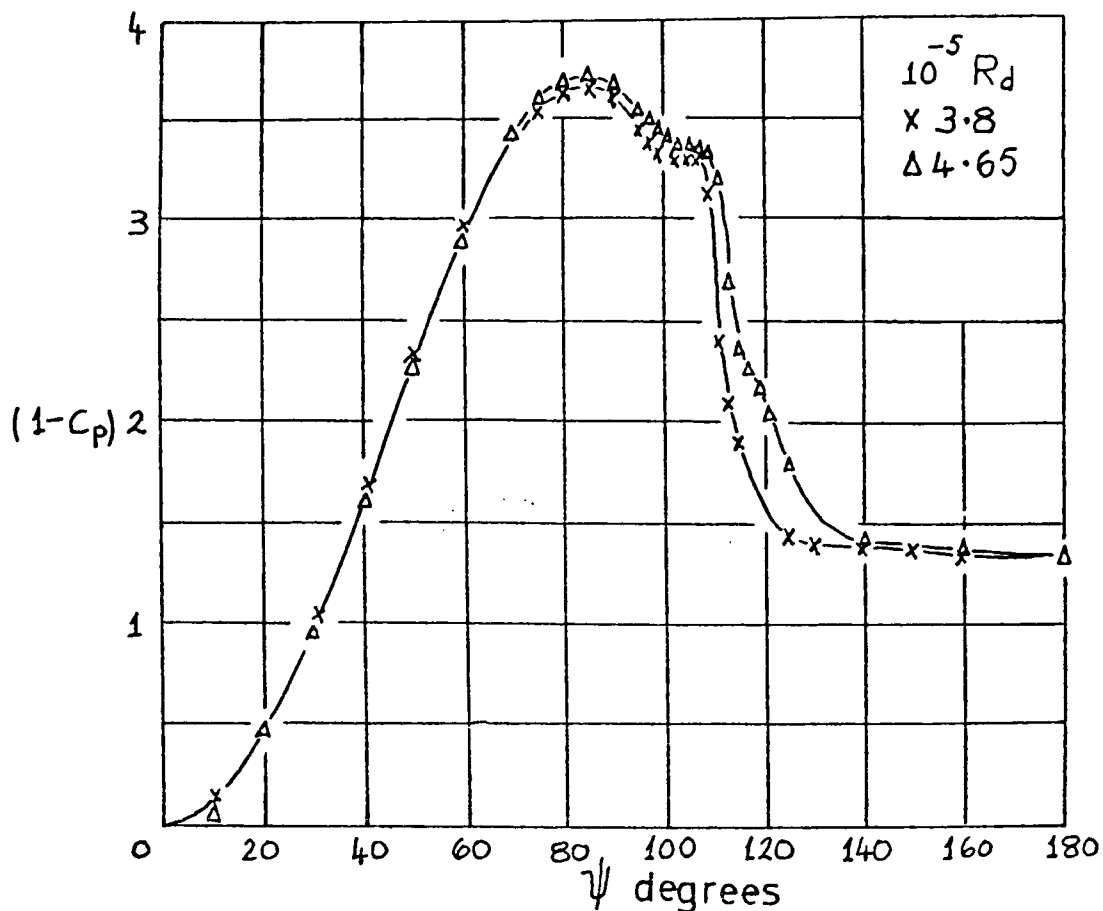


FIG.II.2 PRESSURE DISTRIBUTION ON A CIRCULAR CYLINDER WITH LAMINAR SEPARATION BUBBLES AFTER TANI(84)

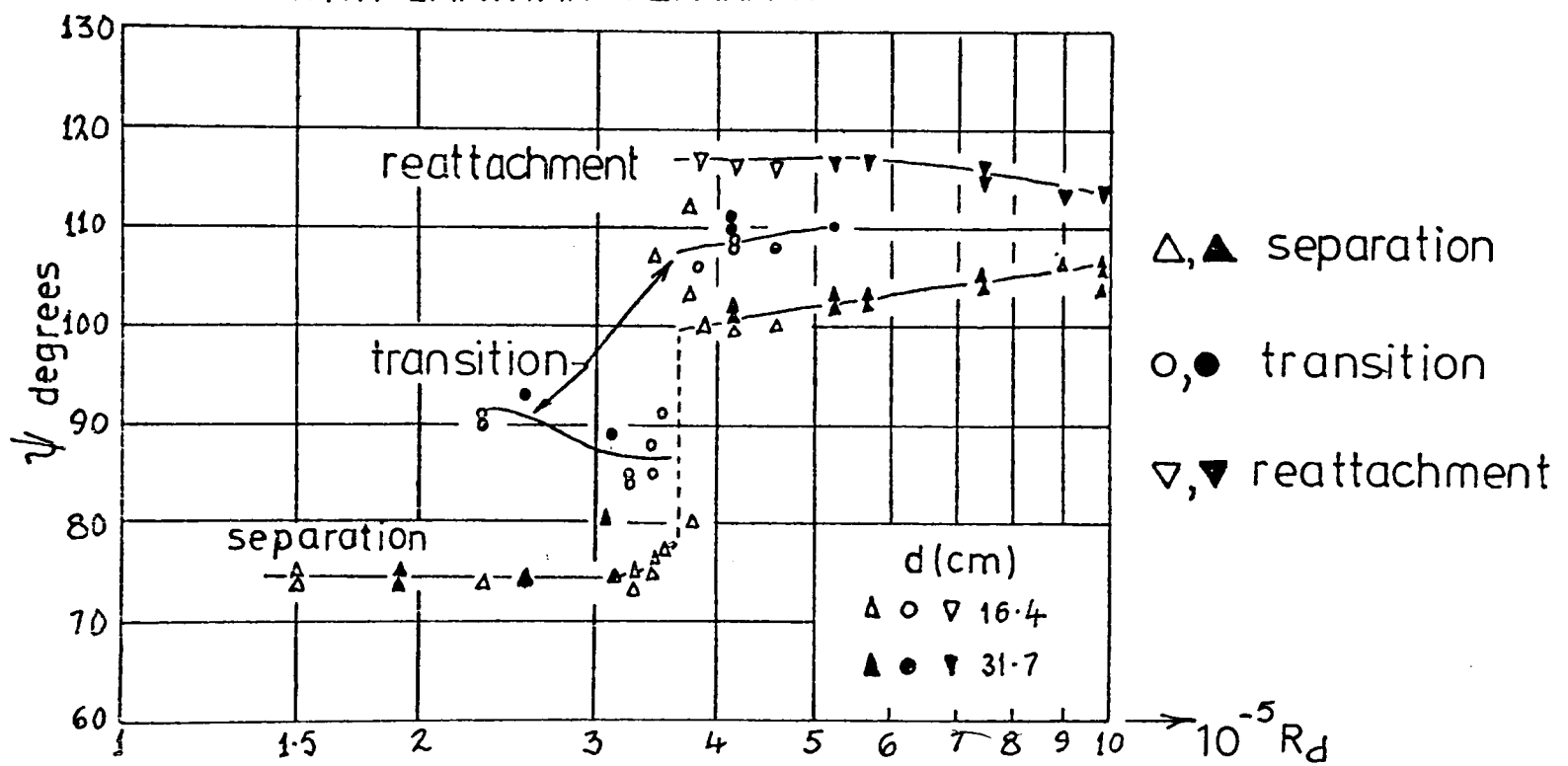


FIG.II.3 POSITIONS OF SEPARATION, TRANSITION AND REATTACHMENT ON CIRCULAR CYLINDERS IN THE CRITICAL REYNOLDS NUMBER RANGE AFTER TANI(84)

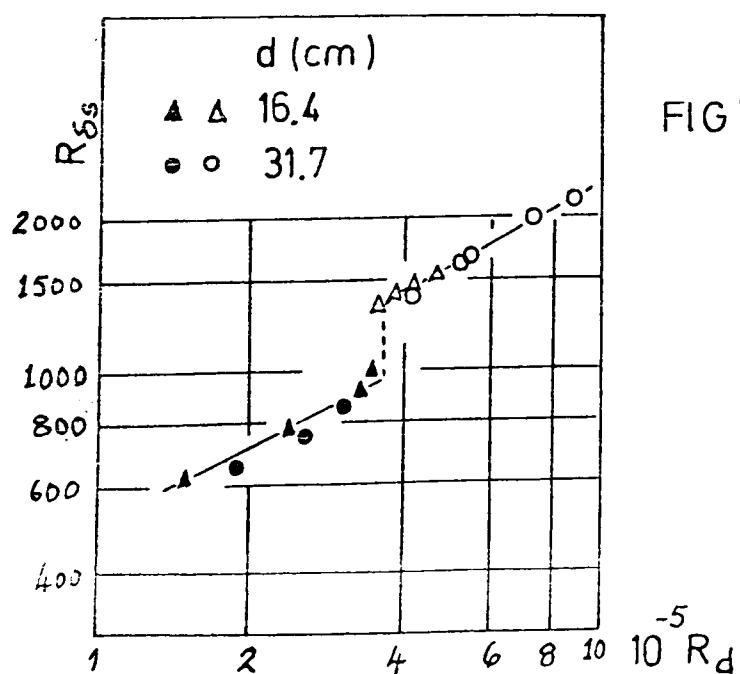


FIG.II.4 CALCULATED BOUNDARY LAYER REYNOLDS NUMBER AT LAMINAR SEPARATION ON A CIRCULAR CYLINDER AFTER TANI(84)

closed symbols ~ laminar separation without reattachment

open symbols ~ laminar separation followed by turbulent reattachment

LENGTH OF SEPARATED REGION AS MULTIPLE OF
BOUNDARY LAYER MOMENTUM THICKNESS AT SEPARATION

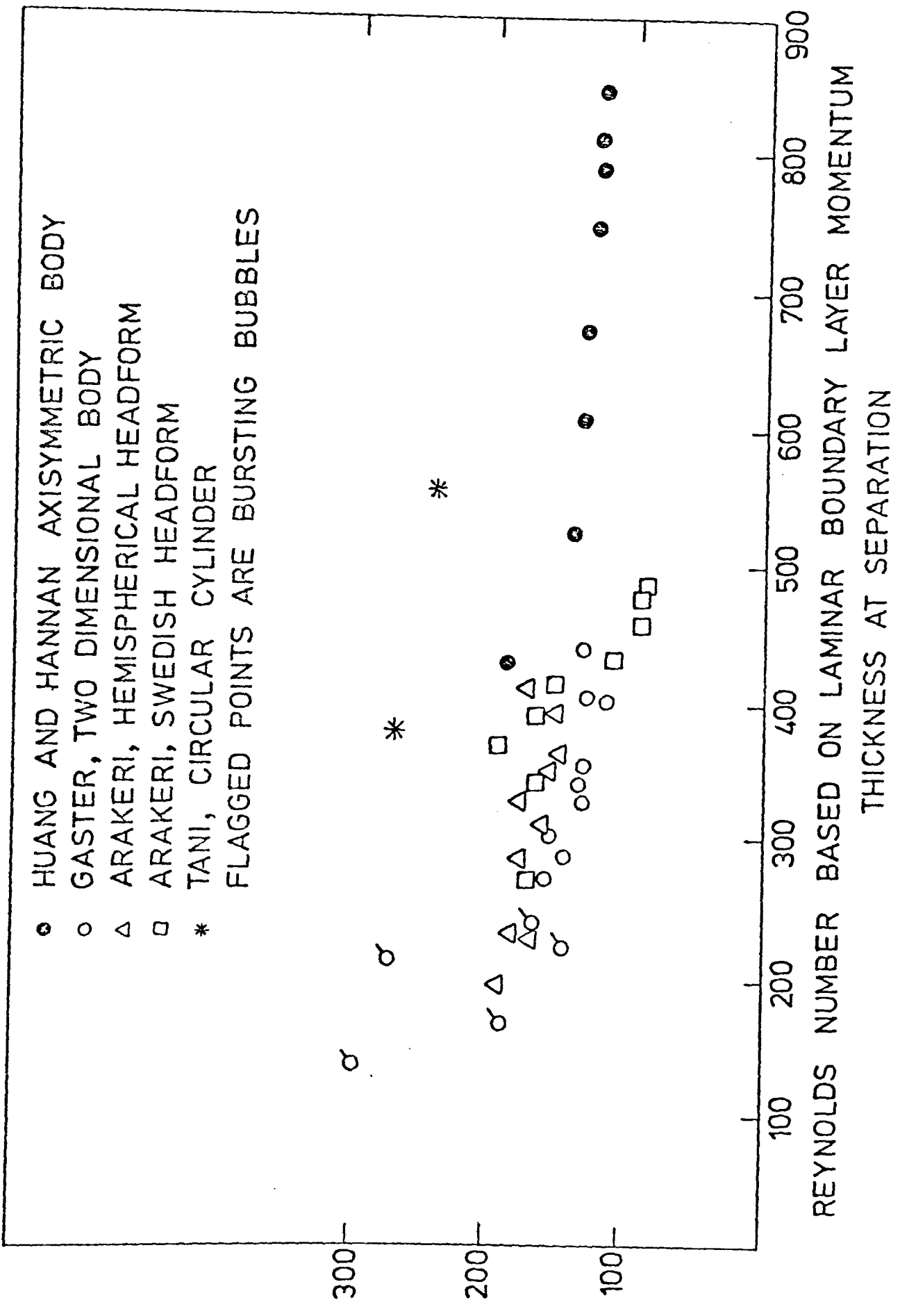


FIG.11.5 DIMENSIONLESS BUBBLE LENGTH VERSUS REYNOLDS NUMBER

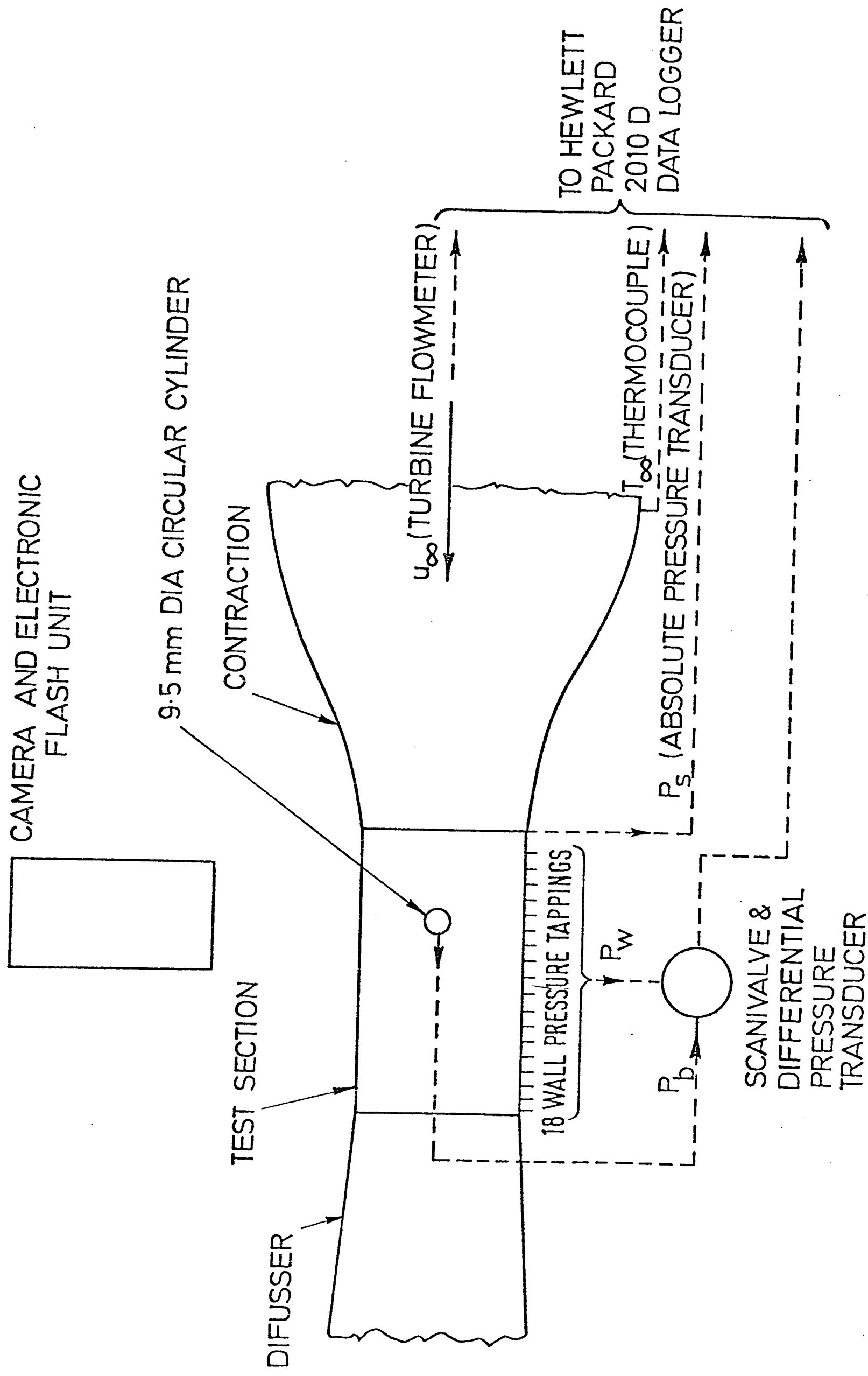


FIG. II.6 SCHEMATIC OF WORKING SECTION SHOWING MEASURED VARIABLES AND EXPERIMENTAL SET-UP FOR THE FULLY WETTED AND NATURAL CAVITATION EXPERIMENTS

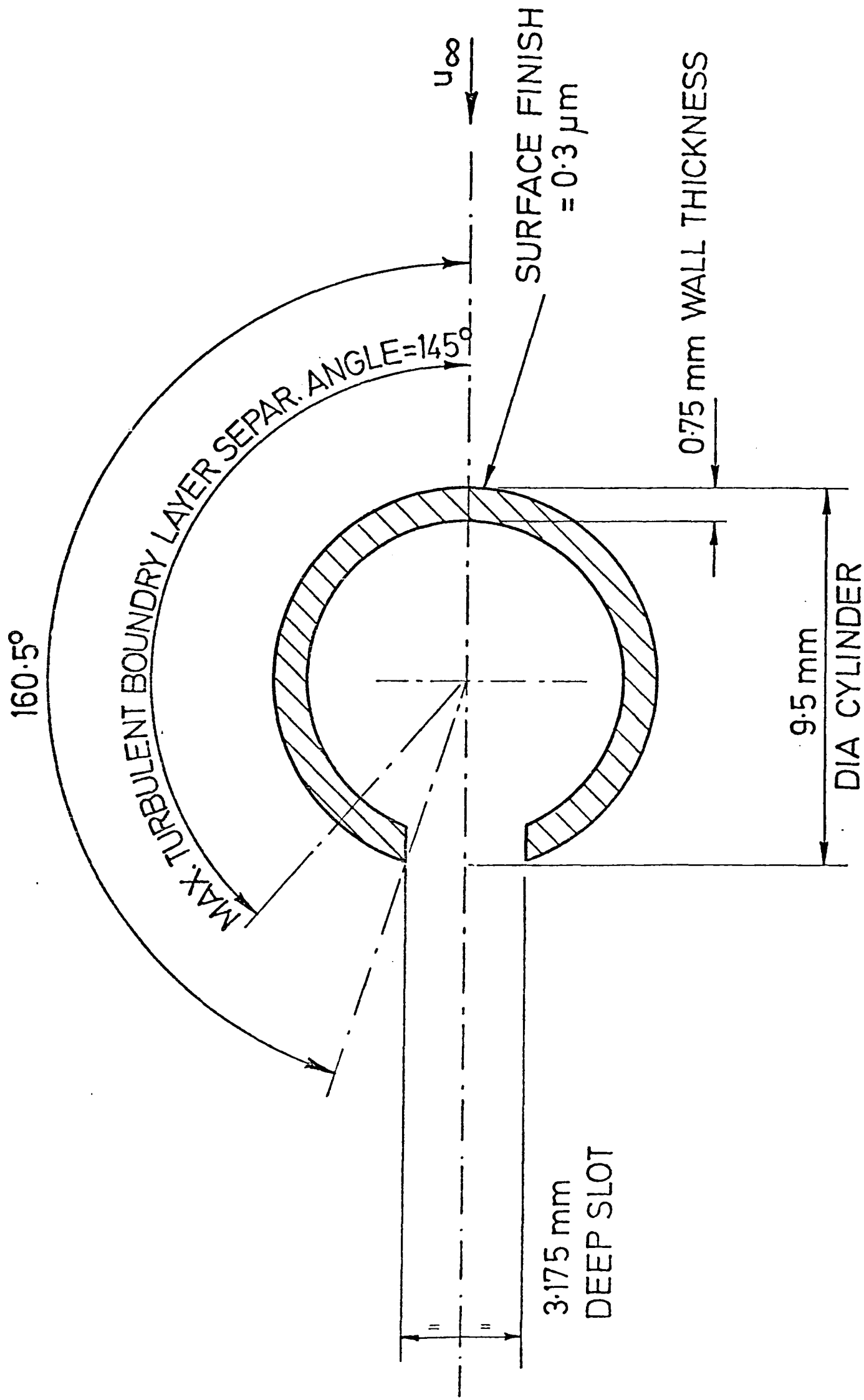


FIG. II. 7 TYPICAL CROSS SECTION THROUGH CYLINDRICAL TEST BODY SHOWING SLOT POSITION IN RELATION TO THE MAXIMUM TURBULENT BOUNDARY LAYER SEPARATION ANGLE

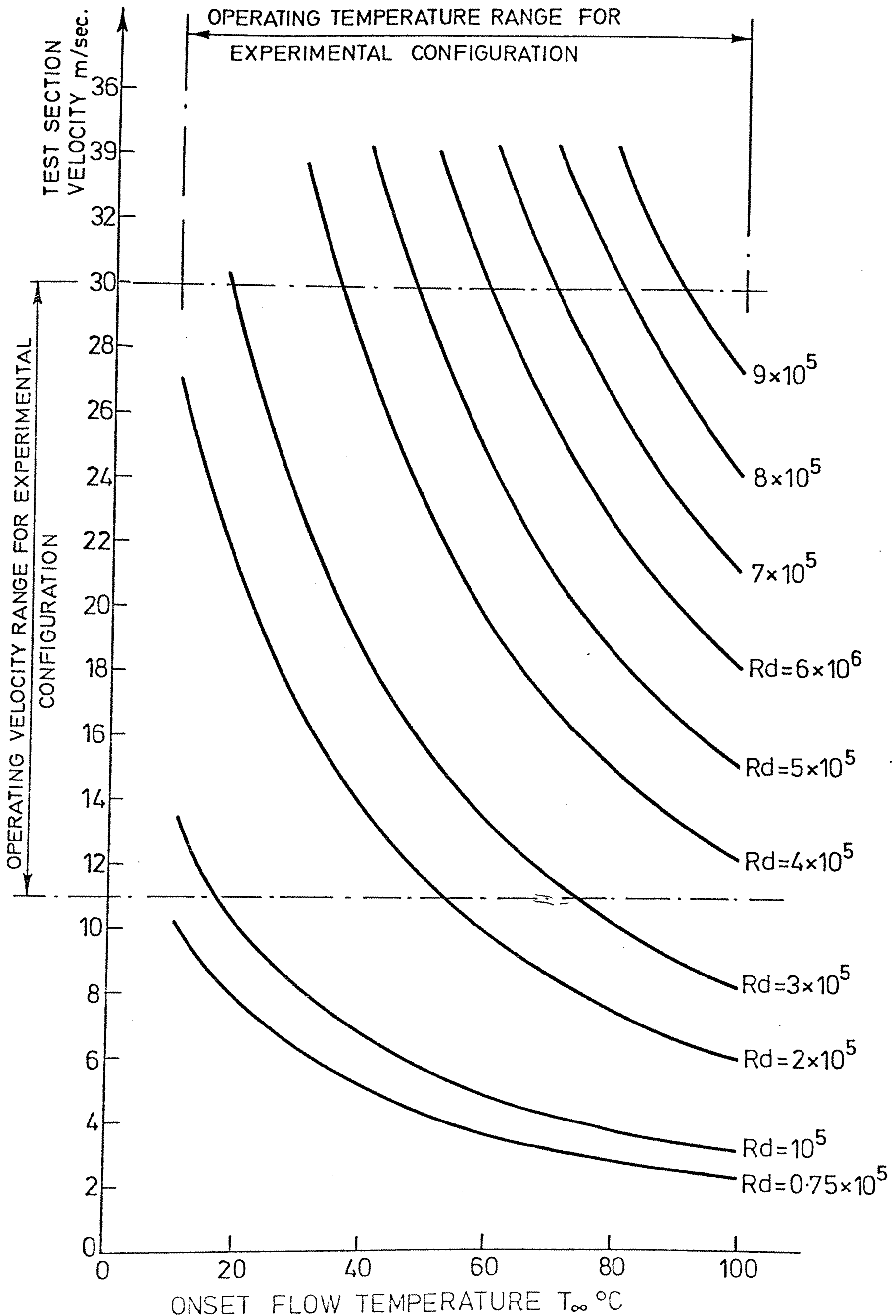


FIG.II.8 OPERATING RANGE OF TEST SECTION VELOCITY, TEMPERATURE AND CYLINDER DIAMETER REYNOLD NUMBER FOR EXPERIMENTAL CONFIGURATION.

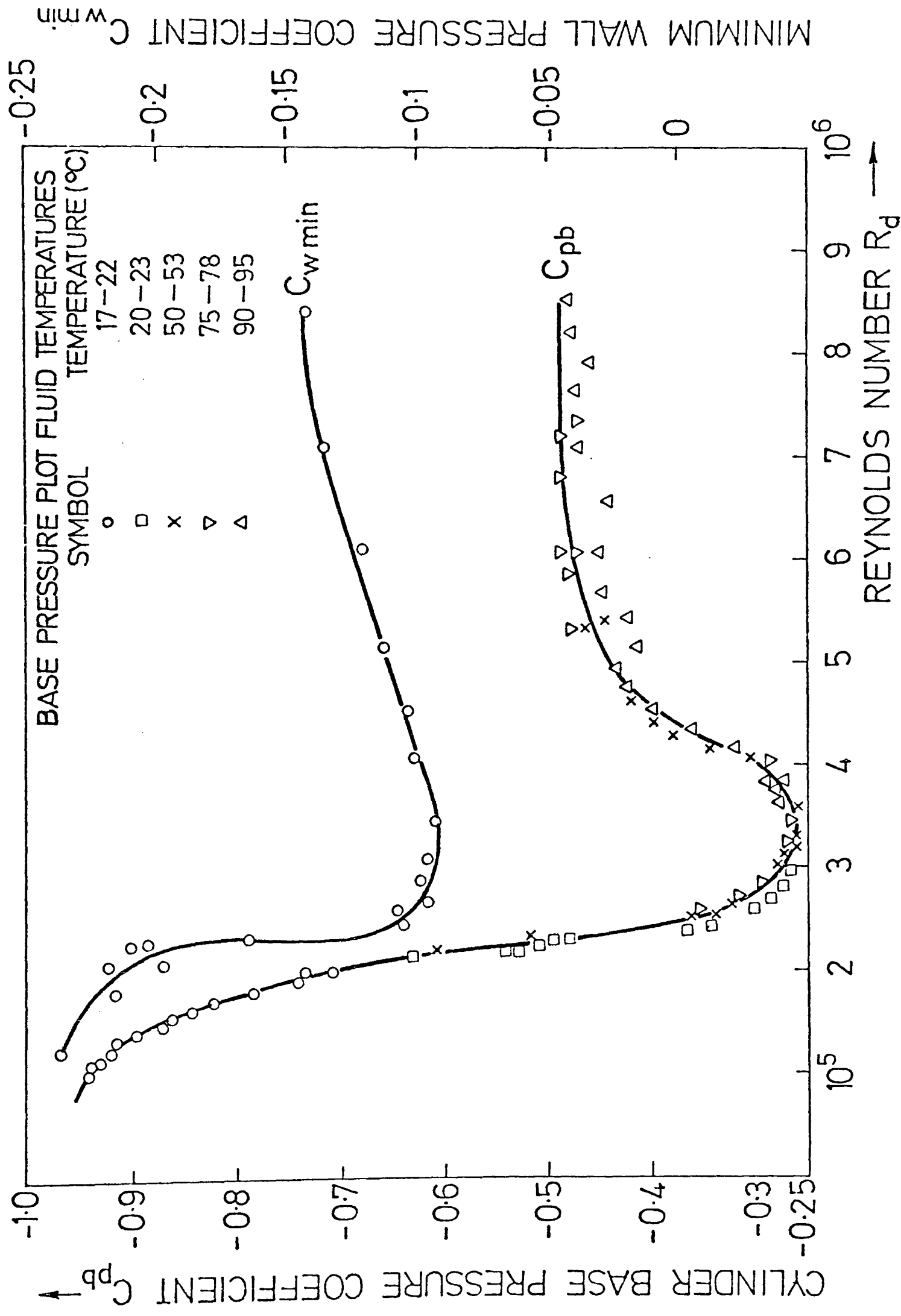


FIG. II.9 CYLINDER BASE PRESSURE AND MINIMUM WALL PRESSURE COEFFICIENTS VERSUS REYNOLDS NUMBER FOR FULLY WETTED FLOW

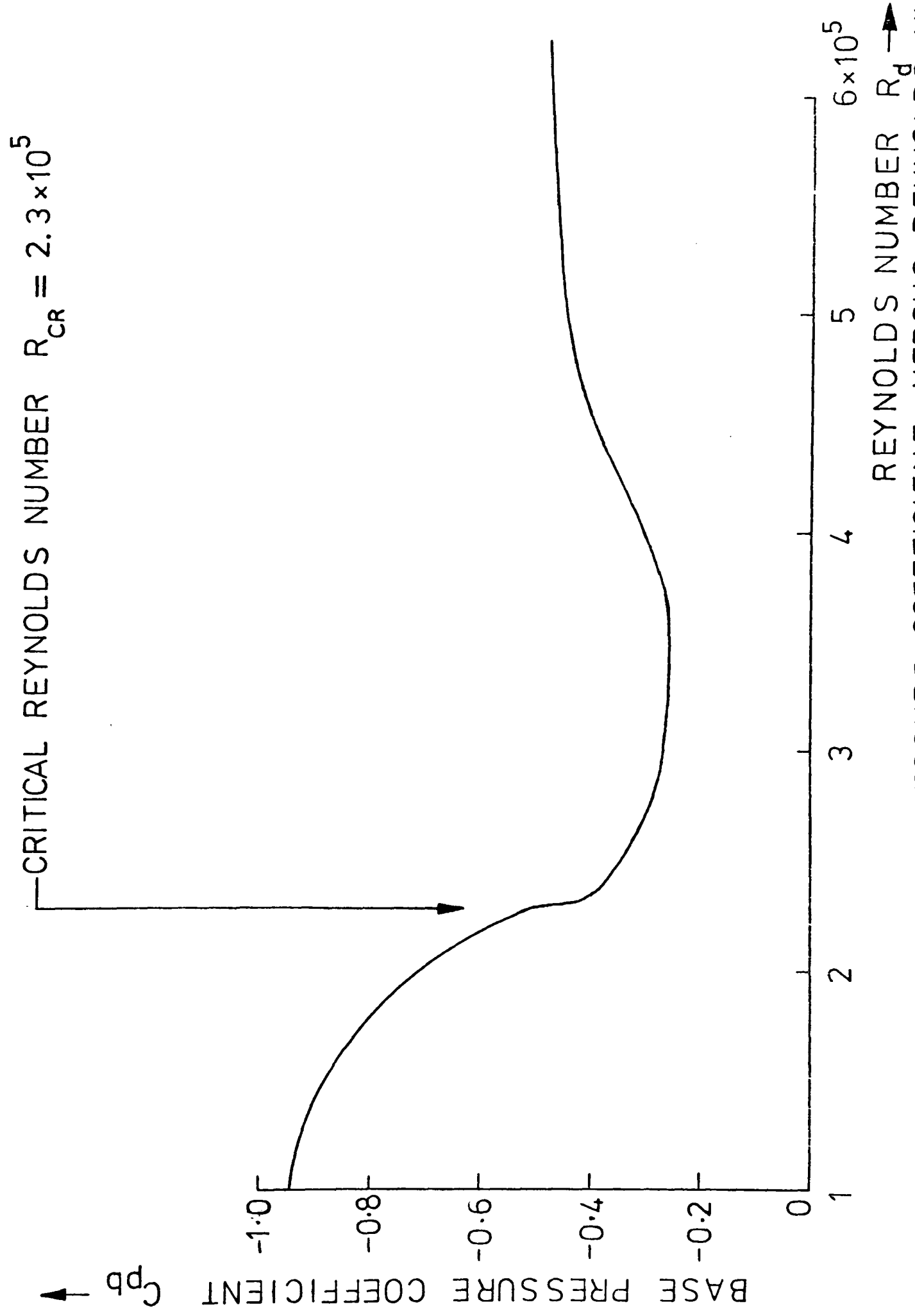


FIG. II. 10 CYLINDER BASE PRESSURE COEFFICIENT VERSUS REYNOLDS NUMBER FOR FULLY WETTED FLOW

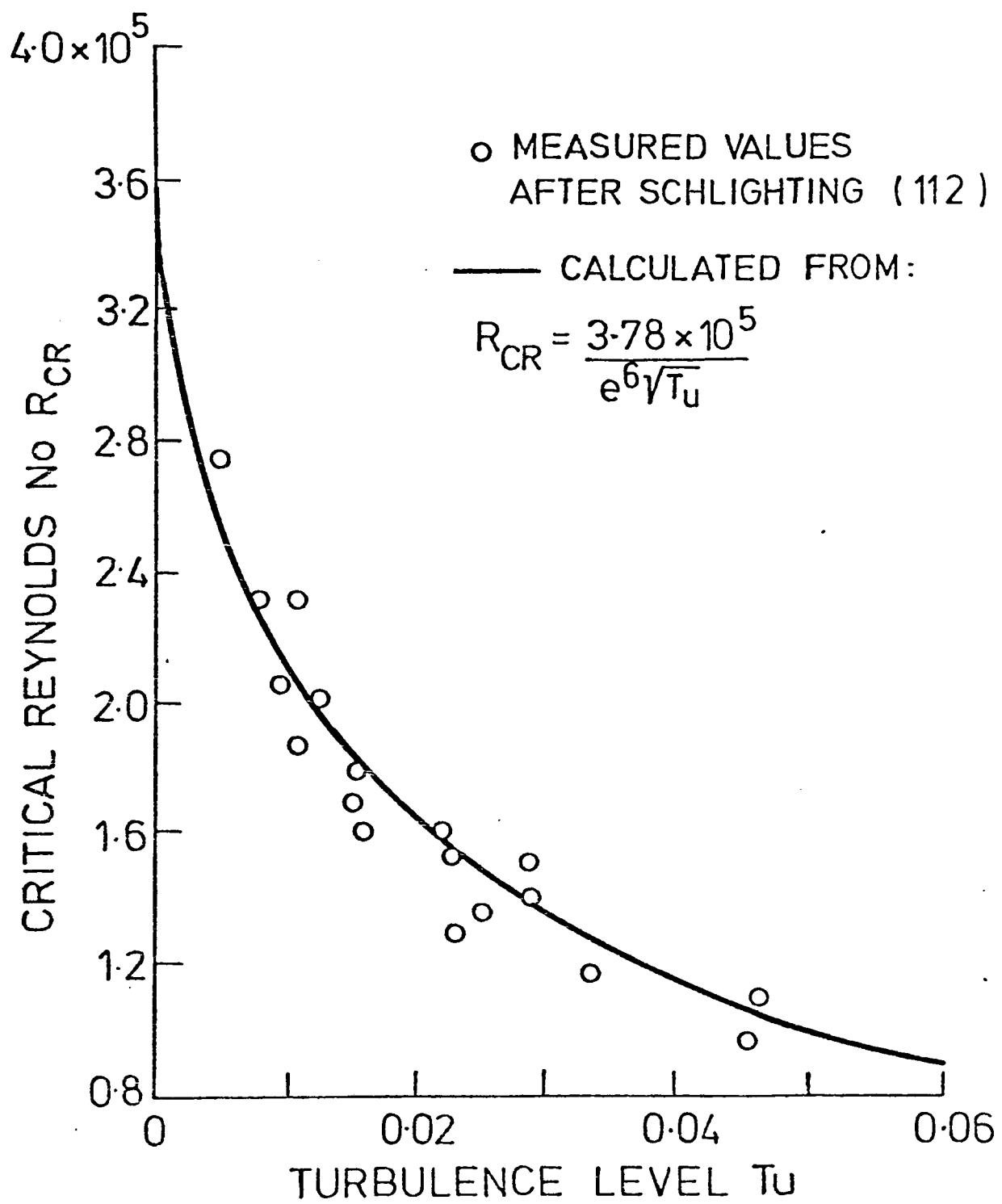


FIG.II.11 VARIATION OF CRITICAL REYNOLDS NUMBER WITH ONSET FLOW TURBULENCE LEVEL FOR A CIRCULAR CYLINDER FROM REF. (111)

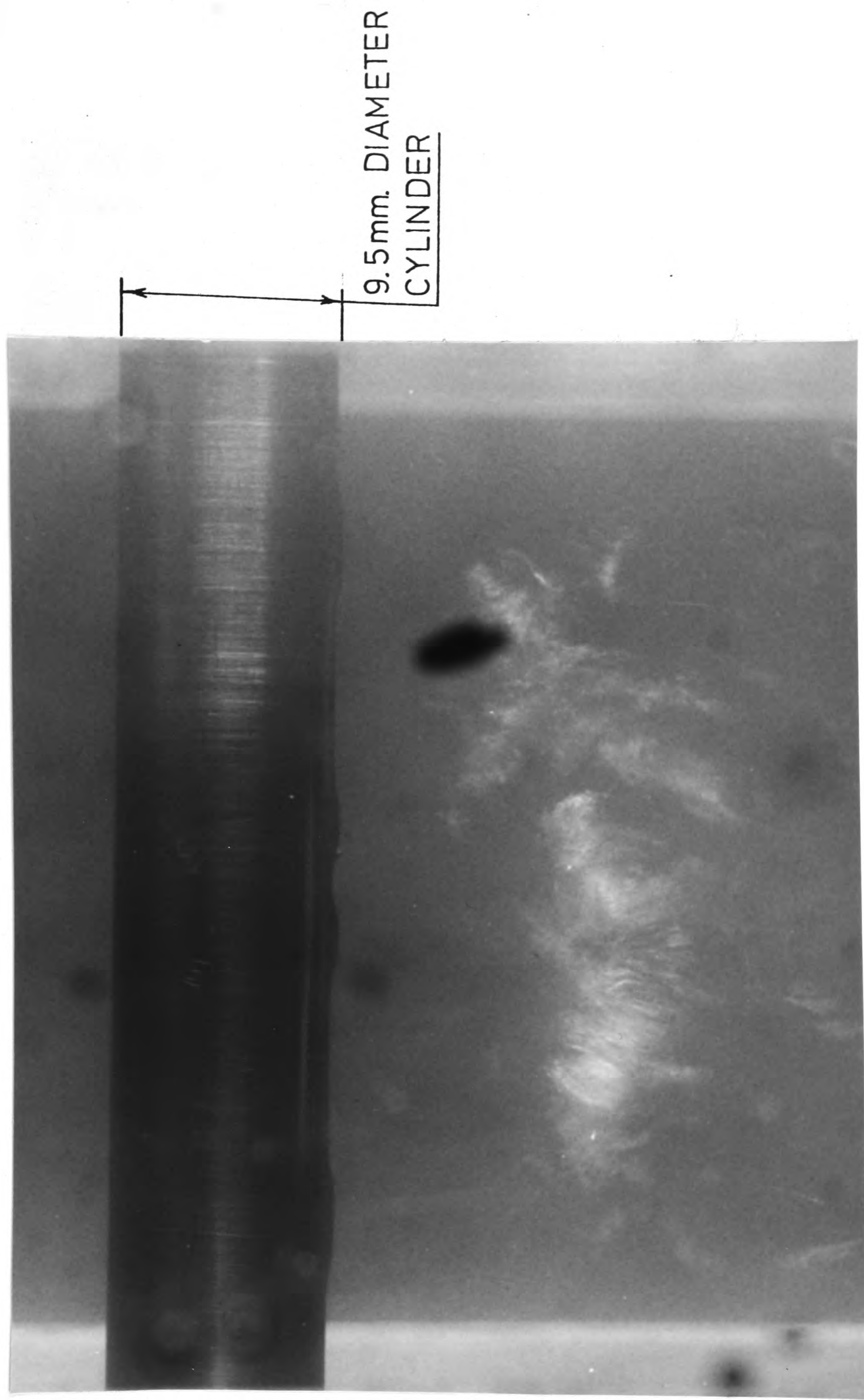


FIG. II.12 OVERHEAD VIEW OF WAKE TYPE CAVITATION INCEPTION
AT $R_d = 1.2 \times 10^5$, $\sigma_w = 2.2$

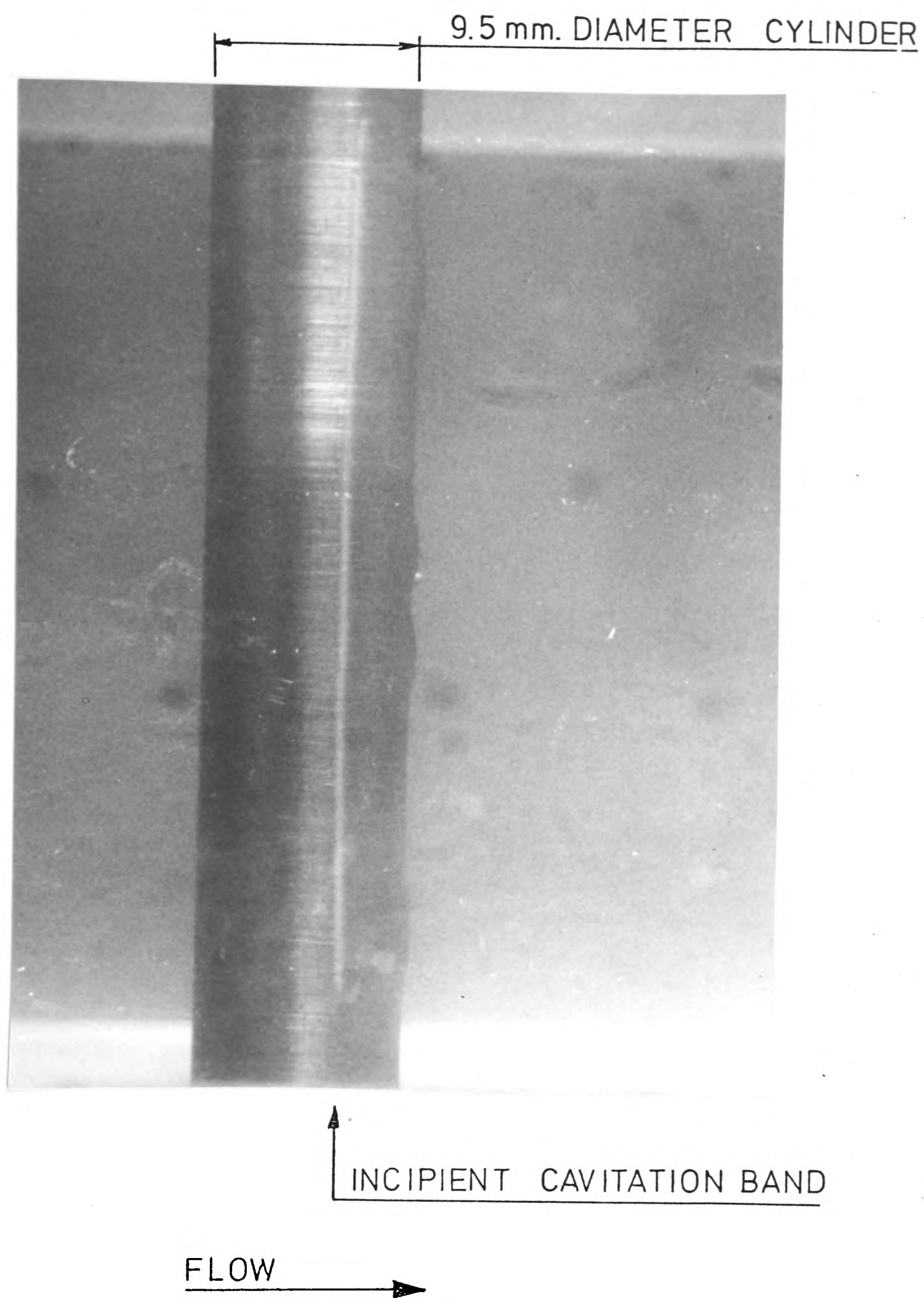


FIG. II.13 OVERHEAD VIEW OF BAND TYPE INCEPTION
AT $R_d = 2.85 \times 10^5$, $\sigma_{ib} = 2.33$

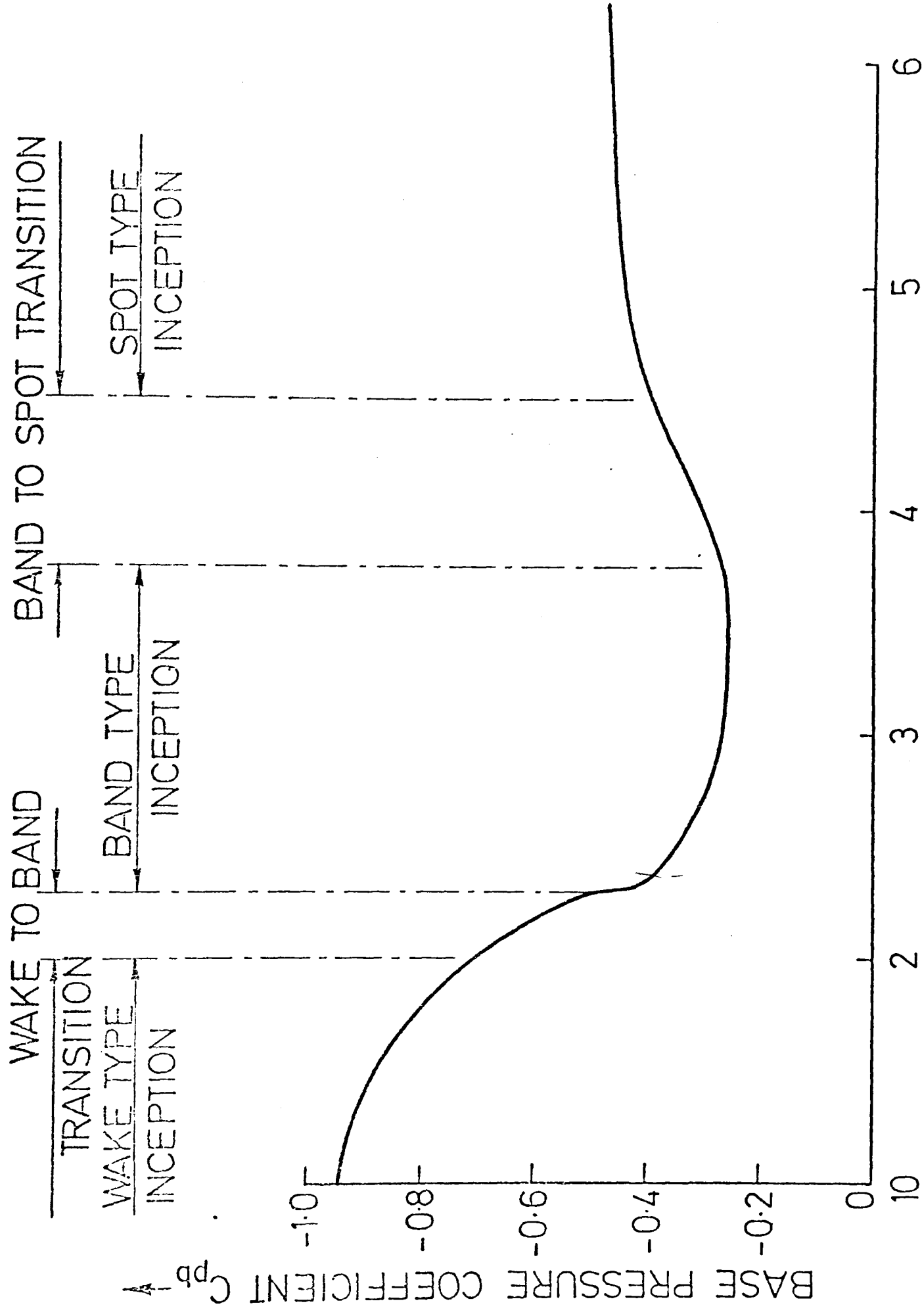


FIG.II.14 FULLY WETTED BASE PRESSURE COEFFICIENT VERSUS REYNOLDS NUMBER PLOT SHOWING CAVITATION INCEPTION REGIMES

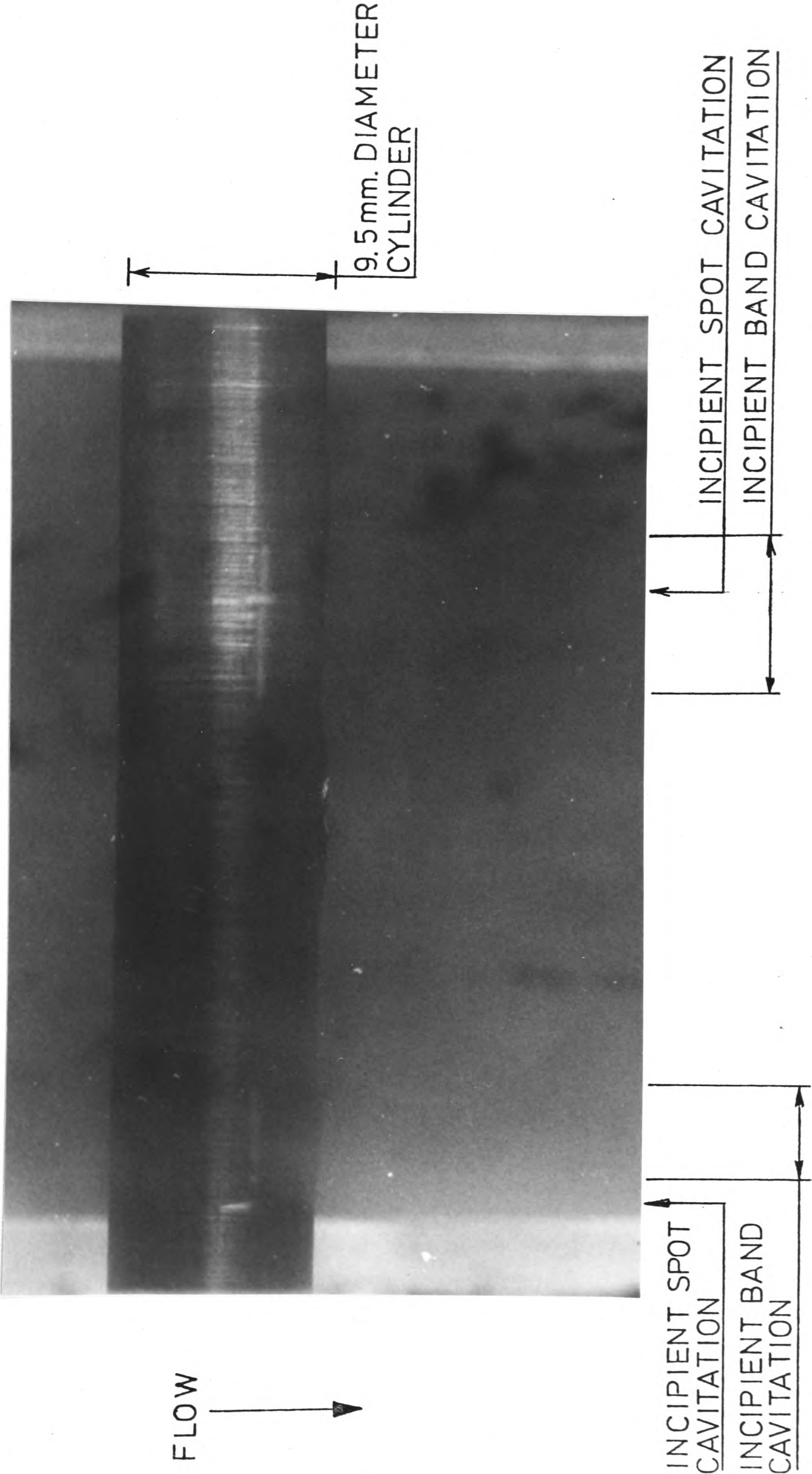


FIG.II.15 OVERHEAD VIEW OF BAND AND SPOT TYPE INCIPIENT CAVITATION AT $R_d = 3.9 \times 10^5$, $Q_v \approx 2.55$

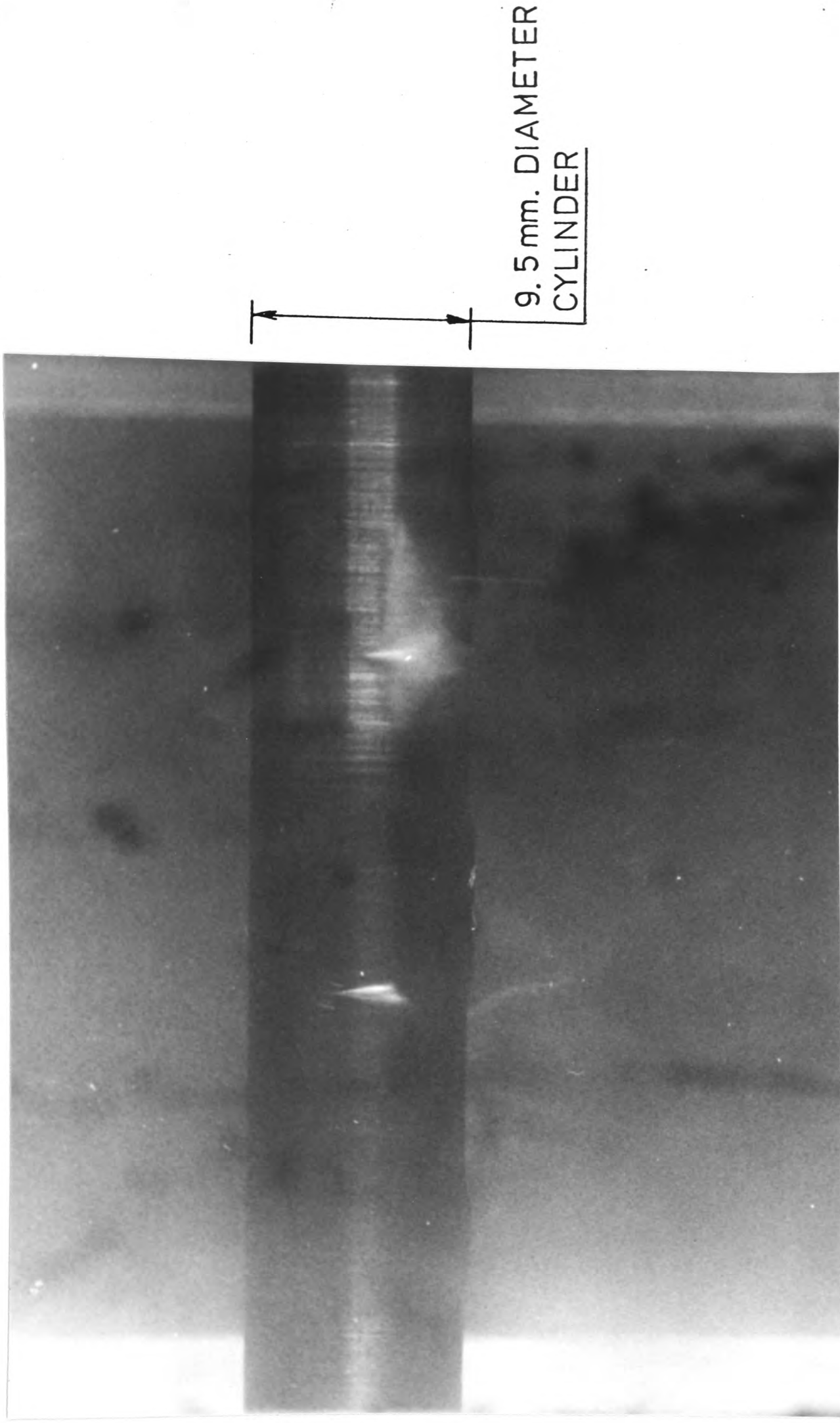


FIG. II. 16 OVERHEAD VIEW OF SPOT TYPE CAVITATION INCEPTION
AT $R_d = 5 \times 10^5$, $\sigma_{is} = 2.45$

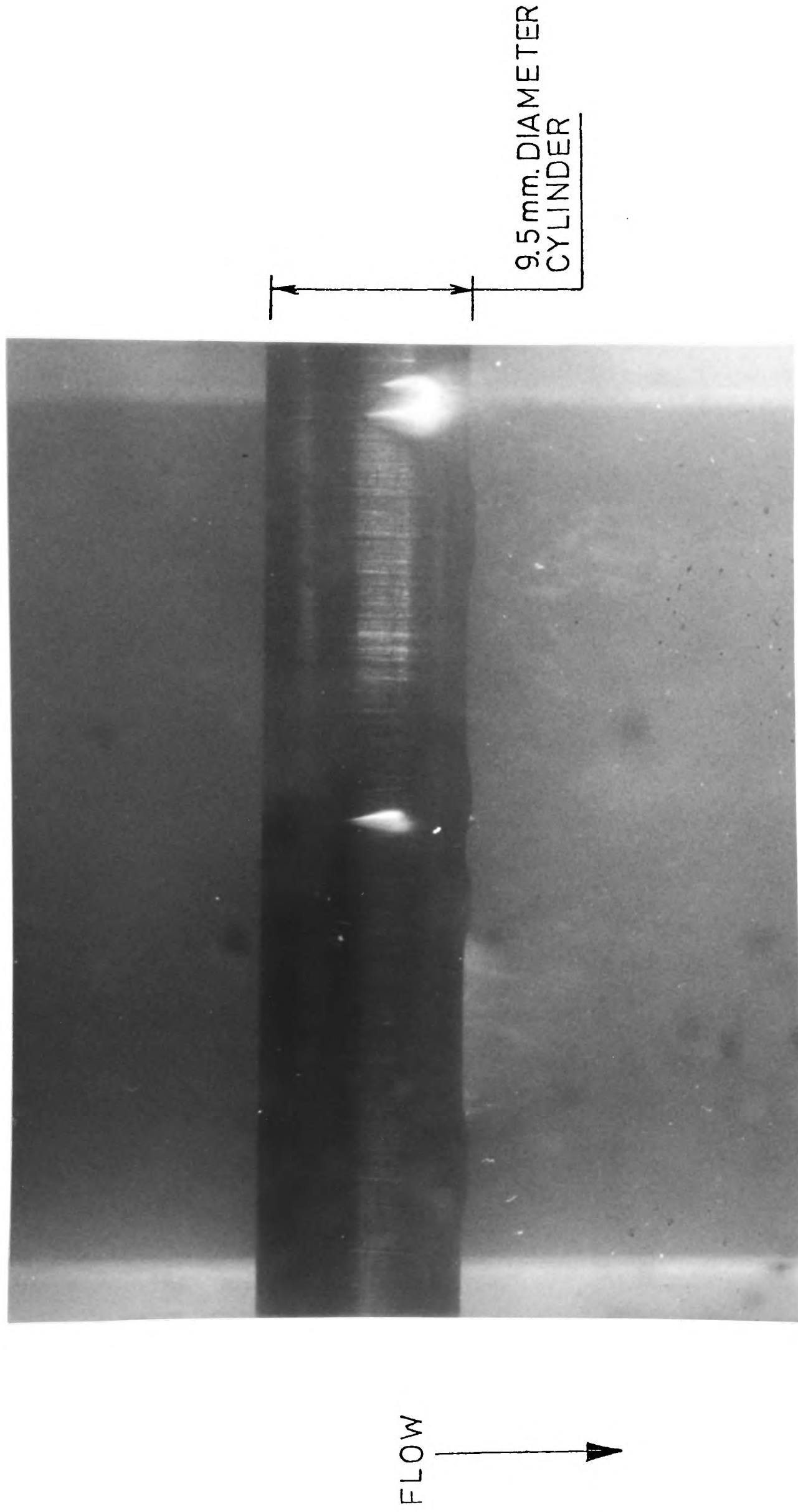


FIG. II.17 OVERHEAD VIEW OF SPOT TYPE INCIPIENT CAVITATION
AT $R_d = 7.1 \times 10^5$, $\sigma_{is} = 2.23$

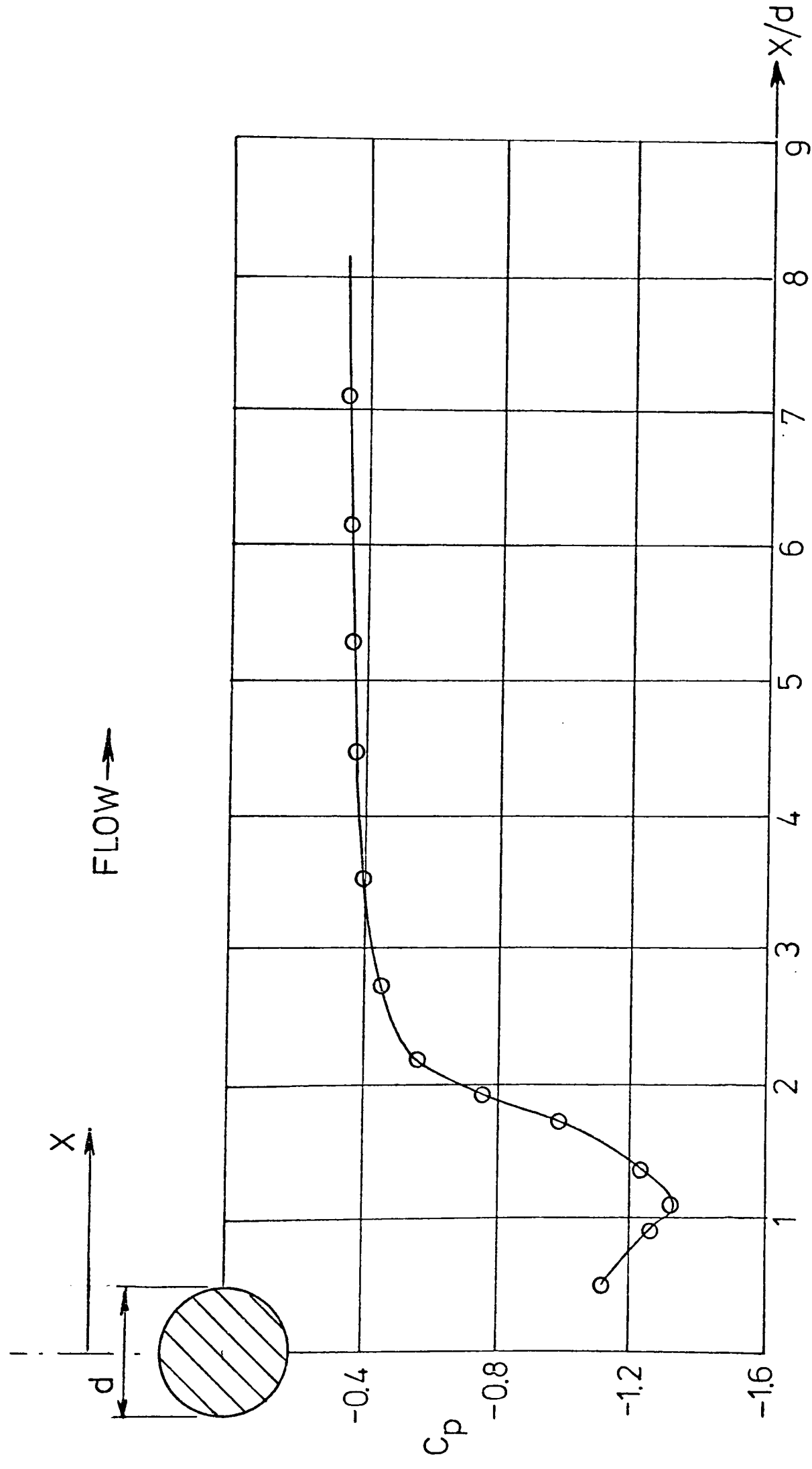


FIG. II.18. STATIC PRESSURE DISTRIBUTION ON THE CENTRELINE OF A WAKE BEHIND A CIRCULAR CYLINDER AT A SUBCRITICAL REYNOLDS NUMBER. AFTER ROSHKO (114)

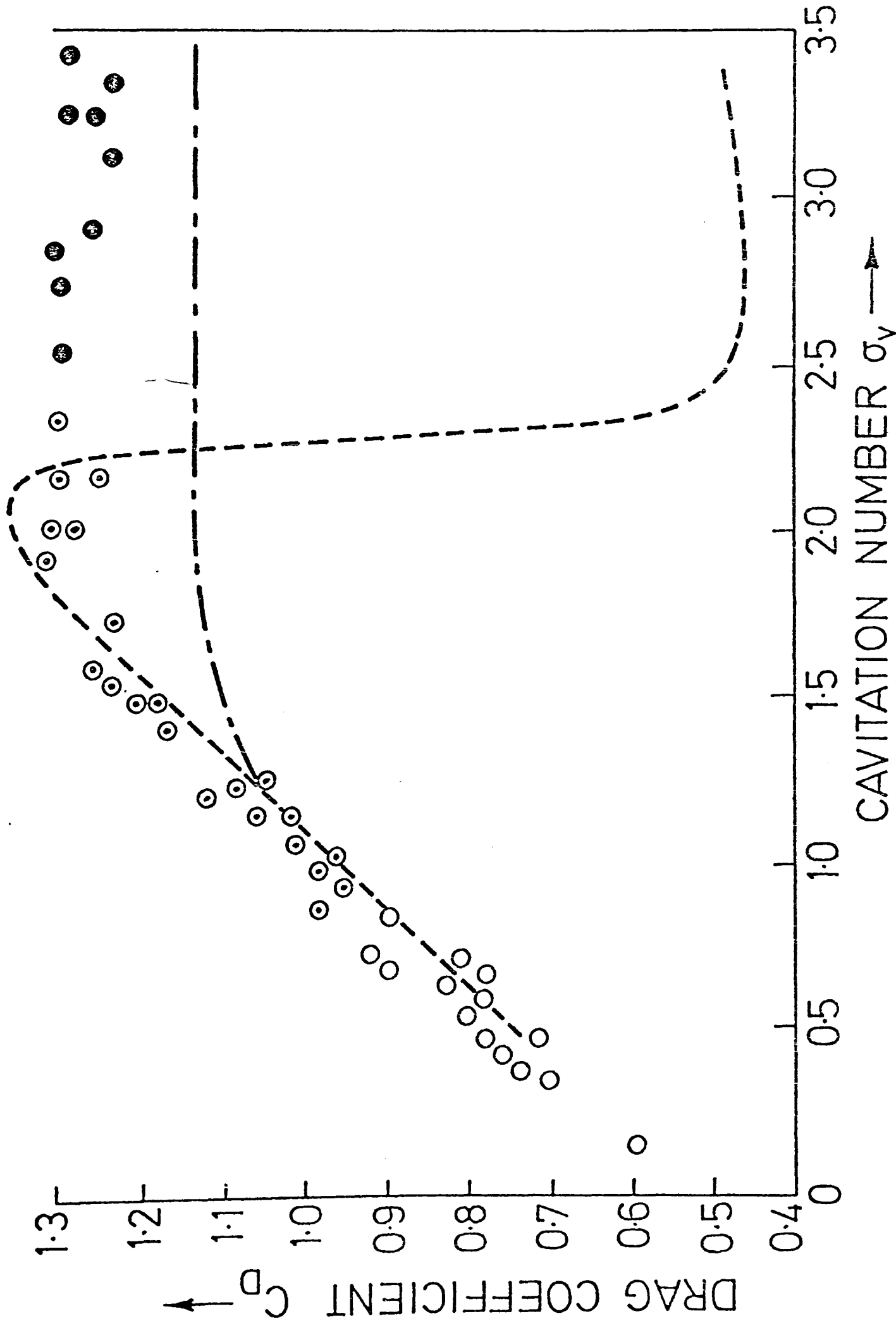


FIG. II. 19 DRAG COEFFICIENTS OF CAVITATING CIRCULAR CYLINDERS. DATA OF MARTYRER (120) AND SILBERMAN (67)
 DATA OF MARTYRER (120) $---$ $R_d < 6 \times 10^4$, $---$ $R_d > 2 \times 10^5$
 DATA OF SILBERMAN (67) $\bigcirc$ DEVELOPED CAVITIES, $\bullet$ BUBBLE CAVITATION, $\bullet$ FULLY WETTED

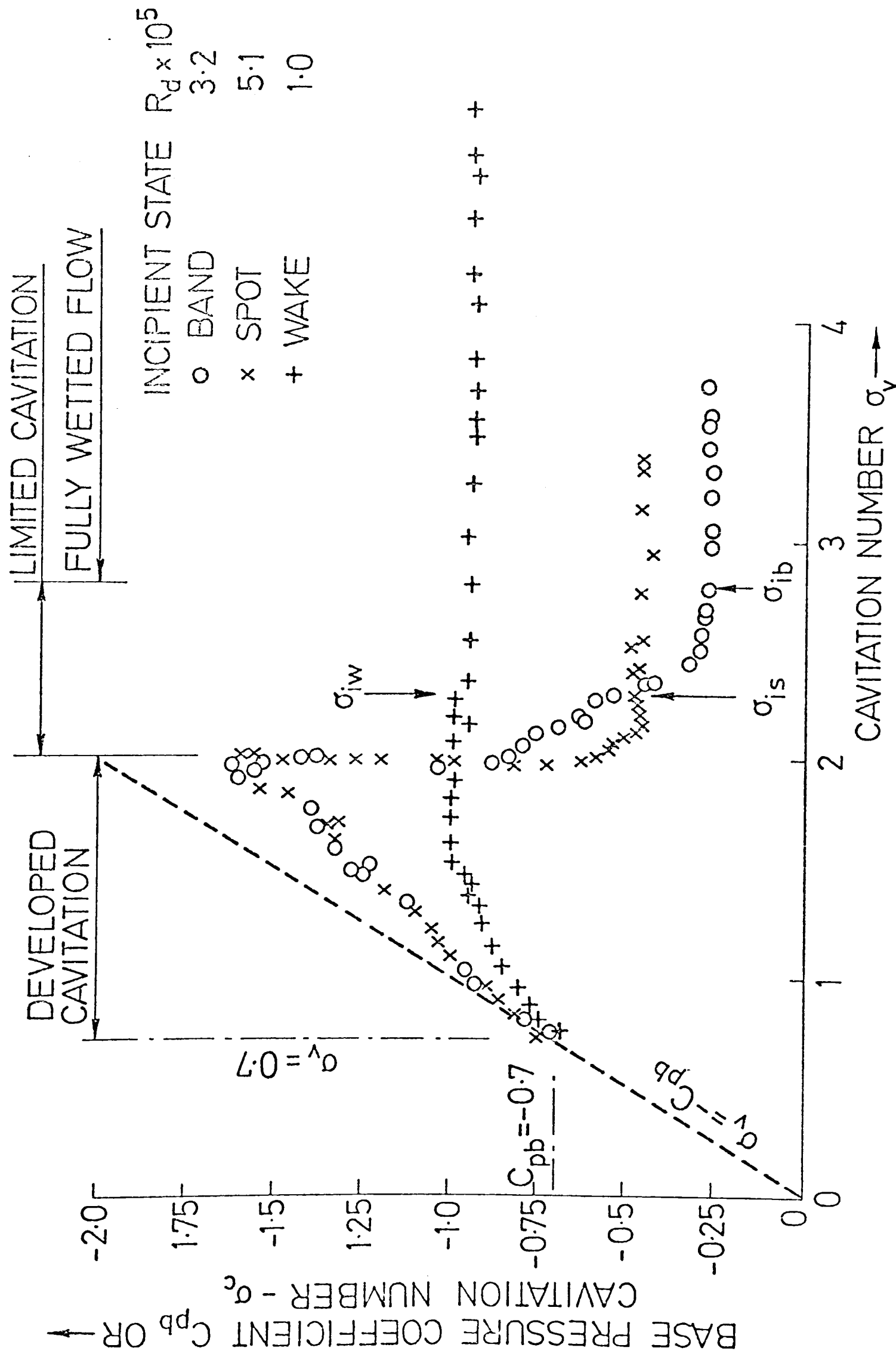


FIG. II. 20 CYLINDER BASE PRESSURE COEFFICIENT VERSUS CAVITATION NUMBER FOR THE DEVELOPMENT OF CAVITATION FROM THREE INCIPIENT STATES

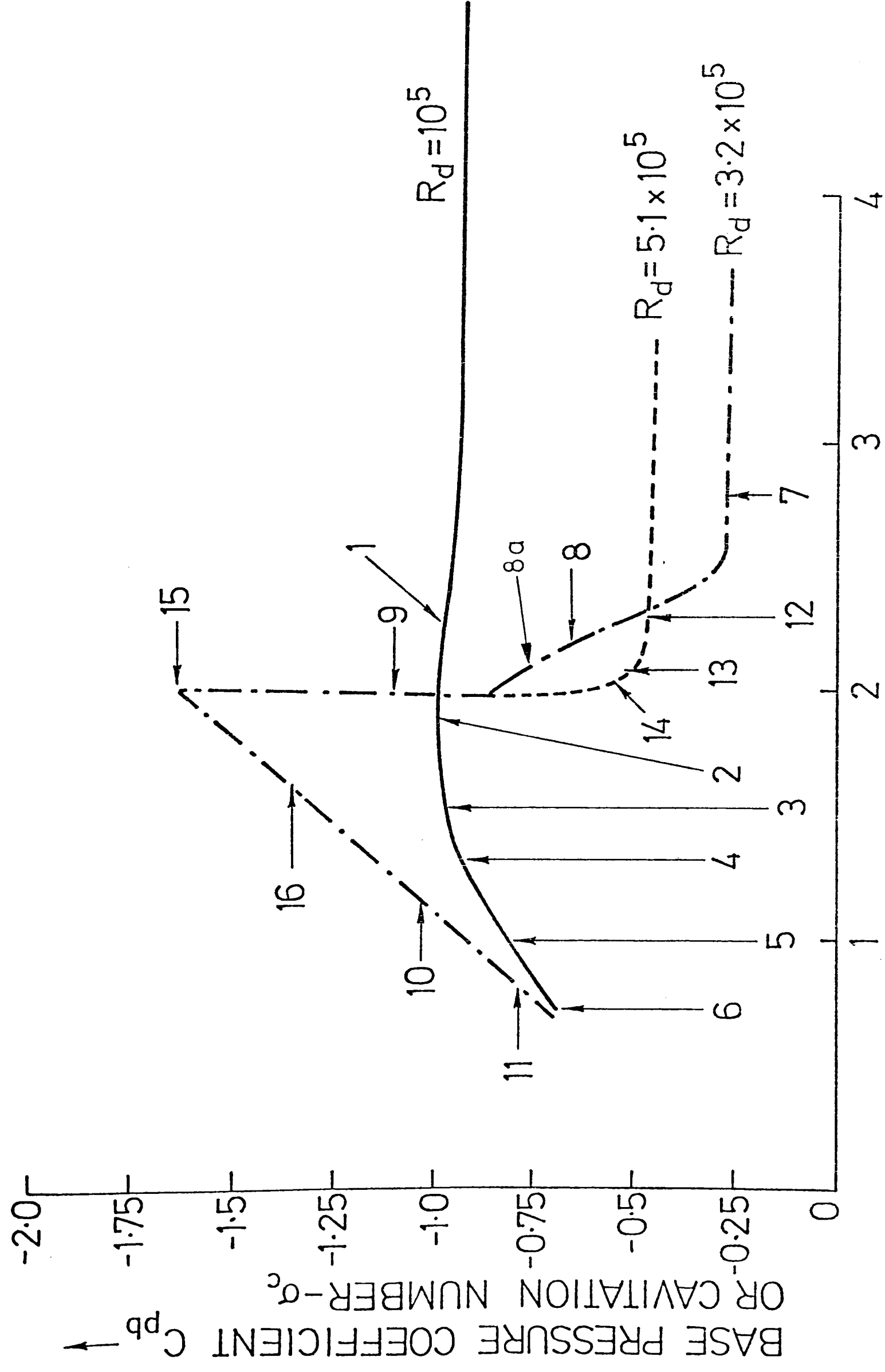
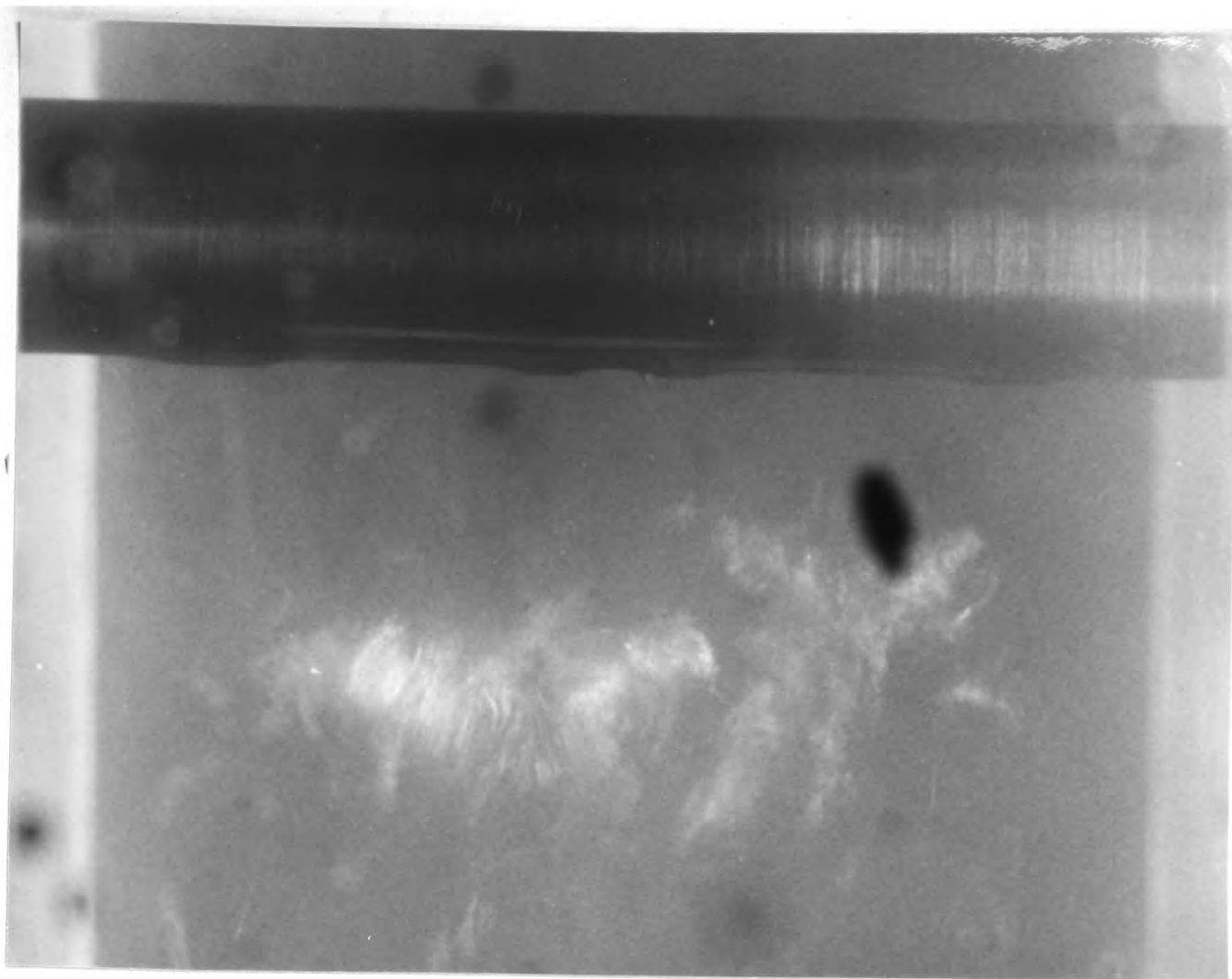
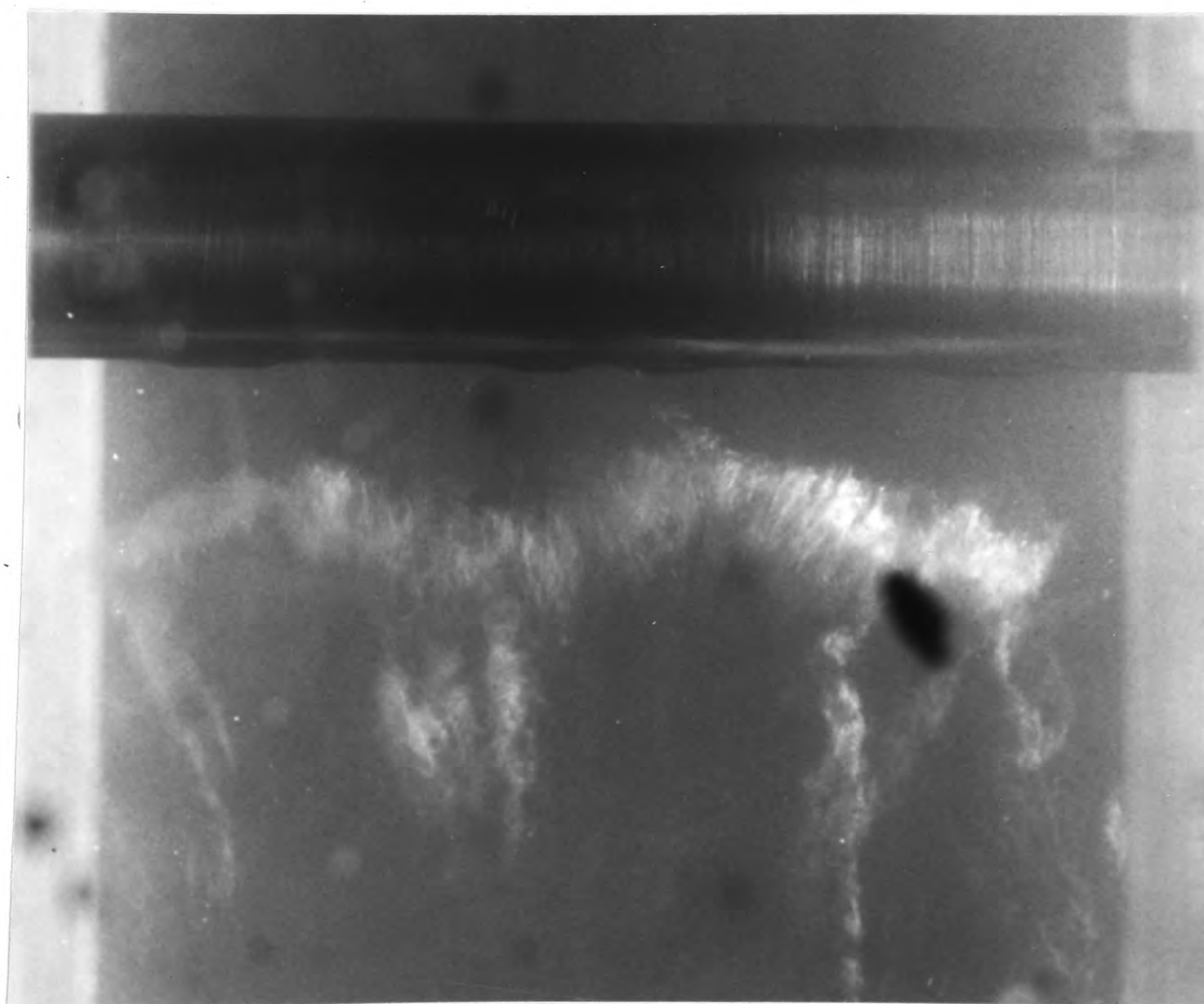


FIG. II.21 CYLINDER BASE PRESSURE COEFFICIENT VERSUS CAVITATION NUMBER.
KEY TO PHOTOGRAPHS IN FIGS. II.22, II.23 AND II.24



FRAME 1 (OVERHEAD VIEW)

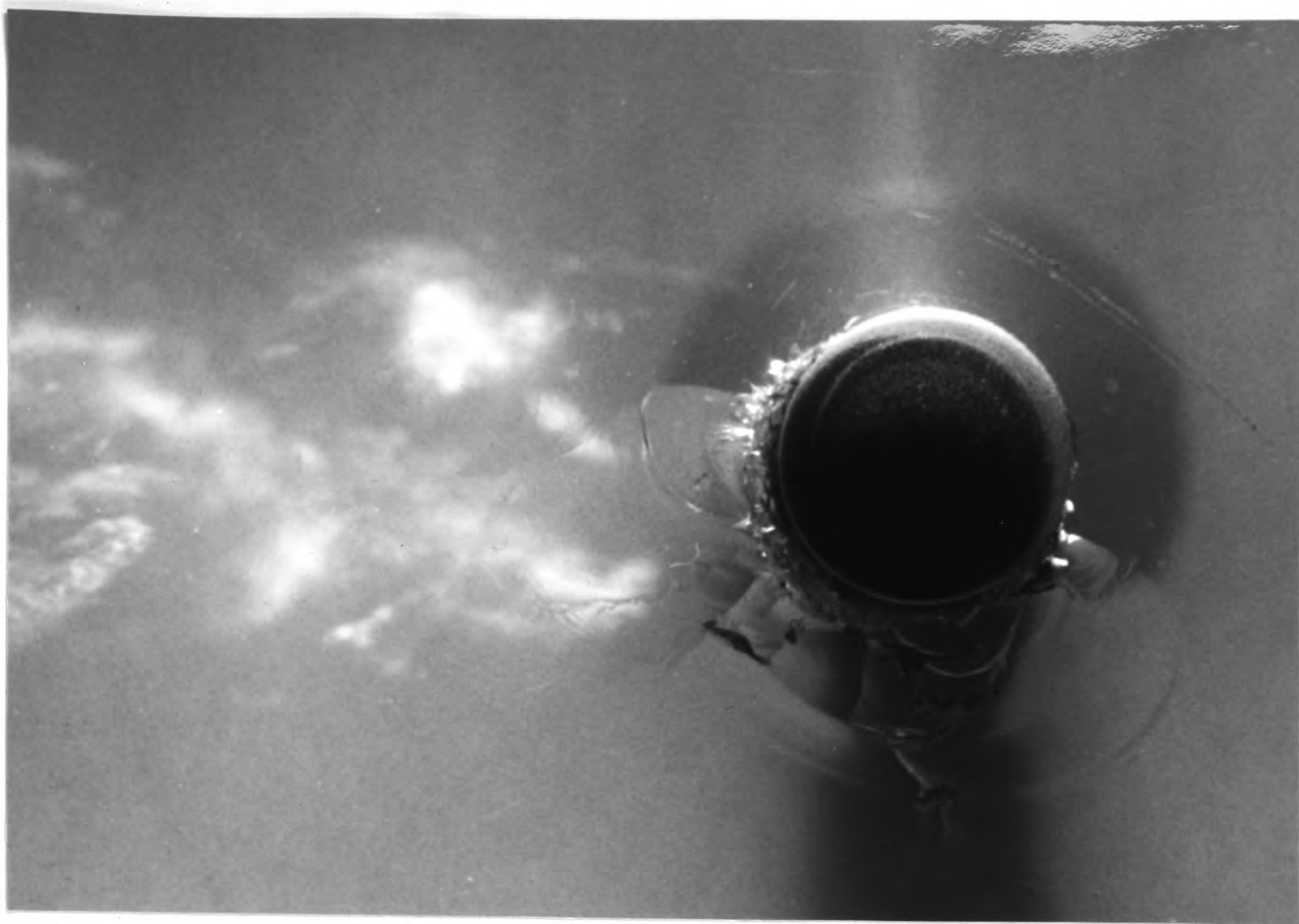
FLOW



FRAME 2 (OVERHEAD VIEW)

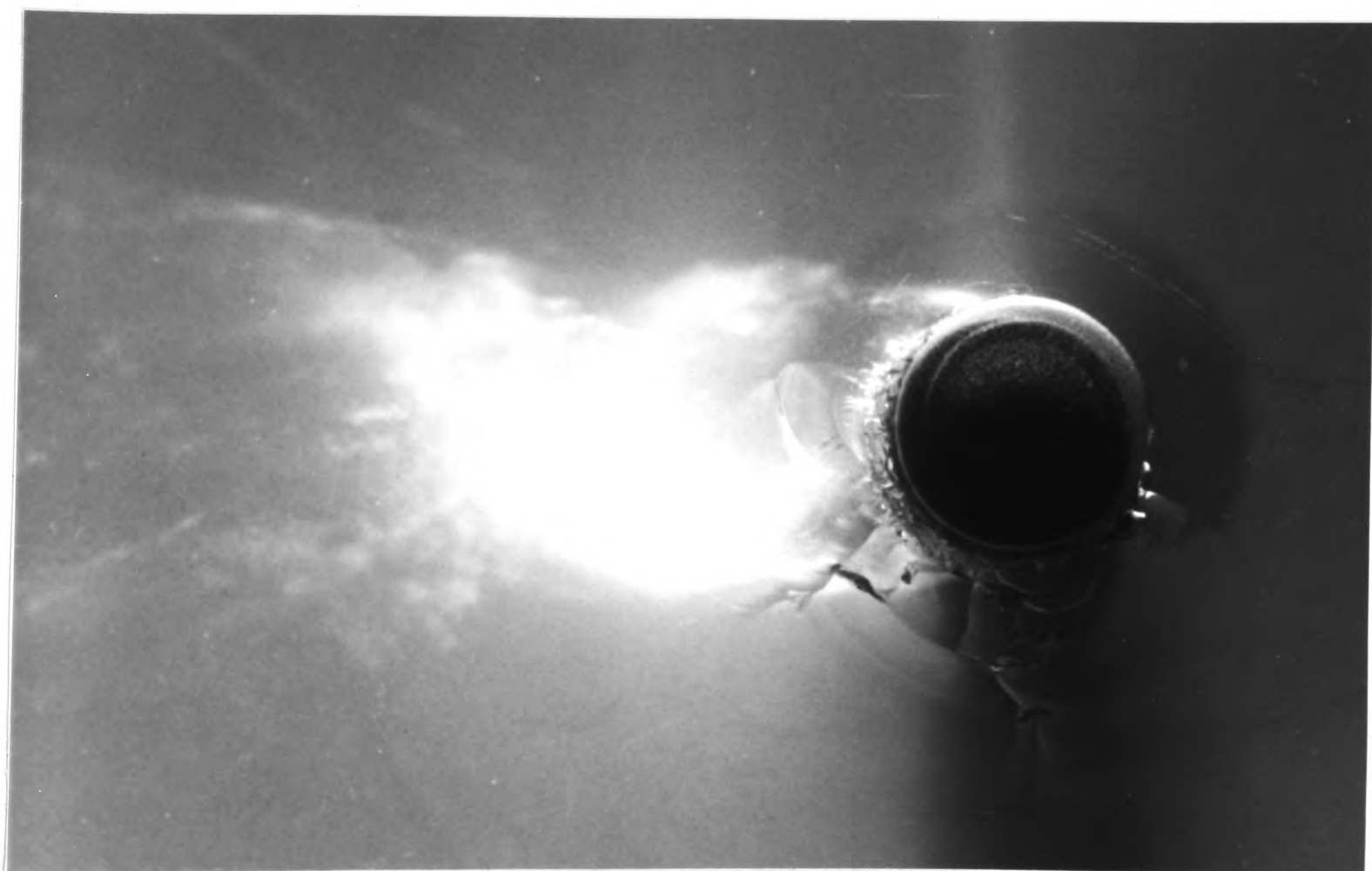
FIG. II. 22 OVERHEAD AND END VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT WAKE STATE

FRAMES 1 AND 2



FRAME 3 (END VIEW)

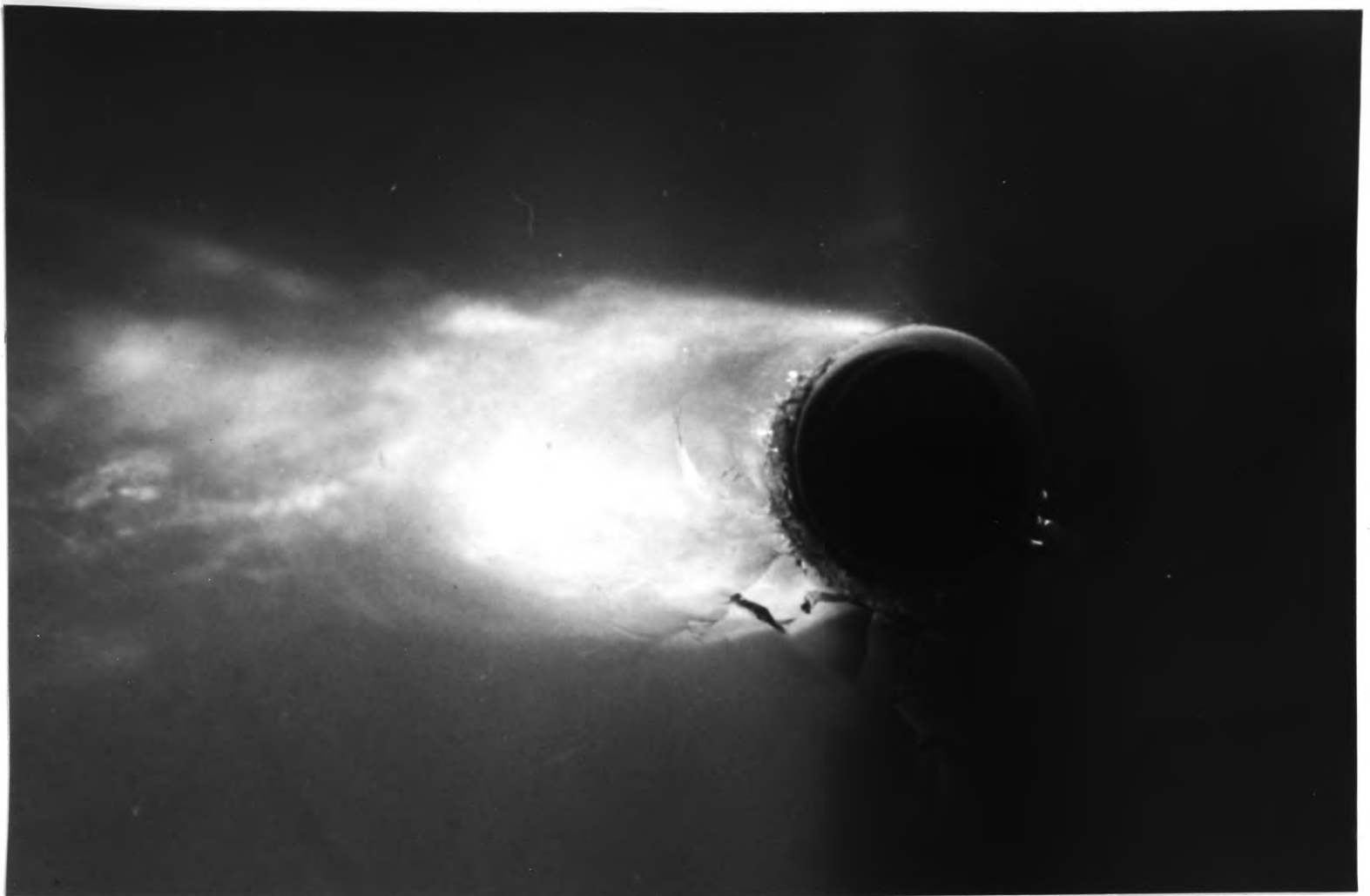
← FLOW



FRAME 4 (END VIEW)

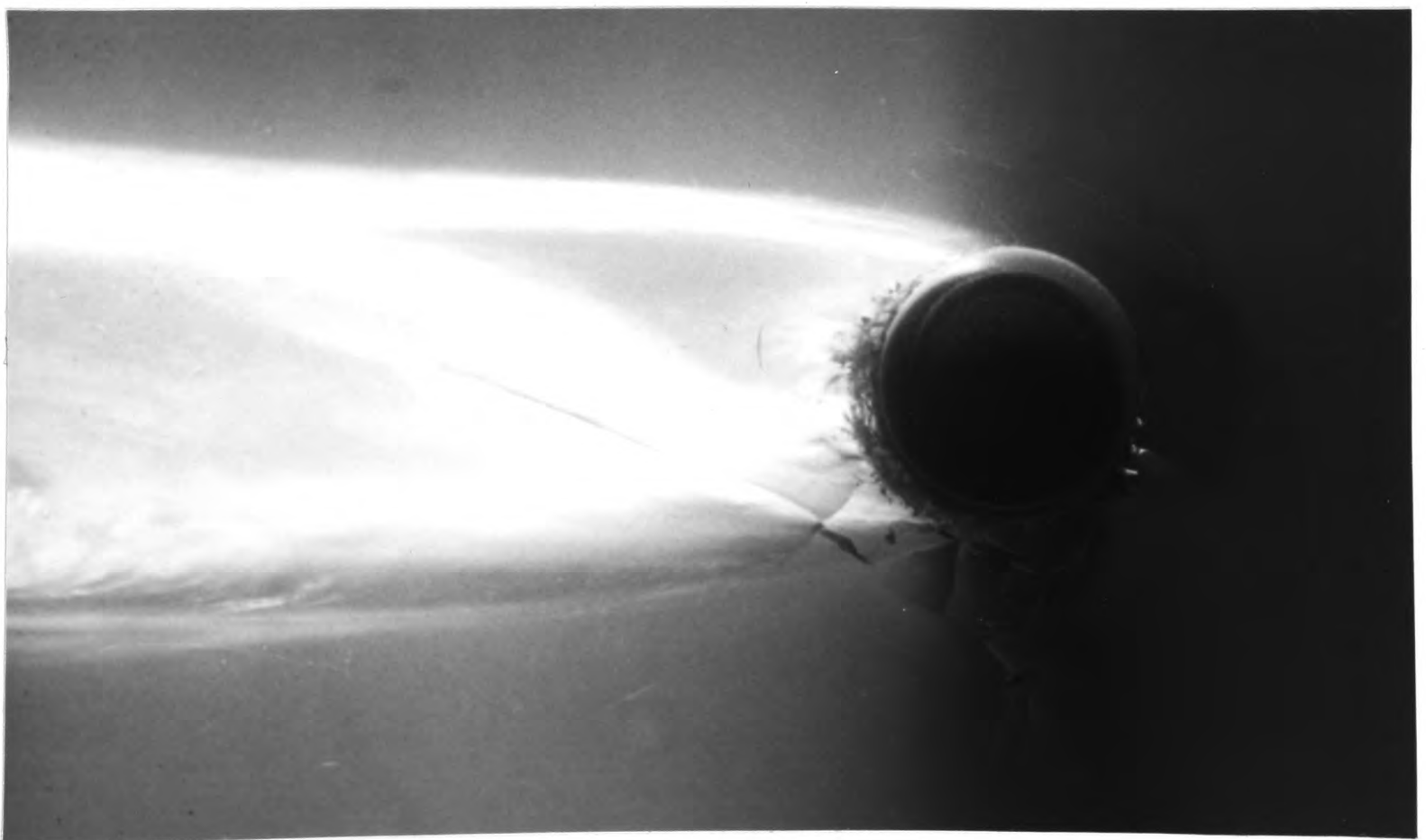
FIG. II.22 OVERHEAD AND END VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT WAKE STATE

FRAMES 3 AND 4



FRAME 5 (END VIEW)

← FLOW

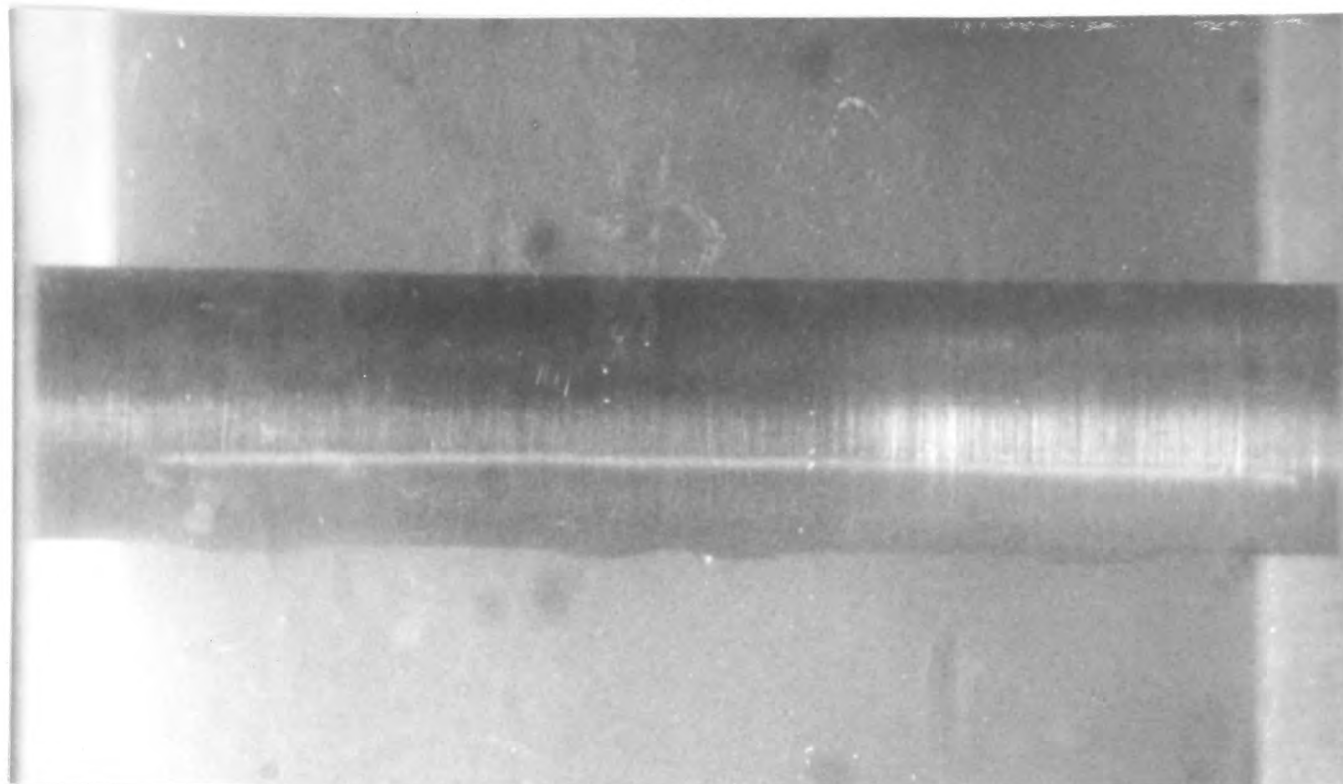


FRAME 6 (END VIEW)

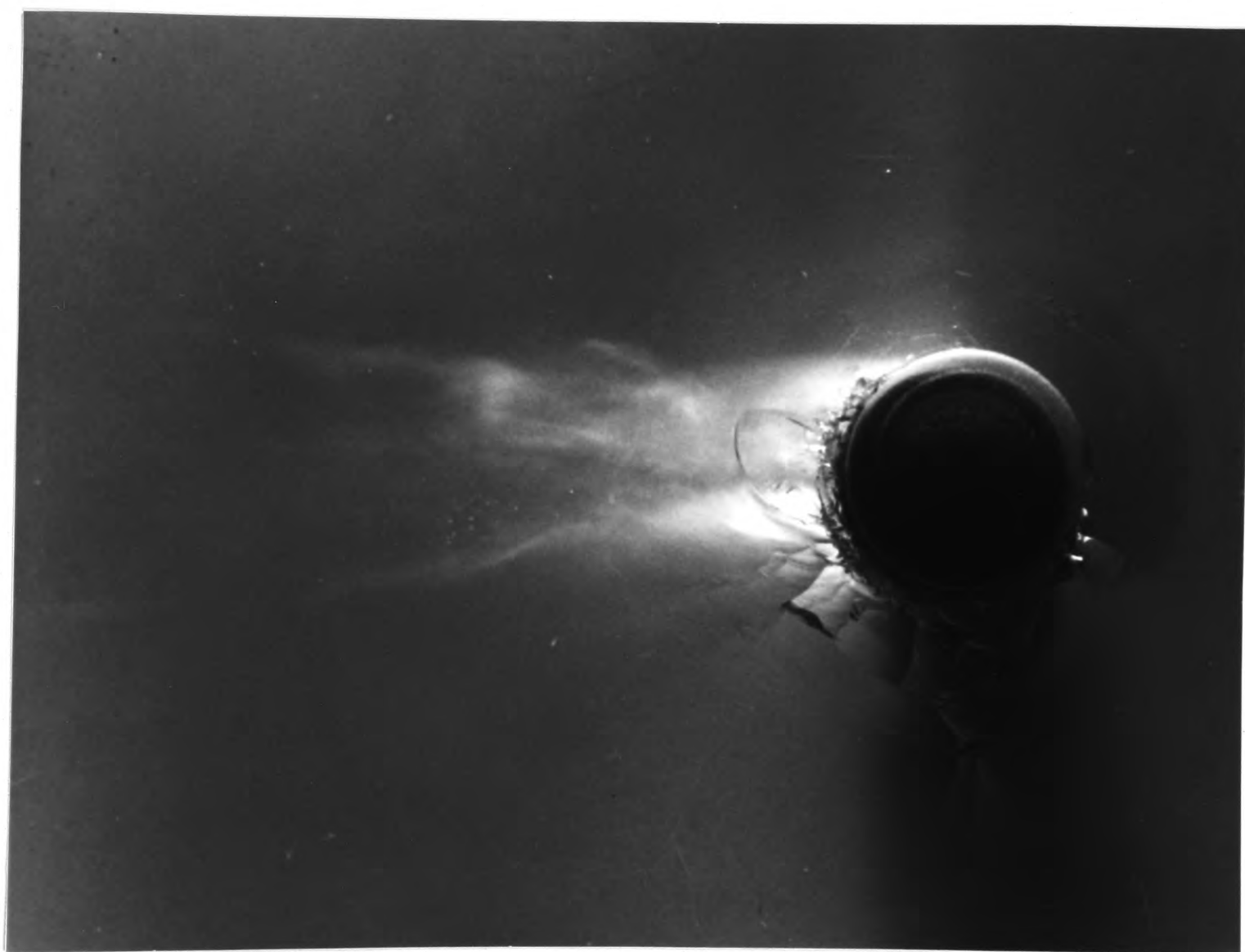
FIG. II.22 OVERHEAD AND END VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT WAKE STATE

FRAMES 5 AND 6

FLOW
↓



FRAME 7 (OVERHEAD VIEW)

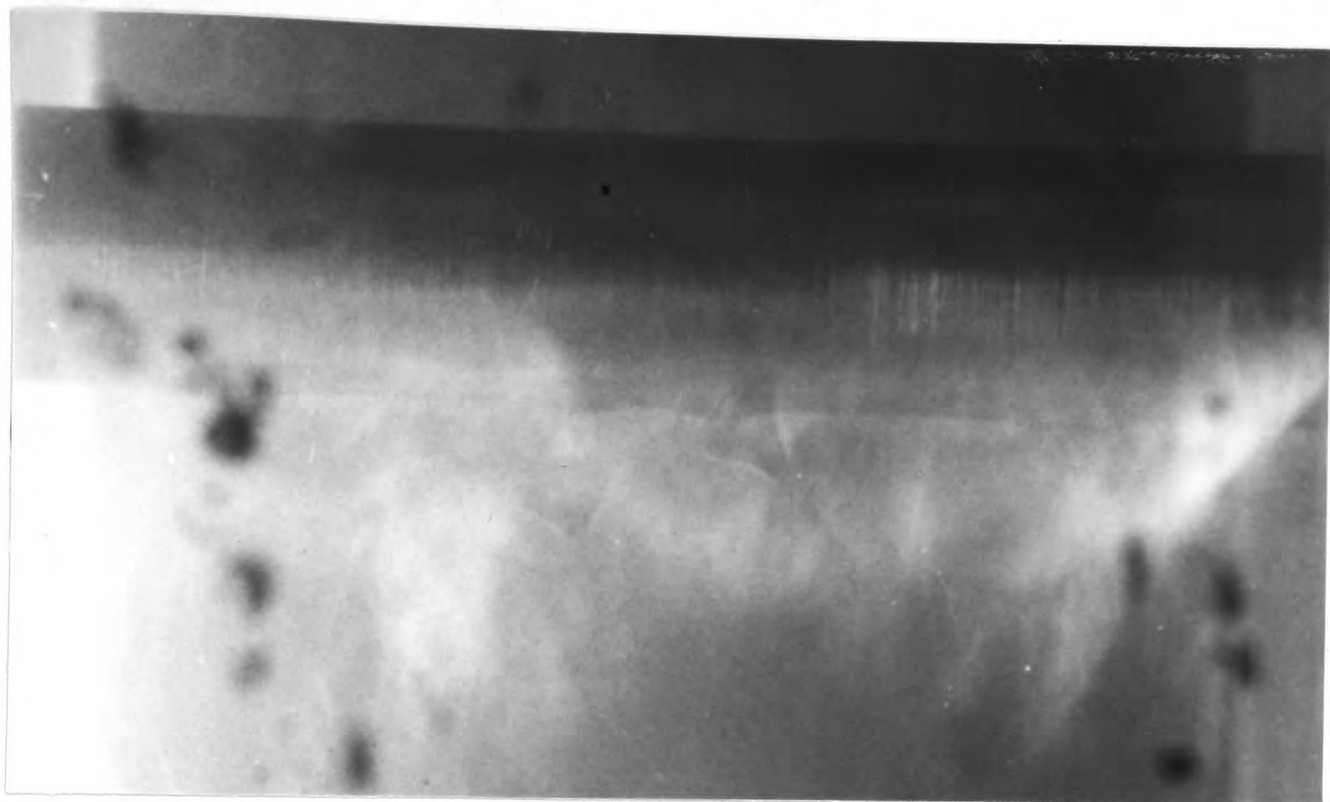


FRAME 8 (END VIEW)

FIG. II.23 OVERHEAD AND END VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT BAND STATE

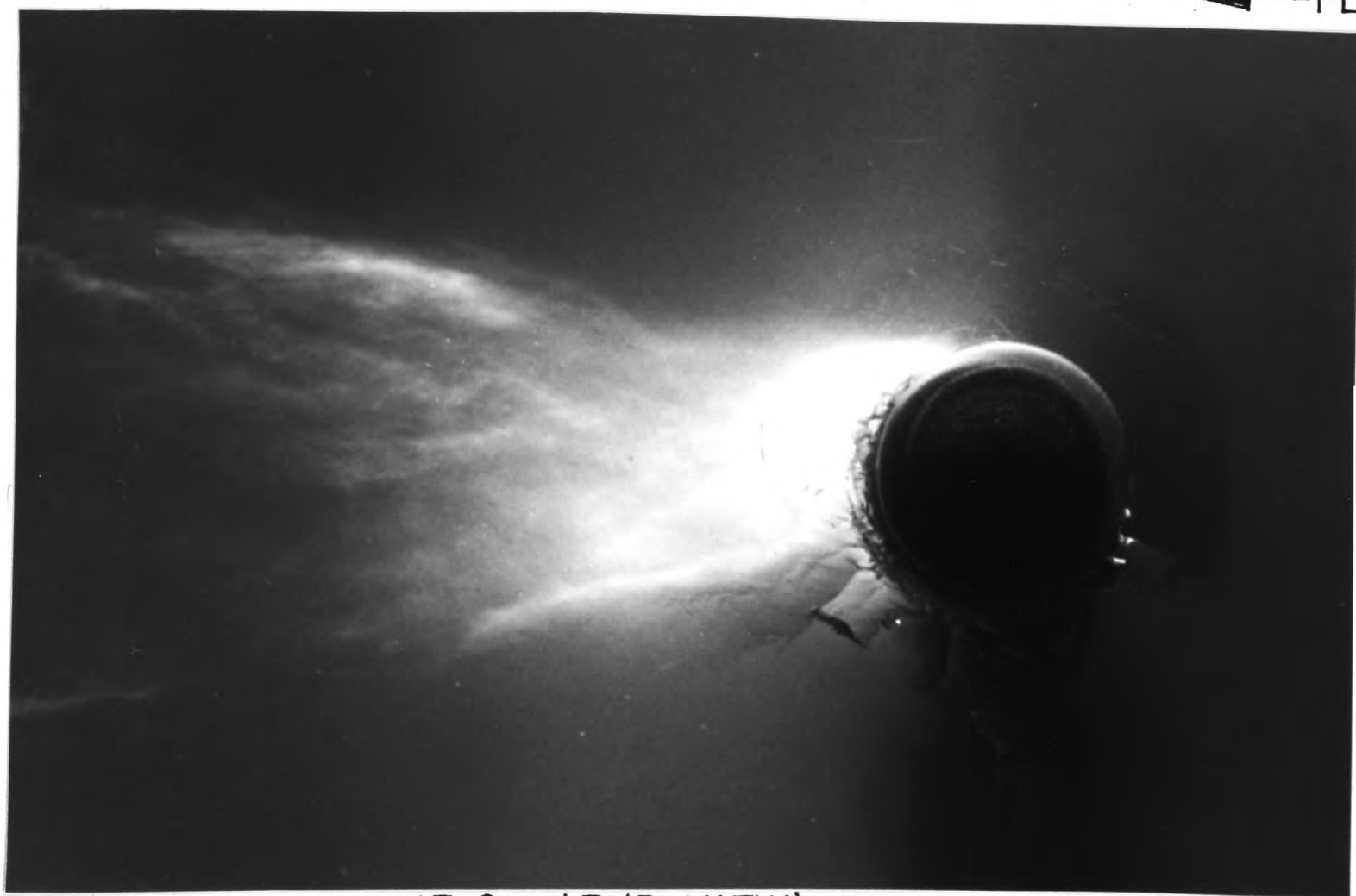
FRAMES 7 AND 8

FLOW
↓



FRAME 8a (OVERHEAD VIEW)

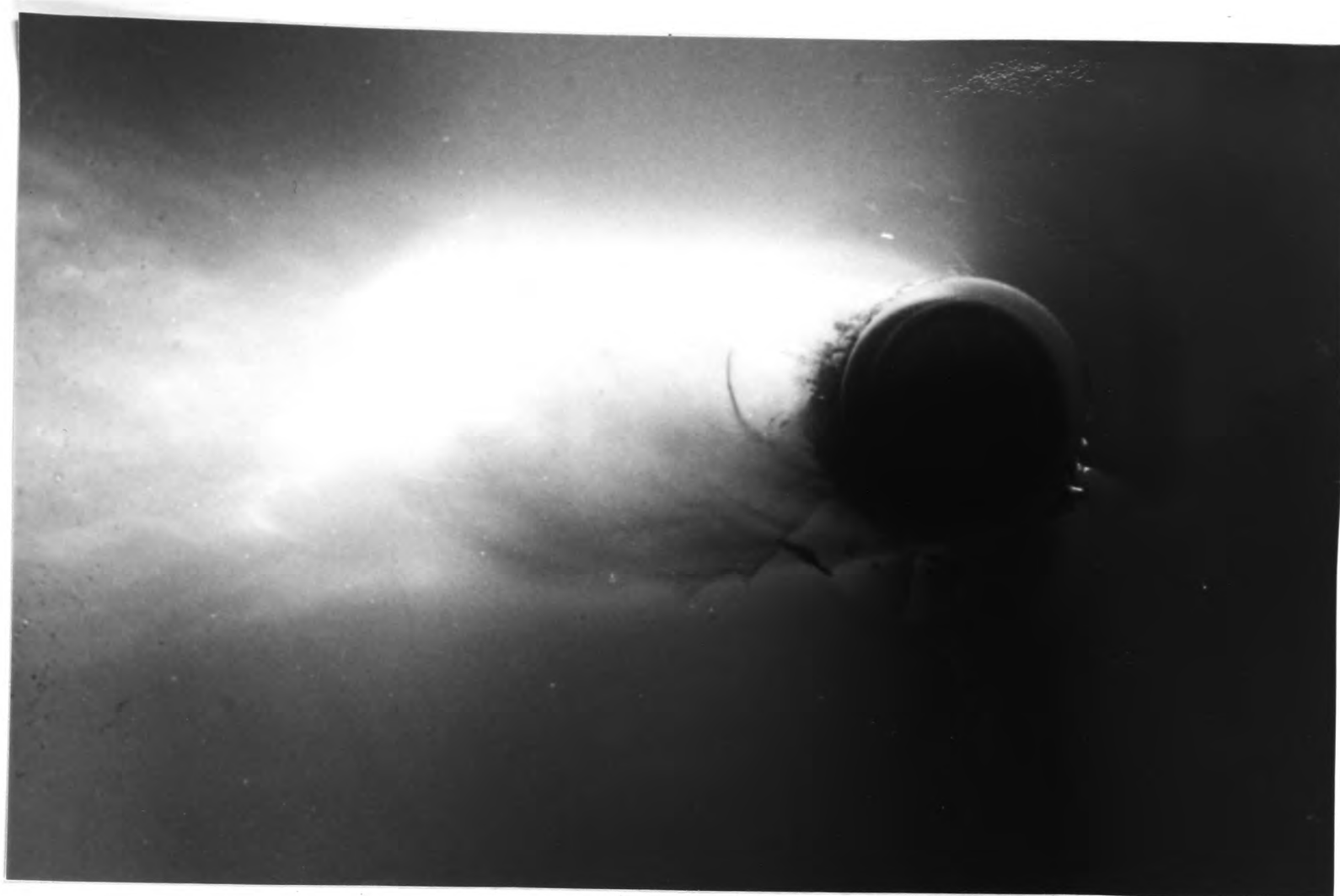
← FLOW



FRAME 9 (END VIEW)

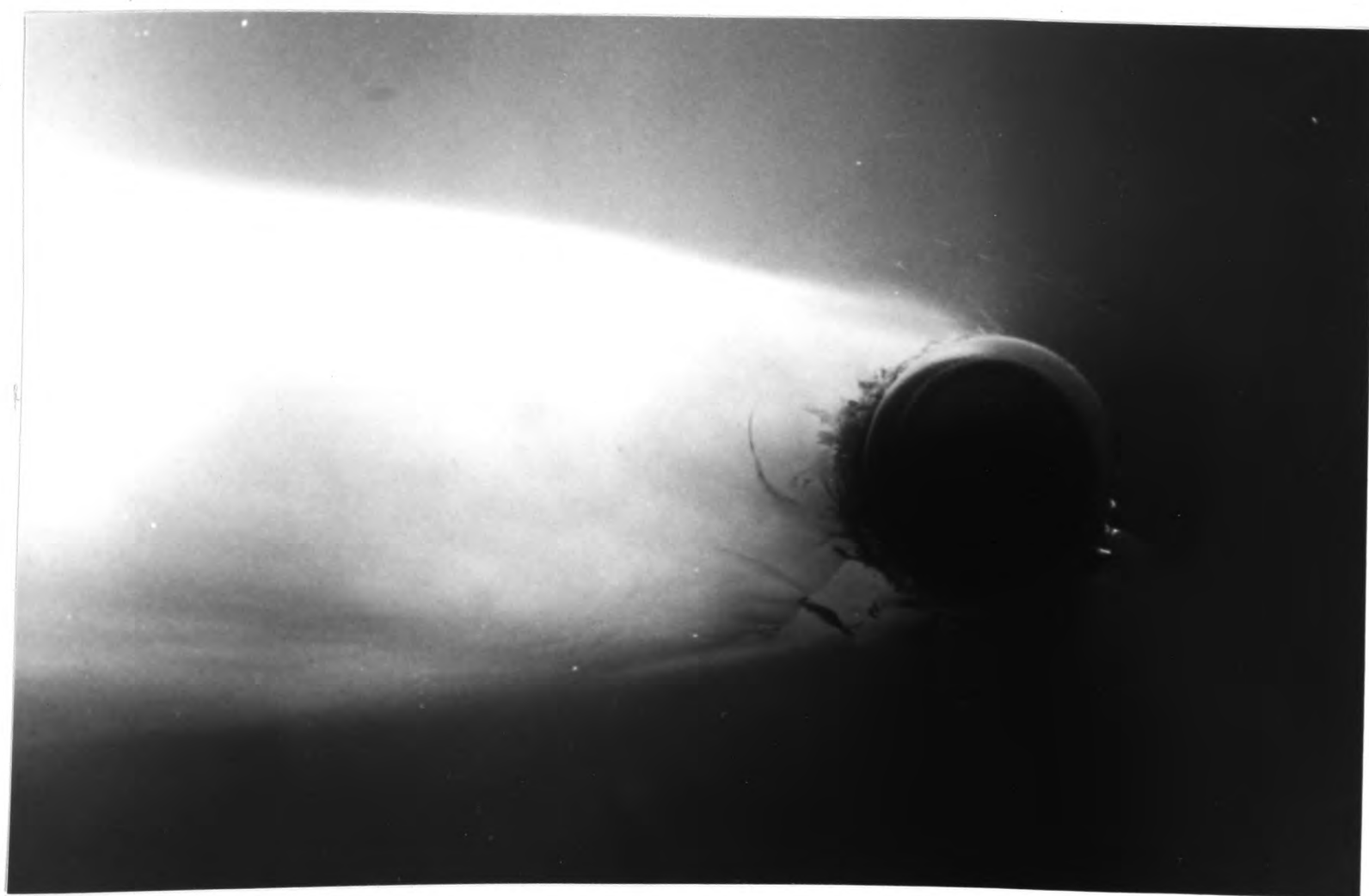
FIG. II. 23 OVERHEAD AND END VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT BAND STATE

FRAMES 8a AND 9



FRAME 10 (END VIEW)

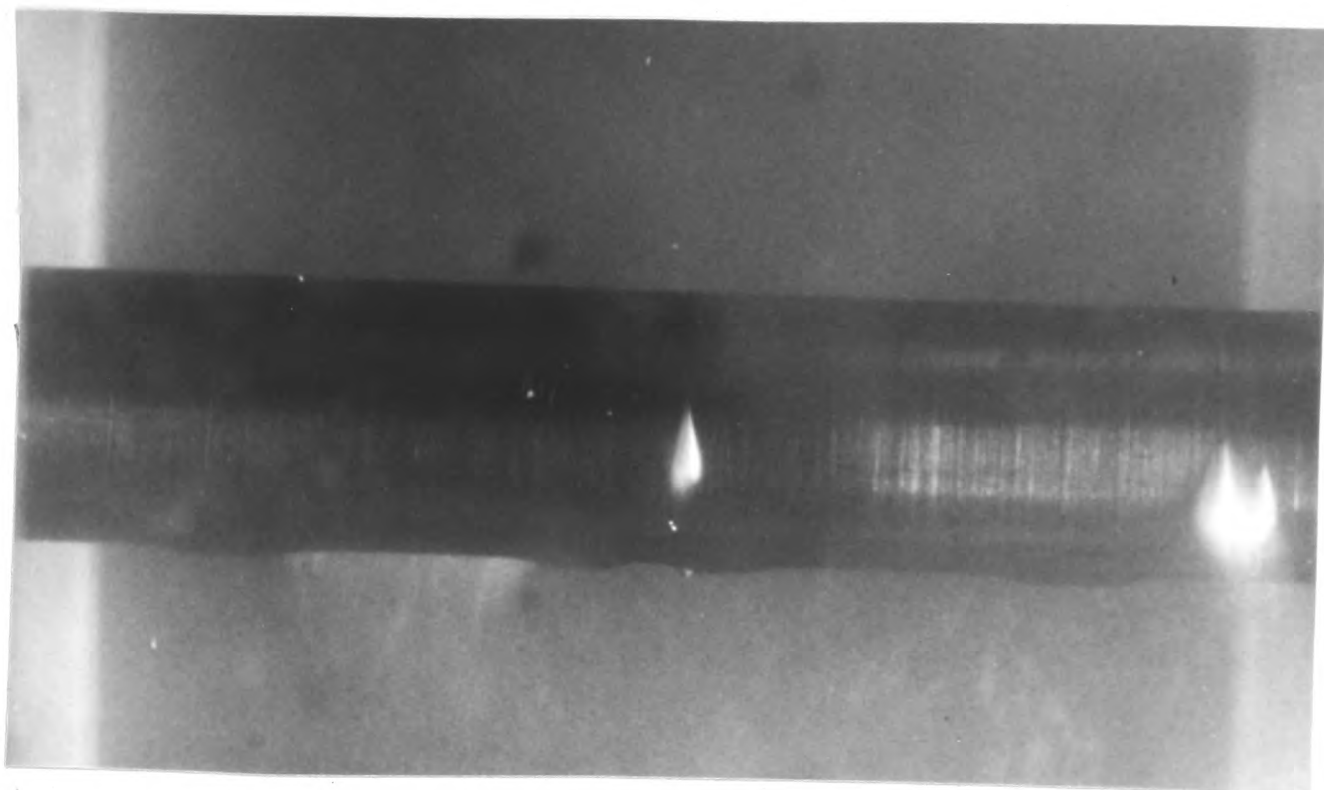
← FLOW



FRAME 11 (END VIEW)

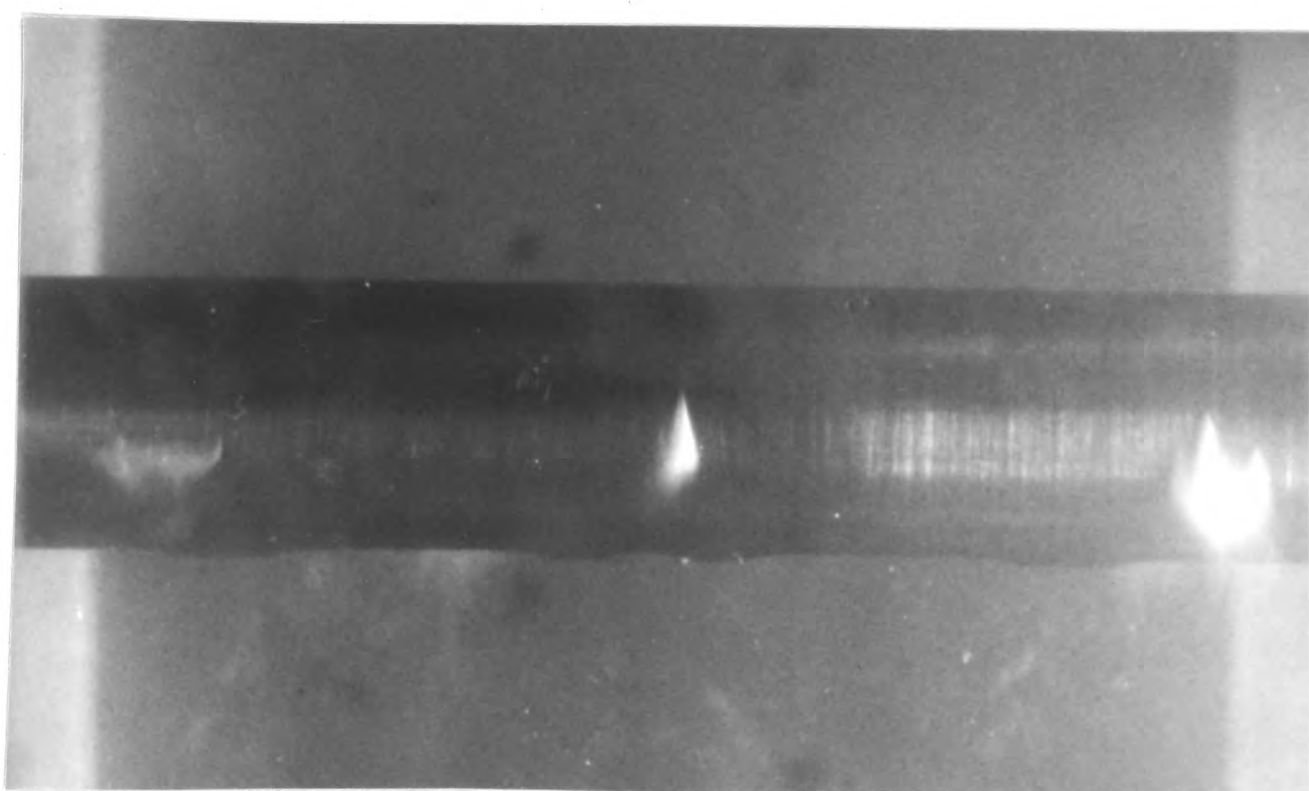
FIG. II. 23 OVERHEAD AND END VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT BAND STATE

FRAMES 10 AND 11



FRAME 12 (OVERHEAD VIEW)

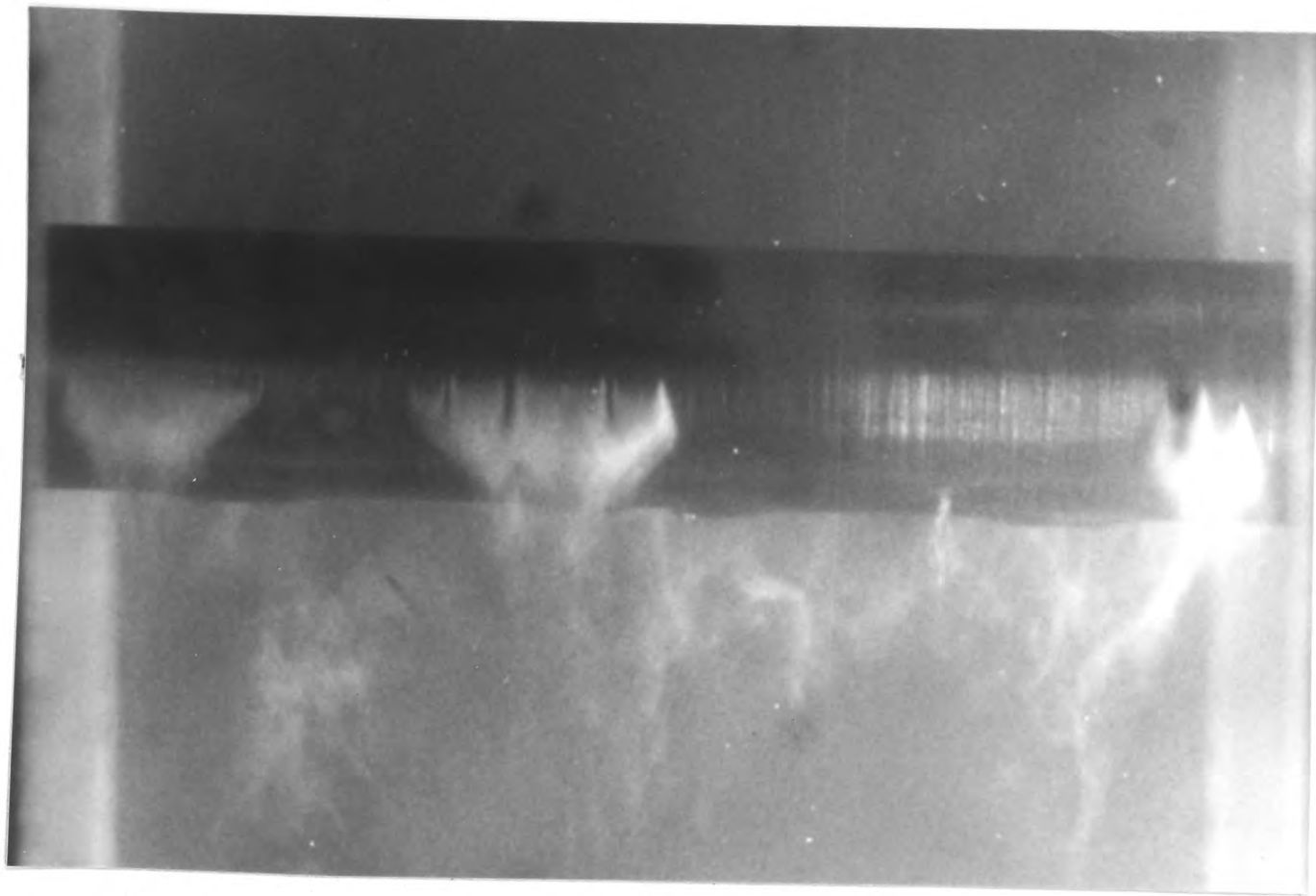
FLOW



FRAME 13 (OVERHEAD VIEW)

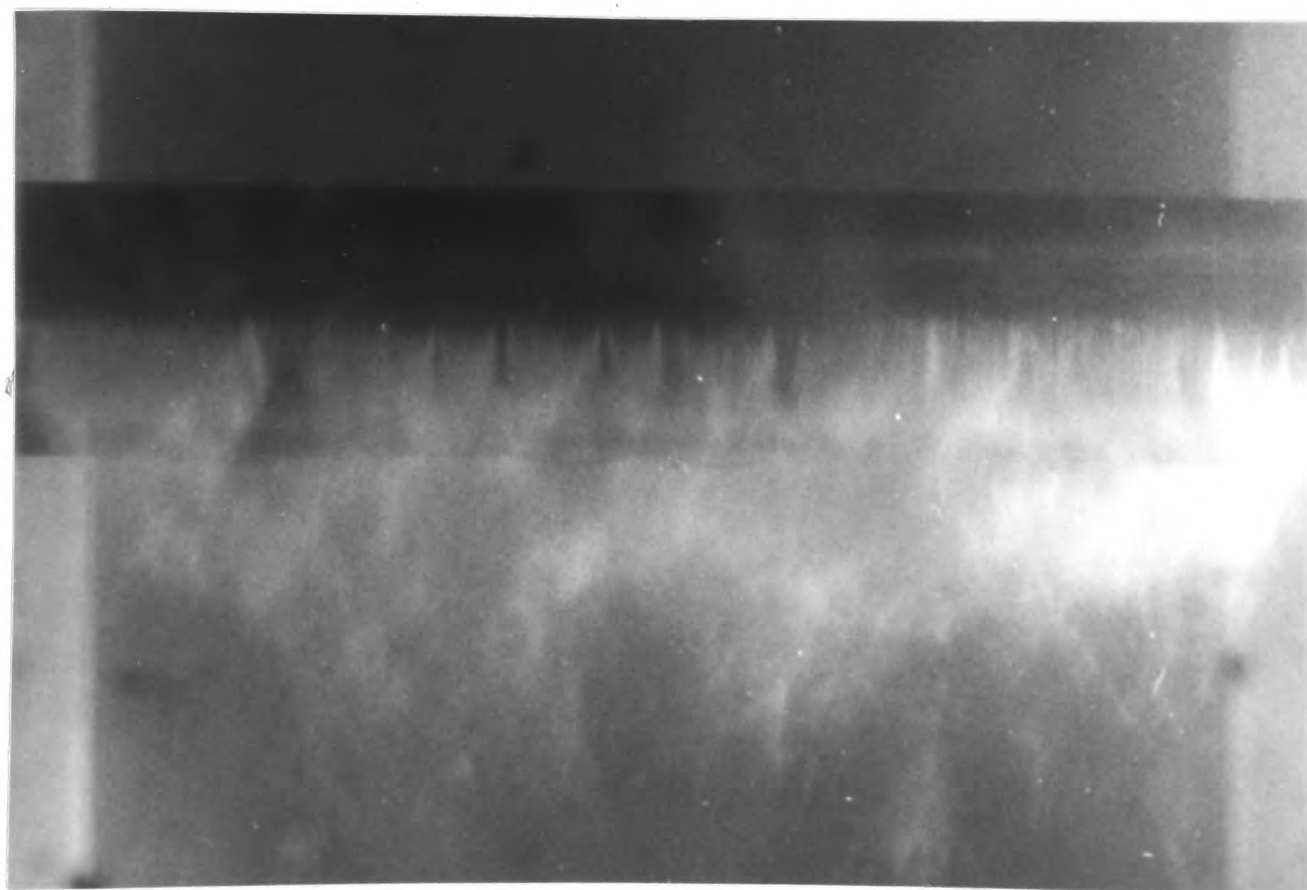
FIG. II.24 OVERHEAD VIEW PHOTOGRAPHS OF THE
DEVELOPMENT OF CAVITATION FROM THE
INCIPIENT SPOT STATE

FRAMES 12 AND 13



FRAME 14 (OVERHEAD VIEW)

FLOW

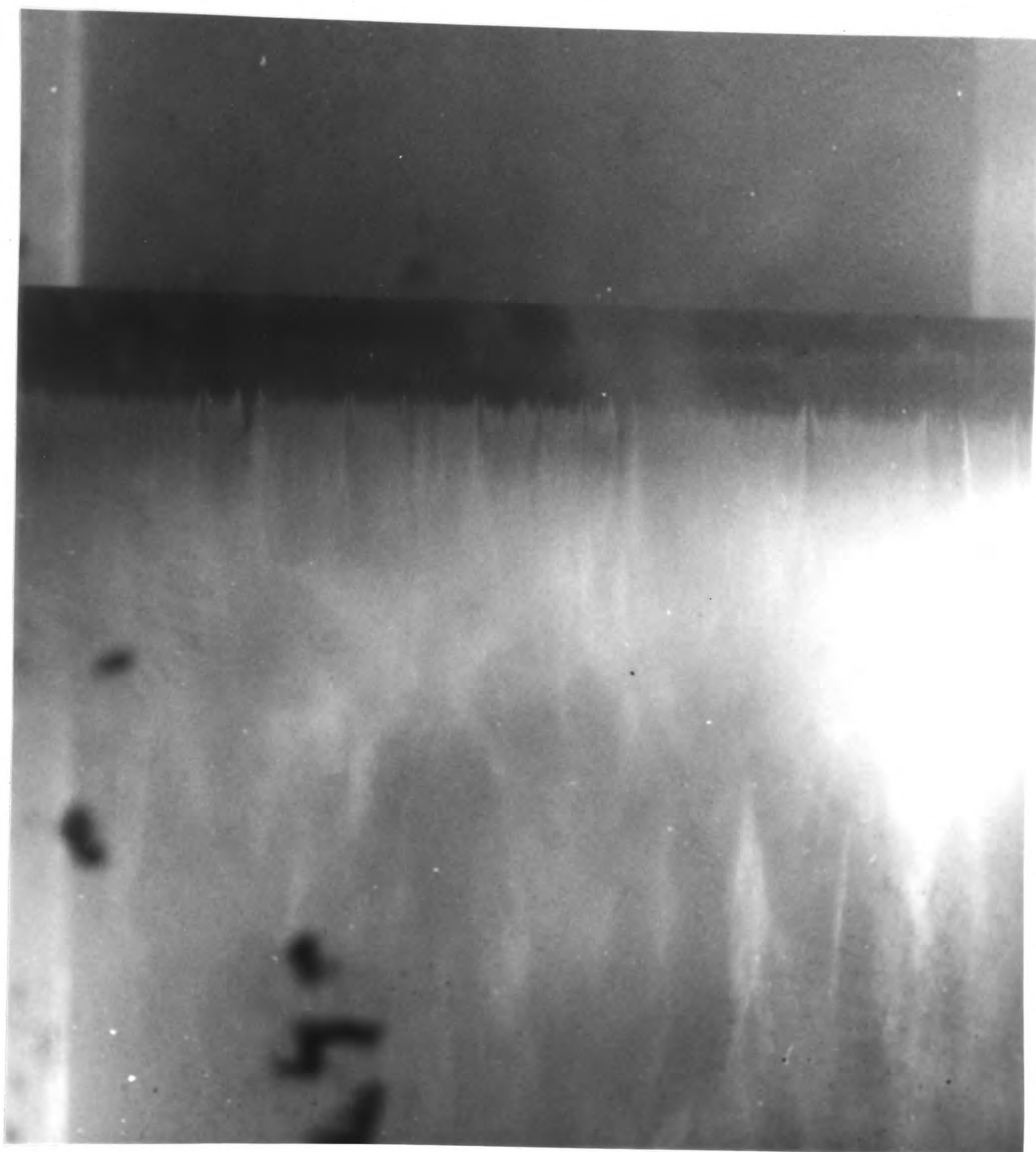


FRAME 15 (OVERHEAD VIEW)

FIG. II.24 OVERHEAD VIEW PHOTOGRAPHS OF THE DEVELOPMENT OF CAVITATION FROM THE INCIPIENT SPOT STATE

FRAMES 14 AND 15

FLOW



FRAME 16 (OVERHEAD VIEW)

FIG. II.24 OVERHEAD VIEW PHOTOGRAPHS OF THE
DEVELOPMENT OF CAVITATION FROM THE
INCIPIENT SPOT STATE

FRAME 16

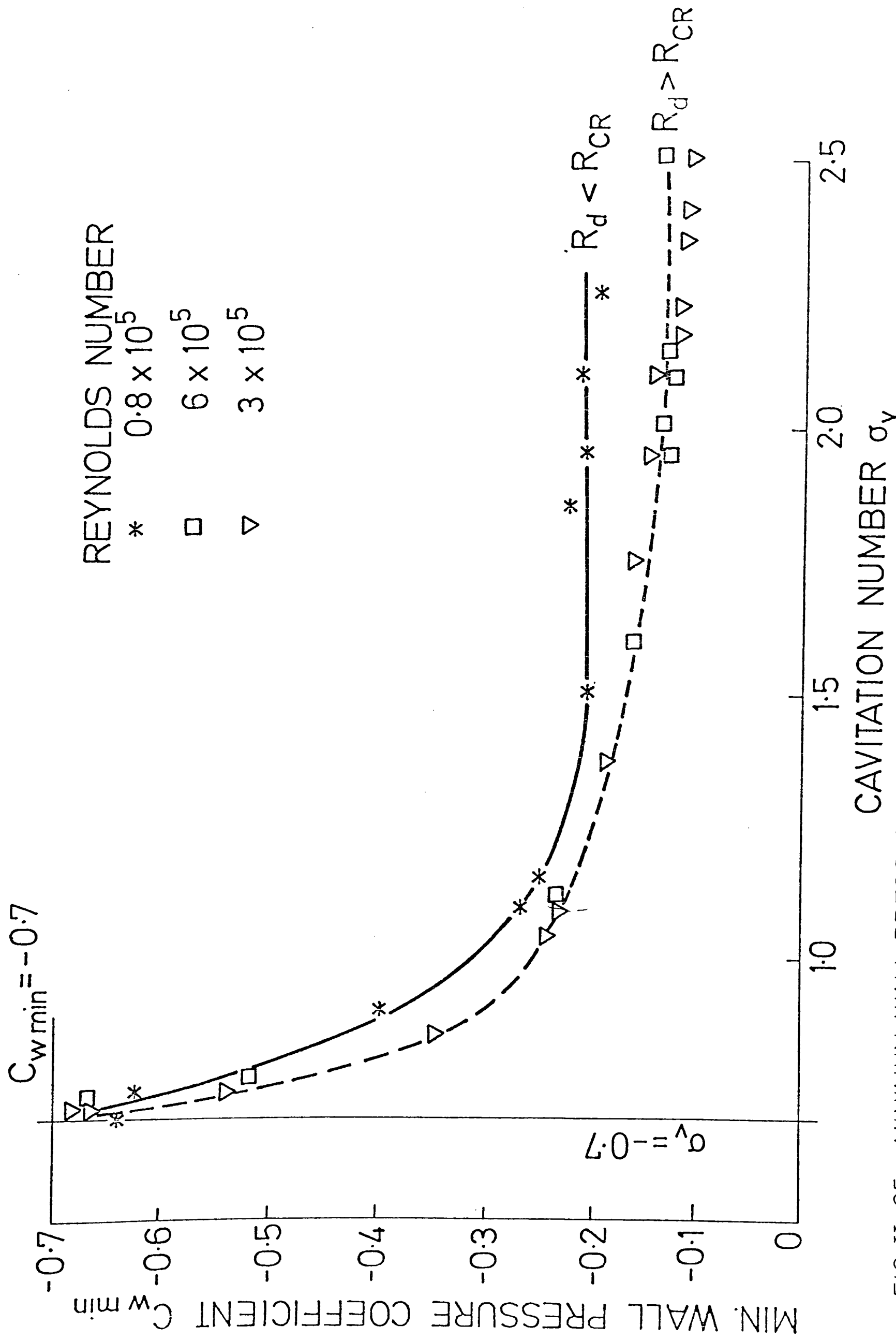
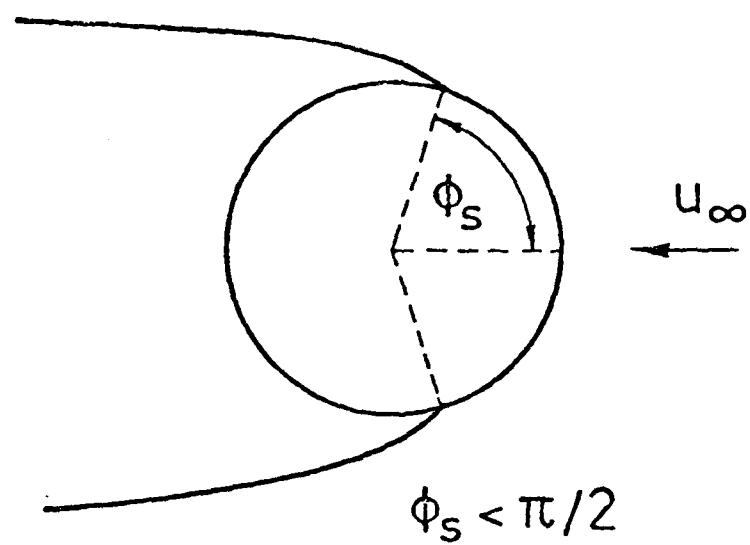
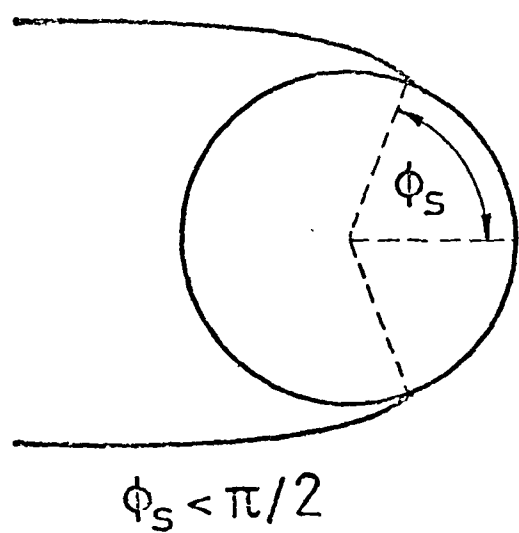


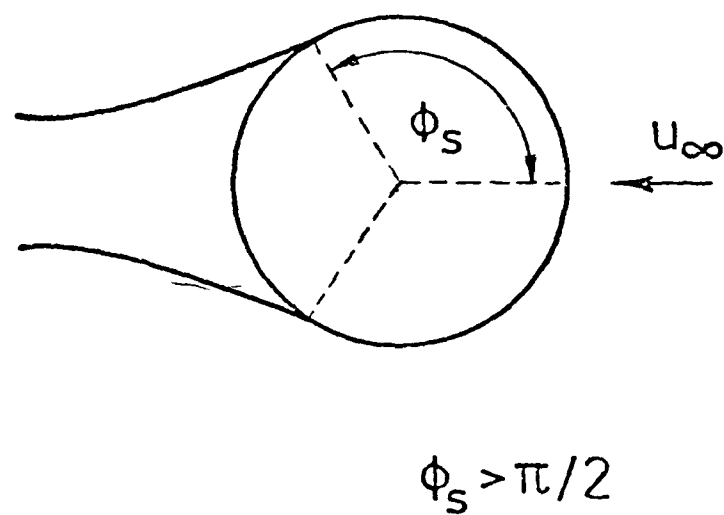
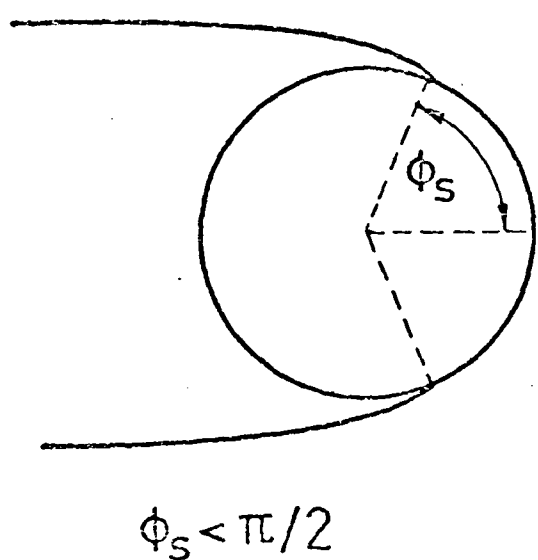
FIG. II. 25 MINIMUM WALL PRESSURE COEFFICIENT VERSUS CAVITATION NUMBER

DEVELOPED
CAVITATING
FLOW

FULLY WETTED
AND LIMITED
CAVITATING FLOW



$$R_d < R_{CR}$$



$$R_d > R_{CR}$$

FIG. II. 26 FREE-STREAMLINE CONFIGURATION FOR FULLY WETTED AND LIMITED AND DEVELOPED CAVITATING FLOWS AROUND A CIRCULAR CYLINDER AT SUBCRITICAL AND SUPERCRITICAL REYNOLDS NUMBERS

