The performance and application of cross flow fans for automotive engine cooling systems

Fernando, L.M

The Performance and Application of Cross Flow Fans for Automotive Engine Cooling Systems

by

L.M. Fernando B.Sc.

Thesis submitted for the degree of Master of Science
in the Department of Engineering, University of Durham

January 1986

Thesis
1986/FER

Dedicated to Chitrani, Natasha
and my parents Mr & Mrs H.M. Fernando

ACKNOWLEDGEMENTS

My gratitude is extended to everyone in the Department of Engineering who contributed towards this research project.

Particular thanks are due to Dr M.J. Holgate for supervising the research and to Ford Motor Company for making this possible through their contract to the University of Durham.

Finally, I wish to extend my thanks to all the technical staff of the department, particularly David Jenkins for operating valuable services.

L.M. Fernando

ABSTRACT

The work reported centres on the application of cross flow fan units to provide forced air flow over the radiator used for engine cooling. It is shown that a cooling system in which the airflow is ducted through the vehicle would lead to a significant reduction in vehicle drag. A ducted system allows more exact prediction of performance, and can be more easily designed in conjunction with a cross flow fan, that has rectangular inlet and outlet sections, and an ability to form 'S', 'L' and 'U' airflow geometries. This shape advantage of the cross flow fan also allows the use of radiators with a higher aspect ratio that could in turn permit lower bonnet lines on vehicles with a further reduction in aerodynamic drag.

At higher vehicle speeds adequate cooling is usually provided by ram airflow through the cooling system. An airflow by-pass arrangement controlled by a balance valve has been devised and tested. The operation of this valve was found to be stable and smooth, and extremely beneficial in reducing the total system resistance and increasing fan performance under ram airflow conditions.

A computer model has been developed and a programme produced to facilitate matching of the air circuit, fan and motor. Examples illustrate the use of this in the design of a system for a 1.6 litre passenger vehicle, with maximum b.h.p. in second gear being taken as the most critical engine condition for cooling performance.

LIST OF CONTENTS

	Page
Acknowledgements	(iii)
Abstract	(iv)
Nomenclature	(viii)
Chapter One: Introduction	1
1.0 Introduction	2
Chapter Two: Engine Cooling Systems	10
2.1 Traditional Configurations	11
2.2 Overall Requirements	13
2.2.1 Heat Dissipation	14
2.2.2 Power Consumption	17
2.2.3 Cooling System Efficiency	18
2.3 Ducted Configuration	20
2.3.1 The Utilisation of Ram Air in a Ducted System	22
2.3.2 Fan Requirement in a Ducted System	25
2.3.3 Inlet and Outlet of a Ducted System	30
Chapter Three: Fan Development, and its Performance and Resistance to Ram Air Flow	44
3.1 The Aim of Fan Development and Testing	45
3.2 Basic Fan Development	45
3.2.1 Adopted Design Features	47
3.2.2 Desirable Features, Limitations and Guidelines of Basic Fan	48
3.2.2.1 The Casing	50
3.2.2.2 Rear Wall	51
3.2.2.3 The Fan Rotor	52
3.3 Test Facilities	53
3.3.1 The Test Airway	53
3.3.2 Measurement Apparatus	57
3.4 Novel 'By Pass' Configurations	57
3.4.1 The 'By Pass' Value	57
3.4.2 Fan Configuration 1	57
3.4.3 Fan Configuration 2	59
3.5 Scaling of the Novel Fan	60
3.6 Fan Performance	62
3.6.1 Fan Configuration 1	62
3.6.1.1 Effect of Flap Mass	63
3.6.1.2 Effect of Tilt Angle to Horizontal	63
3.6.1.3 Flap Operation	63
3.6.1.4 Grains in Performance Due to Flap Operation	64
3.6.1.5 Outlet Arc of Outlet Bend	65
3.6.1.6 Rear Wall Extensions	65
3.6.1.7 Inlet Duct Height	65
3.6.1.8 Inlet Duct Transition	66
3.6.1.9 Curtailed Back Wall	66

List of Contents (continued)

Page

3.6.2 Fan Configuration 2 66
3.6.2.1 Effect of Flap Mass 66
3.6.2.2 Effect of Tilt Angle to Horizontal 67
3.6.2.3 Flap Operation 67
3.6.2.4 Grains in Performance Due to Flap Operation 68
3.6.2.5 Effect of Outlet Diffuser 68
3.7 Resistance of the Unenergised Fan to Ram Air Flow 68
3.7.1 Fan Configuration 1 69
3.7.1.1 Effect of Flap Mass 69
3.7.1.2 Effect of Tilt Angle to Horizontal 69
3.7.1.3 Flap Operation 69
3.7.1.4 Grains Due to Flap Operation 69
3.7.1.5 Outlet Arc of the Outlet Bend 70
3.7.1.6 Rear Wall Extensions 70
3.7.1.7 Inlet Height 70
3.7.1.8 Inlet Duct Transition 70
3.7.1.9 Effect of Rotor Locked and Freewheeling Conditions 71
3.7.1.10 Curtailed Back Wall 71
3.7.2 Fan Configuration 2 71
3.7.2.1 Effect of Flap Mass 71
3.7.2.2 Flap Operation 72
3.7.2.3 Grains Due to Flap Operation 72
3.7.2.4 Effect of Rotor Locked and Freewheeling Conditions 72
3.8 Overall Behaviour of the System with a Radiator at the Inlet 72
3.8.1 Fan Configuration 1 72
3.8.1.1 Overall Performance 72
3.8.1.2 Overall Resistance 73
3.8.1.3 Velocity Distribution at the Radiator Inlet Face 73
3.8.2 Fan Configuration 2 74
3.8.2.1 Overall Performance 74
3.8.2.2 Overall Resistance 74
3.8.3 Effect on Performance Due to Radiator 74

Chapter Four: Computer Aided System Matching 128
4.1 Computer Model 129
4.1.1 Principle 129
4.1.2 Computer Simulation of Fan Dimensionless Characteristics 131
4.1.3 Input Data Structures 133
4.1.3.1 Input Data Structure Type 1 133
4.1.3.2 Input Data Structure Type 2 134
4.1.3.3 Input Data Structure Type 3 134
4.1.3.4 Input Data Structure Type 4 135
4.2 Matching to an Overall Requirement 135
4.2.1 Match 1 136
4.2.2 Match 2 137
4.2.3 Match 3 138

List of Contents (continued)

Page

Chapter Five: Discussion 155
5.1 Discussion of Results 156
5.2 Constructional Features 160
5.3 Design Summary 161

Chapter Six: Conclusions 165
6.1 Installation Advantages 166
6.2 Aerodynamic Advantages 167
6.3 Disadvantages 168
6.4 Future Work 168

References 169

Appendices 174
Appendix 1: Listing of the Computer Programme 175
Appendix 2: Context of Programme Execution 187

NOMENCLATURE

Symbol		SI Units
A	area	m^2
ATB	Air Temperature to Boil (the air temperature that the coolant would boil)	Deg.C
BP	boiling point of coolant	Deg.C
c	fan cord	m
C	Torque at zero motor speed	N/m
d	fan inner diameter	m
D	fan diameter (outer)	m
E	Young's modulus	Pa
F	force	N
F_x	Drag force	N
g	gravitational acceleration	ms^{-2}
H	height	m
HD	heat dissipation	W
I	moment of area	m^4
K	system resistance coefficient	-
K_E	voltage constant of motor	V/rad s^{-1}
l	hinge length (fig.3.12)	m
L	fan length	m
L_v	distance to centre of gravity of flap	m
m	mass	kg
$\dot{m}$	mass flow rate	kg s^{-1}
M	voltage in volts over speed regulation and voltage constant of the motor (voltage/$R_m K_E$)	rpm
P	pressure	Pa

NOMENCLATURE (continued)

Symbol		SI Units
Q	volume flow rate	$m^3 s^{-1}$
R_m	speed regulation constant of motor	rad s^{-1}/Nm
SH	specific heat dissipation	W/Deg.C
T	torque	Nm
U	fan peripheral velocity	ms^{-1}
V	velocity	ms^{-1}
V_o	vehicle speed	ms^{-1}
W	power	W
Z	number of blades	-
α	vortex wall leading edge location	degs
β	blade angle	degs
γ	rear wall leading edge location	degs
ε	vortex wall leading edge clearance	m
ε_2	rear wall clearance, when $\gamma = 0$	m
η	efficiency	-
ΔP	pressure difference	Pa
θ	angle	degs
μ	viscosity of air	$kgm^{-1}s^{-1}$
ρ	density of air	kgm^{-3}
ϕ	dimensionless flow coefficient = Q/LDU	-
ψ	dimensionless pressure coefficient $= \Delta P/\frac{1}{2}\rho U^2$	-
ω	rotational speed	rad.s^{-1}

NOMENCLATURE (continued)

Subscripts

1	fan inner periphery
2	fan outer periphery
a	air
e	exit
f	fan
i	inlet
o	an atmospheric condition
of	overall fan
ot	overall transmission
r	radiator
s	static
t	total
T	the tilt condition
v	balancing valve (flap)
w	water
$\sim$	a fluctuating value

Superscripts

–	vector
.	flow rate

CHAPTER ONE

INTRODUCTION

1.0 INTRODUCTION

The engine cooling system contributes a significant amount to the overall cost and power consumption of today's road vehicles. Power is consumed to overcome cooling drag and drive certain components of the system. And also, these systems afford little lat itude to the vehicle aerodynamic designer, since the radiator tends to have a height approximately equal to its width due to the geometry of the conventional axial flow fan. Congested engine space due to the trend towards a more compact car construction and front end with inefficient stylish grilles and bumpers had restricted the cooling airflow to the radiator, and increased the cooling drag. This complex nature of the air cooling circuit also made very difficult to interpret the installed performance of a fan, hence increased the development work.

Though, it was a known fact that a ducted cooling systems would improve these circumstances, the conflicting requirements of duct volume,cost and conventional fans have not permitted realization of such a system. But a cross flow fan, having its inlet and outlet naturally rectangular and with an ability to form 'S', 'L' and 'U' air flow configurations, and relatively low duct volume is an ideal contender for a ducted system. It also provides another degree of flexibility to aerodynamic and engine compartment designers.

The concept of cross flow fan engine cooling arrangements (fig.1.1 to 1.3) include a fan (or fans) whose axis (or axes) of rotation is transverse to its casing with a radiator of substantial width and low height arranged transversely to vehicle,or alternatively a system with relatively small face area with a high density radiator, so that it can be installed in passenger vehicles at the side of an engine. The latter arrangement would however, have the disadvantage of decreased ram air flow. Both contain ducting to direct air stream from and towards preferred locations. This prevents heated air entering the engine space and also allows application of a fully tailored smooth underpan over engine irregularities which has a bearing on air drag. Substantial simplification of assembly of the engine cooling system may be achieved by combining the components to form a pre-assembled package. This might be constructed in a suitable manner as a front spoiler. As a whole, this engine cooling arrangement offers the advantage of low aerodynamic shaping of the front of the motor car for a given engine space.

A cross flow type fan consists structurally of a multi-bladed rotor and an external casing. The throughflow is a plane at right angles to the rotor axis, and each fan blade passage experiences a continuously varying flow field. The path of a streamline follows through the cascade impeller, first radially inward and then finally flow is discharged radially outwards. When the rotor is in motion, it is known to generate a Rankine type vortex, with an eccentrically located core, away from the fan's axis of rotation close to the inner blade circumference. Due to this eccentricity the two separate zones of suction and discharge occur round the rotor periphery.

An earlier investigation [15] was undertaken to establish the suitability of the cross flow fans for this application. This involved the determinaton of the effect on performance of the inlet flow restrictions and ducting, and also established the resistances of the fan when running as a turbine, and when locked to a stationary state, in the face of a ram airflow when the vehicle is travelling at a speed. These resistances were found to be significant. In these tests a cross flow fan consisting of 24 straight blades of 240mm in length, making up a rotor of 120mm and 150mm inner and outer diameters was used.

Two prototype fan configurations, having different geometries were constructed, based on the results of the above investigations and other available published data, and the design parameters governing the fan performance and efficiency were set as near as possible to optimum. The general rotor and casing parameters were same for both fan configurations, except for the inlet and outlet geometries. At operating speeds corresponding to top gear driving, the fan rotor behaves as an obstruction to ram-induced airflow. This was avoided by incorporating a balancing valve (fig. 1.4) to the vortex wall of the fan configurations to allow ram-induced airflow to bypass the rotor. These two configurations, including changes to some aspects of rear wall, inlet and outlet, were tested for performance and efficiency. The research also concentrated on the purpose of achieving the required airflow for a critical cooling condition, taken as the maximum b.h.p. in the 2nd gear. A computer model was developed for the interaction of air circuit, fan and the motor and a programme produced to facilitate matching of air circuit, fan and motor. Simple mathematical analysis has also shown some appreciations of the system performance and efficiency.

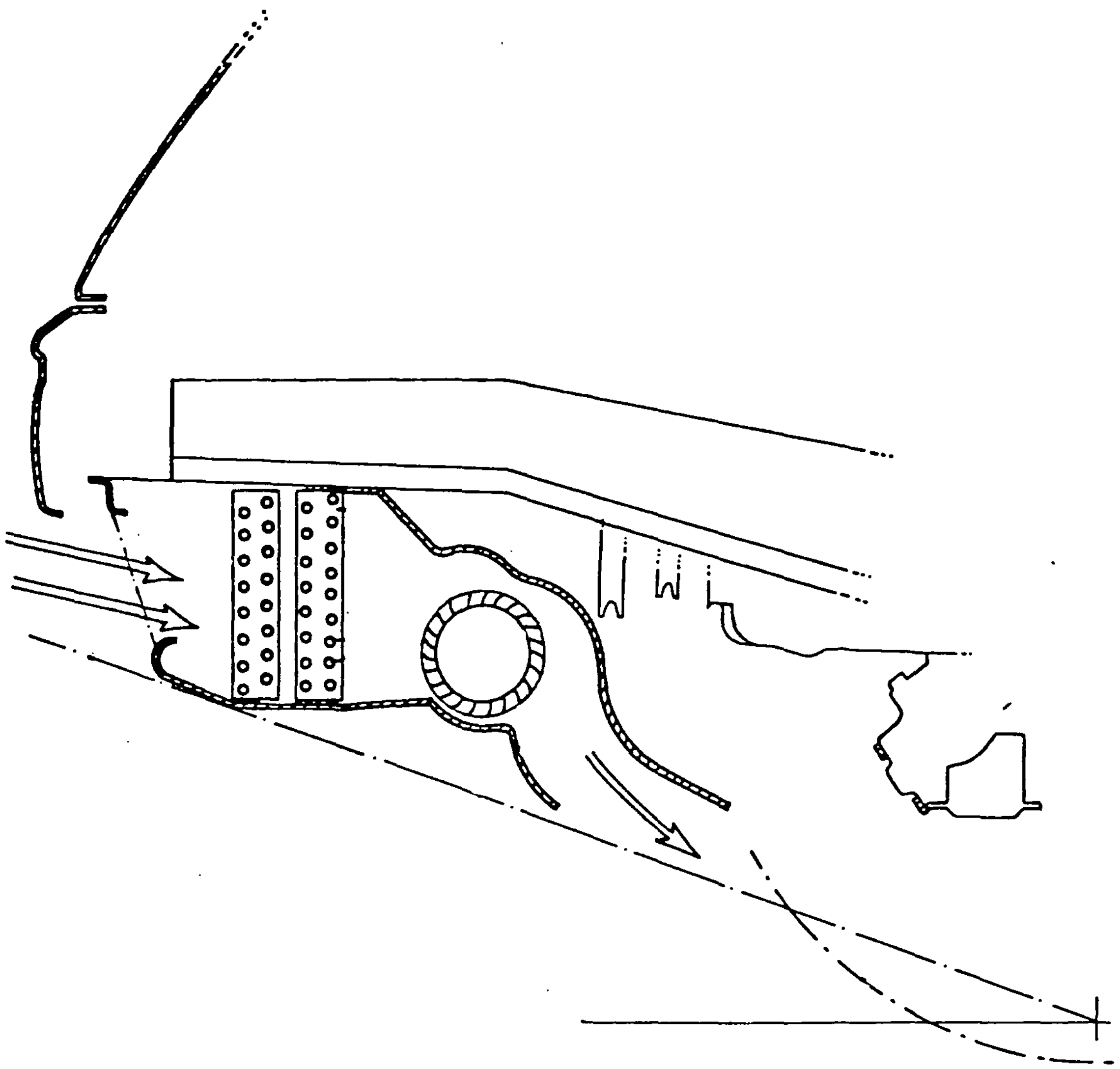


FIGURE 1.1 A CROSS FLOW FAN COOLING ARRANGEMENT

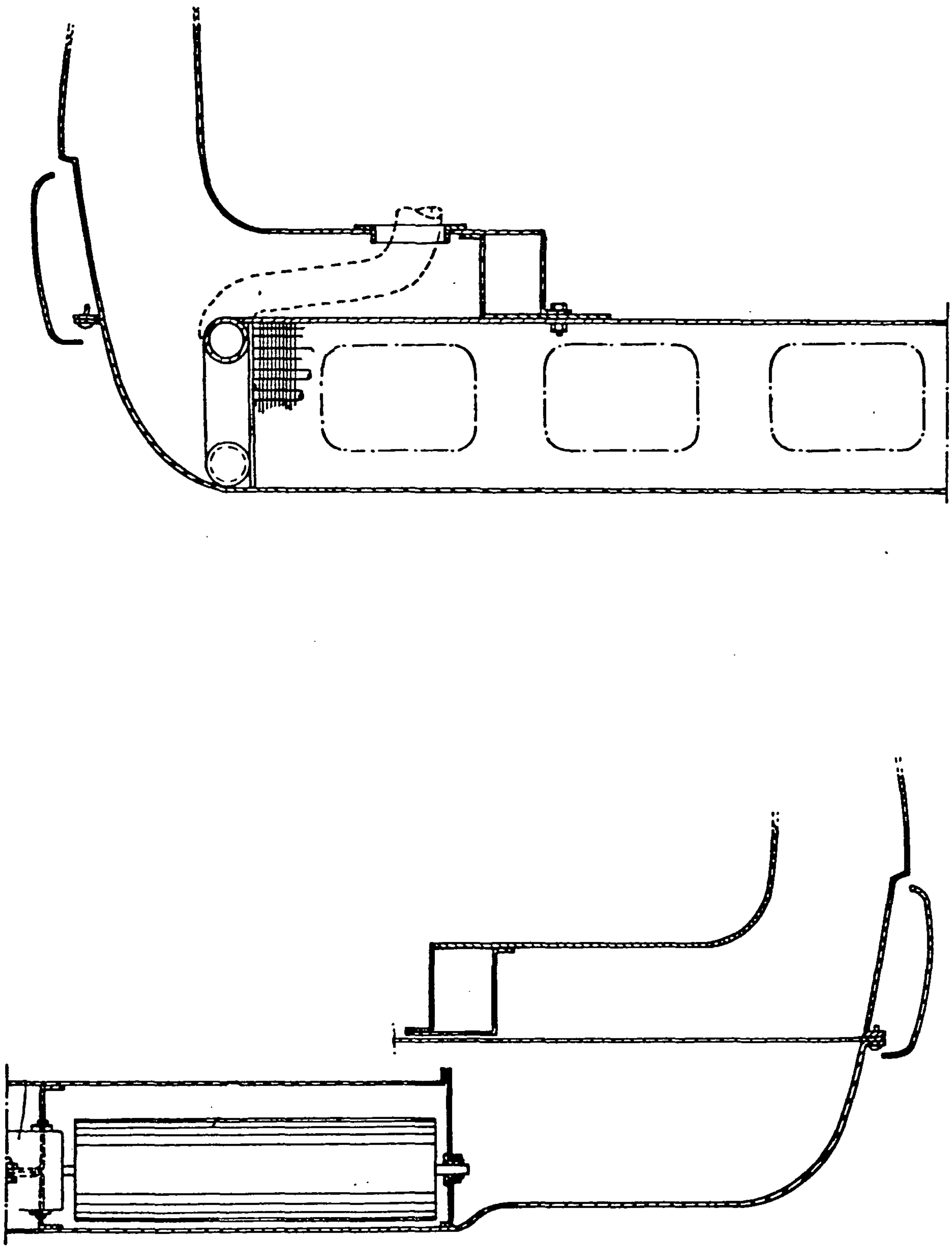


FIGURE 1.2 A CROSS FLOW FAN COOLING ARRANGEMENT

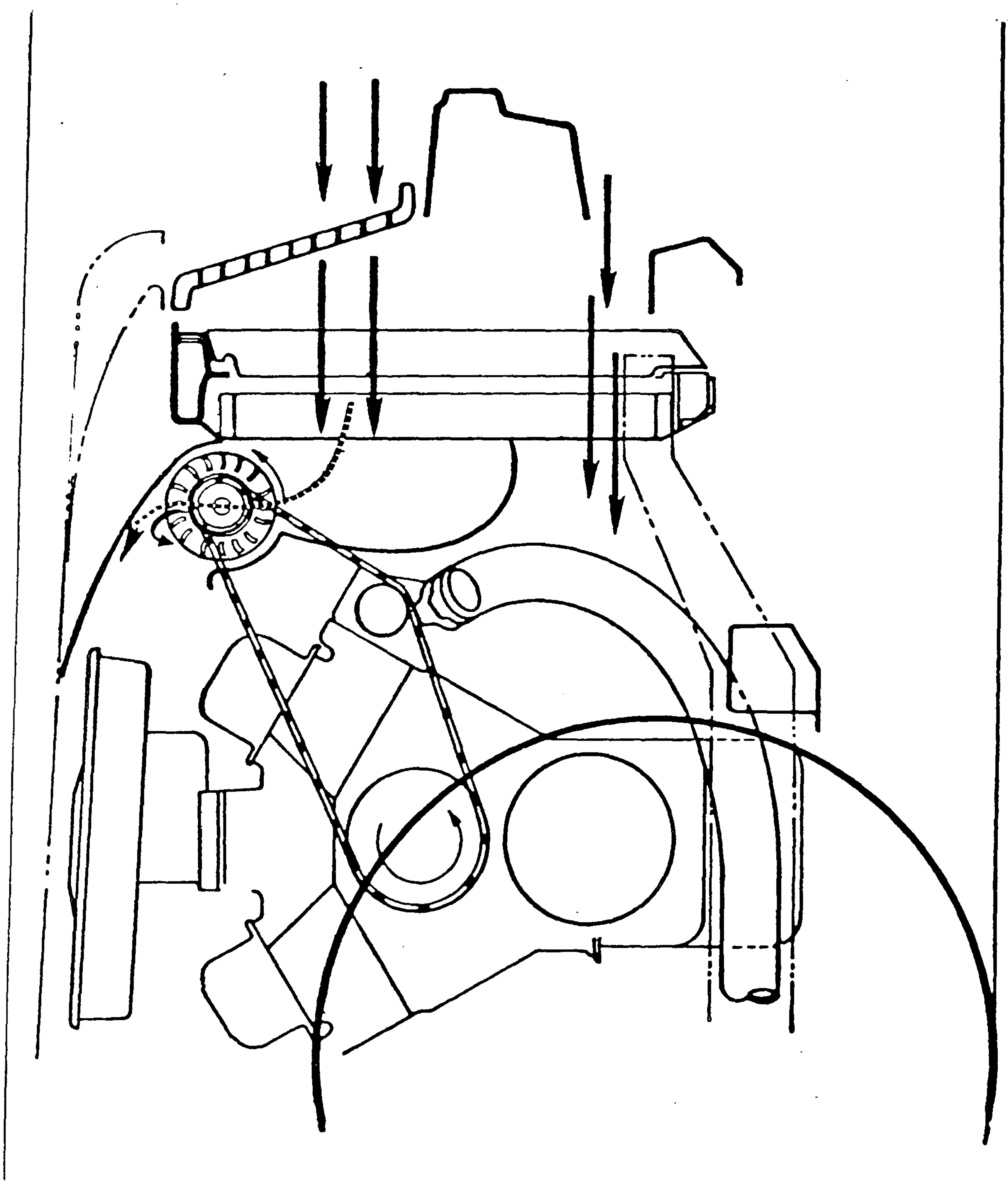


FIGURE 1.3 A CROSS FLOW FAN COOLING ARRANGEMENT

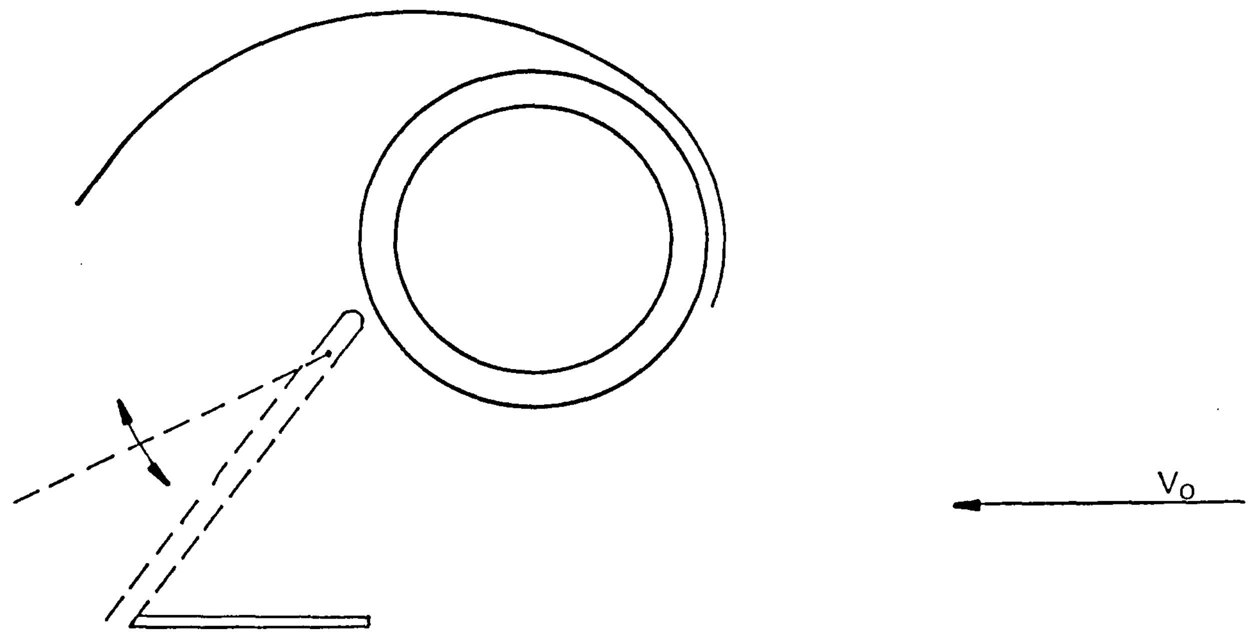


FIGURE 1.4 A SIMPLE 'BY PASSING' ARRANGEMENT

CHAPTER TWO

ENGINE COOLING SYSTEMS

2.1 Traditional Configurations

Though over the years, the space available for the radiator and fan is being steadily decreased as a result of the trend towards more compact car construction, traditional cooling system configurations afford little latitude to the vehicle aerodynamic designer, since the radiator tends to have its height approximately equal to its width due to the geometry of the conventional axial flow fan. This sometimes leads to the installation of a radiator inclined prone at a high angle to the vertical. The flow of air to the front-mounted radiator is also restricted by the bumpers and cross members in the body, and engine block and fan hub behind it, causing uneven velocity distribution at the face of radiator, often with local depression at the centre.

At entry to the cooling system, the airflow is usually dissipated by blocking the approach to the radiator with a sharp-edged grille and a bumper, and often allowed some of it, to stall rather than pass through or to recirculate (fig. 2.1) around the radiator matrix. Up to 15% of total airflow has been identified[46] as being air recirculated around the radiator. The remaining air then wanders, as shown in figure 2.2, through a rough system of cavities experiencing several sudden changes of direction due to obstacles and corners in the engine compartment, before discharges to atmosphere from the underside of the car. This undesirable air flow path results in tremendous loss of momentum.

Engine blockage loss is, of course, greatly dependent on the amount of hardware congestion in the air flow path. SCHAUB and CHARLES[54] have interpreted this blockage to be the pressure drop across the fan exit face and the engine bay, and indicated a value of 4.0 for the blockage loss coefficient based on radiator exit face dynamic head of a 1976 Ford Granada, demonstrating also the extreme negative effect of the engine blockage to cooling air flow. The engine bay pressures were also measured to be positive during the road tests of the ref.54.

The complex nature of the air cooling circuit, specially due to the non-ducted arrangement, the entry and exit conditions of fans installed in any cooling system are vastly different from those on which fan manufacturers' performance data is derived. This makes selection of a fan more difficult. Therefore a suitable fan is usually selected(59) by trial and error, together with experience of a previous or a similar competitive model. When installed, reduction in fan performance from manufacturers' data of 25% have been recorded by IMI Radiators (46) on various vehicle mock-ups and also on vehicles, at manufacturers' premises.

A typical example of a cooling system performance, obtained from ref.6, is presented in graphical form in figure 2.3. The curves, where applicable, refer to a 60°C extreme water/air differential.

More saloon car designers have now recognised the wastefulness of having a continually running fan, driven by a belt from the in-line engine, mounted at the rear of the radiator matrix. This arrangement has often been now replaced and would be taken over completely in the near future by a fan driven by an electric motor which only operates on demand, as sensed by a thermostat mounted on the engine or radiator, and which freewheels or 'windmills' when not being driven.

This eliminates the waste of the power absorbed by continually belt-driven fan to force air through the radiator matrix, and thereby potentially overcooling the system at most road speeds. Also, the wider use of transversely mounted front engines with front wheel drive, made this further necessary.

Today, engine cooling drag represents about 10% of the total drag of the production vehicles. It would represent in the future, in the absence of a specific programme for its reduction, about 20% of the total drag[4] of a vehicle with a coefficient of 0.25. In addition to cooling drag, the cooling fan considered to absorb between 3% and 8% of the engine power. This high grade energy consumption associated with engine cooling systems can amount to 6bhp[57] for a medium size family saloon car travelling at 70mph. Therefore this consumption can reflect on vehicle performance considerably.

2.2 Overall Requirement

Basically, this would be satisfied efficiently when optimum, safe operating temperatures of the engine are achieved through a process of heat dissipation with the minimum of power consumption, acceptable noise and space.

To ensure these safe temperatures are kept to their limits, the engine coolant should not boil under the most severe operating conditions, eg, low speeds, crawling road traffic, towing on a climb, high ambient temperature, high power output. In order to ensure that a car engine can operate reliably in these conditions without either boiling or overcooling the coolant, the cooling system designer must take a number of factors into account. These include radiator and fan performance, engine heat rejected to the coolant, water pump characteristics and the aerodynamics of the whole system, from the air intake grille to engine compartment air outlet. The transfer of waste heat from the hot combustion gases in the engine to the air is a complex multi-stage process. The major design parameters of the final stage in this process, that of transferring heat from the coolant to the air, are

i) the airflow rate determined by the amount of heat dissipation required; and

ii) the power consumption by the system determined by the total installed system resistance that must be overcome to maintain the airflow. The power is mainly consumed in two forms. They are to assist air flow by operating a fan, and to overcome cooling drag which is produced due to loss of momentum, of the ram induced air flow. The power consumed by the water pump can be assumed negligible. One of the principal contributors to the resistance is the radiator core itself.

2.2.1 Heat Dissipation

The complex geometry of the extended surfaces used in the present day radiators make their heat transfer and air pressure drop characteristics very difficult to calculate. In practice, the heat flow characteristics of a basic radiator are best determined (figs 2.4 and 2.5) by rig tests. Such tests enable the relative merits of various fin configurations, the nature and thickness of materials used, and the matrix size to be assessed in terms of their contribution to specific dissipation and resistance to air and water flow. This information provides the basis for choosing a radiator for a given system to meet specified cooling performance targets. Cooling systems are frequently characterized by the quantity air temperature to boil (fig. 2.3), at specified vehicle speeds, gears and loads. This is defined as(59):-

ATB = BP - HD/Specific Dissipation

where BP = boiling point of coolant

HD = heat to be rejected by coolant

and specific dissipation is the quantity of heat rejected by the coolant per degree temperature defence between air and coolant. It is a function of the cooling system and is independent of the heat source although its value will be dependent on the mass flow rate of the coolant. The factors affecting specific dissipation are

i) radiator heat flow characteristics

ii) airflow rate and velocity distribution

iii) coolant flow rate

It is seen from the equation, that the specific dissipation of the cooling system is the quantity which must be determined in order to satisfy a target ATB value for a given heat rejection and boiling point. And that a radiator with a high ratio of specific heat dissipation to pressure drop coefficient is preferred.

Heat rejection to the coolant is a function of the engine and is almost independent of the cooling system configuration. It is therefore sufficient for the cooling system designer to know the heat rejection characteristics of the particular engine used in the vehicle for which a cooling system is being designed.

The known radiator specific dissipation together with maps of 'Ratio of Heat Loss to Power'[59] of the engine would allow design of the cooling system to most critical (fig. 2.6) conditions. Maps of 'Ratio of Heat Loss to Power' over the entire useful range of speed and load are very useful for the purposes of cooling system design. Another very important use for such data is to indicate how to run an engine so as to minimize unnecessary heat losses at a given power output. But engine builders seldom publish such information. Therefore, a good estimate can only be obtained by expressing the amount of heat rejected as a proportion of the brake horse power, and this was indicated by EMMENTAL and HUCHO[11] (fig.2.7). For spark ignition engines the heat rejected to the coolant is between 60 and 80 per cent [57] of the bhp, while for diesel engines it is between 55 and 90 per cent at full load.

Unfortunately, these maps of brake horse power were not readily accessible for the vehicles concerned. Therefore, the following estimate, guided by figure 2.6, was compiled as a basis from which to calculate the cooling air requirement.

Ref.13 indicated following data for the Ford Sierra OHC vehicle-

Maximum power is 55kw at 4900rpm

Engine speed of 4900rpm corresponds to a vehicle top gear speed (fig. 2.9) of 90mph (145km/h) and a second gear speed of 45mph (72.5km/h)

Assuming radiator heat dissipation = $0.6\times$bhp, and also guided by figure 2.8, the estimated power transmission and heat dissipation were plotted against vehicle road speed for Ford Sierra 1.6 OHC in figure 2.10. When the vehicle is consuming maximum power the required heat dissipation is 33kw (20m/s) at top and second gear conditions. Second gear running conditions are normally critical[59] for cooling requirements, because under this gear, vehicle speeds are low and so is ram air cooling, even with high engine power output. Therefore, this condition would be considered as a critical cooling case.

For example, use of radiator 85BB8005BD (ref. 20) would dissipate 35kw of heat, when the extreme temperature difference is 58°C and air and water flow rates are $0.7m^3/s$ and 2L/S.

Since the radiator area is $0.155M^2$, the through air velocity = 0.7/0.155m/s = 4.5m/s.

The target of the cooling system is to satisfy cooling requirement of the critical case, which is the point of maximum bhp in the second gear. In this condition the vehicle's speed is 45mph and required through radiator face velocity was evaluated to be approximately 4.5m/s.

2.2.2 Power Consumption

The power is mainly consumed to overcome cooling drag and to drive the cooling fan. And the power consumption due to the water pump is comparatively negligible. Therefore power consumption by a cooling system

= Cooling drag power + Cooling fan power

$= F_{X}.V_{O} + \Delta P.Q/\text{overall efficiency}$

The amount to which this power consumption can be minimised depends on the type of the vehicle, its external shape, the size of the engine and on the arrangement of the cooling system and its components. However, adoption of the following measures[55] would be generally desirable:

i) placing the inlet area in a region of highest stagnation pressure;

ii) placing the outlet in the region of low pressure and making the area of outlet adjustable;

iii) sealing the whole cooling system, if possible to avoid any leaks to the outside airflow;

iv) arranging the inlet diffuser to recover pressure with low losses;

v) avoiding sudden, but encouraging gradual changes in the internal air flow direction and speed.

Therefore, basically an ideal air circuit for an engine cooling system requires a smoothly edged clean intake opening that first leads to a completely enclosed diffusing duct permitting the kinetic energy to be converted to pressure energy with low losses. Then the air flow experiences the minimum required energy loss in the form of pressure drop across the radiator core and the associated flow path.

2.2.3 Cooling System Efficiency

The aerodynamic efficiency of the cooling system, denoted as η_{cs}, may be defined as the amount of the heat dissipated per unit of power consumed to achieve this. Without any concern for this, the performance of a cooling system cannot be fully appreciated.

Hence $$\eta_{cs} = \frac{\text{Heat}}{W_f + W_p + F_x V_o}$$

where W_f and W_p are power consumed by the fan and water pump. For a typical vehicle radiator, amount of the heat dissipated can be estimated from the formula.[55]

$$\text{Heat (HD)} = K_r A_r \, (T_w - T_a) \, V_r^{0.8}$$

K_r is a constant factor dependent on the type of radiator. W_p is very small compared to other components. Therefore this may be ignored from the expression to give

$$\eta_{cs} = \frac{K_r A_r \, (T_w - T_a) \, V_r^{0.8}}{W_f + F_x \, V_o}$$

i) Consider a case, where the cooling air flow is produced purely by the fan; ie: vehicle stationary.

$$\text{Fan power } (W_f) = \tfrac{1}{2} \rho K_s \, V_r^2 Q/\eta_f$$

where K_s is system resistance coefficient

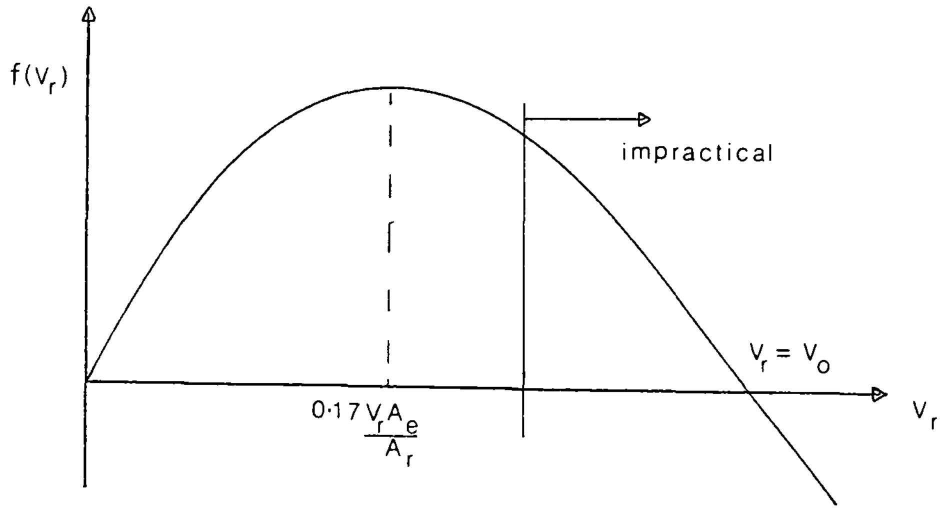
$$\text{Therefore } \eta_{cs} = \frac{2\eta_f \, K_r A_r \, (T_w - T_a)}{K_s \, Q \, V_r^{2.2}}$$

$$= \frac{2\, \eta_f \, K_r \, (T_w - T_a)}{K_s \, V_r^{2.2}}$$

ii) For a case where the cooling air flow is purely ram-induced, the drag force (Section 2.3.1) for a simplified case, exit pressure $(P_e) = P_o$, is given by

$$= \dot{m} \, (V_o - V_e) = \rho A_r \, (V_r V_o - A_r V_r^2/A_e)$$

$$\eta_{cs} = \frac{K_r \, A_r \, (T_w - T_a)}{V_o \rho_r A_r \, (V_o \, V_r^{0.2} - A_r \, V_r^{1.2}/A_e} = \frac{K_r \, A_r \, (T_w - T_a)}{f(V_r)}$$



This shows for both cases, when air flow is produced purely by ram and fan, the efficiency of the cooling system decreases as the airflow velocity (V_r), through a particular radiator increases. Therefore, realization of cooling system performance through an increase in face velocity of a particular radiator is not desirable.

However, EMMENTHAL and HUCHO(11) have compared different radiators and demonstrated (fig. 2.11), that more compact radiators, having a highly effective fin geometry with smaller face area can be used as an alternative to a wider mesh with larger face area counterpart, without deteriorating the efficiency of the cooling system. This indicates that the compactness of a radiator should be obviously achieved through fin effectiveness to heat and air flow, but not purely by increasing the face velocity.

2.3 Ducted Configuration

Cooling drag results from momentum losses experienced by the internal flow as a function of internal flow circuit configurations. For example a ducted systems would reduce this drag by eliminating the friction and momentum losses created due to the irregular airflow path in the engine compartment of a conventional system. Ducted systems also prevent heated air entering the engine space and allow application of a fully tailored smooth underpan (eg, Ford Probe series, Ref. 18) over engine irregularities, improving the aerodynamic efficiency of the underside and the engine environment, and also damp air-borne sound from the engine. As a result, it may also be possible to reduce the warming up phase of the internal combustion engine in winter.

Ducted configurations enable realization towards an idealised system as mentioned in Section 2.2 and the fan to establish more accurately the installed performance. This reduces installation losses and the development work to a minimum, and enable fine control of air through the radiator to avoid overheating or overcooling, which would reflect beneficially on engine performance.

Figure 2.12 shows proposals made by BUCHHEM[3] for cooling air ducts. And each one of these proposals were compared with present conventional arrangement of non-ducted radiator. The vertical cooling air duct running diagronally upwards from below with outlet and inlet placed in the regions of low and high pressures, was most desirable for low drag and high velocity ratio. But conflicting requirements of grille styles, duct volume, cost and conventional fans, currently permit the realization of neither these nor the desirable features, mentioned in Section 2.2.2. On the other hand, a cross flow fan, having its inlet and outlet naturally rectangular and with an ability to form 'S', 'L' and 'U' air flow configurations, has also several other desirable features to be an ideal contender towards achieving these.

One of the disadvantages of a duct system is that there is no direct convective heat transfer from the engine to air, since direct interaction between cooling air and the engine is not possible. This may impose a relatively higher radiator duty than in a non-ducted system; and it may trap and accumulate vaporised fuel inside the engine compartment, if no provisions are made to refresh it with clean air. This may create a dangerous situation, if a fully tailored smooth underpan is also fitted. On the other hand, in a non-ducted system, the application of warm fresh air to engine space may relieve this situation, causing a higher fuel vaporisation penalty.

2.3.1 The Utilisation of Ram Air in a Duct System

The induced ram airflow through the cooling system has a two-fold effect. First, by depriving the external flow it lowers the overall pressure, and second, the mass flow induced through the internal duct loses momentum due to friction and leakage. The first effect, though small, is theoretically beneficial because bleeding a mass of air into the internal duct system reduces the value of the resultant drag due to external airflow. On the other hand, the internal flow also introduces additional drag. Since these losses are always greater than gains,extra energy is consumed from the engine due to this ram induced internal flow. However, avoiding utilisation of external ram air means, the required cooling airflow should be purely induced by the fan, which also consumes useful energy indirectly from the engine.

Therefore, to quantify the overall advantage of this utility, following analysis was undertaken by considering a general air flow circuit as shown in figure 2.13.

Assumptions -

i) flow external to the cooling system reversible;

ii) conditions at all points within the control volume do not change with time (steady).

For the stream tube bounding the flow of the cooling air

Force on fluid in 'x' direction

$$\int_{\text{steam tube}} \rho V.V \, dA = (P_o - P_e) A_e - F_x \qquad - \qquad (1)$$

where F_x = drag force on the vehicle attributable to the cooling air

= - net force on the air due to passage through the ducting

By continuity

$$\dot{m} = \rho_o A_o V_o = \rho_e A_e V_e$$

Therefore $\int_{\text{steam tube}} \rho V.V \, dA = \dot{m} V_e - \dot{m} V_o$

Therefore from (1)

$$F_x = \dot{m} V_o - \dot{m} V_e + (P_o - P_e) A_e \qquad - (3)$$

and generally in subsonic flows the exhaust jet expands to its surrounding pressure.

For a more general practical case, as shown in figure 2.14, this would become

$$F_{xx} = \dot{m} V_{ox} - \dot{m} V_{ex} + (P_o - P_e) A_{ex}$$

Hence consumed useful engine power by the cooling air system due to cooling drag for a simple case of

$$P_e = P_o$$

is

$$= \frac{F_x V_o}{\text{Overall transmission efficiency}}$$

$$= \frac{\dot{m} (V_o - V_e) V_o}{\eta_{ot}} = \frac{2A_r V_r (\frac{1}{2} \rho V_o^2) - A_r \rho V_r V_e V_o}{\eta_{ot}}$$

If the system is not using ram air and the power consumption from the engine by the fan to provide same dynamic head (same total pressure difference across the system).

$$= \frac{(\frac{1}{2} \rho V_o^2) A_r V_r}{\eta_{of} \text{ (overall fan)}}$$

(It is assumed isentropic flow up to the inlet of the cooling system and same entry loss for both cases.)

Therefore the power consumption factor (PF) of a system using ram air to that does not

$$= 2(1 - V_e/V_o) \eta_{of} / \eta_{ot}$$

In the case of an electric fan, η_{of} comprises individual efficiencies of fan, electric motor, drive, generator and battery. Therefore $\eta_{of}/\eta_{ot} < 0.5$ is the most possible case. Hence PF<1.

But for an engine driven fan HAWES[23] has quoted values of 0.75 and 0.5 for η_{ot} and η_{of}, which would result 0.67 for η_{of}/η_{ot}.

Hence PF >1

The above comparison indicates that the use of ram air would most probably be desirable in an electric fan system but not in a system where the fan is driven by the engine.

In a duct system, the total to static pressure drop (ΔP_s) represents the overall pressure drop across the unit, since the dynamic pressure at duct outlet would be lost almost completely. If the system resistance coefficient, K_s, is based on ΔP_s and radiator velocity, and fan unenergised condition, then

$$\Delta P_s = \tfrac{1}{2}\rho K_s \cdot V_r^2$$

$$= \tfrac{1}{2}\rho V_o^2 \text{ at } V_o \text{ vehicle speed}$$

Hence $V_r/V_o = 1/\sqrt{K_s}$

EMMENTHAL and HUCHO[11] have confirmed this experimentally.

2.3.2 Fan Requirement in a Ducted System

Installed fan performance can be more accurately predicted with low installation losses in a ducted system, from general manufacturers' performance data. Therefore this would provide selection of a fan much easier and better control of air, which would cut down development time and cost, and improve the vehicle performance.

a) Fan/Duct Interaction

Basically, in a ducted system, the pressure rise created by the fan must be identical with the pressure drop of the whole cooling system. If coincidence of this nature is not immediately fulfilled, there would be an acceleration and deceleration due to high pressure or low pressure in the ducting, but eventually equilibrium will be achieved.

Theoretically, there are various possible ducting characteristics, according to the physical character of the resistance:

i) pressure drop = KQ for pure laminar friction (often across parallel radiator core cavities)

ii) pressure drop = KQ^2, for pure turbulent friction as the most frequent case

iii) pressure drop = KQ^n, for polytropic resistance

Practically all intermediate positions occur for instance a resistance which lies between laminar and turbulant flow. This is the case of the flow across a conventional vehicle's cross flow radiator.

Dimensionless coefficients are often employed for the design, comparison and critical assessment of fans. These coefficients are of considerable assistance to manufacturers and users. The dimensionless parameter associated with the interaction of a fan and duct is called Throttling Coefficient (ψ/ϕ^2), which is pressure coefficient divided by square of flow coefficient. There is an interaction between a fan and a duct to which it is connected, because of the retarding or throttling action of the resistance set up by the duct upon the air discharged by the fan. And in general, the resistance coefficient of the duct system is proportional to the pressure developed divided by the square of the output volume flow.

Therefore $$\frac{\psi}{\phi^2} = \frac{2\,\Delta P_f L^2 D^2}{\rho \quad Q^2} = \frac{2\Delta P_f L^2 D^2}{A_r^{\,2} V_r^{\,2} \rho}$$

$$= \frac{2\Delta P_f}{\rho V_r^{\,2}} \cdot \frac{D}{H_i}^2$$

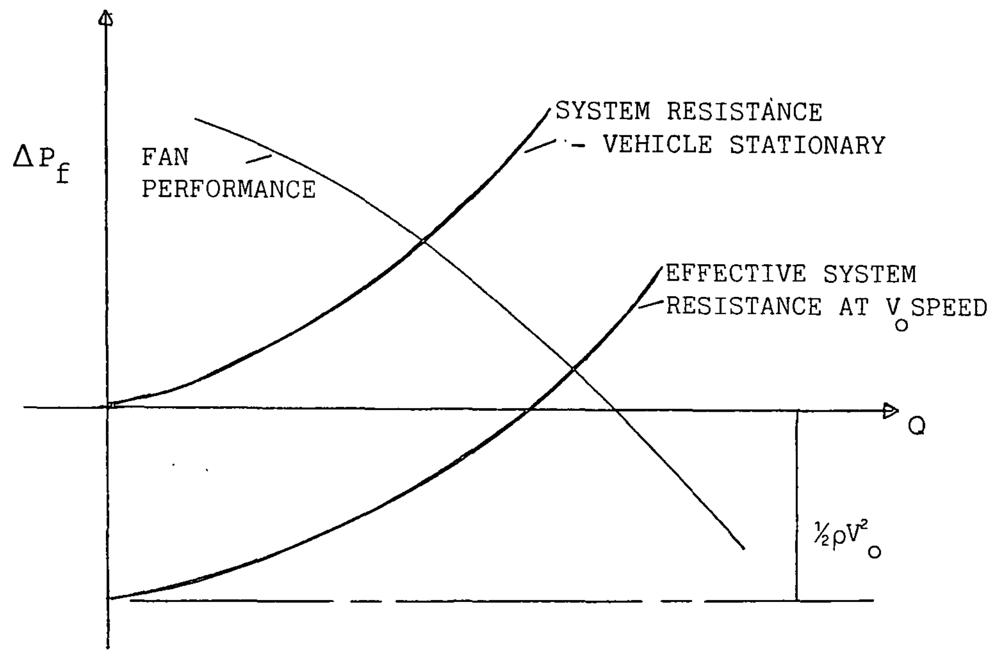
since $\frac{2\ P_f}{\rho V_r^{\,2}} = K_S$ (system resistance coefficient)

ie: assuming the flow through the radiator also obeys the square law, though in reality this is not exactly the case.

Hence $\psi/\phi^2 = K_S\ D^2/H^2{}_i$

This above expression shows how convenient the throttling coefficient is, in automotive applications, as an independent variable. Because it is associated with the interaction of the fan and the duct, and also with available space.

A fan is normally designed to operate at the point of optimum efficiency. This obvious requirement cannot be fulfilled easily in this particular application, because in practice, the effective resistance of a vehicle cooling system fluctuates considerably over the vehicle speed range of stationary to maximum operable, as a result of ram-induced airflow.



Therefore $\Delta P_f = \frac{1}{2}\rho K_d V_r^2 = \frac{1}{2}\rho K_s V_r^2 - \frac{1}{2}\rho V_o^2$, where Kd is the effective system resistance. And the fan pressure rise (ΔP_f) should be carefully defined to suit the duct installation.

Hence $\psi/\phi^2 = 2\Delta P_f/\rho V_r^2 = (D/H_i)^2 (K_s - V_o^2/V_r^2)$ - (1)

represent the general equation of the fan/duct interaction.

This indicates that there is no definite throttling line available but a range, in which the effective throttling line can fluctuate. In such cases one may incline to select the mean throttling line of a given range as the guide line and to use the point of interaction with the fan characteristic for designing. It could easily occur that in spite of this layout the power demand would rise considerably at the end of the range, as in the case of some fans (eg, cross flow fan, forward curved centrifugal). That is maximas of the efficiency and the power demand do not coincide. In these cases it is often advisable to design to suit the maximum power demand and not to go by the maximum efficiency.

b) Fan/Motor Interaction

The permanent magnet electric motors have a linear torque/speed relationship (fig.2.4) and if this is represented by:

$$T = -MN + C$$

then $M = \frac{1}{R_m} \times \frac{2\pi}{60}$ and $C = \frac{1}{R_m} \frac{V}{K_E}$

where R_m = Speed regulation constant of the motor

K_E = Voltage constant

V = Applied voltage in volts

Power input to impeller $= \Delta P_f Q / \eta_f$

$$= \eta_b \frac{\pi}{30} (T.N)$$

where η_b is transmission efficiency of fan drive.

Therefore $\frac{\psi Q}{\eta_f} = \eta_f \frac{\pi TN}{15\rho V^2} = 240\, \eta_b \frac{(-MN + C)}{\rho \pi D^2 N}$

substituting $N = \frac{60}{\pi} \frac{Q}{LD^2 \phi}$ gives

$$\frac{\pi}{60} . \frac{CLD^2}{MQ} . \phi = \frac{\pi}{240\, \eta_b \eta_f} . \frac{1}{M} . \rho D^2 Q \psi . + 1$$

Evaluating this for positive values of 'Q' gives

$$Q = -\eta_b \eta_f \left(\frac{120\ M}{\pi \rho D^2 \psi} \right) + \frac{1}{2} \left[\left(\eta_b \eta_f \frac{240M}{\pi \rho D^2 \psi} \right)^2 + \left(\eta_b \eta_f \frac{16CL\phi}{\rho \psi} \right) \right]^{\frac{1}{2}}$$

This equation combines with equation (1) of the section 2.3.2 to establish the interacting point of the Fan, Motor and Duct.

2.3.3 Inlet and Outlet of a Duct System

Inlet and outlet should be selected in the regions of high stagnation pressure and low static pressure. The inlet should be in the form of a diffuser, ideally representing the boundaries of the inlet stream tube. PAISH and STAPLEFORD[47] have experimentally ascertained that nothing would be lost if corresponding flat-sided walls are used instead of these ideal curved boundaries. The diffuser can be designed to recover pressure[26,37] with minimum of losses. HAWES[23] has indicated that the size of the intake opening should ideally vary with the requirement of radiator corresponding to different operational conditions, so that excess cooling air is efficiently diverted outside the vehicle. This would give an advantage of power saving.

The outlet area may be projected in order to provide maximum possible air momentum to horizontal (ie, direction of vehicle speed) without exceeding the system resistance correspond to the required air flow rate, or alternatively, the outlet may be incorporated with a diffuser to recover pressure effectively, before air is discharged to atmosphere. The outlet can also be made adjustable and used as a flow control means to produce the same effect indicated in ref.23 for an adjustable inlet.

Unnecessary high pressure losses often associated with conventional stylish grilles may be replaced with flow stabilising[26] 'splitter vanes' (fig.2.15) in the inlet diffuser. The addition of these vanes can prevent stall in a diffuser and cause the pressure rise to increase so that the wall boundary which were stalled are more highly loaded than before, yet they do not separate.

Diffuser inlet may also be formed in a suitable manner, with well rounded thick edge lip (fig.2.15) for efficient air entry at low vehicle speeds and to capture air flow (eg, with the add of a valance panel) effectively at high vehicle speeds.

2.4 Summary

1. Previous work has shown that for a spark ignition engine running at full load, the heat rejection to coolant is approximately 60% of the brake horse power developed.

2. This figure is taken as the cooling load to be handled by the cooling system when the engine is running at maximum b.h.p. in second gear.

3. A ducted air system not only reduces the momentum loss associated with the flow of cooling air, but also permits better design prediction thereby reducing development cost.

4. Heat exchanger design and development is better directed towards improved fin effectiveness rather than increasing radiator face velocity if air system pressure losses are not to become too high.

5. Electrically driven fans are to be preferred to directly driven from the engine if ram-air cooling at higher road speeds is to be optimised.

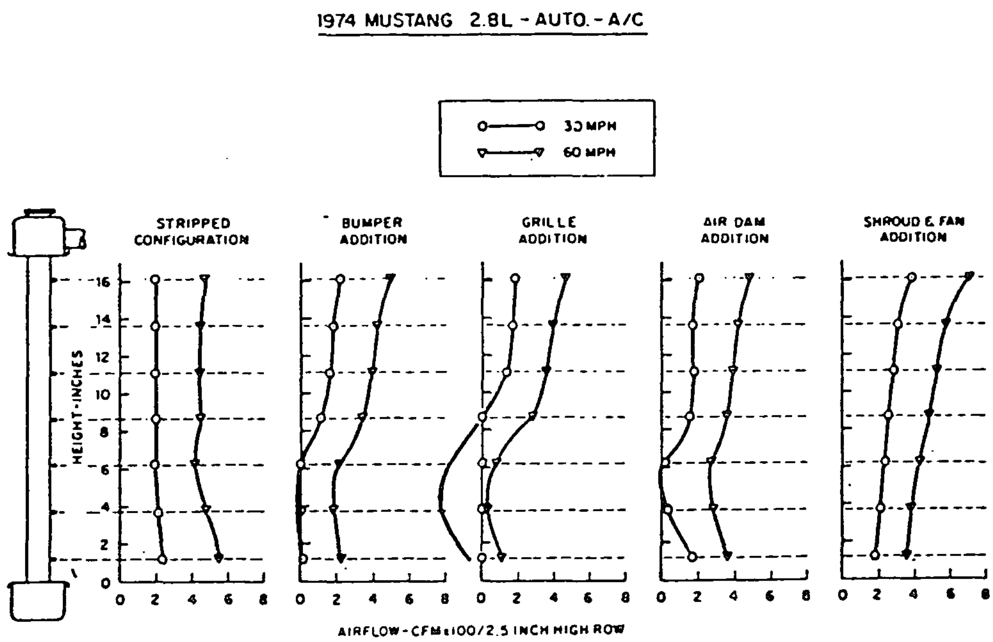
6. Fan/Duct interaction can be approximated by an equation of the form

$$\psi/\phi^2 = g(K, D, H_i)$$

7. Fan/Motor matching can be accomplished by use of an equation of the form

$$Q = f(\psi, \phi, C, M.\eta)$$

8. Care is needed in the design of the ducting upstream of the radiator/fan combination as well as in the exit side. Entry grilles with high blockage should be avoided, and 'splitter vanes' (fig.2.15) would be beneficial in improving diffuser action.



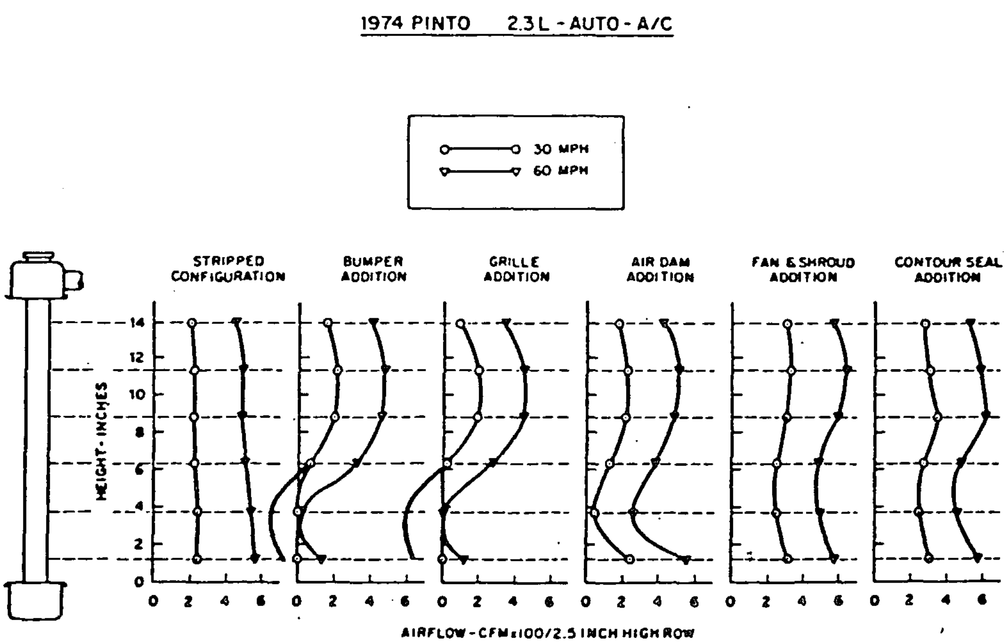


FIGURE 2.1 RADIATOR AIRFLOW - VERTICAL DISTRIBUTION (43)

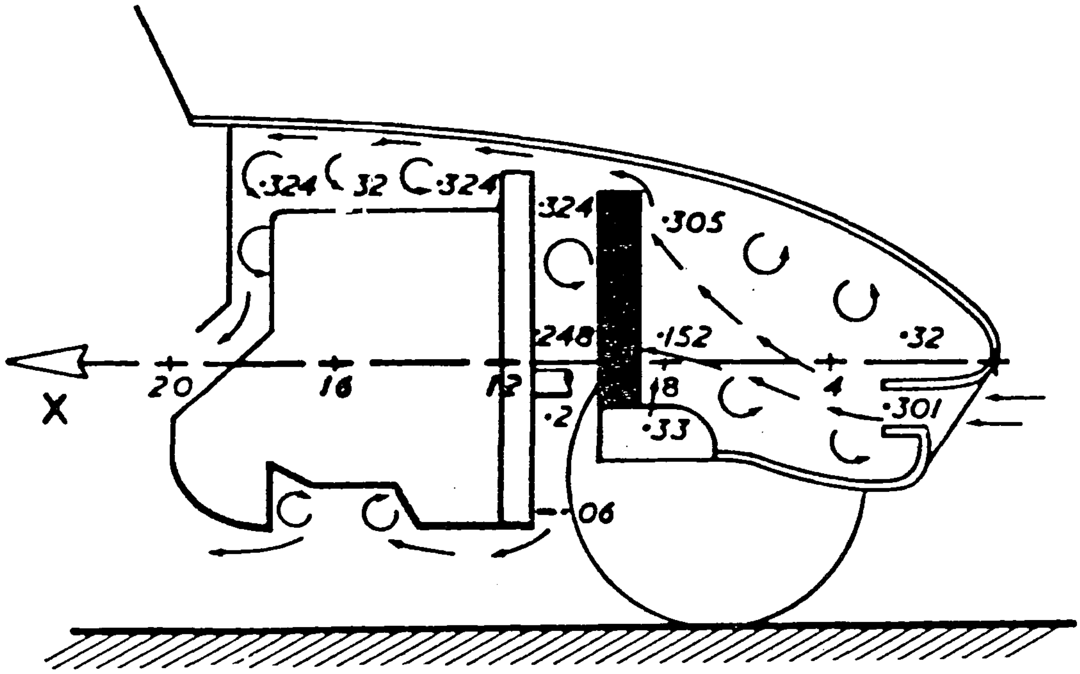


FIGURE 2.2 AIRFLOW PATTERN INSIDE A TYPICAL ENGINE COMPARTMENT (55)

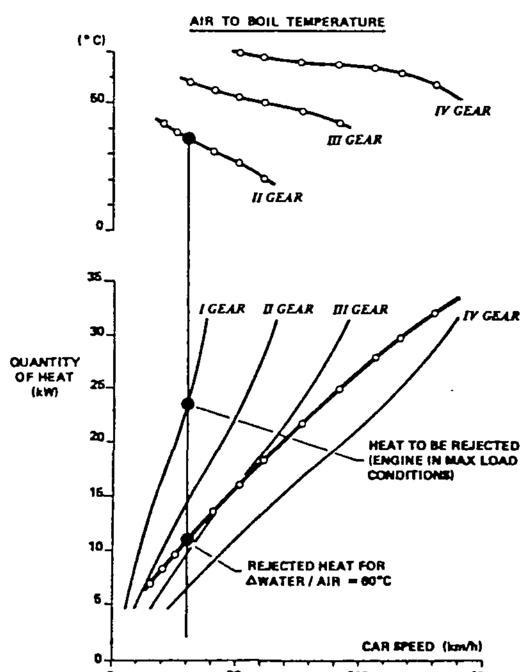
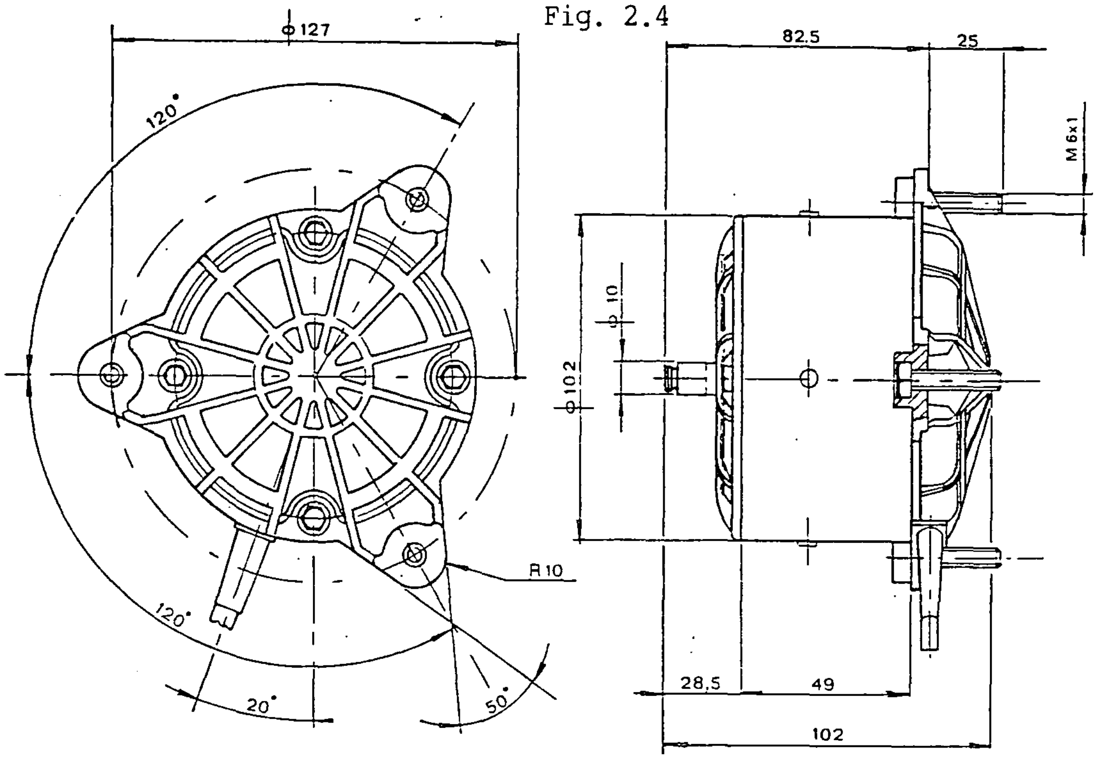


FIGURE 2.3 A.T.B. TEMPERATURES AND COMPARISON BETWEEN HEAT TO BE REJECTED AND ACTUALLY REJECTED HEAT (6)

GATE
ASTI - ITALY

MP 8025/238

Fig. 2.4



Caratteristiche

Tensione nominale 12 V
Giri nominali 2250
Corrente nominale 15 A
Potenza resa 130 W
Senso di rotazione reversibile
Servizio continuo
Temperatura di funzionamento
−40°C +80°C

Note
Con ventola dis. nr. 825.0018 - 825.0016

Caractéristiques

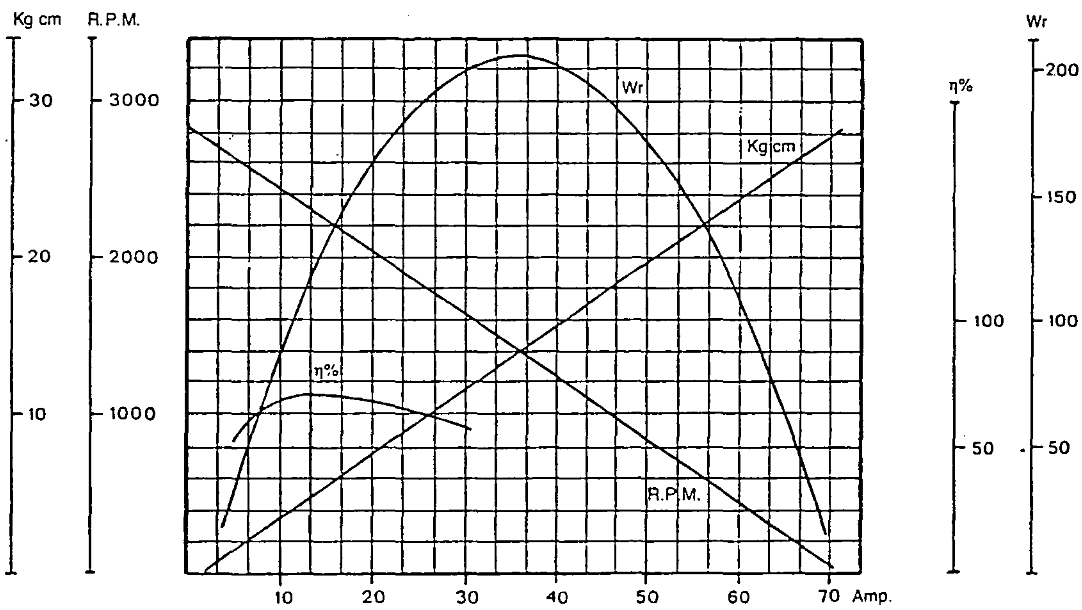
Tension nominale 12 V
Tours minute 2250
Courant nominale 15 A
Puissance rendue 130 W
Sens de rotation réversible
Service continu
Temperature de travail
−40°C +80°C

Notes
Avec hélice plan nr. 825.0018 - 825.0016

Features

Rated voltage 12 V
R.P.M. 2250
Rated current 15 A
Effective Horsepower 130 W
Direction of rotation reversible
Service continuous
Working temperature
−40°C +80°C

Notes
By fan dwg. nr. 825.0018 - 825.0016



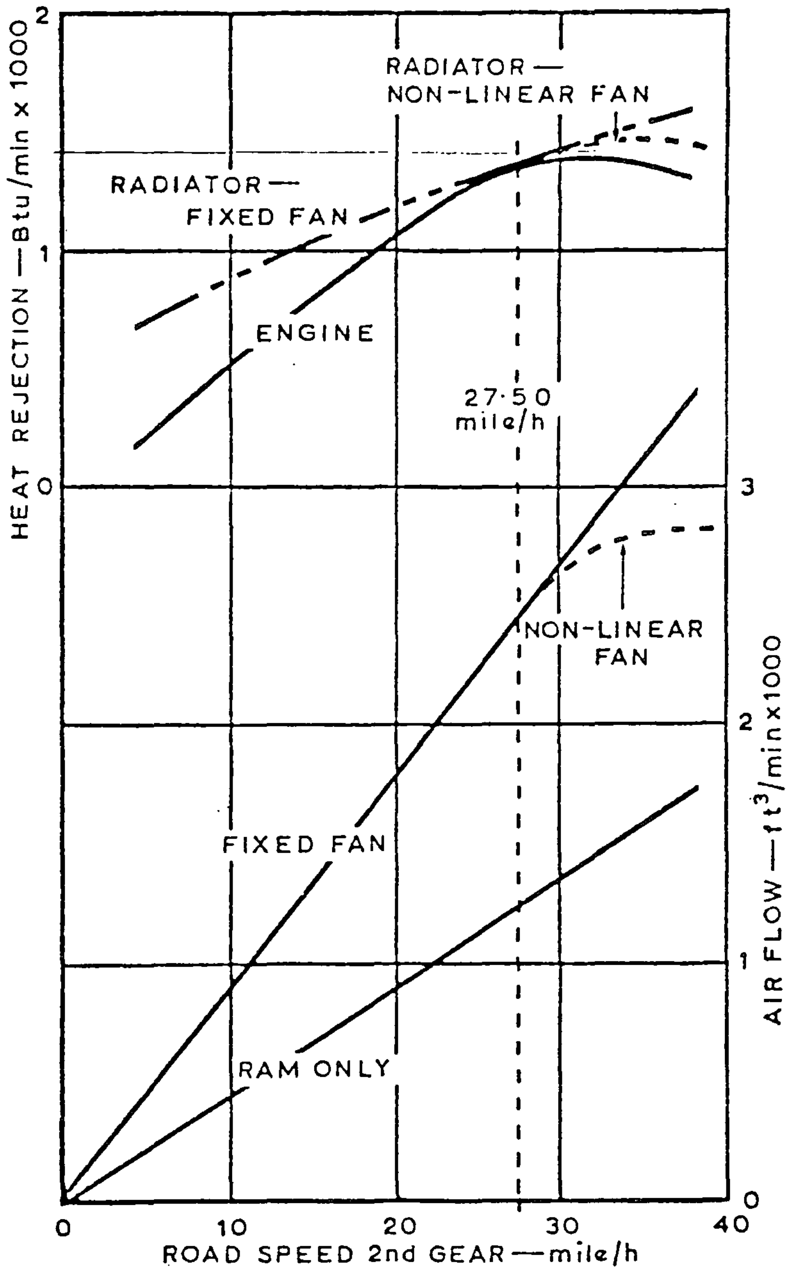


FIGURE 2.6 CRITICAL COOLING CASE [59]

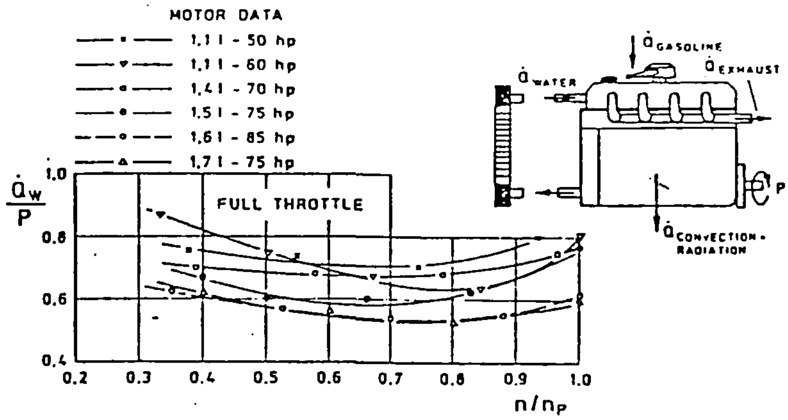


FIGURE 2.7 HEAT REJECTION OF DIFFERENT ENGINES (11)

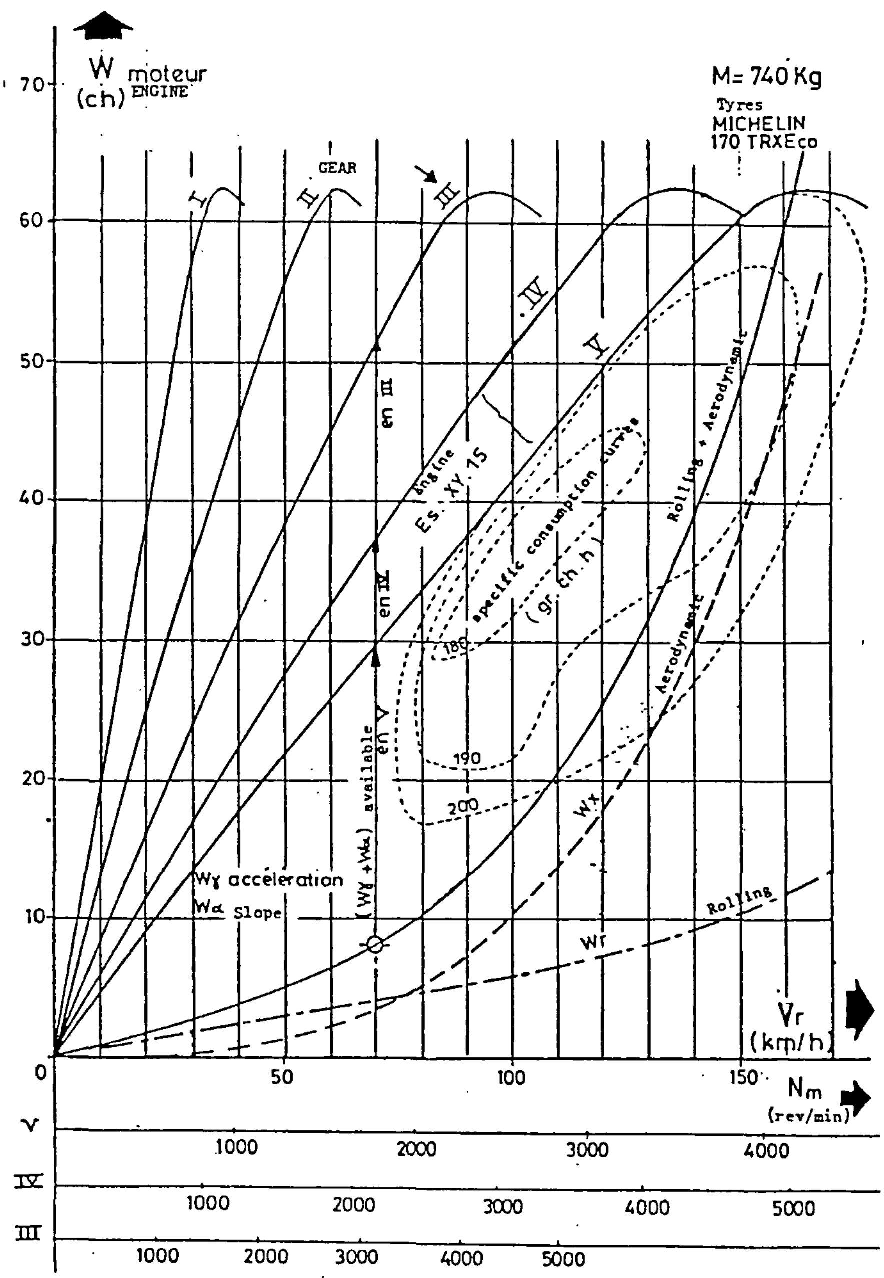


FIGURE 2.8 POWER ANALYSIS (52)

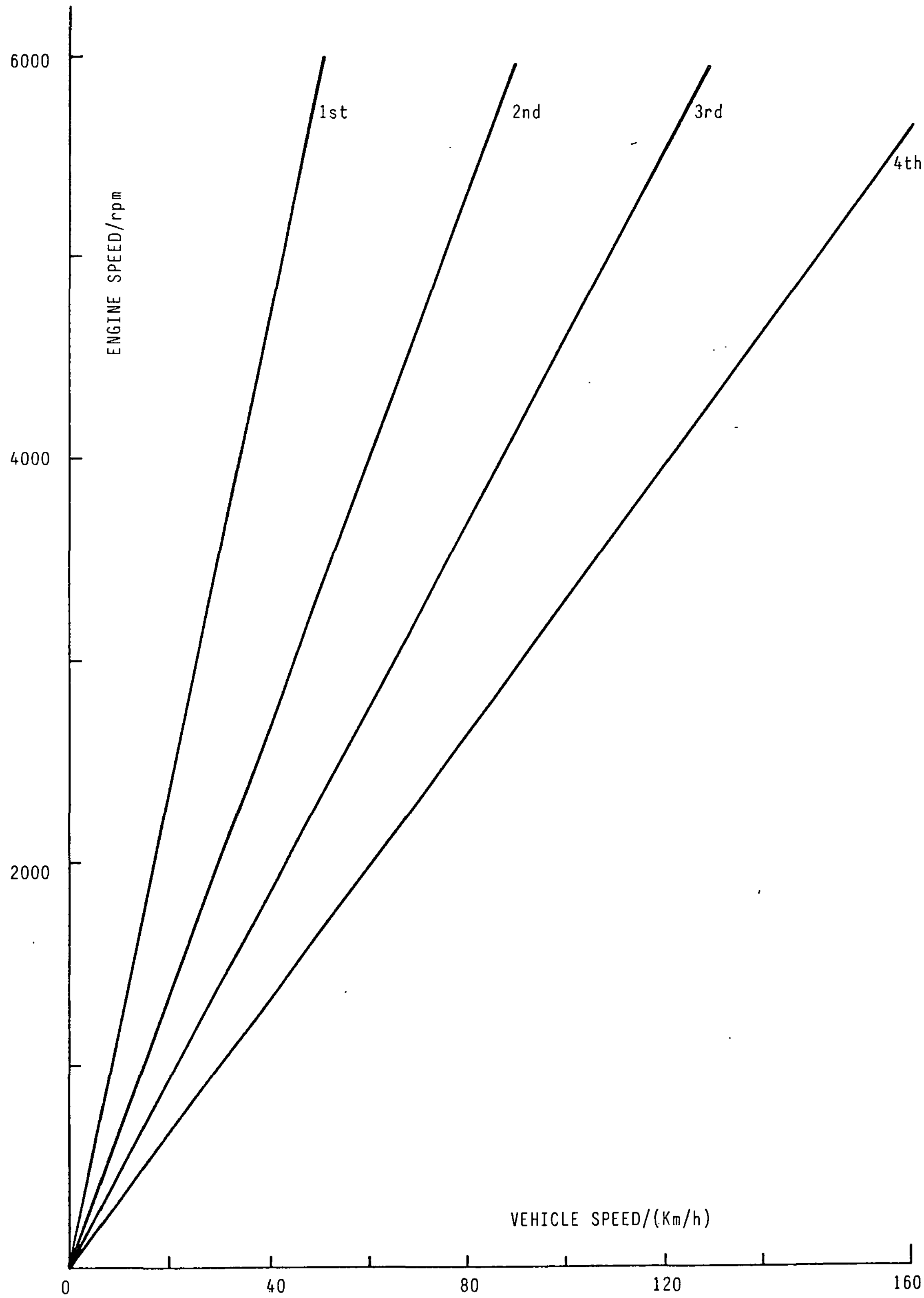


FIGURE 2.9 SIERRA 1.6 OHC - GEAR RATIOS

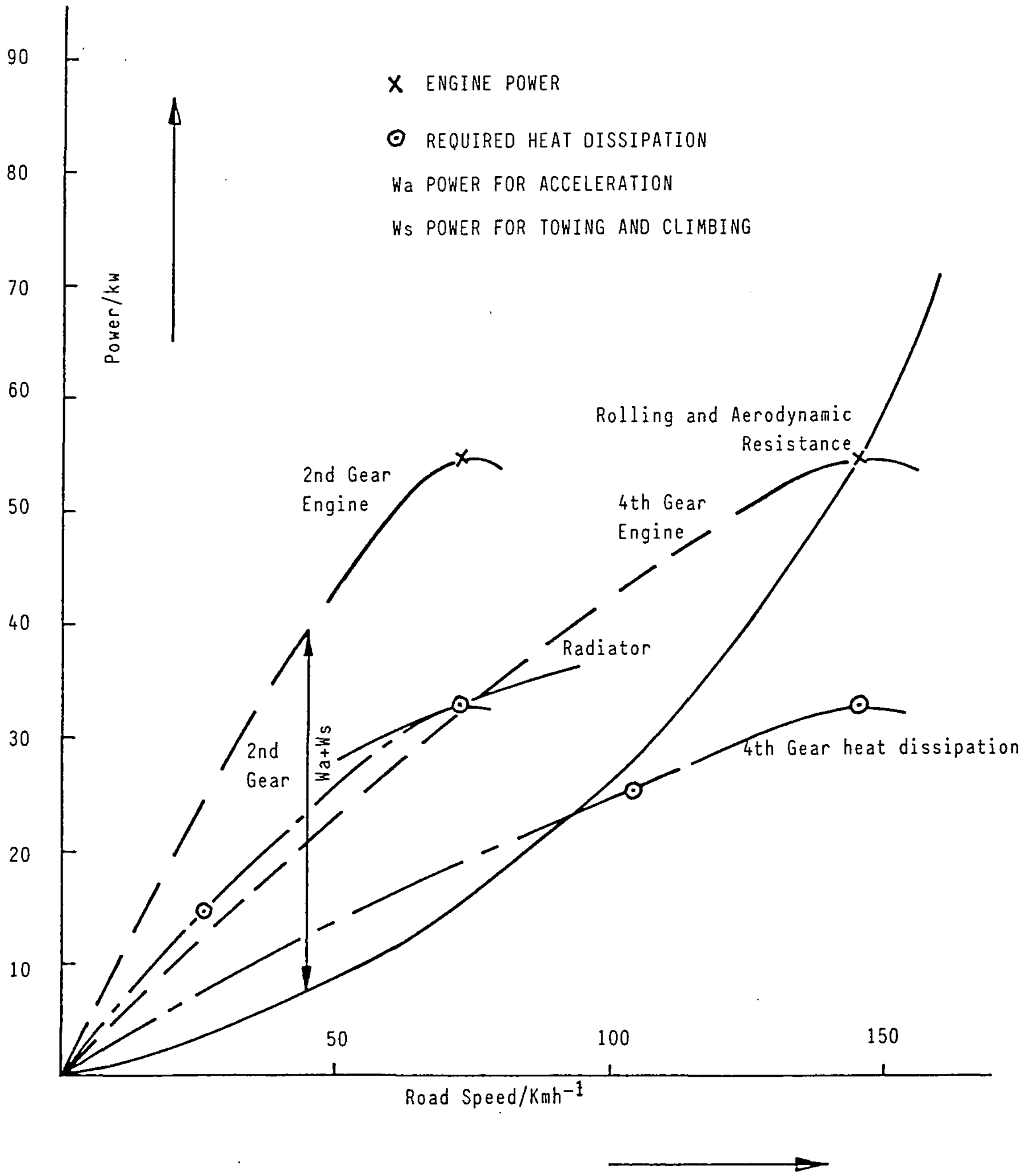


FIGURE 2.10 PREDICTED POWER TRANSMISSION AND HEAT DISSIPATION AGAINST ROAD SPEED FOR SIERRA 1.6 OHC

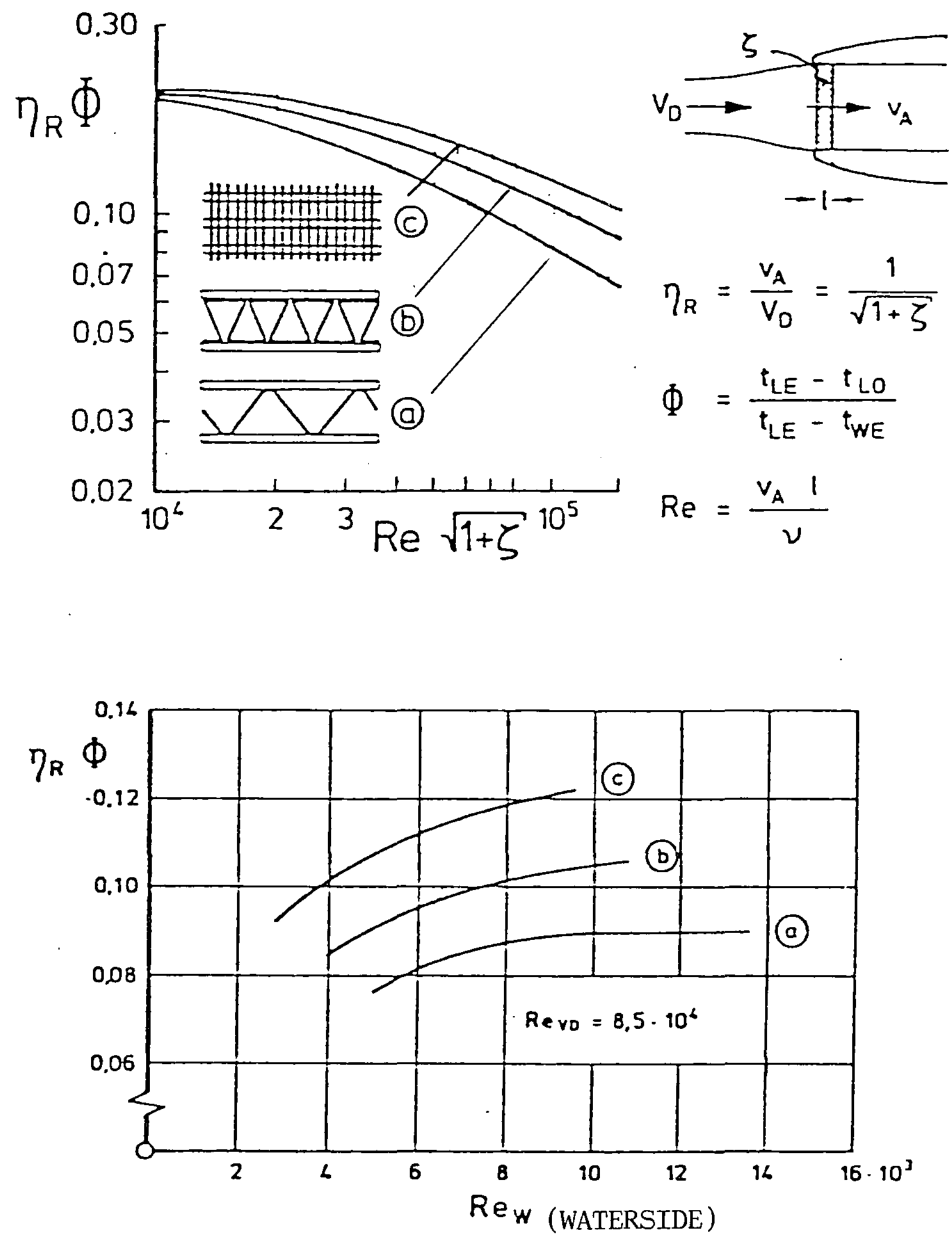


FIGURE 2.11 COMPARISION OF DIFFERENT MATRIX DESIGNS(11)

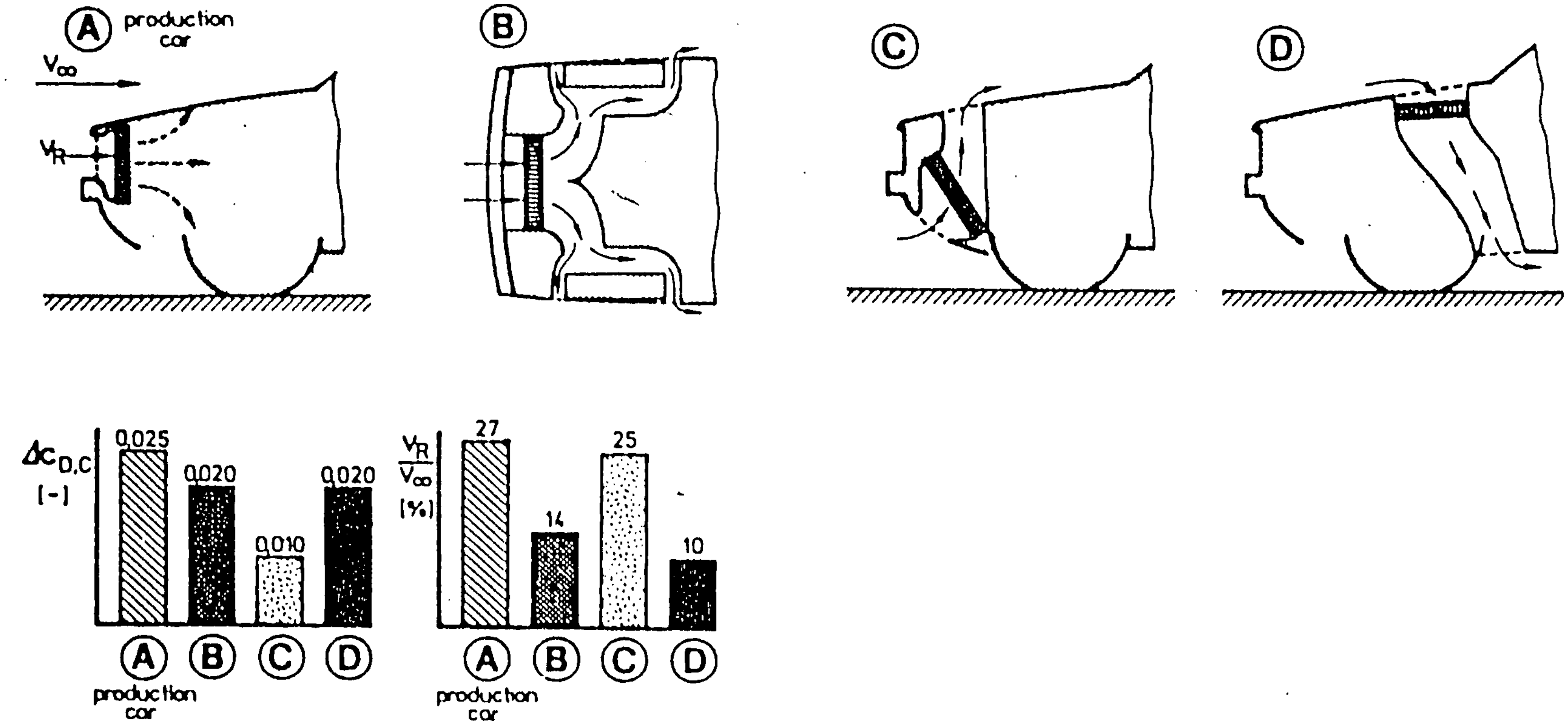


FIGURE 2.12 PROPOSALS FOR COOLING AIR DUCT (3)

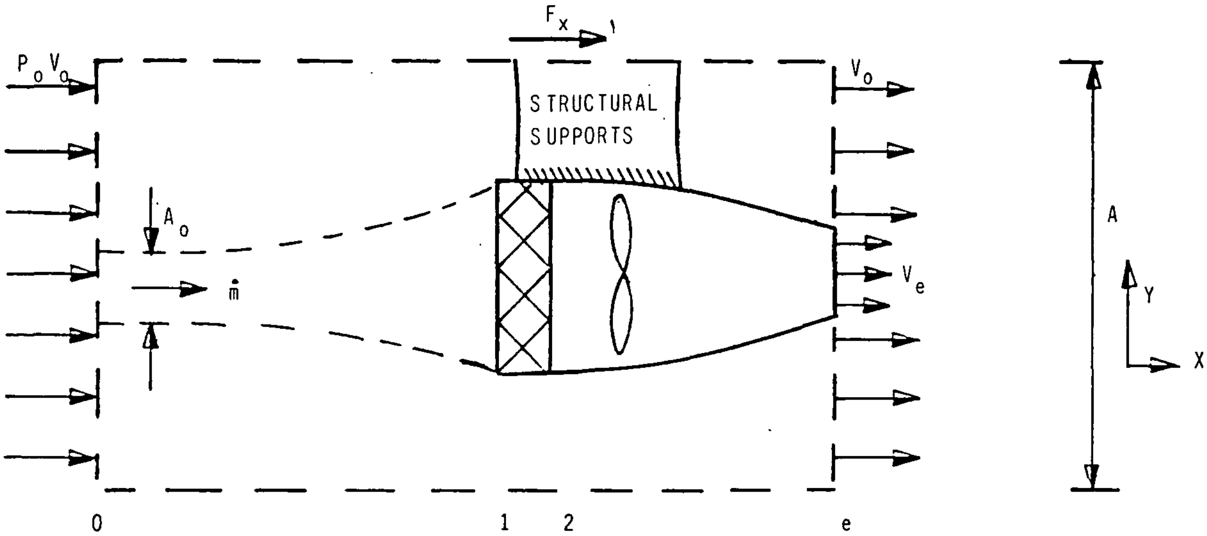


FIGURE 2.13 DRAG DUE TO MOMENTUM LOSS

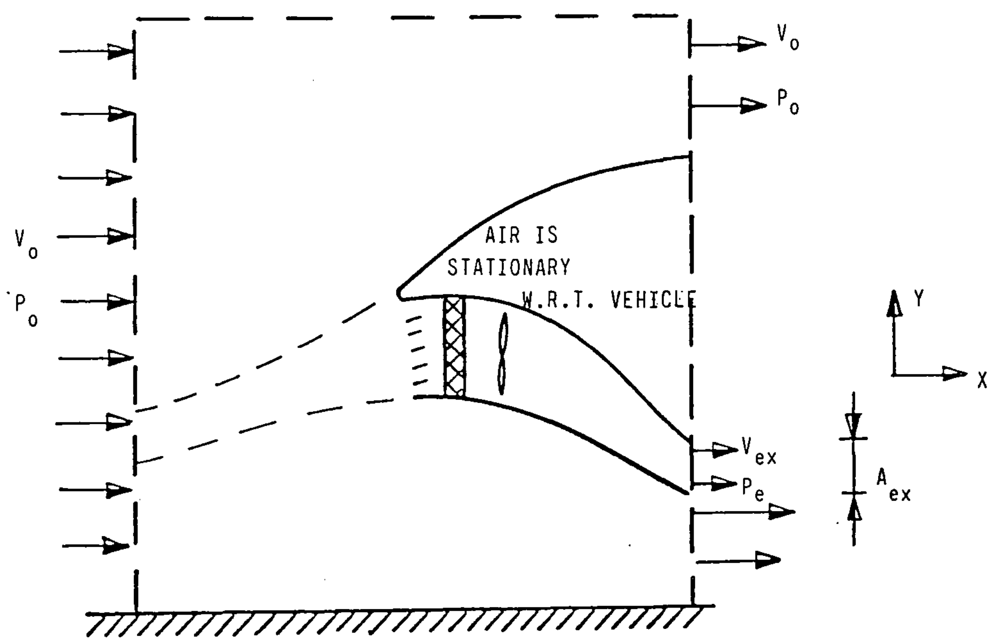
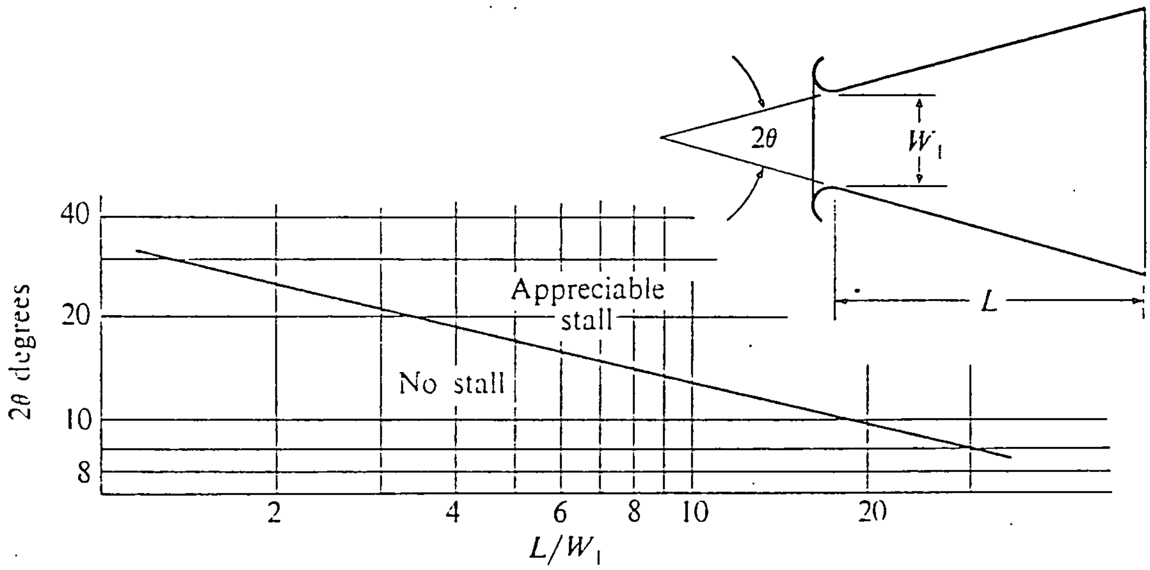


FIGURE 2.14 A PRACTICAL SYSTEM



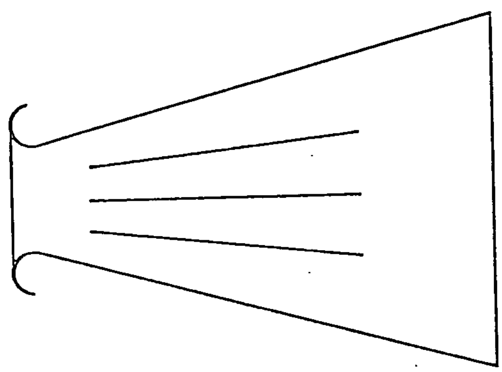


FIGURE 2.15 LIMITS AND PREVENTION OF STALL IN A DIFFUSER (26)

CHAPTER THREE

FAN DEVELOPMENT, AND ITS PERFORMANCE AND RESISTANCE TO RAM AIR FLOW

3.0 FAN DEVELOPMENT, AND ITS PERFORMANCE AND RESISTANCE TO RAM AIR FLOW

3.1 The Aim of Fan Development and Performance Testing

The aim here is to design, develop and manufacture two viable prototype fans, incorporating also some means of a balancing valve for the ram induced air flow to bypass fan rotor obstruction. At higher vehicle speeds, the presence of the fan rotor is known to be unnecessary and disadvantageous. The general rotor and casing parameters were chosen to be the same for both prototypes, except for the inlet and outlet geometries, forming 'L' and 'S' overall casing shapes for different in-vehicle installation philosophies. Two of these viable installation options were shown in figures 3.4.1 and 3.4.2.

For each fan configuration, with and without a vehicle radiator at the inlet, a complete performance test, aimed at determining volumetric flowrate, pressure head, aerodynamic efficiency and flap stability, was made. A slave fan was available to provide simulation of ram air conditions, where entry pressure exceeds exit pressure even with the test fan running.

3.2 Basic Fan Development

An optimum fan design is a compromise between the goals of low noise, high efficiency and adequate flow for a given speed. In addition, the fan should meet size, weight and cost restrictions. Present, cross flow fans use various design philosophies and features, but the simple fan casing geometries with simple construction methods are the best to use for automotive engine cooling applications, to be commercially acceptable.

A cross flow type fan consists structurally of a multi-bladed rotor and an external casing. The throughflow is in a plane normal to the rotor axis. And each fan blade passage experiences a continuously varying flow, each time it makes a revolution through the flow field. The path of a streamline follows through the cascade impeller, first radially inward and then finally is discharged radially outwards. When the rotor is in motion, it generates a Rankine type vortex, with eccentrically located core away from the fan axis of rotation and within the inner blade circumference. Due to this eccentricity, the zones of suction and discharge occur around the rotor periphery.

The selection of an optimum design for a cross flow fan is somewhat difficult with a large number of geometric variables which have an influence on performance. Previous experimental work at Durham by HAINES,[27] ALLEN[1] and TUCKEY[62] has established an optimal design for free air entry, and this was taken as the starting point for this study.

This adopted design of the fan with free air entry and its constructional features were laid down first, followed by a list of desirable features, limitations and guidelines served in selection of this design and on the development of the novel casings, were prescribed. This list will also serve to throw some light, when a definite selection or modification of a design is required for a similar application with different installation, performance and constructional constraints.

3.2.1 Adopted Design Features

The basic fan (fig.3.3A) uses a rear wall of logarithmic spiral form, which is tangential to the outlet duct, where they meet. The vortex wall forms the opposite face of the parallel sided outlet duct, and has a simple sharp edge at the rotor.

The fan rotor is constructed with 24 straight forward curved blades of length 400mm (fig.3.2) with slotted, end plates and a mid-stiffener ring to accommodate blade ends (fig.A1) and the blades were held in place by Belzona liquid metal. The cascade of blades forms an outer diameter of 100mm. The blades were pressed from 20 gauge plate, to a simple arc section of radius 11mm and chord of 13mm, using a jig having a radius of curvature somewhat smaller than that required for final blades, to allow for spring back. Although the blades were made from stainless steel 430 grade for higher strength, for this testing purpose, this might be degraded to a suitable cheaper material for the application concerned. The rotor end plates and the mid-stiffener ring were constructed from aluminimum to minimise rotor moment of inertia. The casings were of 18 gauge mildsteel, continuous welded construction.

The design parameters (fig. 3.3A) governing the basic fan performance were set as follows by referring to previously published work of viable designs.[1, 10, 27, 62]

i) Diameter ratio = d/D = 0.76

ii) Blade angles $\beta_1 = 90$, $\beta_2 = 26°$

iii) Shape of vortex wall is plane and flat

iv) $\gamma = 0$

v) Rear wall clearance $\varepsilon_2/D = 0.07$

vi) Vortex wall tongue clearance $\varepsilon/D = 0.04 \simeq 0.05$

vii) Height of outlet duct $H_e/D = 0.8$

viii) The vortex wall location $B/D = 0.29$ (ie: $\alpha = 36)^\circ$

ix) The thickness of the vortex wall tongue $T/D = 0.01 \backsim 0.015$

x) Pitch chord ratio $(\pi D/Zc) = 1.0$

The rotor (fig.3.2) was belt driven at one end by a pneumatic motor. Driven rotor end and the shaft end were supported by self-aligning, four-bolt flange bearing units. And a self-aligning bearing was built into the far end plate of the rotor (fig.A2).

3.2.2 Desirable Features, Limitations and Guidelines of Basic Fan

The design features given below are mainly a combination of prescription by ALLEN[1] and TUCKEY,[62] who based these on their own findings and on other research by HOLGATE and HAINES,[27] IKEGAMI and MURATA,[30] MURATA and NISHIHARA,[41] PORTER and MARKLAND,[49] PRESZELER and LAJOS,[57] and ECK.[10]

Many researchers generally agree that changes in details of the impeller geometry have a far smaller effect on performance than a change in the geometry of the surrounding casing. TUCKEY[25] has established (fig. 3.3B) the general behaviour and the pattern of the flow, of basic cross flow fan from a flow visualisation study. Identification of the flow behaviour in suction and discharge regions would particularly be useful to appreciate the constraints on the design of inlet and outlet ducting.

The suction arc region divides into three zones:

- the inflow zone (a) covering the arc towards vortex wall leading edge. And the strength of the inflow also increases towards this edge. Therefore obstruction to this region should be avoided by all means.
- the discharge jet (b) from the leading edge of the rear wall. The flow rate associated with this jet, increases as the throttling increases. Attempting to prevent this loss by extending the rear wall height has little benefit and over doing it, can generate a worse situation.
- A zone (c) of entrained flow resulting from the jet over the rear wall.

The flow zones identified in the discharge region were

- a turbulent recirculation zone from the fan interior.
- a diffusing throughflow zone.
- a flow path under the vortex wall, returning essentially laminar flow to the fan. TUCKEY[62] has illustrated these (fig.3.3B) as regions j, k and l respectively, and indicated that the cause of pulsations found during the use of many of these units stems from the shed vortices and recirculation in the outlet duct.

The aerodynamic performance of a cross flow fan is principally dependent on five design parameters (fig.3.3A), diameter ratio, angular position of the rear wall inlet edge, rear wall, vortex wall tongue clearance and the vortex wall location.

3.2.2.1 The Casing

a) The Vortex Wall

This is one of the most important components of the fan, and its location is a main geometrical parameter. An optimum performance can be reached with a straight edged vortex wall, with the leading edge position at $\alpha = 36°$ to a maximum clearance of 5% of the diameter. Original research pursued by HOLGATE and HAINES[27] indicated a trade-off between the performance and efficiency for clearance below 5% of the diameter, with associated increase in noise generation. There is no doubt now that a simple straight wall with a rounded leading edge produces good fan performance and an acceptable level of noise for a clearance to diameter ratios of not less than 0·05. The thickness of the wall will generally be determined by the material stiffness, but less than 4% of the diameter, and a rounded leading edge is recommended. Previous researchers and designers have frequently incorporated a wedge shaped tongue for leading edge. This has been found to have no effect on performance, but might be used purely as a stiffener.

The main advantage of a thicker Eck type wall (fig.3.8) is that it does not necessarily require a small running clearance. Therefore by increasing this clearance to an acceptable level, the sound generated at the vortex wall edge could be considerably reduced. This Eck wall type of design is predominantly used in relatively quieter domestic HVAC compact systems and packages.

IKEGAMI and MURATA[30] proposed sloping of the wall longitudinally as a noise reduction technique and by doing this up to 10°, only slight reduction in total pressure was seen.

3.2.2.2 Rear Wall

PORTER and MARKLAND[49] have clearly shown that a logarithmic rear wall shape with leading edge set at $\gamma = 30°$ will give acceptable levels of performance and efficiency. HOLGATE and HAINES[27] developed a logarithmic rear wall with its leading edge at $\gamma = 0$, and a circumferential extension was attached to the leading edge to extend it forward by $\gamma = 20°$. They also suggested a value between 3% to 5% of the diameter for rear wall clearance, for optimum conditions. But generally the location of the rear wall leading edge as high as $\gamma = 40°$ has little effect on efficiency.

Rear wall clearance was known to be a function of other parameters and is best determined by the particular design. A value of 2 to 10% of the diameter is generally acceptable. A high value of this also reduces the noise level.

All in all the best, basic cross flow fan casing design consists of unobstructed suction arc, a plane, flat vortex wall, and a logrithmic rear wall with an acceptable fan rotor running clearance for performance and noise. The suction arc near the vortex wall, especially, should be left unobstructed since most of the flow enters from this region.

Investigations by TRAMPOSCH[61] indicated that only minor improvements could be achieved by using an outlet diffuser. For effective diffusion a uniform velocity profile is required. But since this is hardly the case at the outlet section adjacent to fan rotor, effectiveness of a diffuser is greatly reduced in this vicinity.

3.2.2.3 The Fan Rotor

a) The Diameter Ratio - With general agreement between researchers, it is possible to suggest a lower limiting ratio of 0.7 and an upper limit of 0.85 for satisfactory operation. Otherwise, the influence of this ratio has little significance.

b) The Blade and its Angles - Eck has shown that β_1 should be 90° although only little effect over the range 60° to 100° was usually observed. IKEGAMI and MURATA(30) and PRESZLER and LAJOS(51) has suggested β_2 should be greater than 22° for stability with values around 26° representing an optimum. This was also later confirmed by MURRAY(30) in his final year BSc project report.

The blade profile generally chosen is a circular arc camber, since each blade experiences flow reversal. It was felt that any attempt to add variable thickness profile to the blades, to suit either the first or second pass (Moore),(38) unlikely to improve performance. But this might be used to the advantage of the structural strength of the rotor.

c) The Length to Diameter Ratio (L/D) - The change in flow rate is directly proportional to length; this is a feature of the cross flow fan. When the fan is too long, however, it tends to behave as a number of fans in parallel with the line vortex breaking into a number of cellular segments. Alternatively, if the rotor is too short, secondary flows developing close to the end plates may distort main flow through rotor. TUCKEY(62) sees these effects to be minimal, in many applications of the past. HOLGATE and HAINES[27] have suggested a maximum value of 4 for this value, for the dimensionless parameters to model the behaviour to a sufficient accuracy. However, there are also strength constraints such as torsional and bending rigidity together with mid-span blade deflection, imposed on the selection of this ratio. Tests by YAMAFUJI[65] showed that large reductions in pressure ratio and efficiency would occur, if for structural reasons a central through-shaft is employed, because of the wake generated by it.

d) Number of Blades and Blade Pitch to Chord Ratio - The number of blades had appeared to be of secondary importance. But the structural strength of the rotor needs careful consideration, when selecting this number. The past researchers have had a preference for rotors between 18 to 36 blades. MURATA and NISHIHARA[41] have optimised a pitch to chord ratio ($\pi D/Zc$) of $0.94 \sim 1.0$. But some workers have found that it is not very important for this ratio to be in this range.

3.3 Test Facilities

3.3.1 The Test Airway

The test airway (fig.1) comprised two plenum chambers, an auxiliary axial flow fan, honeycomb flow strengthener, and an opposed-vane type flow control damper. The auxiliary axial fan which forms part of the control means to overcome the flow resistance, produces a suction effect across the test fan. Located in opposite walls of the rig's outlet plenum are two outlet orifice plates of 139.7mm internal diameter, where the flow is discharged to the atmosphere. These were made to British Standard specifications. There were no walls or obstructions to downstream of the outlet orifice and the flow metering system satisfies BS1042 part one for an orifice plate separating two infinitely large spaces and BS848 part one - 1980, section 22 as a method of determining the flow rate by orifice plate, for testing performances of fans. The plenum chamber is cubic, each dimension being 1.2m, the enclosed volume is sufficient to ensure low flow velocity and low dynamic pressure within the chamber for the testing range. The lower limiting Reynolds number of the orifice plates occur at flow coefficients less than 0.25 for the testing speed ranges of 100mm diameter fan. This limiting value would reduce to 0.125 for one active orifice plate. The flow metering plenum was designed as part of a previous research project by ALLEN.[1] Ref.1 uses the relatively old BS1042 part one; 1964 to obtain an expression for flow rate

$Q = 0.0809 \sqrt{h/\rho}$, but BS848 part one - 1980 section 22 deduce this to be

$Q = 0.0814 \sqrt{h/\rho}$ from $\dot{m} = \alpha \varepsilon \pi \frac{d^2}{4} \sqrt{2 \rho \Delta P}$

where $\alpha \varepsilon = 0.6$ density of water $= 1000 kg/m^3$

standard gravitational acceleration $= 9.807 \, m/s^2$

The conversions in this thesis were based on the former.

A settling chamber between the test fan and the slave fan is of 500mm x 500mm in cross section and 1300mm in length and has a face board at the test fan side to accommodate the fan in question. Static pressure was measured with a piezometer ring of four tappings placed approximately 900mm downstream of the face board. A static pressure check along the height and width showed no considerable (not more than 2%) variation in this region. All pressure tappings used in the ductwork were of 2mm internal bore and fit flush with duct internal surface and were in accordance with BS848 part 1, section 6. The auxiliary axial fan overcomes the resistance of the test airway, enabling the fan under test to be operated through zero to negative static pressure rises, to simulate the effect of high ram air pressures. Cooling air would normally discharge through the outlet of the installed fan to the cavity between the engine underfloor and the ground, experiencing also a sudden enlargement. Therefore, fan outlet chamber was also selected to make grounds of more accurately establishing the installed performance, with least installation losses. Due to the sudden enlargement from fan outlet section to chamber section, there would be a static pressure recovery factor of 0.2 based on fan outlet dynamic pressure for an area enlargement ratio of 7.8,[37] if the small duct discharges with uniform velocity. Therefore, one might attempt to indicate the fan total pressure as the sum of the measured fan static pressure and 0.8 of fan outlet dynamic pressure. But a dynamic pressure check along height and width of fan outlet has shown, this velocity distribution is not perfectly uniform. In a case of discharging to a large space with non-uniform velocity distribution, the total pressure loss coefficient can amount to as high as 3.7.[27] Therefore, the total pressure rise was taken as simply the sum of measured static pressure rise and dynamic pressure based mean velocity at the fan outlet, as in BS848 type A installations. The fan outlet chamber, incorporating a honeycomb, nearer to its downstream face, was connected to the auxiliary fan via the opposed vane type throttling damper. In all, the whole test airway was designed and built to guidelines laid down in BS828 part 1, 1980.

The fan was driven by a pneumatic motor, via a belt drive. The pneumatic motor was pre-calibrated against supply line pressure for fan shaft torque. This was done by replacing the impeller with a brake drum type dynamometer, where mass of the brake drum and connecting shafts were kept approximately equal to the impeller to provide similar bearing loading. The brake drum was manufactured from a Fenner Taper Lock Weld - on Hub (type WH25) machined to accept a friction torque arm. The torque was measured from a spring balance connected to the torque arm. The advantage of obtaining the shaft power by this method is that it excludes losses due to bearing and belt drive mechanism. The resulting calibration chart was drawn in fig.A3 of the Appendix.

Air propellant was supplied to the pneumatic motor via an air filter, a lubricator and a regulator. The lubricator was set to feed oil droplets at fixed regular intervals and the motor and the associated bearings were run for a sufficient time to allow them to reach the normal working temperature.

The system also employed a timer-counter to count the impulses from a 60 segment mild steel timing disc, scanned by a magnetic transducer, which formed the basis of the speed measuring system. The timing disc was constructed from 5mm mild steel plate, machined to a disc of 60 peripheral teeth. This number of teeth were chosen to facilitate the measurement of speed in RPM, directly from the timer-counter.

3.3.3 Measurement Apparatus

The manometer used for the orifice plate's differential pressure measurement was a sensitive, oil filled (SG = 0.821) micromanometer, the vernier scale enables reading to 0.01mm. The pressure differential across the fan was measured to 0.1mm from a digital micromanometer type FC004 by FURNESS. The velocity meter used to scan the inlet face, was by BIRAL. The timer-counter and the probe were type 9902 and G308 by RACAL-DANA and SOUTHERN INSTRUMENTS respectively.

3.4 Novel By Pass Configurations

3.4.1 The 'By Pass' Valve

The two different types of fan inlet geometries and the concept of 'by pass' flap evolved from the earlier investigations into feasibility studies (ref.15). The novel 'by pass flap' replaces part of the static vortex wall. This allows some of the air flow to by pass the rotor under ram air conditions. As a result the desired effect of reduction in pressure drop across the fan is achieved. Use of simple, cost effective construction methods were also one important area of concern here.

3.4.2 Fan Configuration 1

Figure 3.5.1 shows the general composite features of the casing and discharge ducting. Difficulties arise at this stage, in designing a perfect outlet duct geometry for all the settings of the flap, in order to minimise losses that might occur due to flow separation. Therefore the fixed bend geometry of this section was chosen with care, to avoid the adverse values of centre line radius to duct height ratios.[37] Some variation of the outer arc of the outlet bend on the rig was made possible by incorporating a flexible section (fig.3.6.2), constructed in light gauge mild steel. A similar means can be used to vary the inlet height of the 'by pass' duct (fig.3.6.1).

Extensions to rear wall were in two forms. These were demonstrated in figures 3.6.1 and 3.6.2, and were made as integral parts to the main casing. The form 1 uses a circular arc attachment, to log wall leading edge of $\gamma = 0$, as in the case of HOLGATE and HAINES.[27] But has an arc radius and length of 0.08 and 0.085 of the diameter respectively, and can be set to various attitudes around the fixed logwall leading edge, making an angle (γ) between 50° to 85°. The form 2 uses a straight flat piece of 85mm long. Both forms can be positioned to various attitudes, around the fixed log wall leading edge.

The 'by pass' flap was constructed from light gauge plywood and a rubber strip was pop riveted along the top edge, to operate (fig. 3.12) as a hinge. The rubber hinge was most appropriate for the application, though it was thought that the plywood flap needs upgrading to a similar, but more durable material such as light weight plastic. Use of rubber for the whole flap was observed to be unsuccessful with initial small pulsations integrating to heavy amplitude ripples. When the flap closes, it sits airtight, against thin gasket strips bonded to the casing.

The casing had four, quarter circular arc lugs, two diagonally opposite on each side, to be located in place to aluminium discs (fig.3.2). The lugs with self contained slots, slide around the peripheral surface of the discs, allowing adjustment of the casing tilt angle to ground surface. Finally, the whole unit was painted with lacquer to a smooth surface finish.

Hereafter, unless otherwise stated, the test fan configuration 1 refers to a fan having basic design parameters as in section 3.2.1., satisfying also extra conditions of (fig. 3.5.1):

i) circular arc extension set to $\gamma = 55°$

ii) mass of the bypass flap = 100g

iii) tilt angle (θ_T) set to 60°

iv) inlet height to diameter ratio (H_i/D) = 2.85

3.4.3 Fan Configuration 2

Features, shown in figure 3.5.2 were determined from the earlier investigation into effects on variable inlet geometries.[15] This geometry has some similarities to MORTIER fan (fig.3.7) casing. It was ascertained from the earlier study that the fan performance and its resistance to ram air varies little as the entry became progressively more restricted by changing inlet duct height to diameter ratio from 1.7 to 2.8. This ratio was chosen to be 2.65 in this prototype test fan.

As in the previous fan configuration, the 'by pass' flap was constructed from light gauge plywood with a rubber strip as a flexible hinge. But use of rubber for the whole flap observed also to be viable. Fan unit outlet section has means to accommodate a diffuser. This diffuser, 260mm long, is of trumpet shape type, and allows setting of its walls to various divergence. It was mentioned before that use of diffusers in cross flow fans were not effective. But a diffuser located 200mm downstream from the rotor axis, might behave differently.

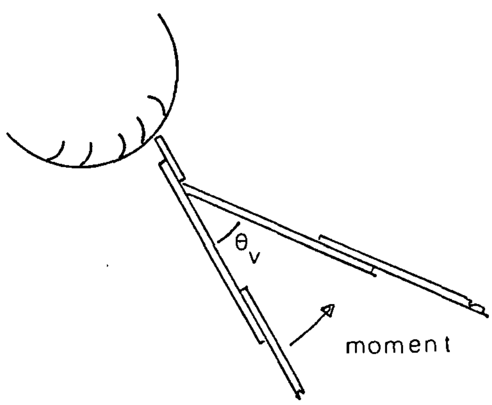
Hereafter, unless otherwise stated, the test fan configuration 2 refers to a fan having basic design parameters as in section 3.2.1, satisfying also extra features as in figure 3.5.2 with a 'by pass' flap having a mass of 90g.

3.5 Scaling of the Novel Fan

Use of fan non-dimensional parameters greatly simplifies the determination of performance and the design of a fan. The scaling of cross flow fan for dimensionless analysis has been researched by HOLGATE and HAINES.(29) Their results indicated that cross flow fan satisfy the accepted similarity relationships. But one might doubt whether this is the case of the novel fan with 'by pass' flap, when operating below zero pressure rises. To clarify this both fan configurations were run at several different fan Reynolds numbers ($Uc\rho/\mu$) between 4.7×10^3 and 6.2×10^3. The results obtained were plotted as pressure coefficient against flow coefficient in figures 3.10 and 3.11. These plots illustrated the dimensionless parameters of the novel fans do model the behaviour to a sufficient accuracy over this Reynolds number range. This also confirms angle of the flap opening as a superfluous variable for one particular fan. The opening angle is superfluous because the flap operates only as a balancing valve, depending on the pressure difference across the unit, when all other physical parameters are fixed.

The above variation in fan Reynolds numbers were obtained by changing speed at constant characteristic length (c = 13mm) but the application in question might require geometrically similar fans of different sizes. Therefore geometrically similar fan of rotor diameter 60mm, 24 blades and chord of 11mm was constructed.

The fundamental variables, for a fixed tilt angle (θ_T), for the novel fan can be written in the form of $Q = f\ (D,L,N,\rho,\Delta P,\ \theta_V)$, where θ_V is the flap opening angle, and θ_V is also a function of 1, EI and flap loading. The flap loading occurs due to weight of flap and 'ΔP'. Since 'ΔP' is already a fundamental variable, θ_V would only be a fundamental variable when it is only a function of 1, EI and mg.



$\tan\theta_V$ = Moment. 1/EI

Moment $=(\Delta PA_VL_{cp} - \rho_Vt_VA_VL_VgCos\theta_V)$ for fan configuration 1 where, see fig. 3·12

L_{cp} = distance to centre of pressure

ρ_v = density of the flap material

t_v = thickness of the flap material

$$\text{Hence, } \tan\theta_v = \left(\frac{\Delta PA_v . L_{cp}}{\rho_v t_v A_v g L_v \cos\theta_v} - 1\right) \rho_v t_v A_v g L_v \cos\theta_v \, 1/EI$$

For same gauge flap material and fixed θ_v

$A_v g L_v l/EI$ = constant

Since A_v and L_v are proportional to DL and D

$D^2 Lgl/EI$ = constant

Therefore rubber hinge stiffness of the smaller fan was scaled down, $D^2 Lgl/EI$ to be equal. Then the smaller fan was tested for

Re = 5.0×10^{3} and 6.8×10^{3}

and the results were plotted in figure 3.9. Though this illustrated a little deviation, these dimensionless parameters still indicated to model the behaviour quite well.

Hinge length (l) of the smaller fan was reduced by four times and then the results of these and figure 3.13 were superimposed in figure 3.9. An effect similar to that of flap mass, indicated in Section 3.6.11, was demonstrated.

The material testing machine, INSTRON type 1195, was used to measure the Young's modulus of the rubber hinge. This value was ascertained to be 11.03×10^{6} Pa.

3.6 Fan Performance

3.6.1 Fan Configuration 1 -(fig 3·5·1)

This fan behaves like a normal cross flow fan, for all positive pressure rises, with the flap fully closed to form the flat and straight vortex wall, immediately downstream to rotor. The transition region from positive to negative pressure rises was observed to be smooth, stable and free from any fluctuations, and the action of flap opening was counter to its weight.

3.6.1.1 Effect of Flap Mass

The effects of flap mass on fan performance and efficiency were plotted in figs 3.13 and 3.14. Mass of the original flap was gradually increased by bonding together six layers (fig. 3.12) of thin gauge mild steel plates across the flap with care, to cause least disturbances to the flow at the surface of the flap. The centres of gravity of the flap and the attaching sheet were made to coincide. Strangely, the performance seems to improve with increasing flap mass, for the flap operating range; ie below zero pressure rises. But it was observed that when the flap mass exceeds 500g, the flap operation became unstable with violent fluctions.

3.6.1.2 Effect of Unit's Tilt Angle to Horizontal

If the flap mass was an important parameter for performance, the tilt angle (Θ_T) of the unit also thought to be worth considering. Increasing the unit's tilt angle to horizontal would also reduce the flap closing moment as in reducing mass. Therefore as one might expect the graph of figure 3.15 confirms this by illustrating a drop in performance for an increased tilt angle.

3.6.1.3 Flap Operation

The least heavy flap balances to almost fully open, swiftly with flow rate, for below zero pressure rises and remain in that position well stationary for any increase in flow rate beyond. As the flap became heavier, more gradual opening of it, correspond to an increase in flow rate was observed. When the flap has taken up its position for a certain flow rate, a severe instanteous impact movement to the flap by a foreign means, was followed by subsequent heavy damping, back to its original position. This reflects the stable character of the flap for example, when a vehicle is under sudden braking, acceleration, impact wind effects when overtaking, or other similar adverse atmospheric conditions. This rate of damping also tend to decrease marginally with flap mass.

Flow behaviour visualised by a smoke generator at the vicinity of inlet side of the flap surface has shown two well defined zones for flow streamlining towards rotor suction arc and the by pass ducting. Streamlines separating the two zones approach smoothly towards a 'stagnation point flow' region near the flap surface, then separate into streams proceeding to opposite directions, showing no signs of disturbance adjacent to flap surface.

The outlet side of the flap, probably disrupted the regular vortex shedding of this region. This influenced by the smooth inlet flow at flap surface, has eliminated the undesired effect of flap pulsations, that one would expect from shedding vortices.

3.6.1.4 <u>Gains in Performance Due to Flap Operation</u>

Throughout the test range the flap was first held closed by a foreign means. These results were then compared with ones obtained, when the flap was left free to operate, in figure 3.16. Gains in performance were extremely beneficial, when the fan operates under negative pressure rises with the 'by pass' valve. For example, when the pressure coefficient was -3.0 the indicated gain in flow coefficient was 22% of static vortex wall fan. Obviously, no changes in performance were demonstrated for positive pressure rises, since then the operating geometrical environments are similar for both cases.

3.6.1.5 Outlet Arc of Outlet Bend

Effective reduction of this arc curvature, as shown in figure 3.6.2, would increase the outlet bend. As a result an additional resistance to the fan would be formed. This was responsible for the marginal reduction in positive pressure coefficient, indicated in figure 3.17.

3.6.1.6 Rear Wall Extensions

The graphs of figure 3.18 illustrate effects of rear wall extensions both of circular arc and flat forms (see figs 3.6.1 and 3.6.2). The performance slightly tends to improve, when the extension piece opens away from the rotor.

3.6.1.7 Inlet Duct Height

When the inlet height was reduced by 40mm as shown in figure 3.6.1, the effect on performance was demonstrated (fig.3.19) to be non-existent.

3.6.1.8 Inlet Duct Transition

Employing a triangular transition duct piece at the fan inlet, as shown in figure 3.20, has been proved to improve the air entry efficiency of the by pass duct by indicating an increase in performance for negative pressure rises.

3.6.1.9 Curtailed Back Wall

Back wall of the outlet duct was cut back to a length of 125mm as shown in figure 3.40 with care not to shift the outlet section too close to the rotor, where the flow behaves in a disturbed pattern. This alteration indicated to be favourable for the fan, when it is running under negative pressures, but unfavourable for the fan under positive pressure rises.

3.6.2 Fan Configuration 2

This fan configuration also behaves like a normal cross flow fan, for all positive pressure rises, with the flap fully closed to form the flat and straight vortex wall immediately downstream to rotor. In this case, the flap opened, assisted by its weight.

3.6.2.1 Effect of Flap Mass

Fan performance and efficiency were plotted in figures 3.21 and 3.22 for flaps having three different masses with 100g apart. The same technique as in fan configuration 1 (fig.3.12) was adopted to increase the mass of the original flap. Since the flap opened, assisted by its weight, the effect of mass on performance was opposite to that of previous case and has indicated a drop in performance and efficiency for an increase in flap mass, in the flap operating region.

3.6.2.2 Effect of Unit's Tilt Angle to Horizontal

Increasing unit's tilt angle by dropping the outlet end 10°, from horizontal, also reduces the flaps opening moment as in reducing mass. Therefore, the graphs drawn in figure 3.23 for these two positions illustrated that the unit tilted to 10° has performed better.

3.6.2.3 Flap Operation

When the least heavy flap (original) was in position, the unit behaved like a normal cross flow fan for all positive pressure rises, with the flap fully closed to form the flat and straight vortex wall. But when the flap mass was increased by 100g and 200g, its operation became somewhat vulnerable to back pressures at a point around, just above the zero pressure level, by performing a jump from this point to zero level, without ever achieving any points in between. This point was detected to be approximately ψ = 1.1 and 0.5, when the original flap mass was increased by 200g and 100g respectively. But the least heavy flap operated smoothly and stably and no undesirable effects similar to that above were observed. It also performed well, when the 'instantaneous impact loading' test was carried out as in fan configuration 1. But the other two heavy flaps performed feebly around the proximity of just above the zero pressure level, for this test, and demonstrated prolong low frequency fluctuations, before it ultimately settled down.

The observed flap stability is a result of smooth flow at one side of the flap surface and the disruption of the regular vortex shedding by the other side.

3.6.2.4 Gains in performance Due to Flap Operation

Throughout the test range, the flap was first held closed by a foreign means. These results were then compared with ones obtained when the flap was left free to operate in figure 3.24. Gains in performance were extremely beneficial, when the fan operates under negative pressure rises with the 'by pass' valve. For example, when the pressure coefficient was -3.0 the indicated gain in flow coefficient was 21% of the static vortex wall fan. There were no performance gains for positive pressure rises, since then the operating geometrical environments were similar for both.

3.6.2.5 Effect of Diffuser at Fan Outlet

Use of a diffuser has indicated in figure 3.25 to be undesirable in the operating range of negative pressure rises. In this range the effect on by passed air flow due to the diffuser is destructive. But the performances in general, when operating in the range of positive pressure rises, were marginally favourable.

3.7 Resistance of the Unenergised Fan to Ram Air Flow

Unless otherwise stated, tests below refer to unenergised stationary fan.

3.7.1 Fan Configuration 1

3.7.1.1 Effect of Flap Mass

Pressure drop across the unit against flow rate was plotted in figure 3.26 for flaps with various masses. When the fan rotor was unenergised and stationary, the resistance due to the fan unit appears to be well independent of the flap mass (tested up to 750 g) for high volume flows (ie, $Re = 1.8 \times 10^5$ based on outlet). But this independence breaks down a little for low Reynolds number flows, as the mass was progressively increased.

3.7.1.2 Effect of Unit's Tilt Angle

Increasing fan tilt angle (θ_T) from 60° to 70° would reduce the flap's closing moment as in reducing mass. But this change in moment seemed to be insufficient to show any considerable deviation in figure 3.2.7.

3.7.1.3 Flap Operation

Operation of the flap was as in the fan energised case, except the flap opens relatively faster.

3.7.1.4 G ains due to Flap Operation

Throughout the test range, the flap was first held closed by a foreign means. These results were then compared with ones obtained when flap was left free to operate, in figure 3.2.8. The associated gains were extremely beneficial with resistance dropping to half, when flap was under operation.

3.7.1.5 Outlet Arc of the Outlet Bend

Outlet arc curvature of the outlet bend was effectively reduced as in figure 3.29. The effect due to this change indicated to be marginal.

3.7.1.6 Rear Wall Extensions

The resistances of the fan were tested, when values of 'Y' were set, as before in 3.6.16, for both circular arc and flat extensions. The resistance of the fan did not demonstrate any signs of dependence on this parameter.

3.7.1.7 Inlet Height

The effect on flow resistance, when the inlet height was reduced by 40mm from the bottom, indicated in figure 3.30 to be almost non-existent.

3.7.1.8 Inlet Duct Transition

Employing a transition duct piece at the inlet as shown in figure 3.31, seems to improve the inlet efficiency by reducing slightly the overall resistance to flow. Therefore the use of this duct transition would be advantageous because it has slightly reduced the resistance to air flow. On the other hand, with this transition the ram air and inlet would be interacted more desirably for ram induced air flow and also the test simulate better for this interaction, to predict the resistance of the unit to ram induced air for installations such as in figure 3.4.1.

3.7.1.9 Effect of Rotor Locked and Freewheeling Conditions

Figure 3.32 shows plots of pressure drop against flowrate for the least heavy original flap, when the rotor was both locked and free wheeling. Though the fan was only set to free wheel under bearing friction, the attained rotational speeds were low for the fan rotor to absorb a significant portion of the useful flow energy. For example, when the flow rate through the unit was 0.55 m^3/s the rotor turned, in a direction counter to normal driven rotation when acting as a fan, at a speed of only 85rpm. Therefore the difference in the resistance, when the rotor was freewheeling to that of stationary was not significant, with the former having the slightly higher value. This would seem to indicate very little flow passing through the fan itself, the vast majority taking the bypass route as intended.

3.7.1.10 Curtailed Back Wall

Curtailed back wall, as in Section 3.6.1.9, indicated in figure 3.41 to be favourable.

3.7.2 Fan Configuration 2

3.7.2.1 Effect of Flap Mass

Unenergised fan was tested for three flaps, each having 100g apart. A tendency of the resistance to increase with the flap mass was indicated in figure 3.33. This tendency was insignificant though not beneficial, with the least and the heaviest flaps deviating by around 4 per cent of the flow rate.

3.7.2.2 Flap Operation

Flap always stationed stably to almost fully open throughout the operation.

3.7.2.3 Gains Due to Flap Operation

As in 3.7.1.4 before the results were compared when the flap was restricted to closed position and left free to operate, in figure 3.34. The associated gains were extremely beneficial, indicating nearly a 100 per cent reduction in resistance when the flap was in operation.

3.7.2.9 Effect of Rotor Locked and Freewheeling Conditions

When the rotor was allowed to freewheel under bearing friction, it only began to do so for flow rates above 0.51 m^3/s (Re = 2.1×10^5 based on outlet), counter to usual driven rotation with very low speeds, reflecting very little on absorption of flow energy. Hence, the difference in resistances, when fan stationary and freewheeling was negligible (fig. 3.35).

3.8 Overall Behaviour of the System with a Radiator at the Inlet

3.8.1 Fan Configuration 1

3.8.1.1 Overall Performance

A conventional crossflow radiator (single row, 65 FPD, Al, BEHR) was installed at the fan inlet in three different attitudes. Two of these employ a triangular transition duct piece in between (figs 3.36 and 3.37), the other was directly fitted to the fan inlet (fig. 3.43).

The triangular transition duct piece, when installed, has two triangular opposite vertical planes, with their bases at the top and bottom of the duct to form the two attitudes. The performances were plotted in figure 3.42 and then compared with the free inlet in figure 3.43.

3.8.1.2 Overall Resistance

The same three fan/radiator combinations as in 3.8.1.1 above, were tested. Graphs plotted in figure 3.44 for these and corresponding free inlets, illustrated that they were nearly independent of the three installation attitudes.

Pressure drops of fan/radiator combinations were varified to be proportional to flow rate to the power 1.8 as

$$\Delta P/\mathrm{KPa} = 1.04\ (Q/\mathrm{m}^3\,\mathrm{s}^{-1})^{1.8}$$

It is a fact that the pressure drops of individual components of this combination are proportional to flow rate to the power of various factors. But log/log plot of the above combination deduced this factor to be 1.8.

A graphical pattern of the overall behaviour was shown in figure 3.45.

3.8.1.3 Velocity Distribution at the Radiator Inlet Face

The velocity profiles were also scanned at fan inlet for these radiator installation attitudes. The positions of the traverse lines and measuring points were selected in accordance with Log Tchebycheff Rule. The measurements of the top row velocities were unpredictable with velocity meter indicating fluctuating values. Therefore except this top row, all the other traverse points indicated a well uniform velocity profile as shown in figures 3.36 to 3.41.

3.8.2 Fan Configuration 2

3.8.2.1 Overall Performance

Same radiator, as before, was directly connected to fan inlet and the performance of this combined unit and the one without the radiator were compared in figure 3.46.

3.8.2.2 Overall Resistance

Pressure drop against flow rate for the fan/radiator combination and the fan with free inlet, were plotted in figure 3.47. The pressure drops were varified to the following relationships:

$(\Delta P/KPa) = 1.735\ (Q/m^3/s)^{1.95}$ with radiator

$(\Delta P/KPa) = 1.125\ (Q/m^3/s)^{2}$ - free inlet

The flow rate to the power 1.95 and 2 indicates the predominating resistance was created by turbulant flows.

A graphical pattern of the overall behaviour was shown in figure 3.48.

3.8.3 Effect on Performance due to Radiator

The basic cross flow fan suction arc identifies three zones with the most important, inflow zone having streamlines curving towards the rotor. The novel fan has another one extra zone, directing flow to the bypass duct (ie, for negative pressure rises). Therefore a vehicle radiator with straight, parallel throughflow channels, would force the original flow pattern to alter. Especially in the case of fan configuration 1 which has its inlet face geometry oriented in a discreet flow zone (around vortex wall) of the suction arc. This effect on configuration 2 can be regarded as negligible. Therefore severity of this situation was assessed by comparing the additional pressure/pressure coefficient drop due to the radiator on each configuration, which indicated to be insignificant.

3.9 Summary

A summary of design conclusions arising from discussion of these experimented results is presented in Section 5.3, page 161.

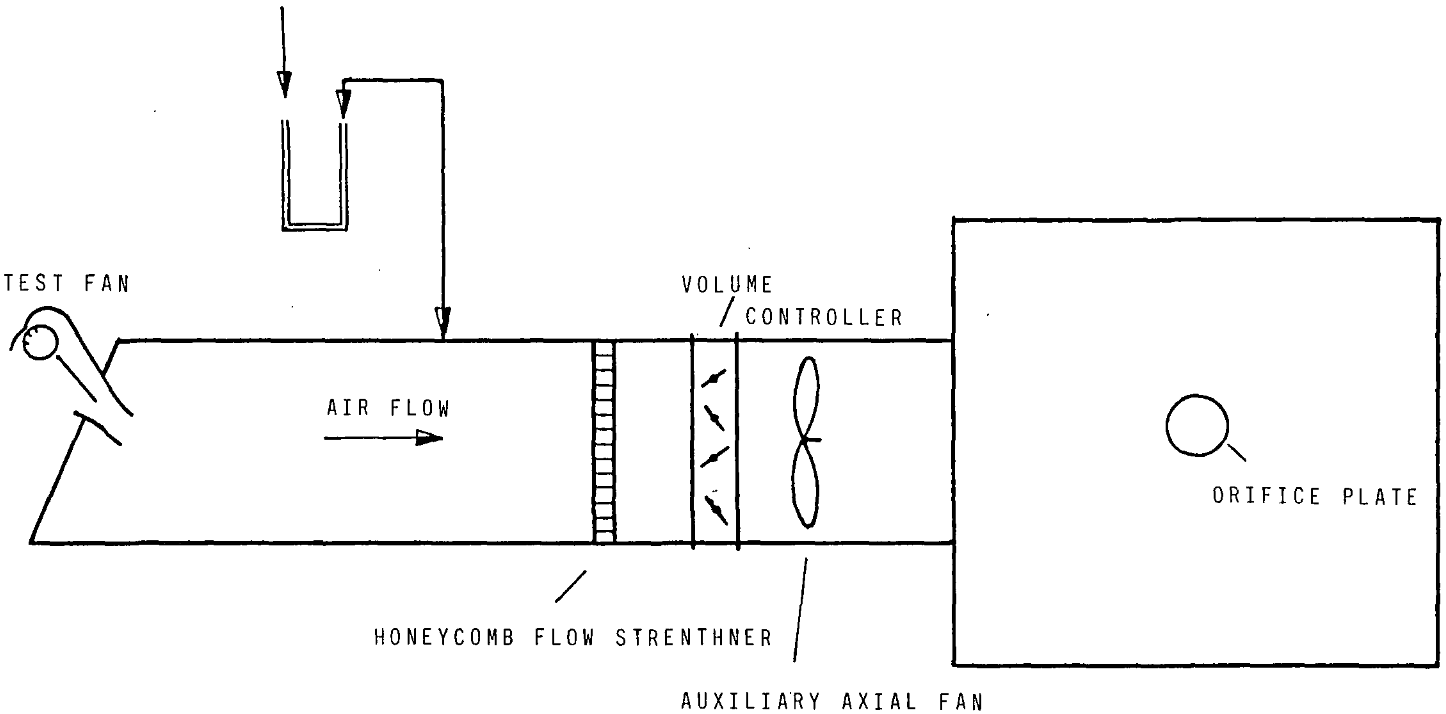


FIGURE 3.1 THE TEST AIRWAY

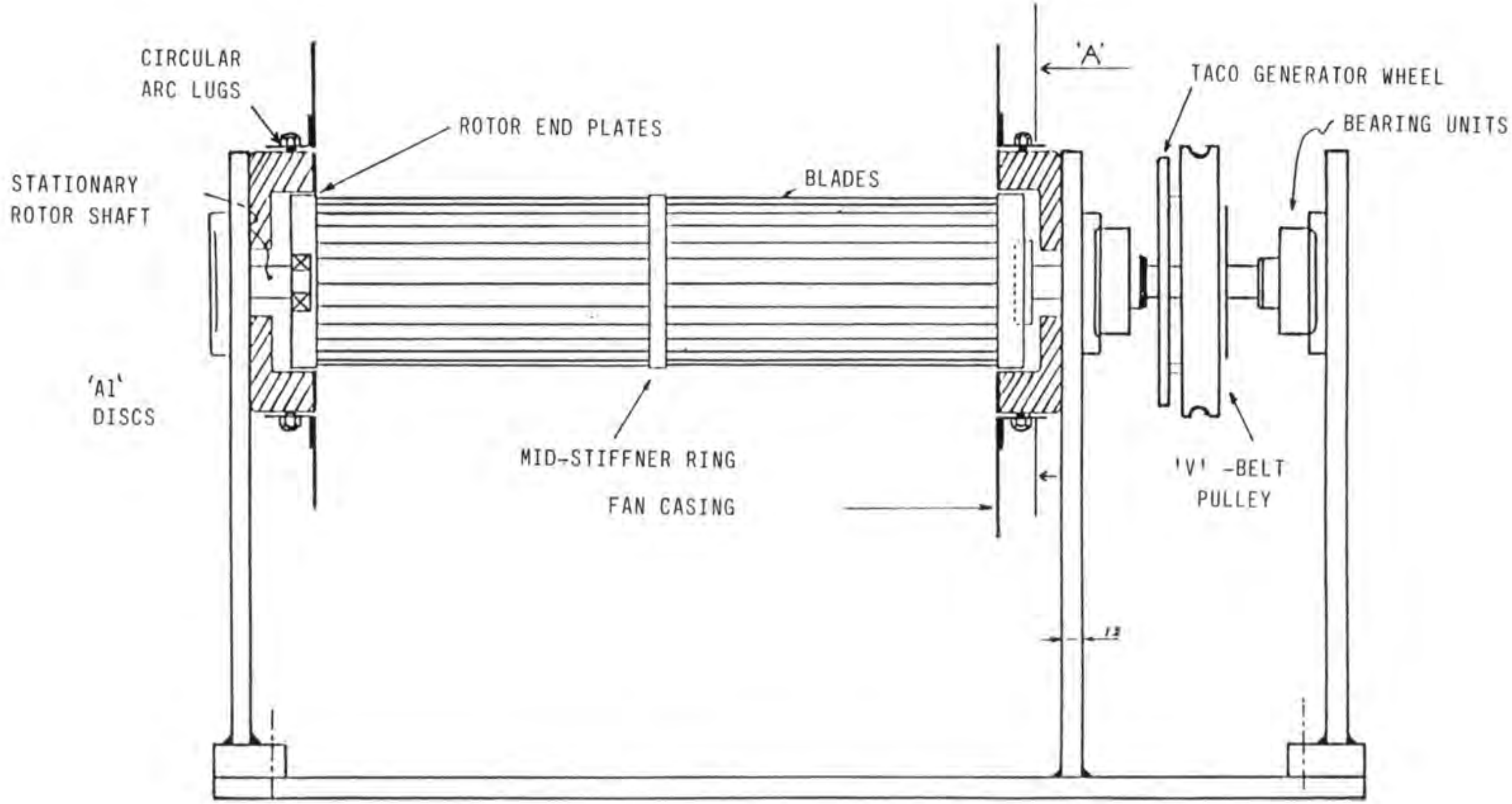


FIGURE 3.2 FAN ROTOR

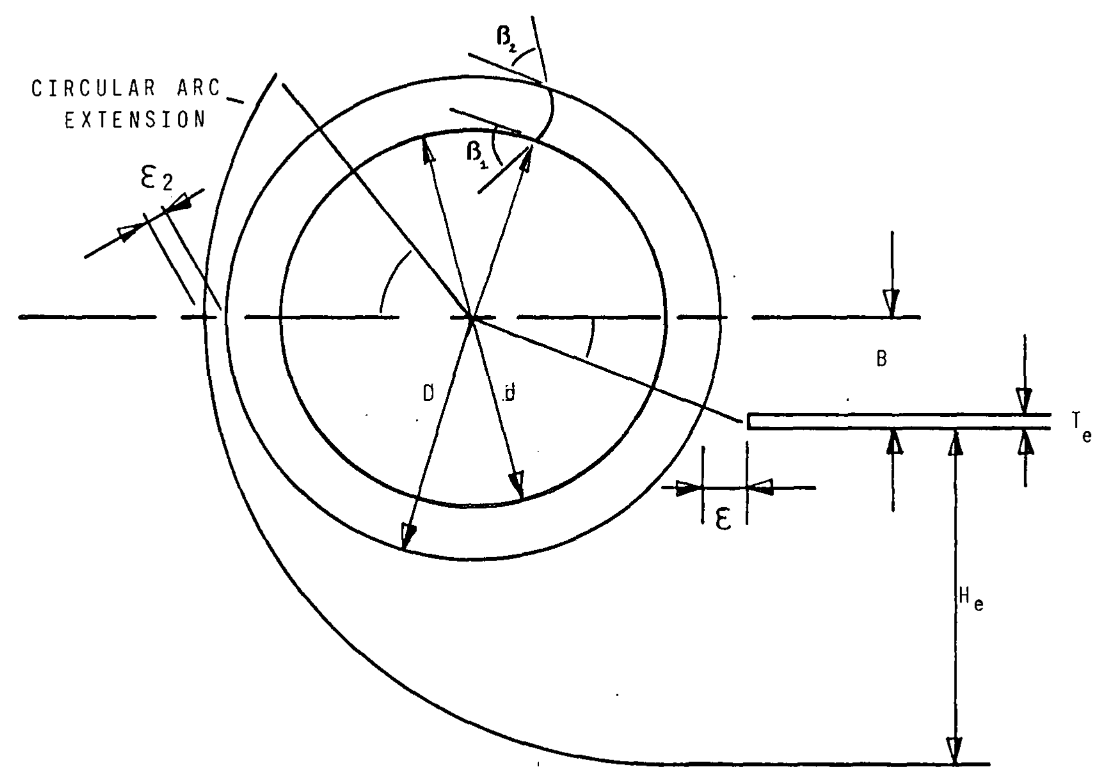


FIGURE 3.3A PRIMARY VARIABLES

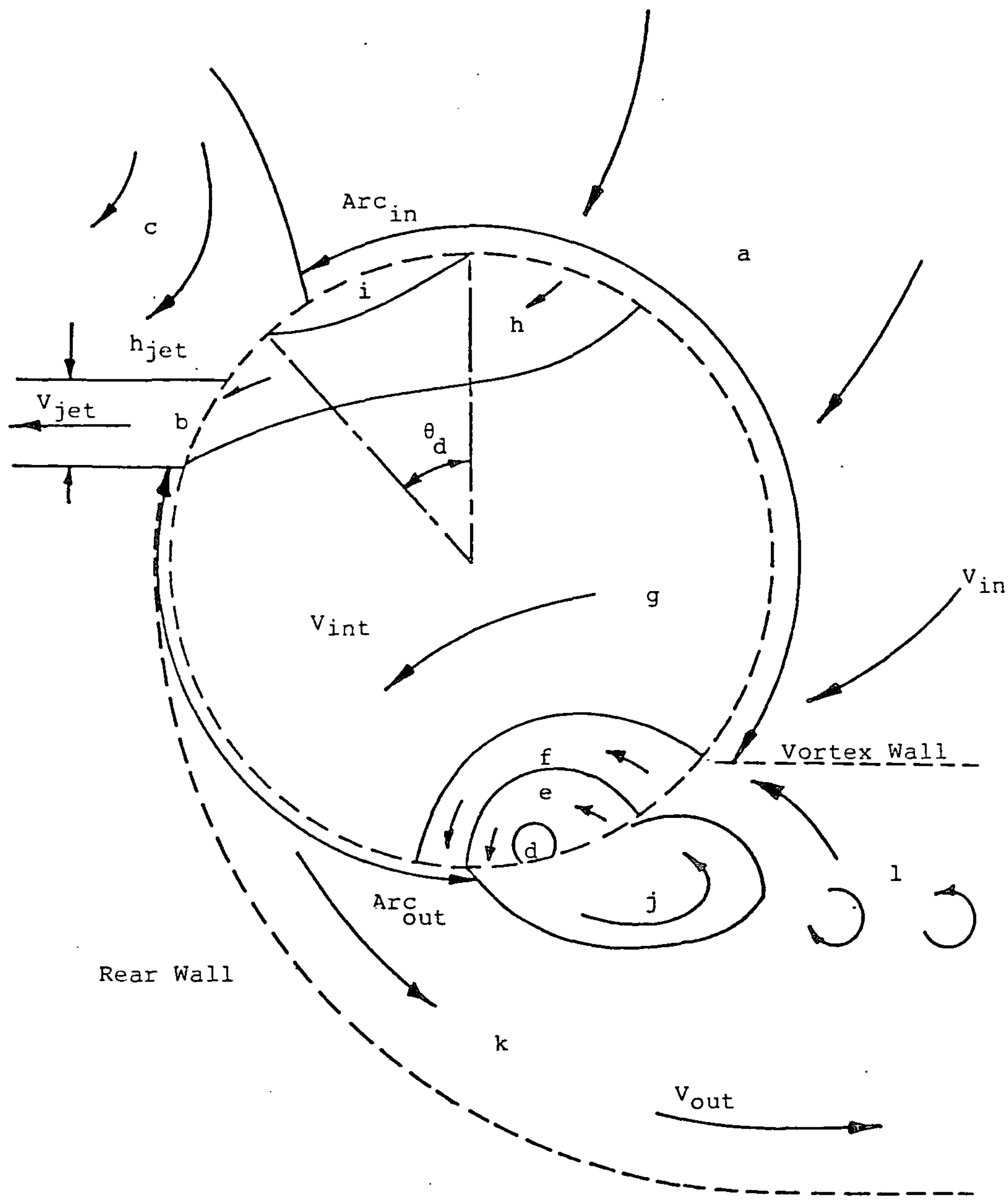


FIGURE 3.3B REGIONS OF FLOW [62]

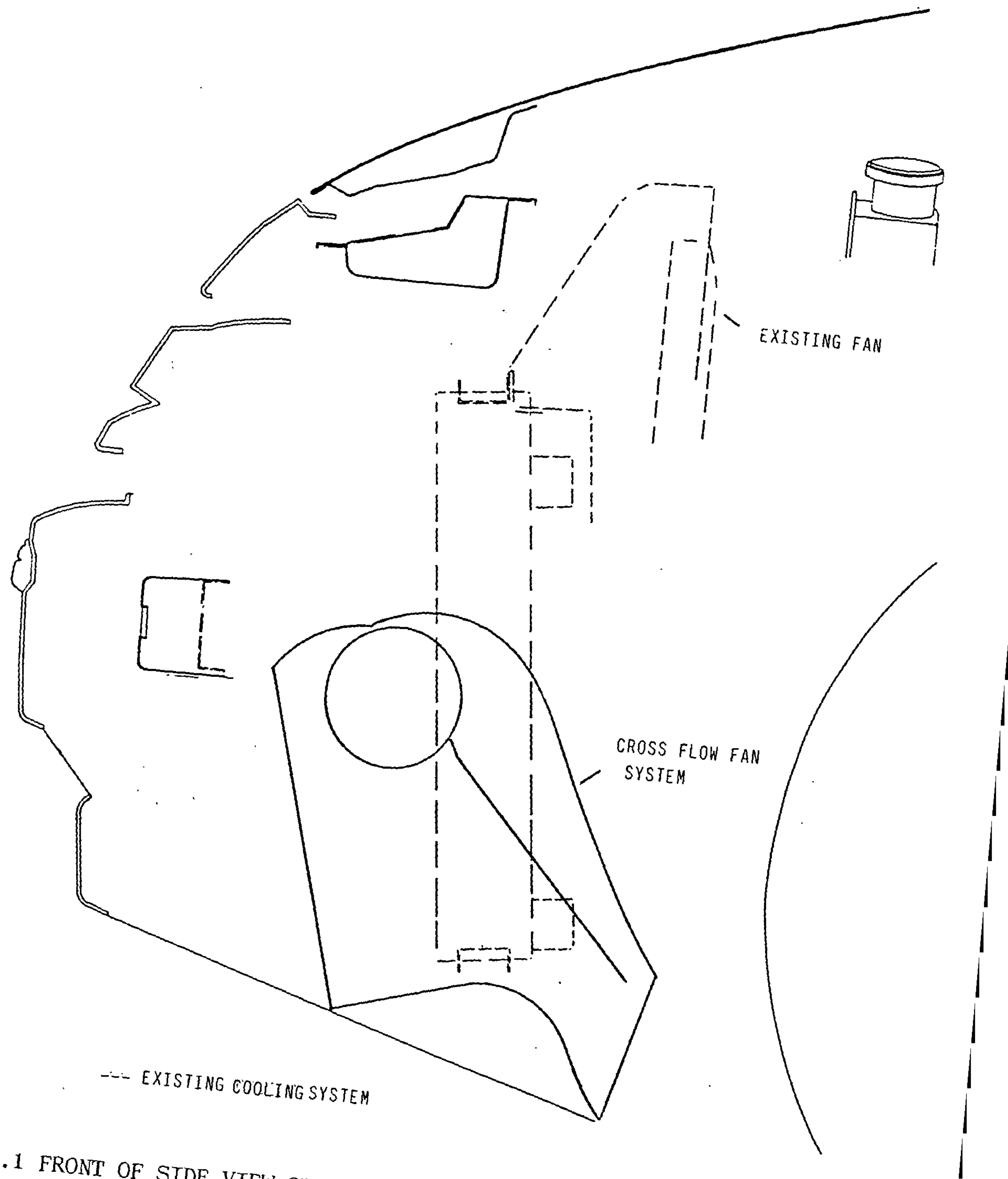


FIGURE 3.4.1 FRONT OF SIDE VIEW OF SIERRA 1.6 OHC

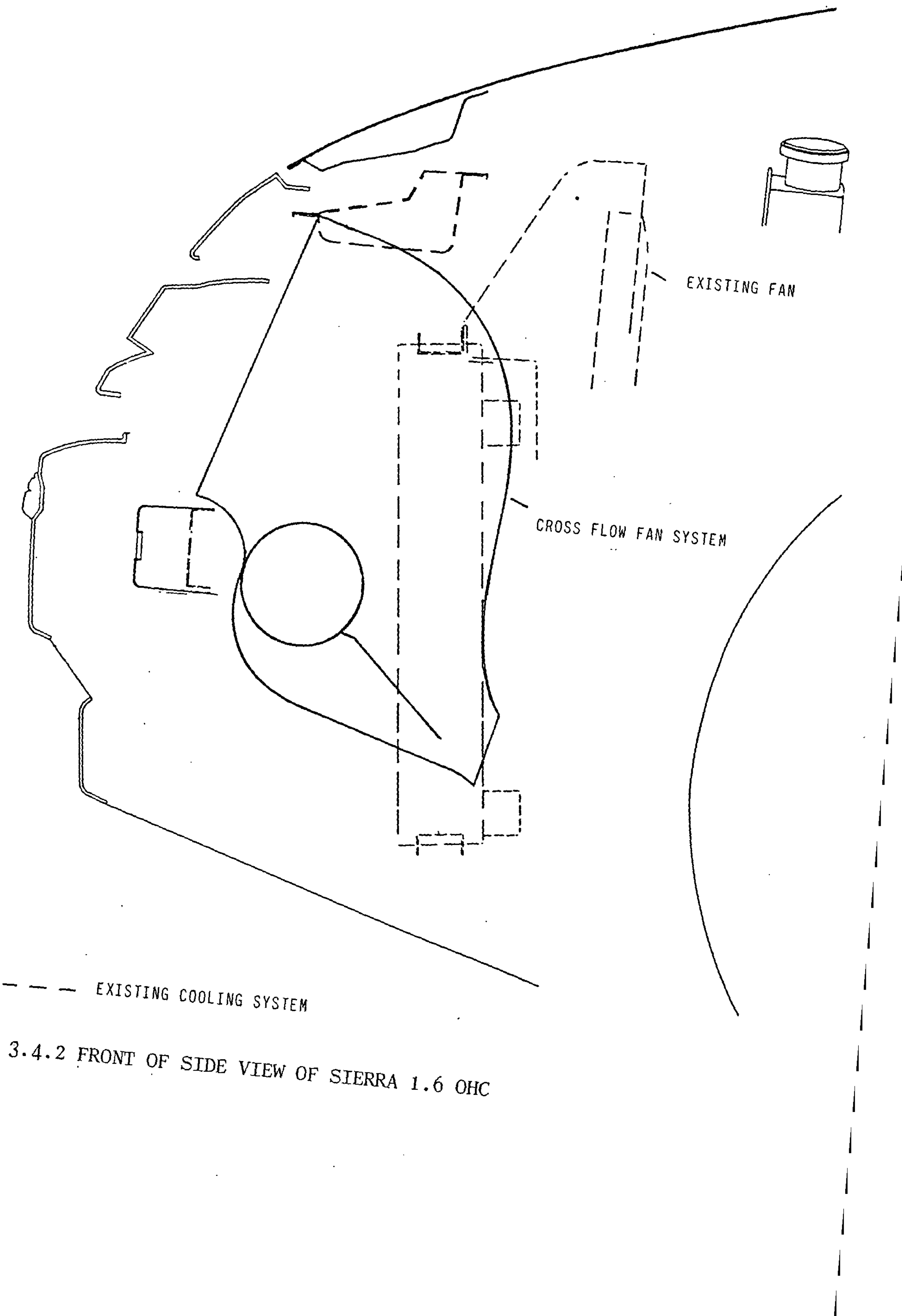


FIGURE 3.4.2 FRONT OF SIDE VIEW OF SIERRA 1.6 OHC

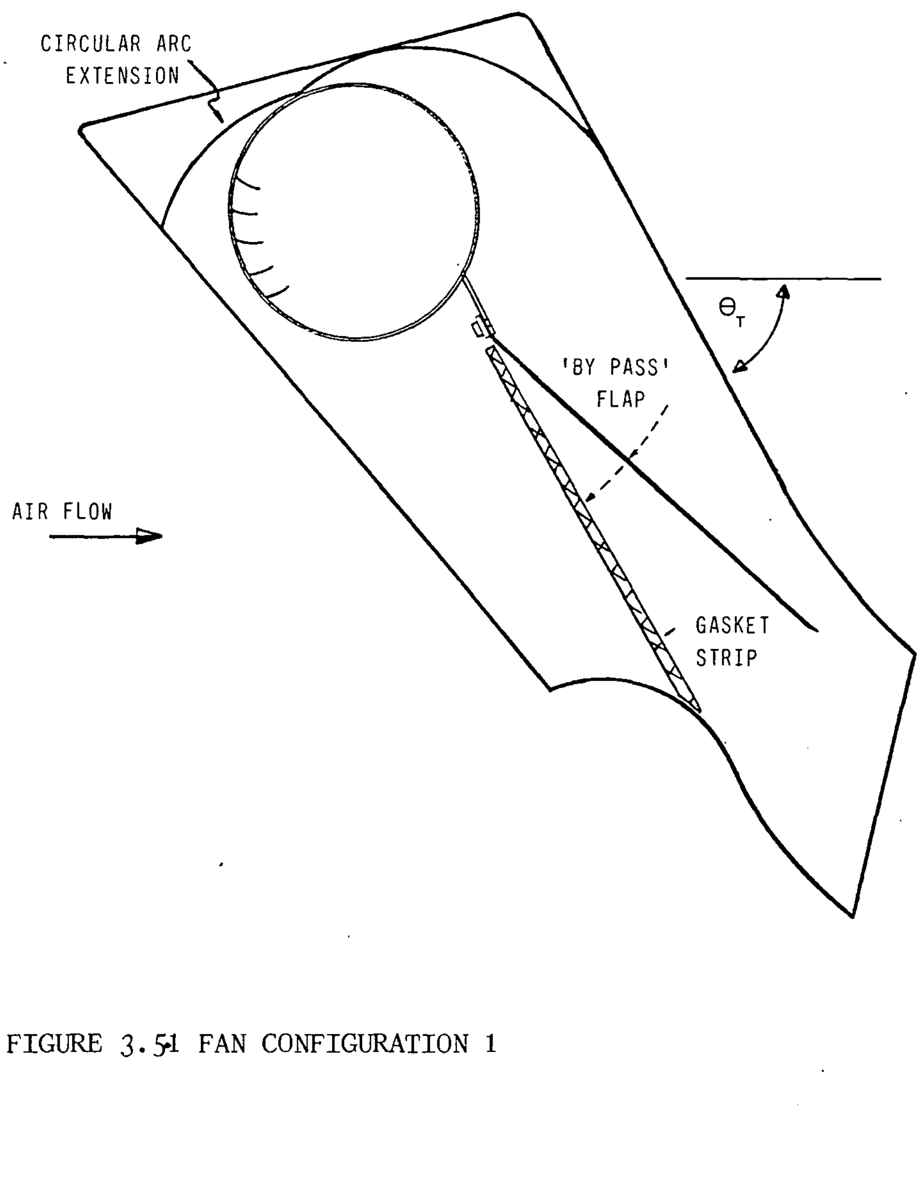


FIGURE 3.51 FAN CONFIGURATION 1

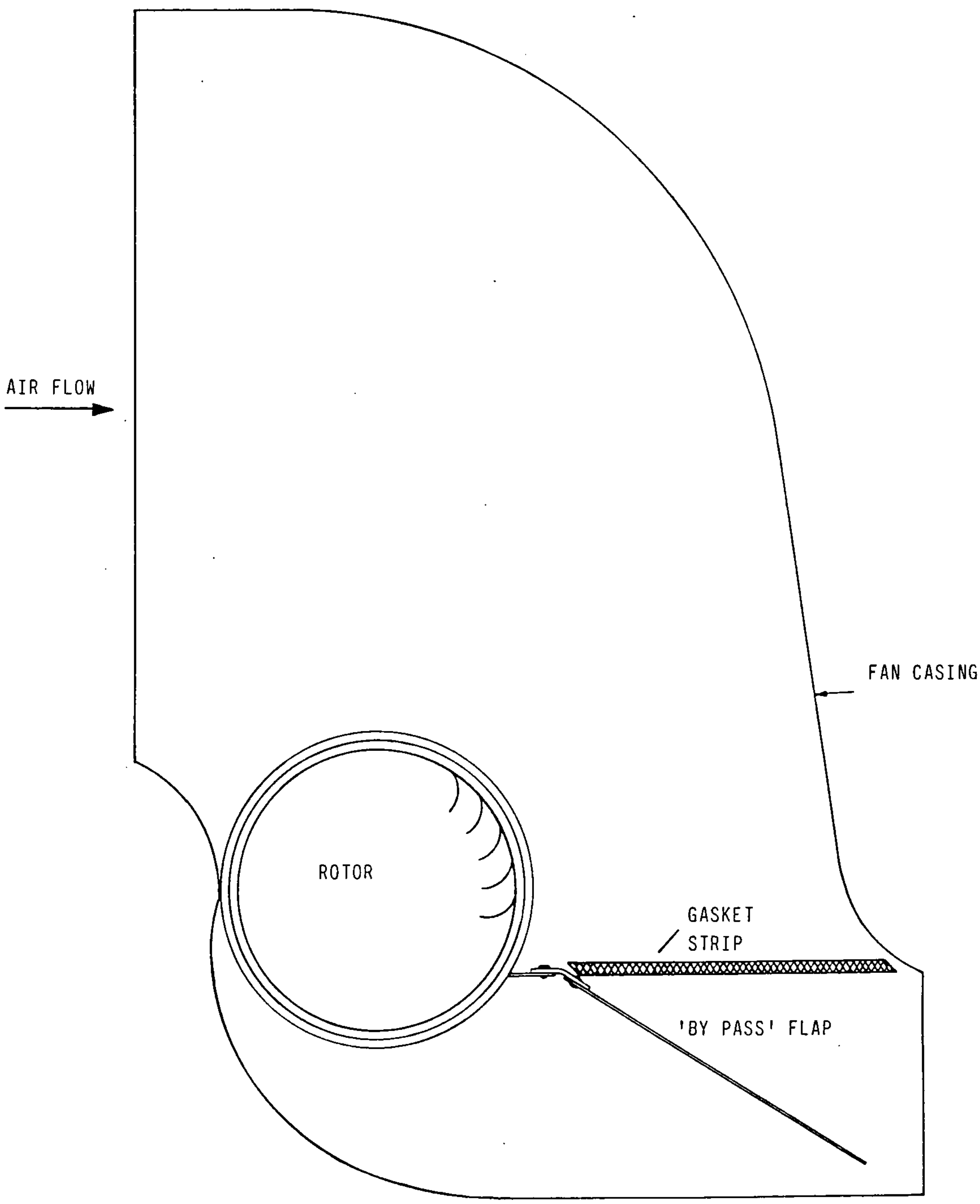
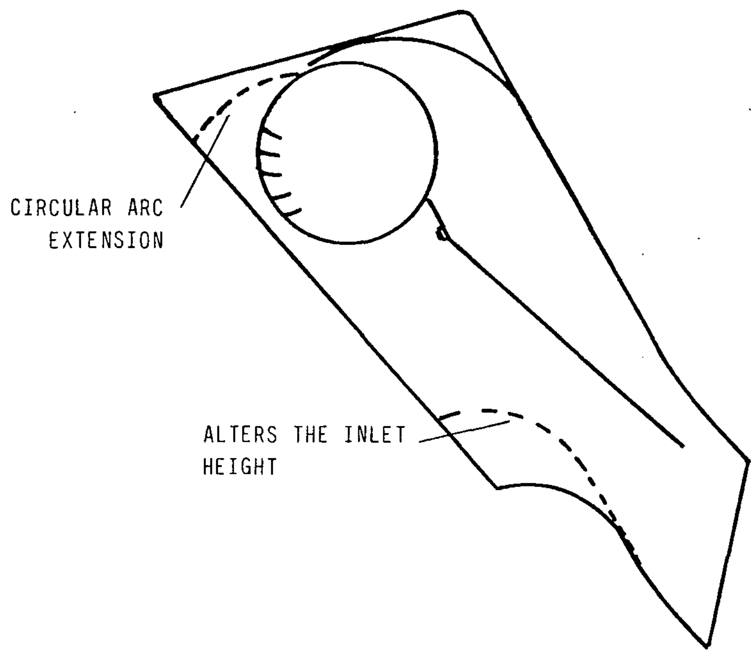
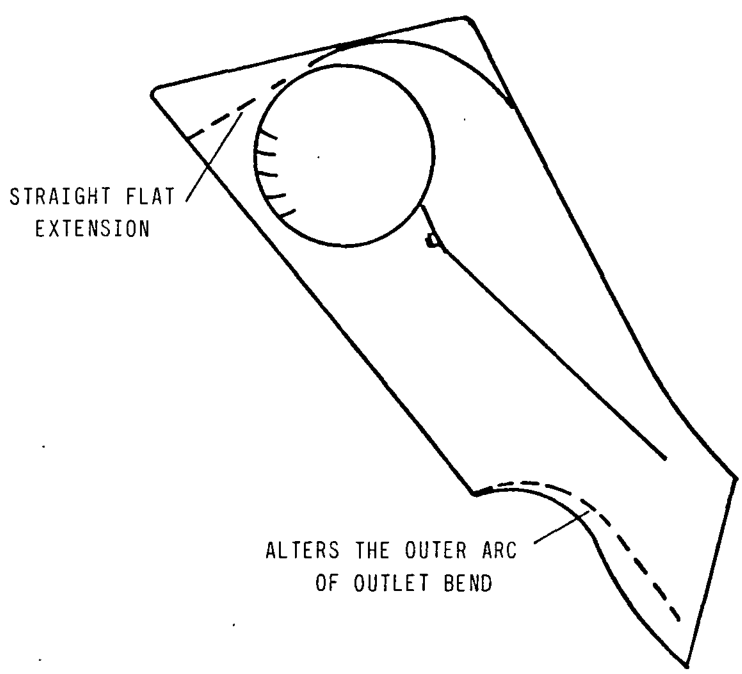


FIGURE 3.5.2 FAN CONFIGURATION 2



3.6.1 INTEGRAL COMPONENTS



3.6.2 INTEGRAL COMPONENTS

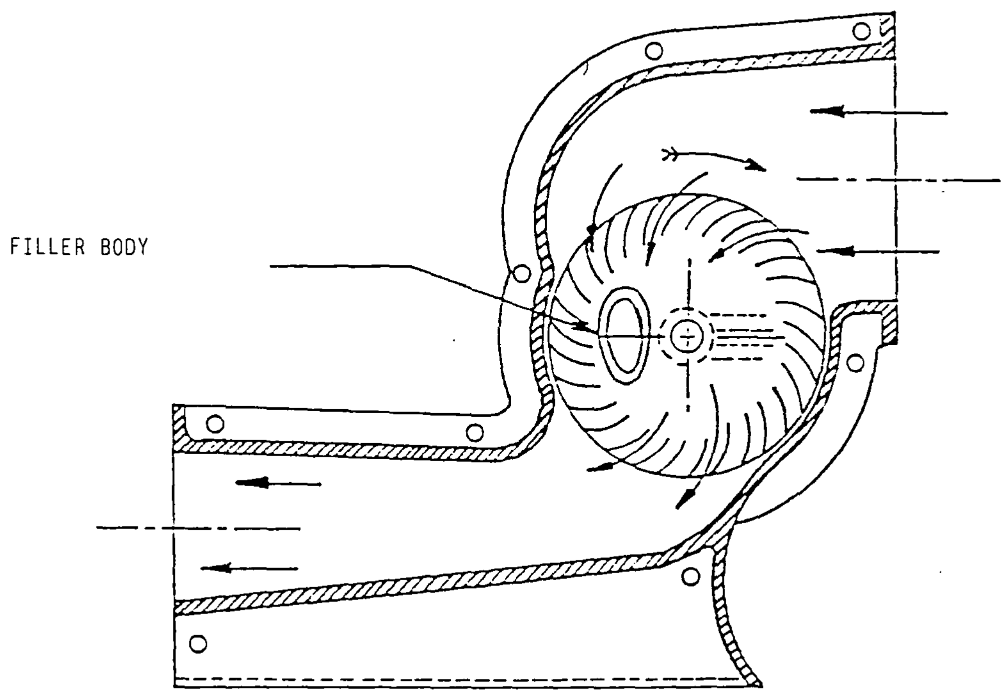


FIGURE 3.7 MORTIER FAN (U.S. PATENT 507, 445, 1893)

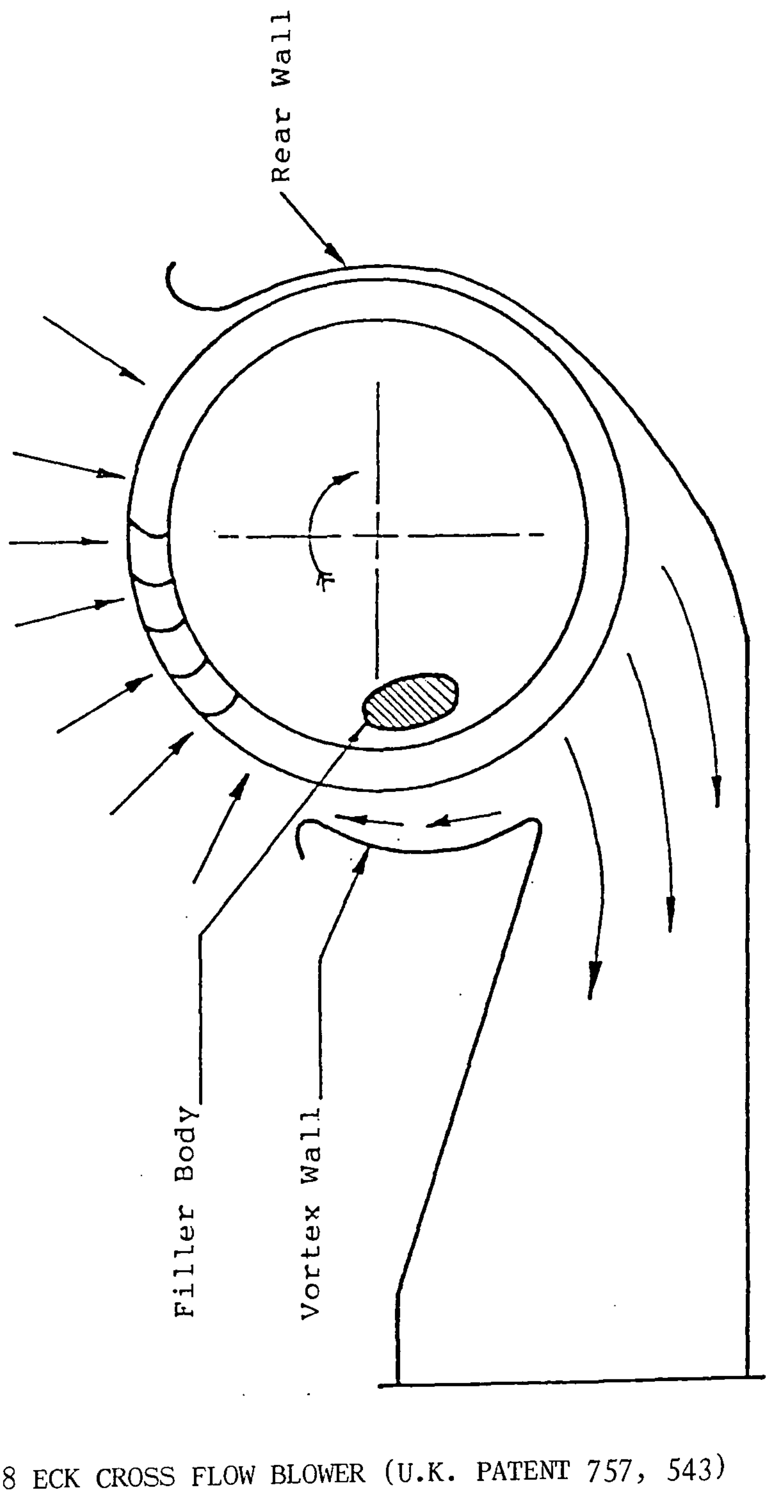


FIGURE 3.8 ECK CROSS FLOW BLOWER (U.K. PATENT 757, 543)

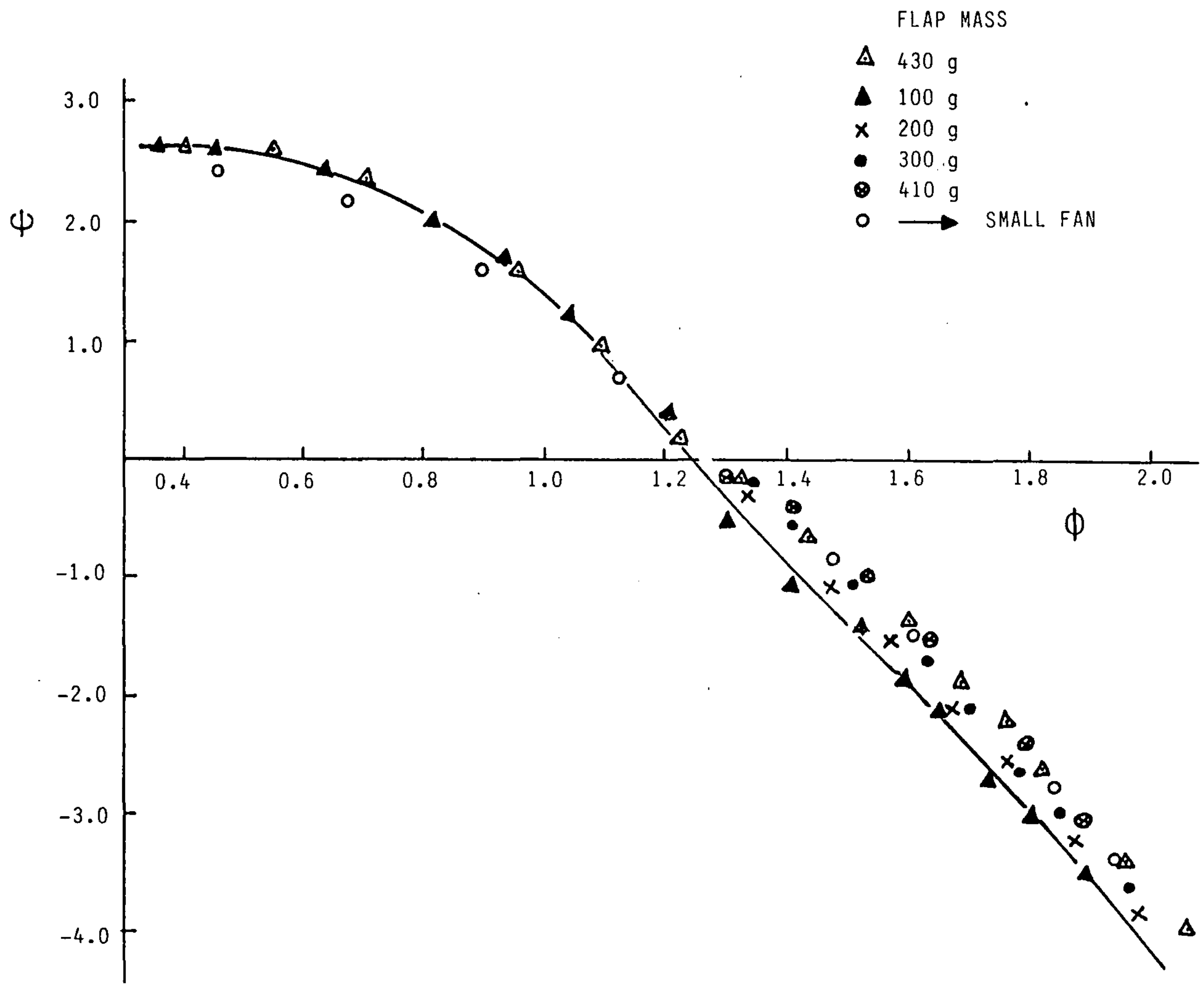


FIGURE 3.9 FAN CONFIGURATION 1

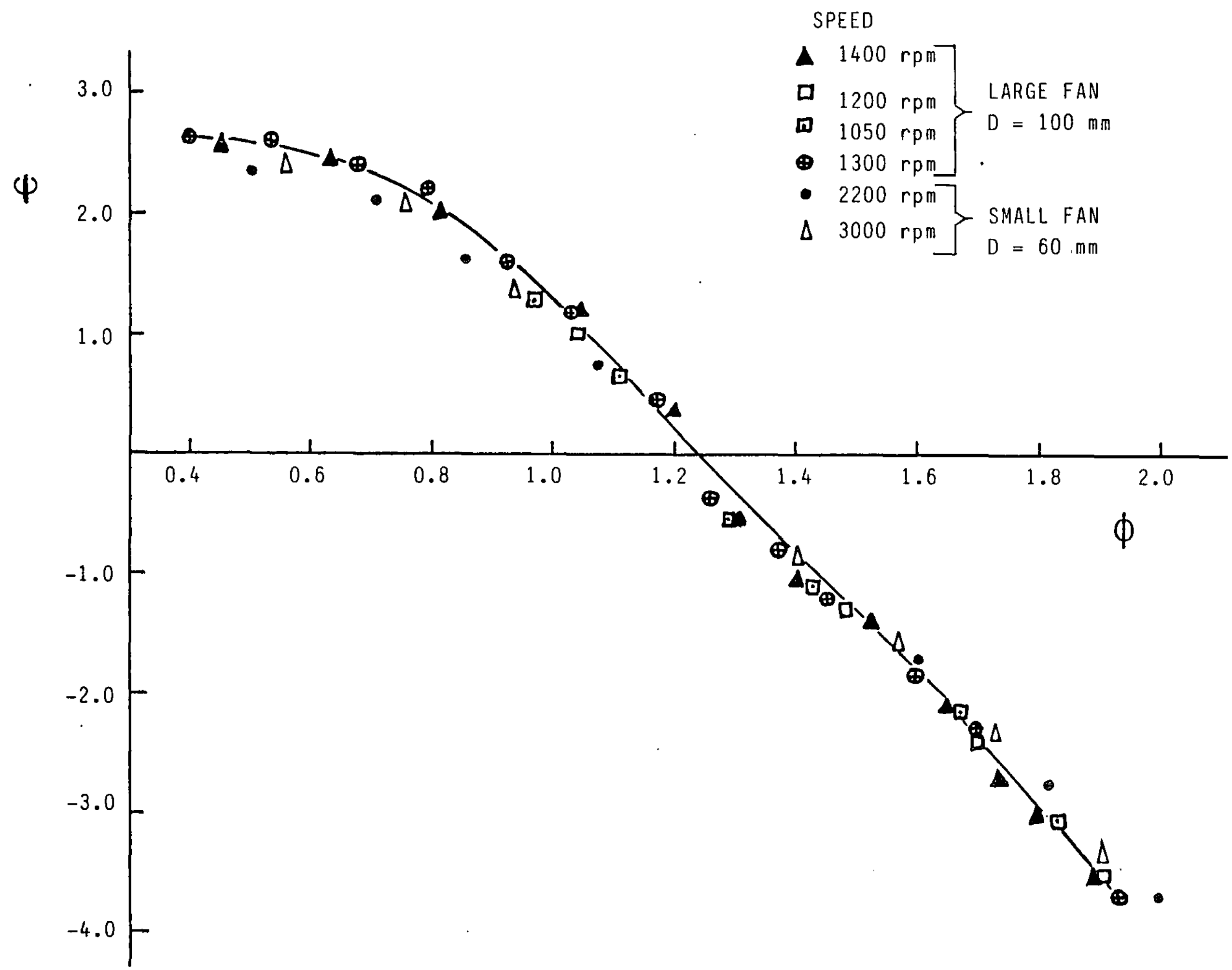


FIGURE 3.10 SCALING OF FAN CONFIGURATION 1

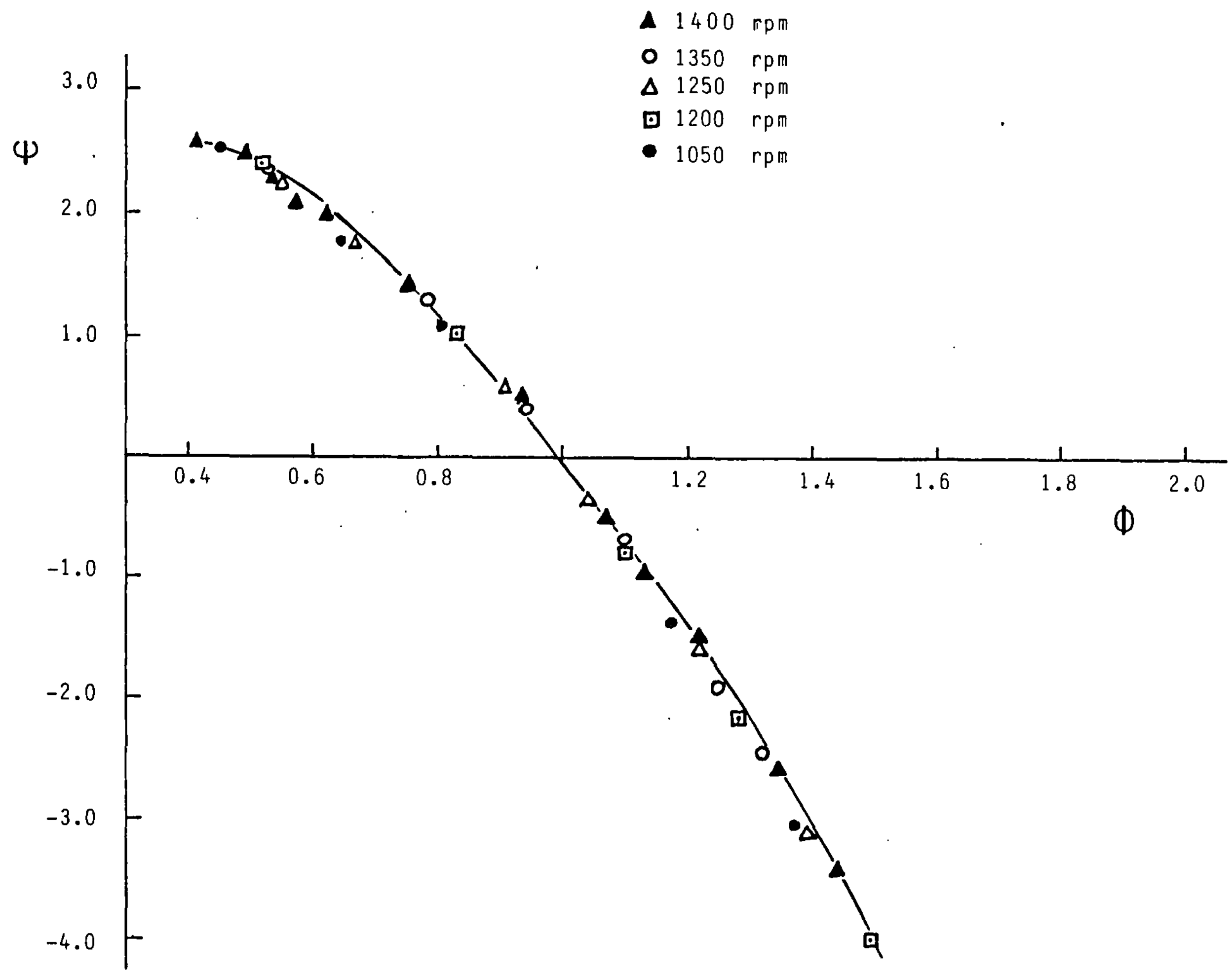


FIGURE 3.11 SCALING OF FAN CONFIGURATION 2

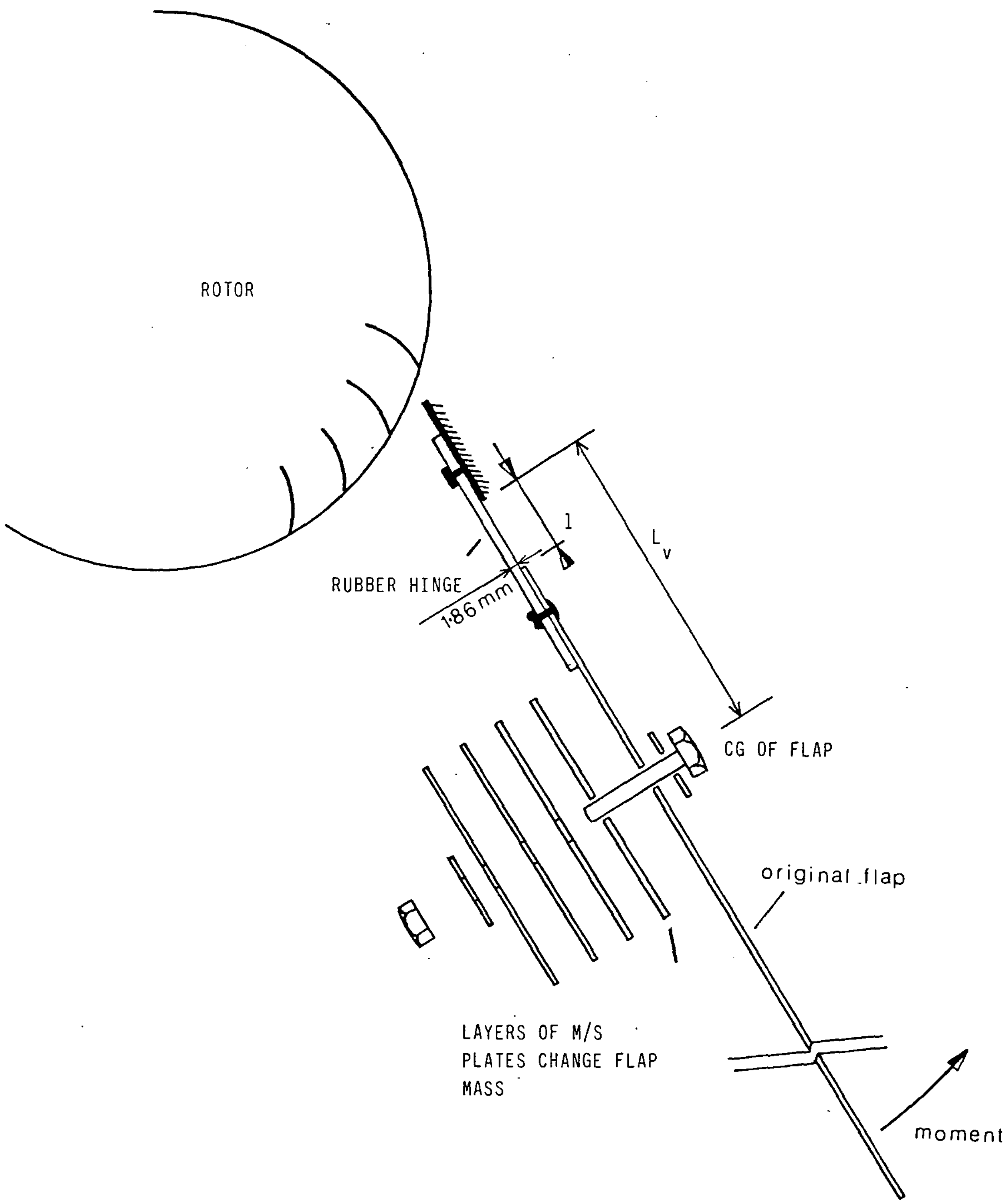


FIGURE 3.12 VORTEX WALL/'BY PASS' FLAP

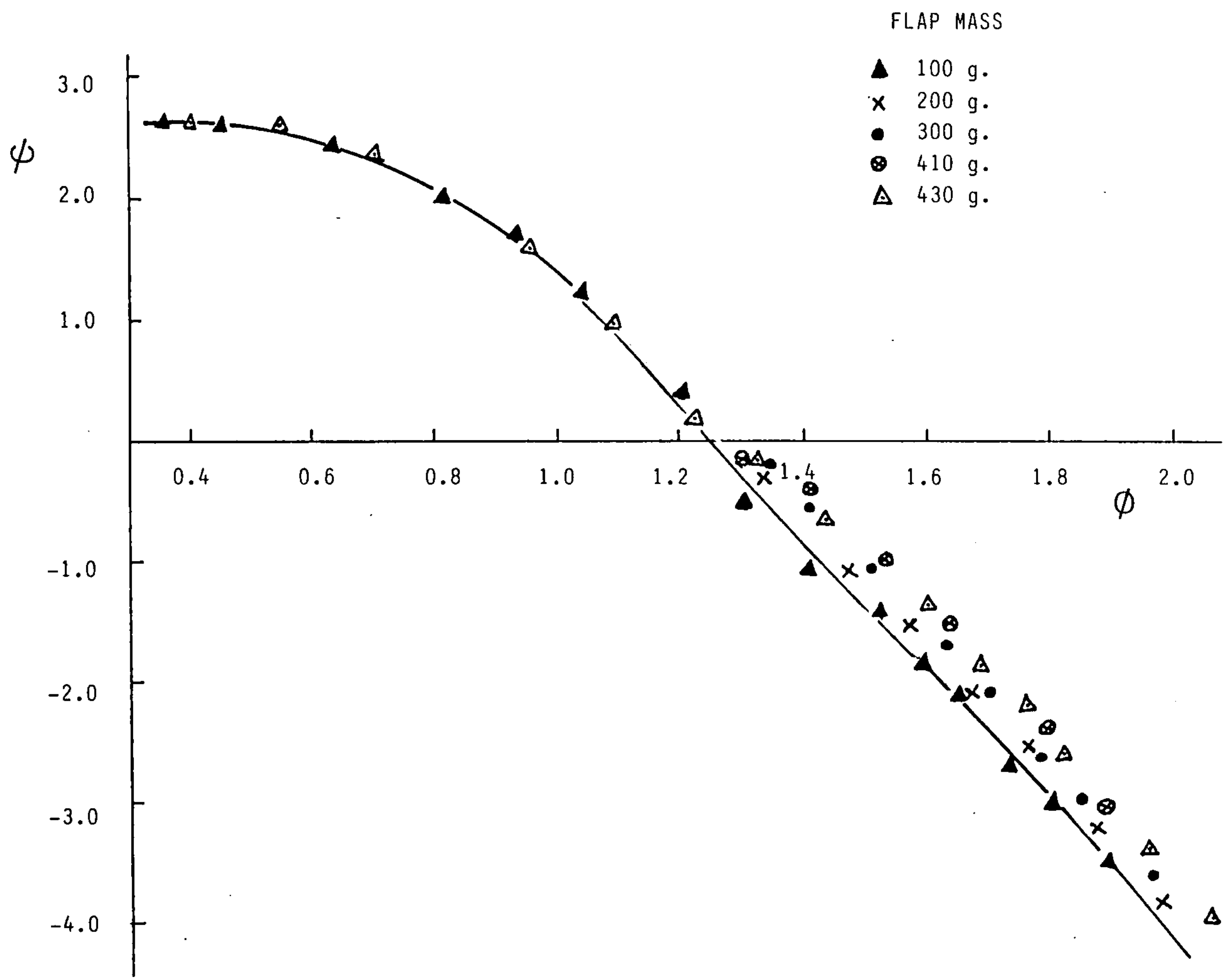


FIGURE 3.13 FAN CONFIGURATION 1 - EFFECT OF FLAP MASS

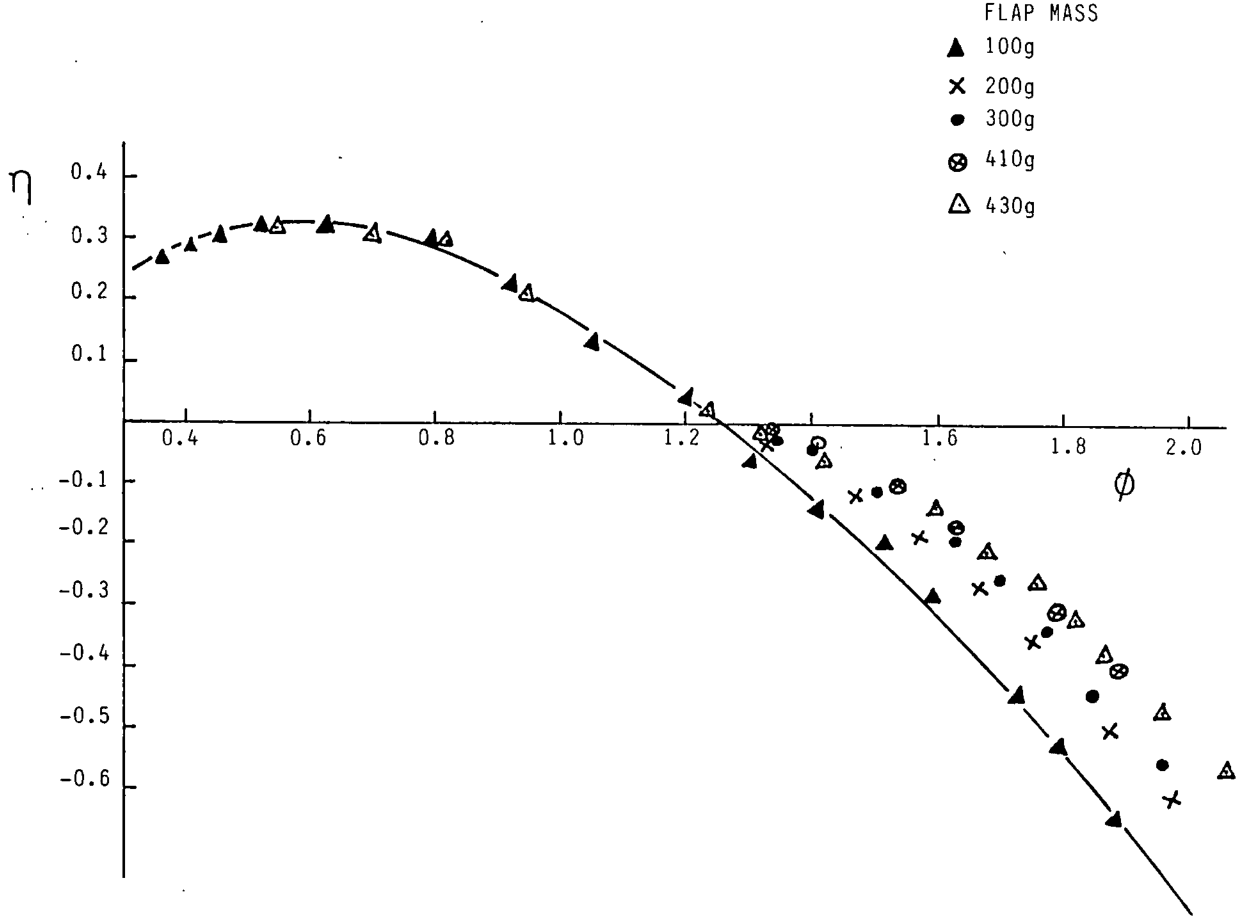


FIGURE 3.14 FAN CONFIGURATION 2 - EFFECT OF FLAP MASS

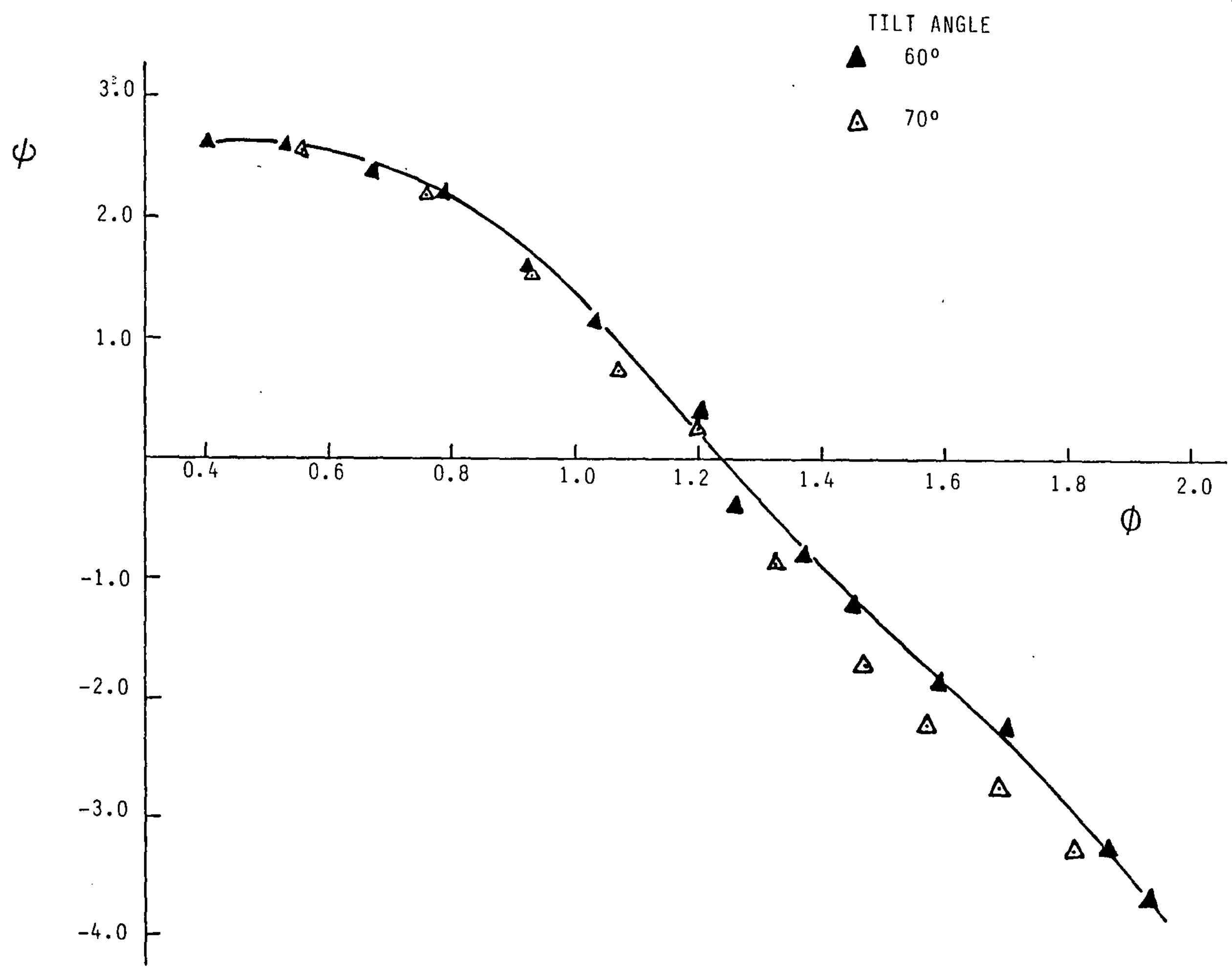


FIGURE 3.15 CONFIGURATION 1 - EFFECT OF UNIT'S TILT ANGLE

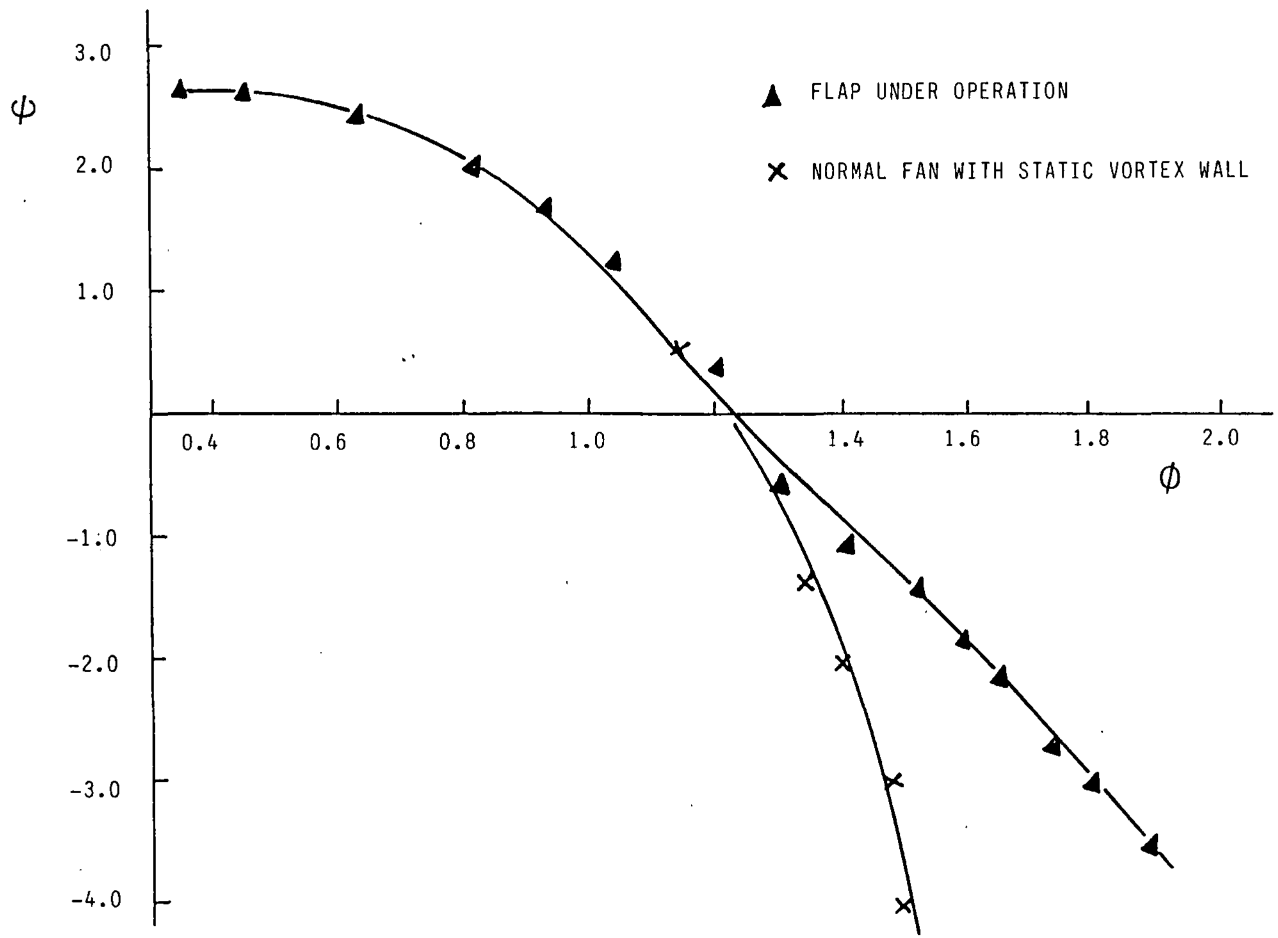


FIGURE 3.16 FAN CONFIGURATION 1 – GAINS DUE TO FLAP

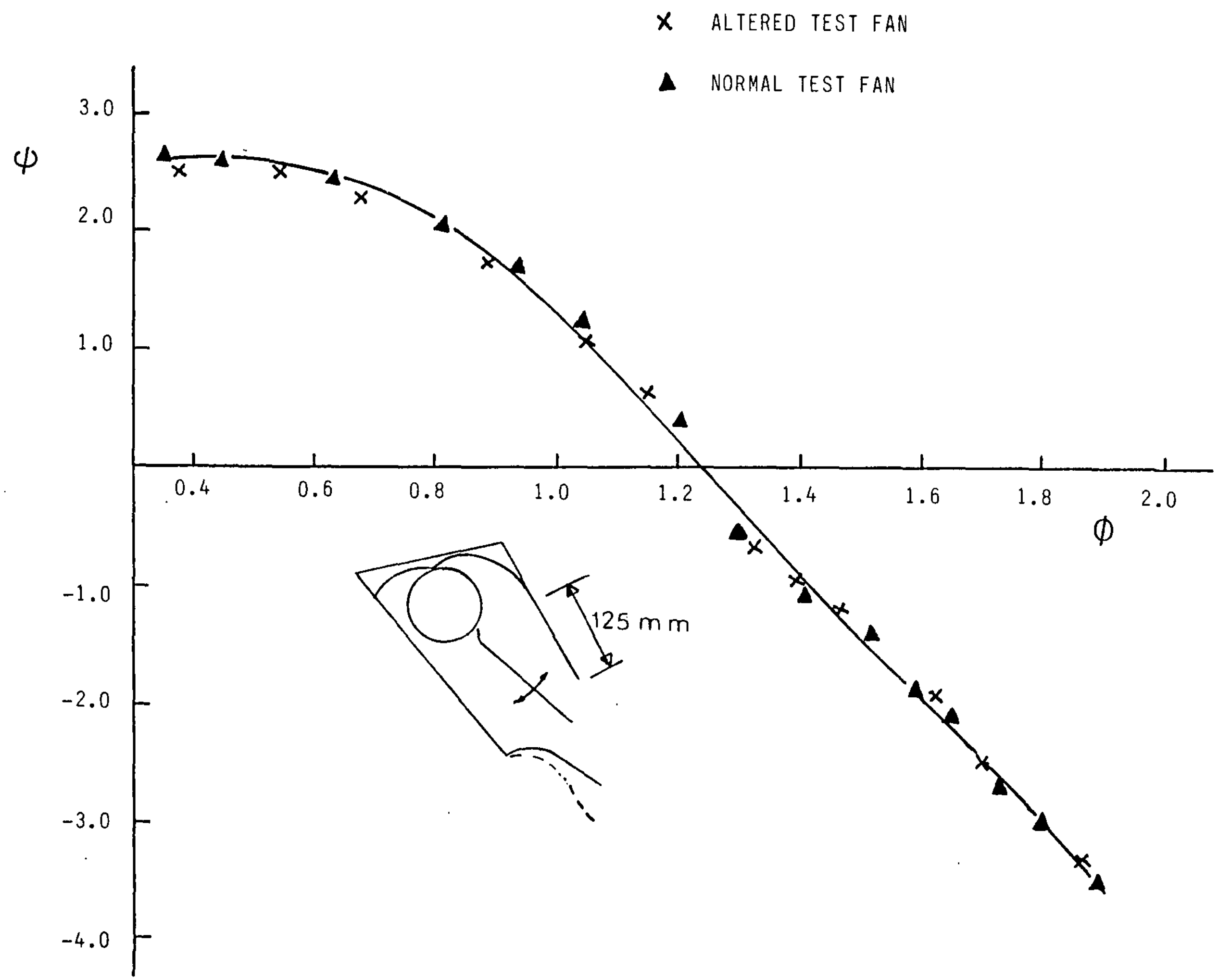


FIGURE 3.17 FAN CONFIGURATION 1

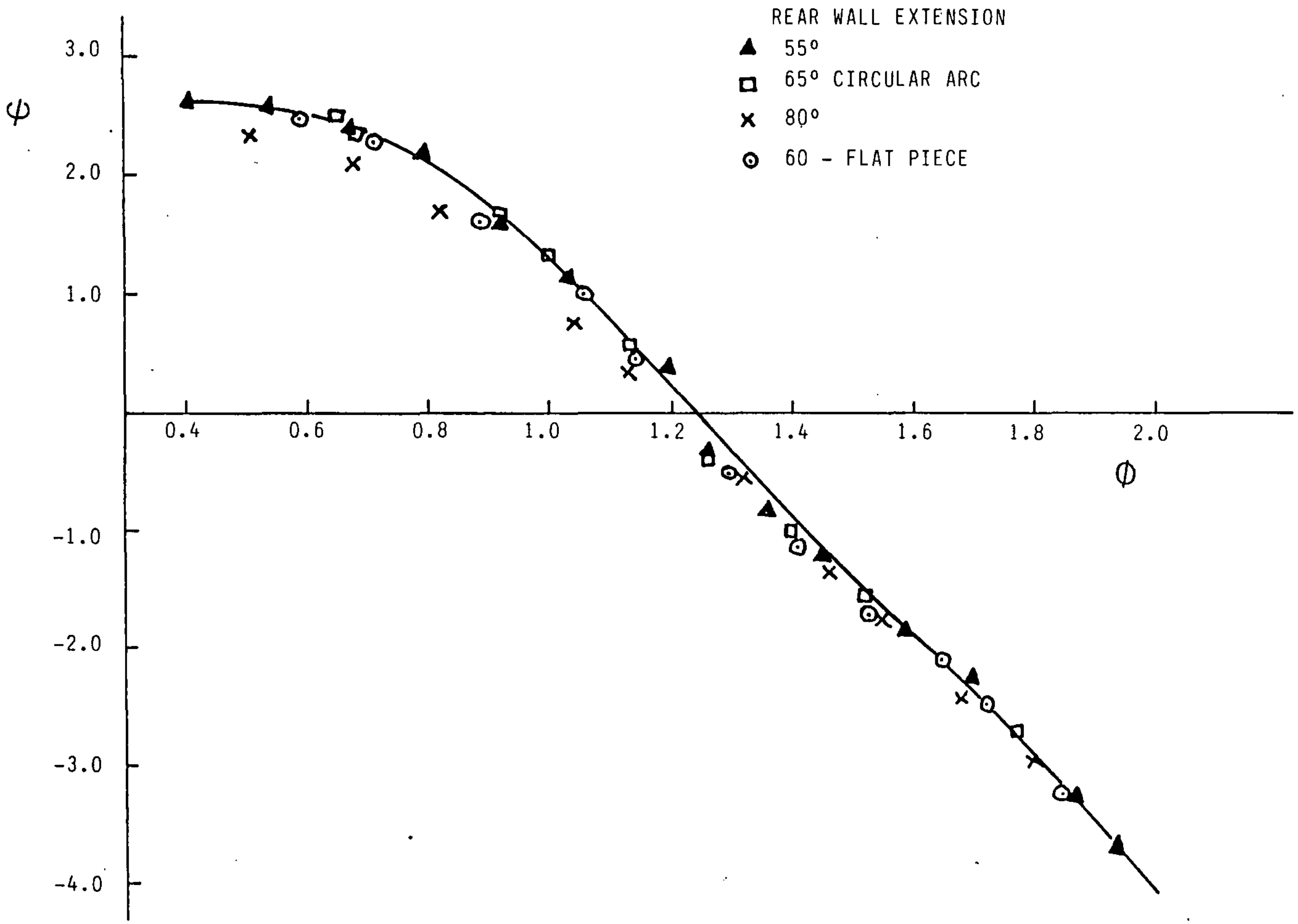


FIGURE 3.18 FAN CONFIGURATION 1 - REAR WALL EXTENSION

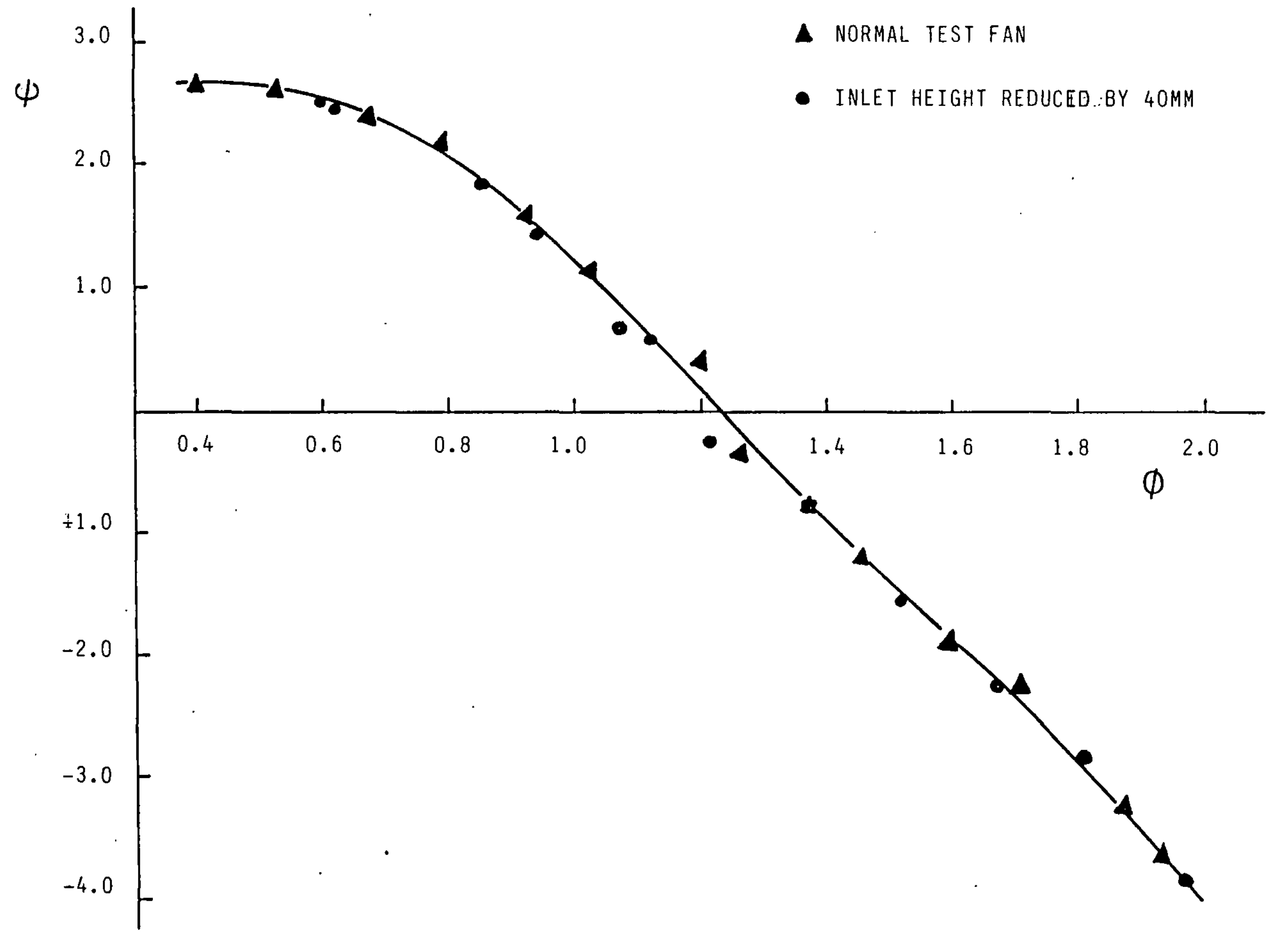


FIGURE 3.19 FAN CONFIGURATION 1 - INLET HEIGHT REDUCED

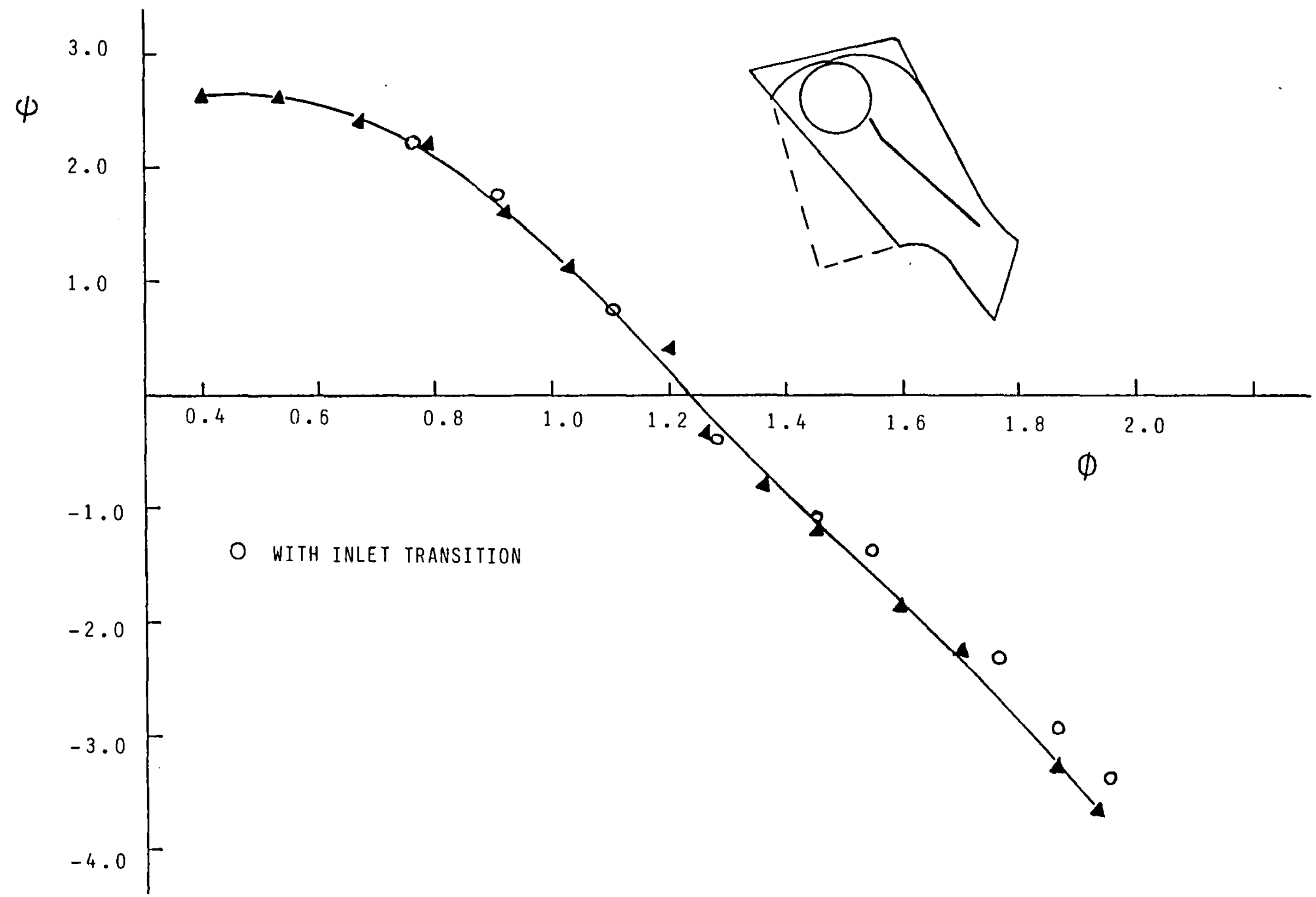


FIGURE 3.20 FAN CONFIGURATION 1 - INLET DUCT TRANSITION

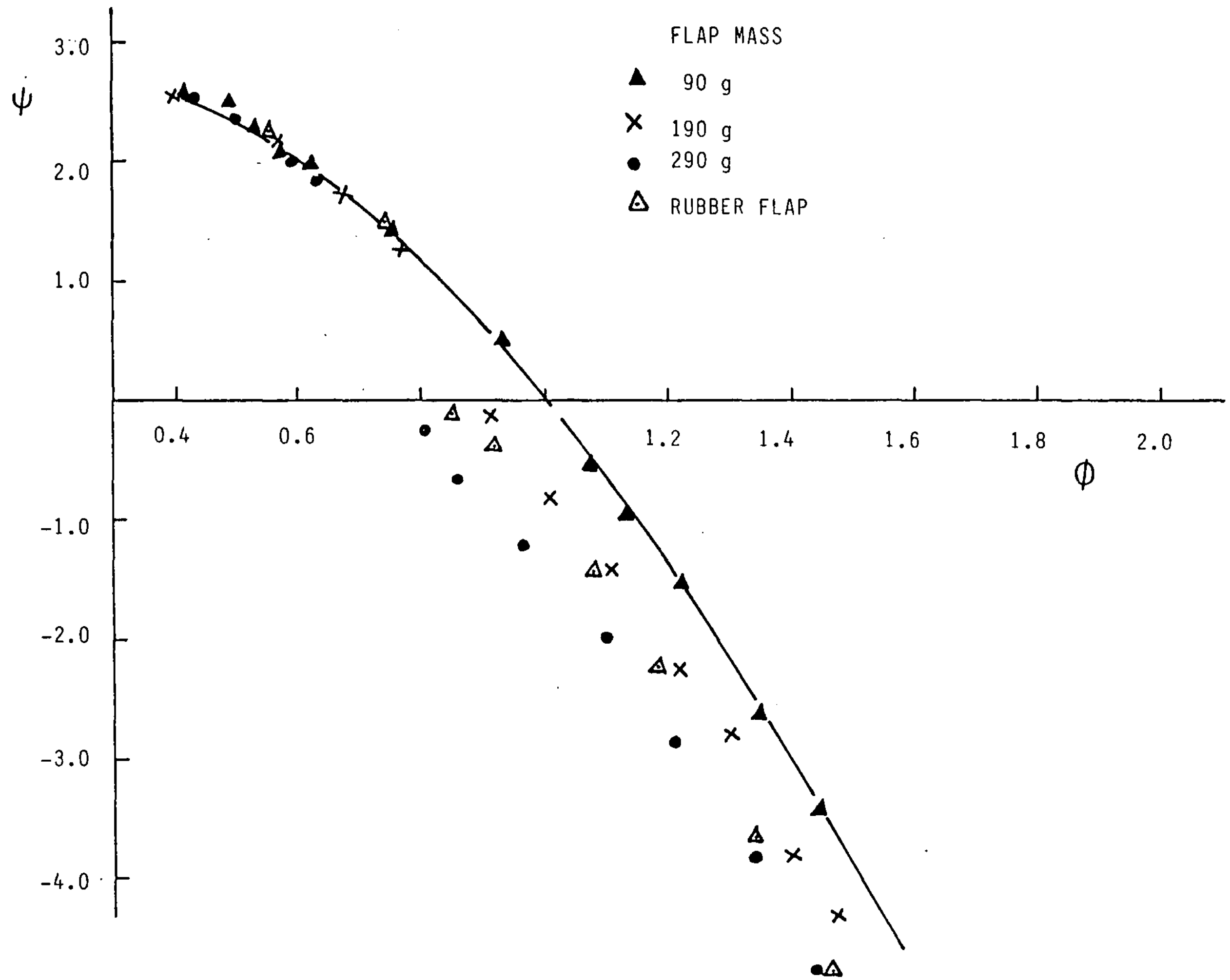


FIGURE 3.21 FAN CONFIGURATION 2-EFFECT OF FLAP MASS

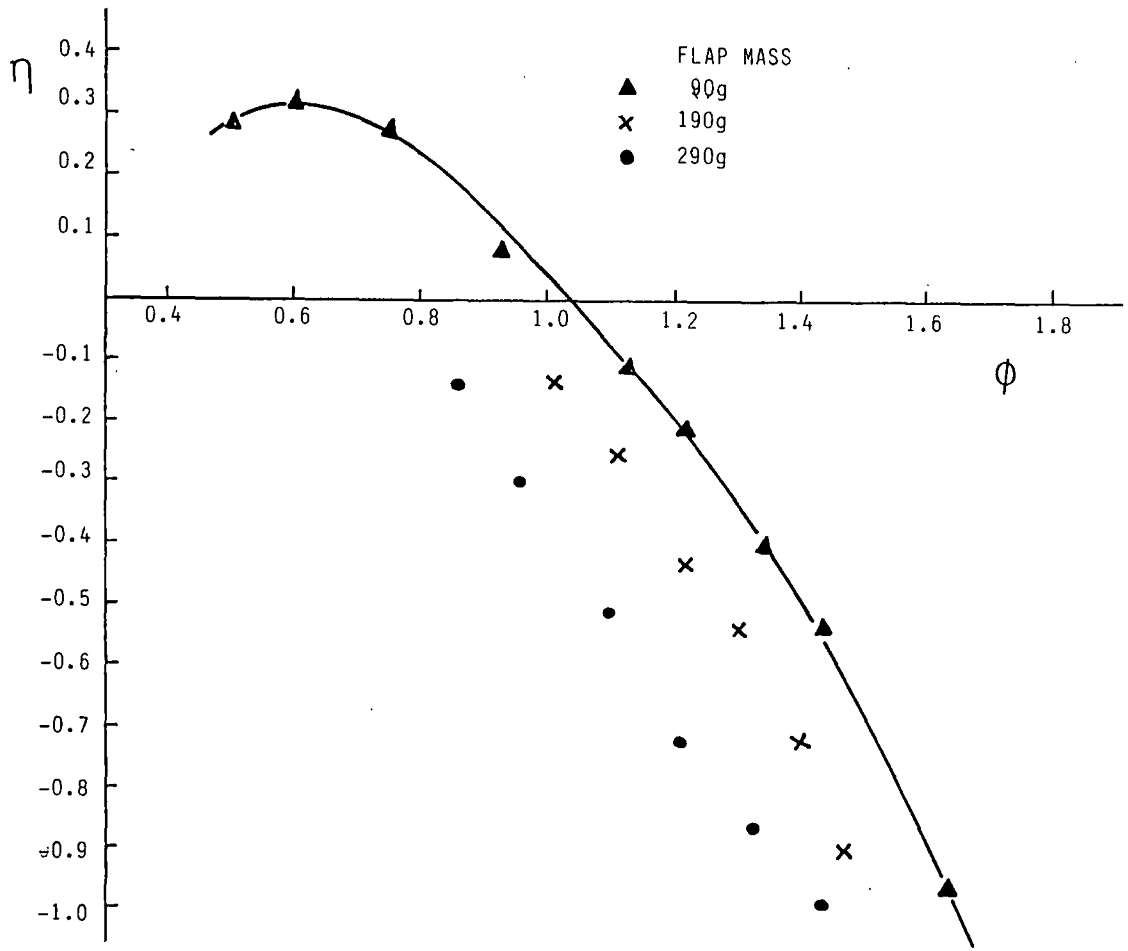


FIGURE 3.22 FAN CONFIGURATION 2 - EFFECT OF FLAP MASS

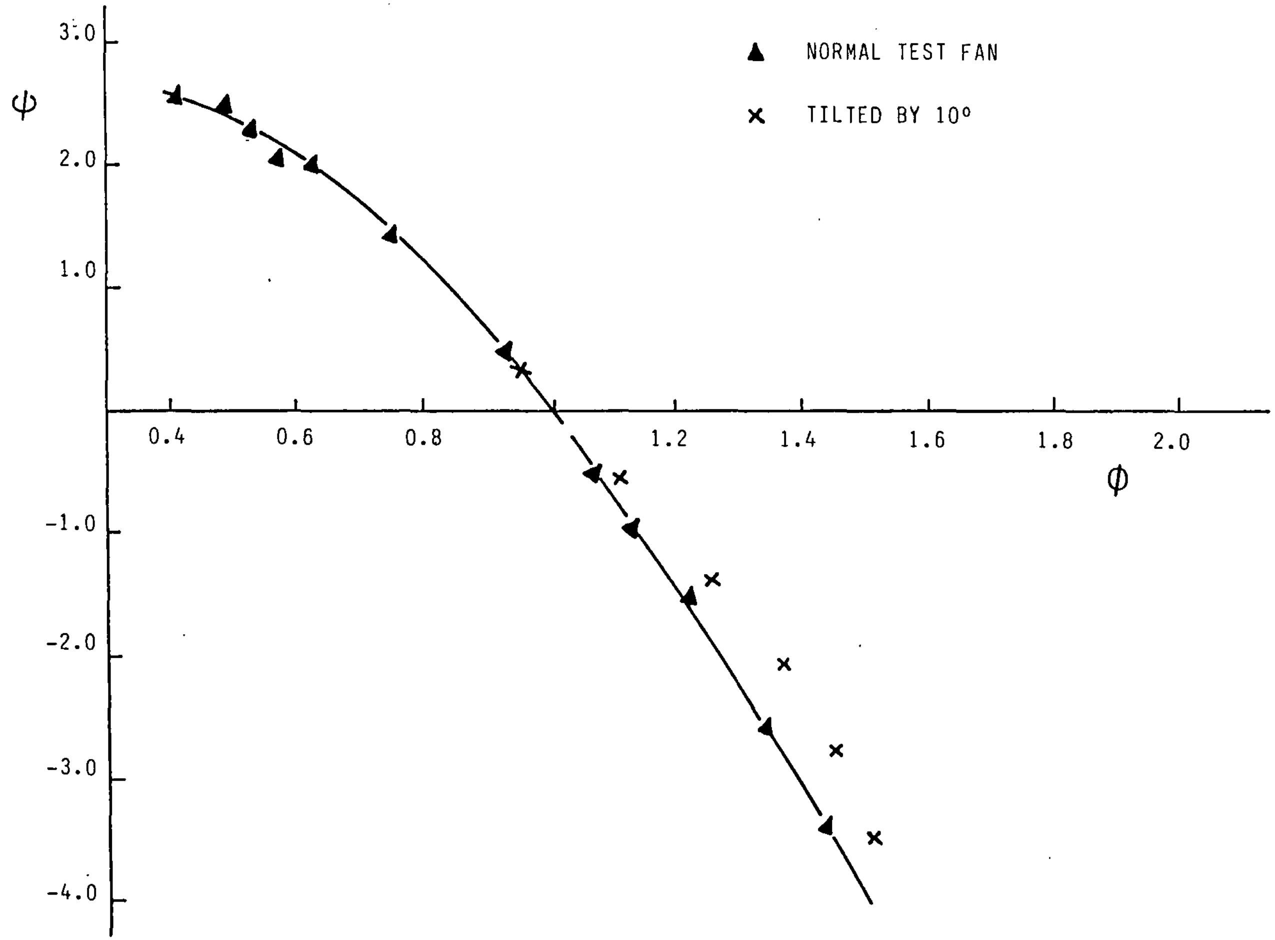


FIGURE 3.23 FAN CONFIGURATION 2 - EFFECT OF TILTED UNIT

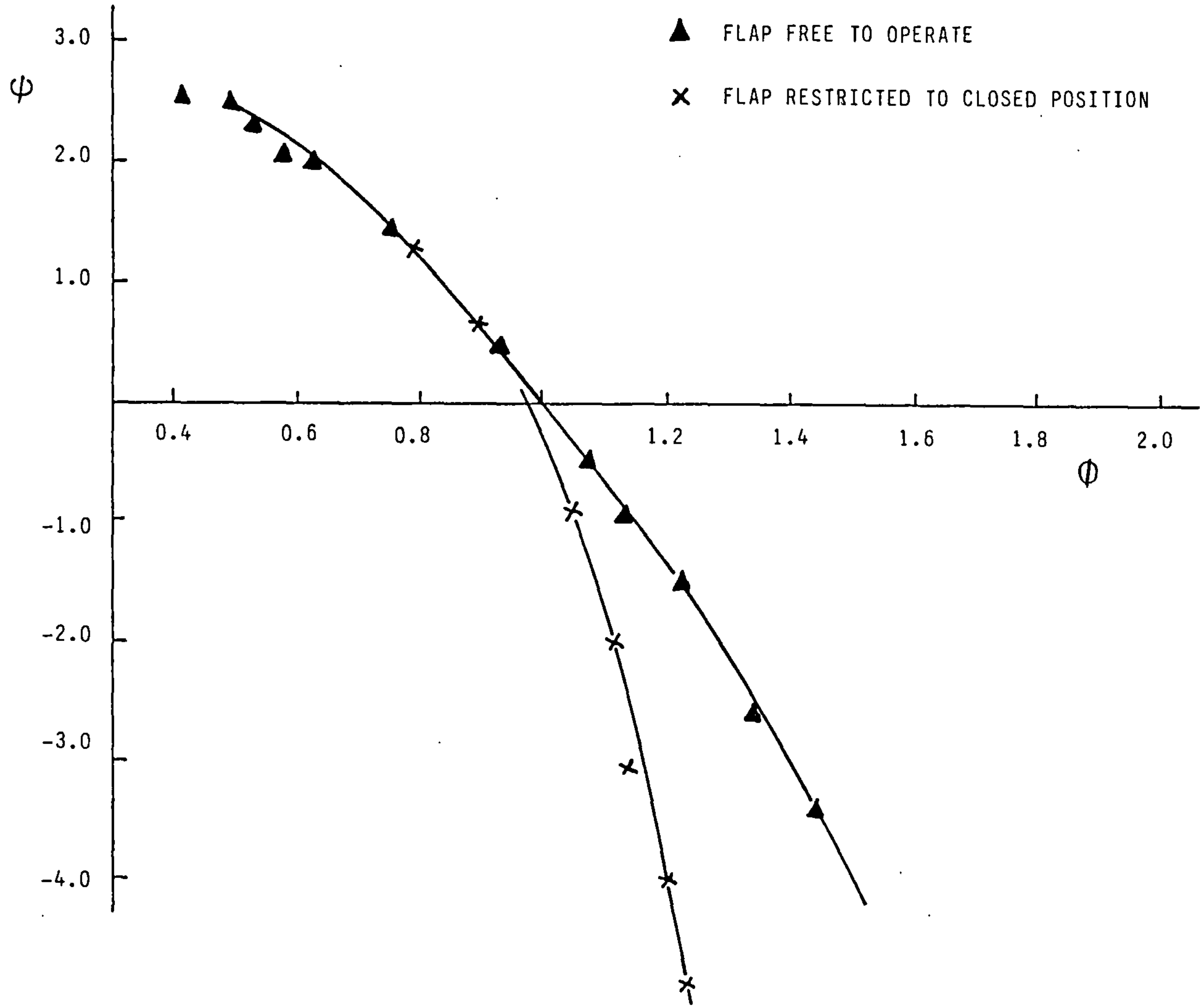


FIGURE 3.24 FAN CONFIGURATION 2 - GAINS DUE TO FLAP OPERATION

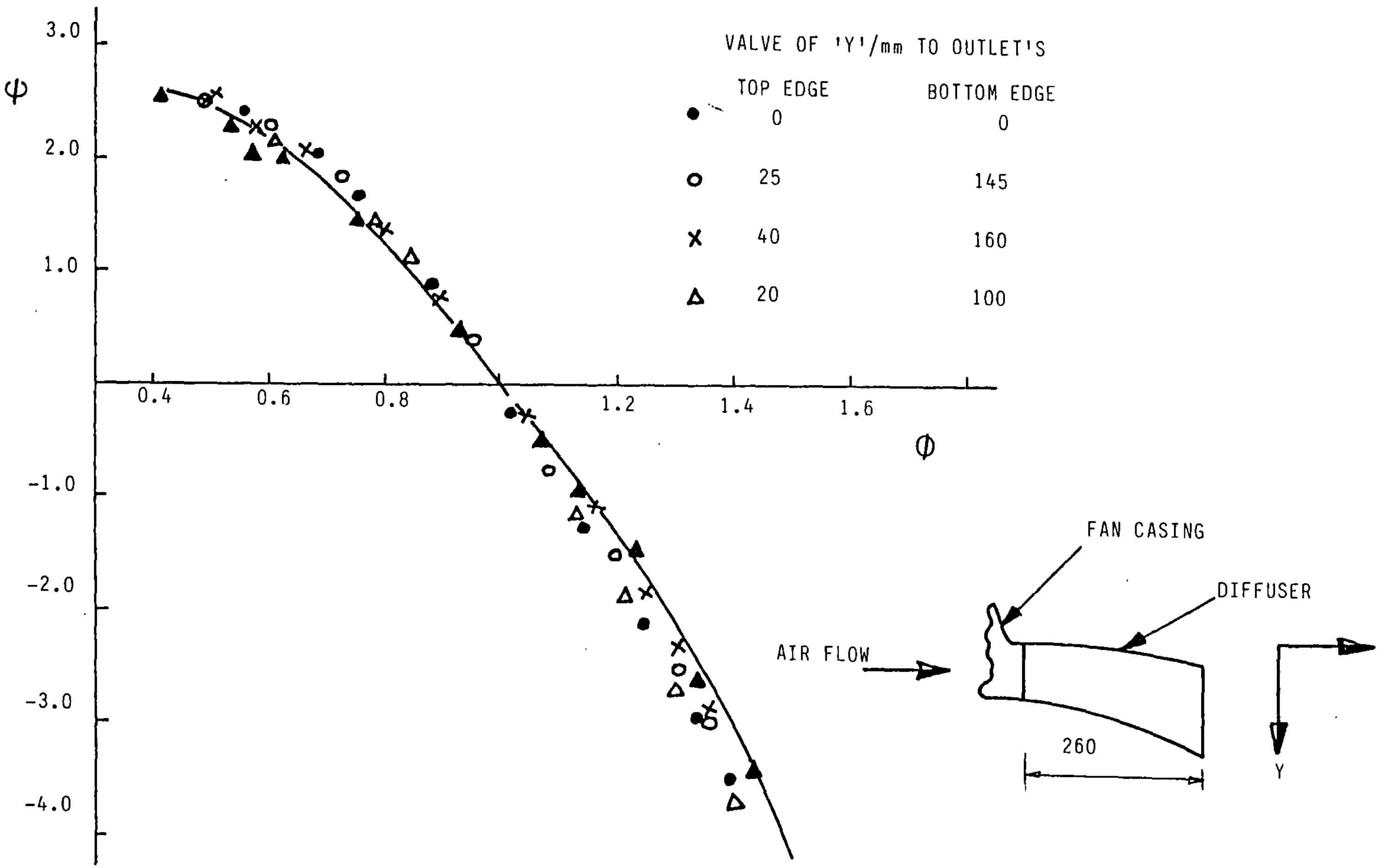


FIGURE 3.25 FAN CONFIGURATION 2 - OUTLET DIFFUSER

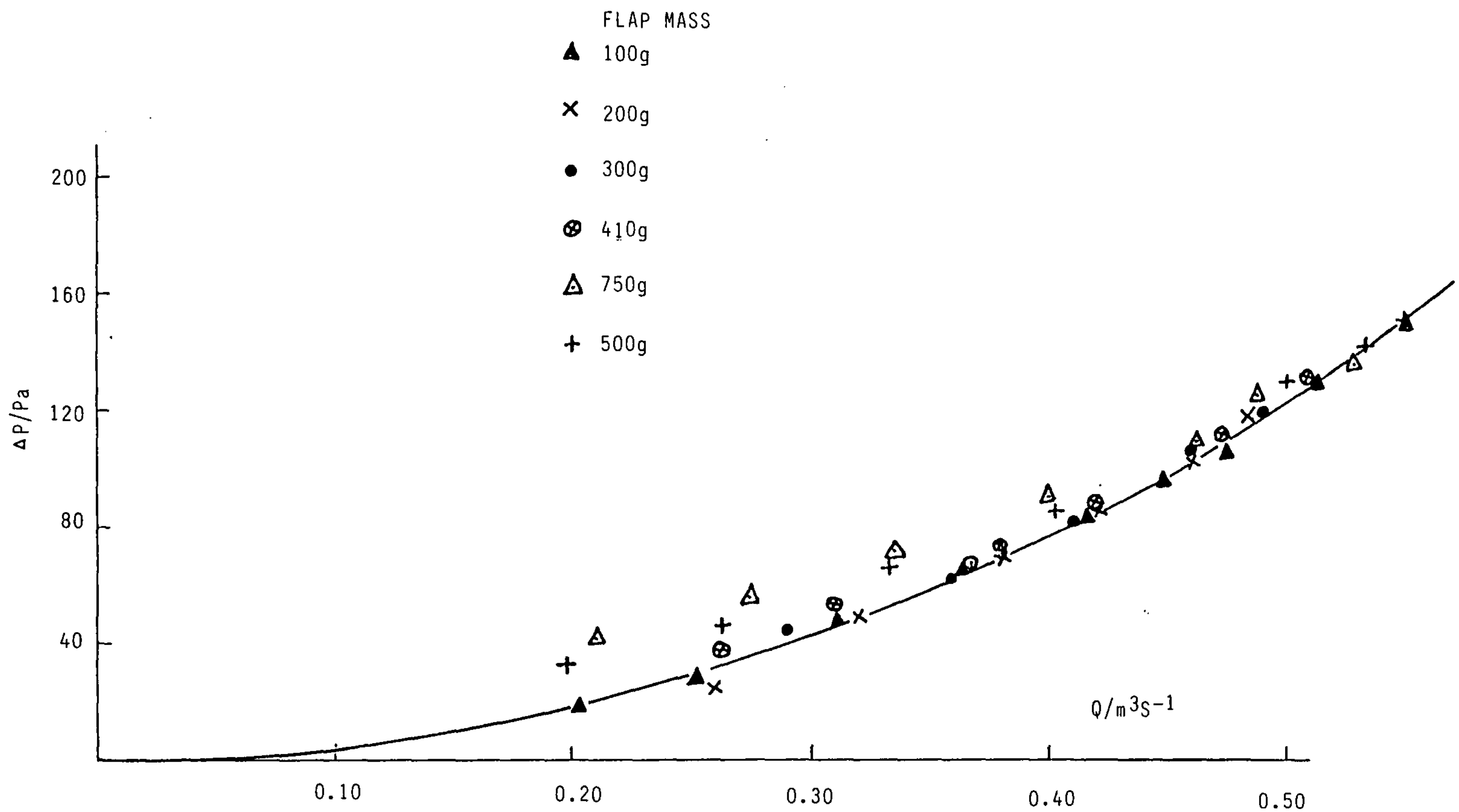


FIGURE 3.26 FAN CONFURATION 1 - EFFECT OF FLAP MASS

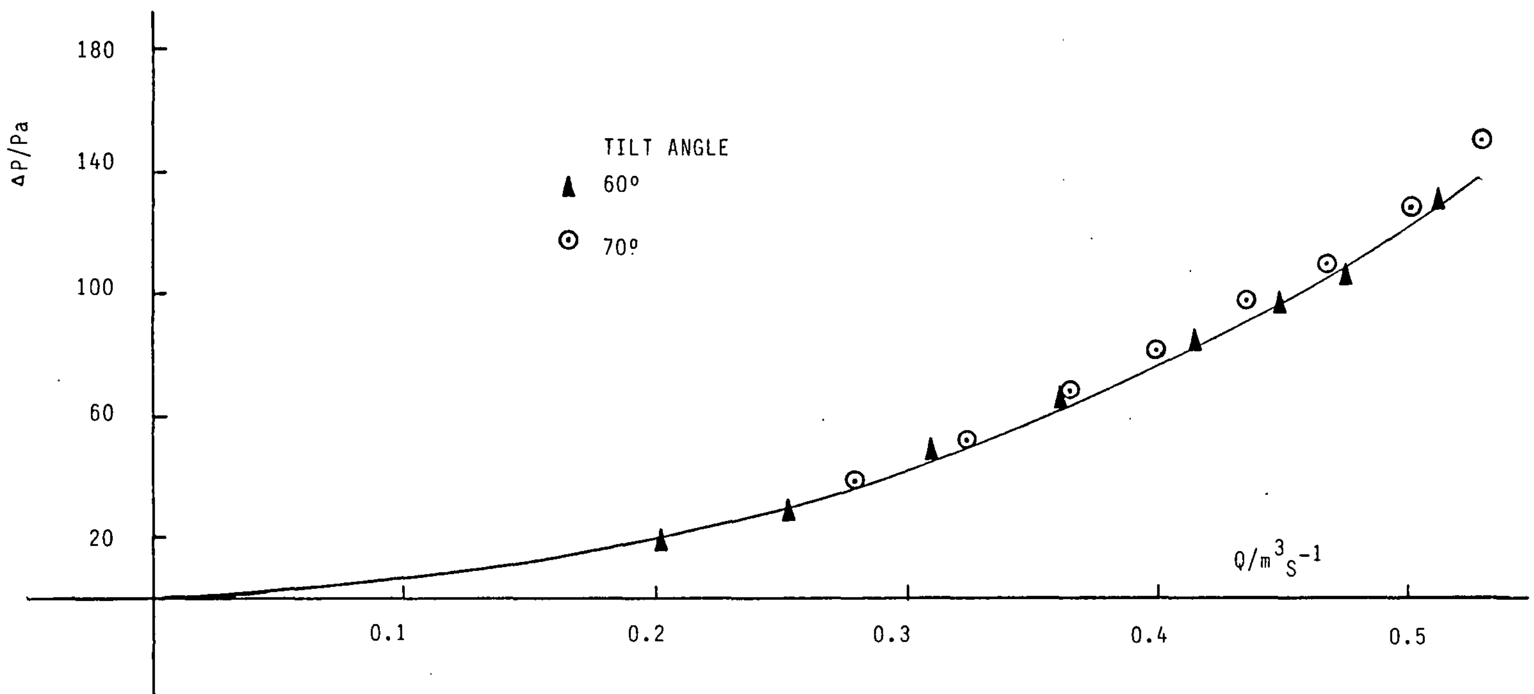


FIGURE 3.27 FAN CONFIGURATION 1 - EFFECT OF UNIT'S TILT ANGLE

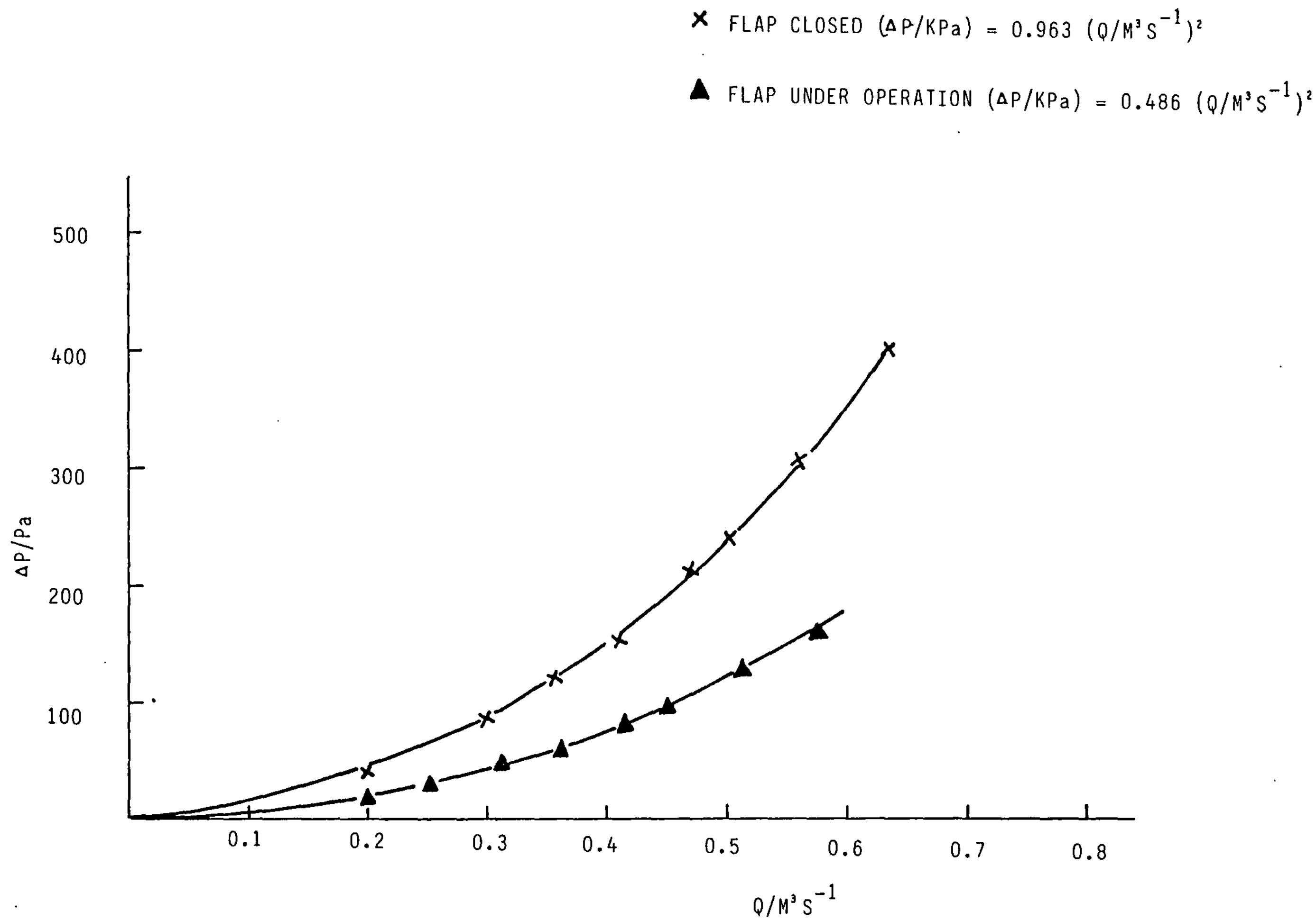


FIGURE 3.28 FAN CONFIGURATION 1 – GAINS DUE TO FLAP OPERATION

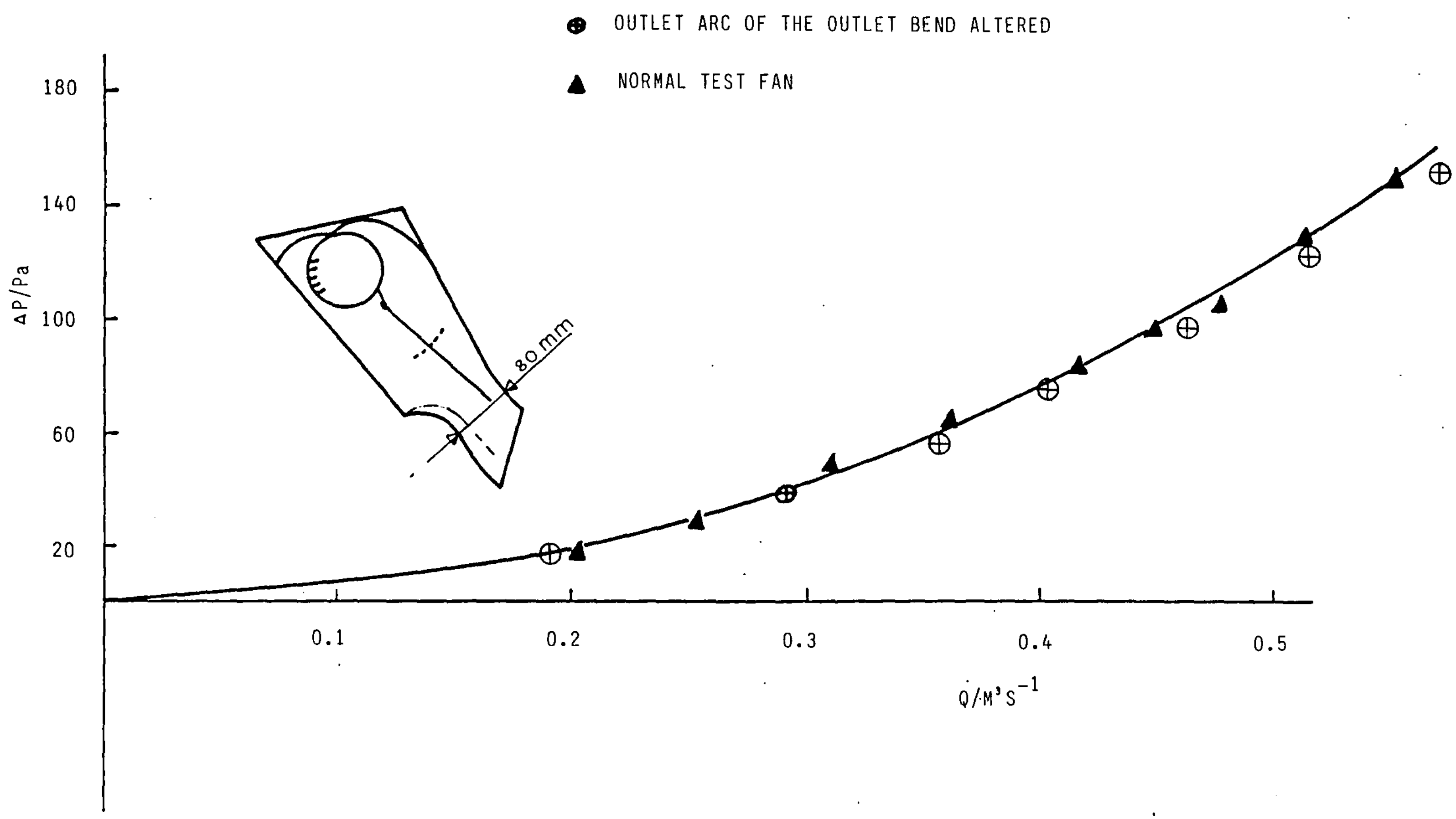


FIGURE 3.29 FAN CONFIGURATION 1

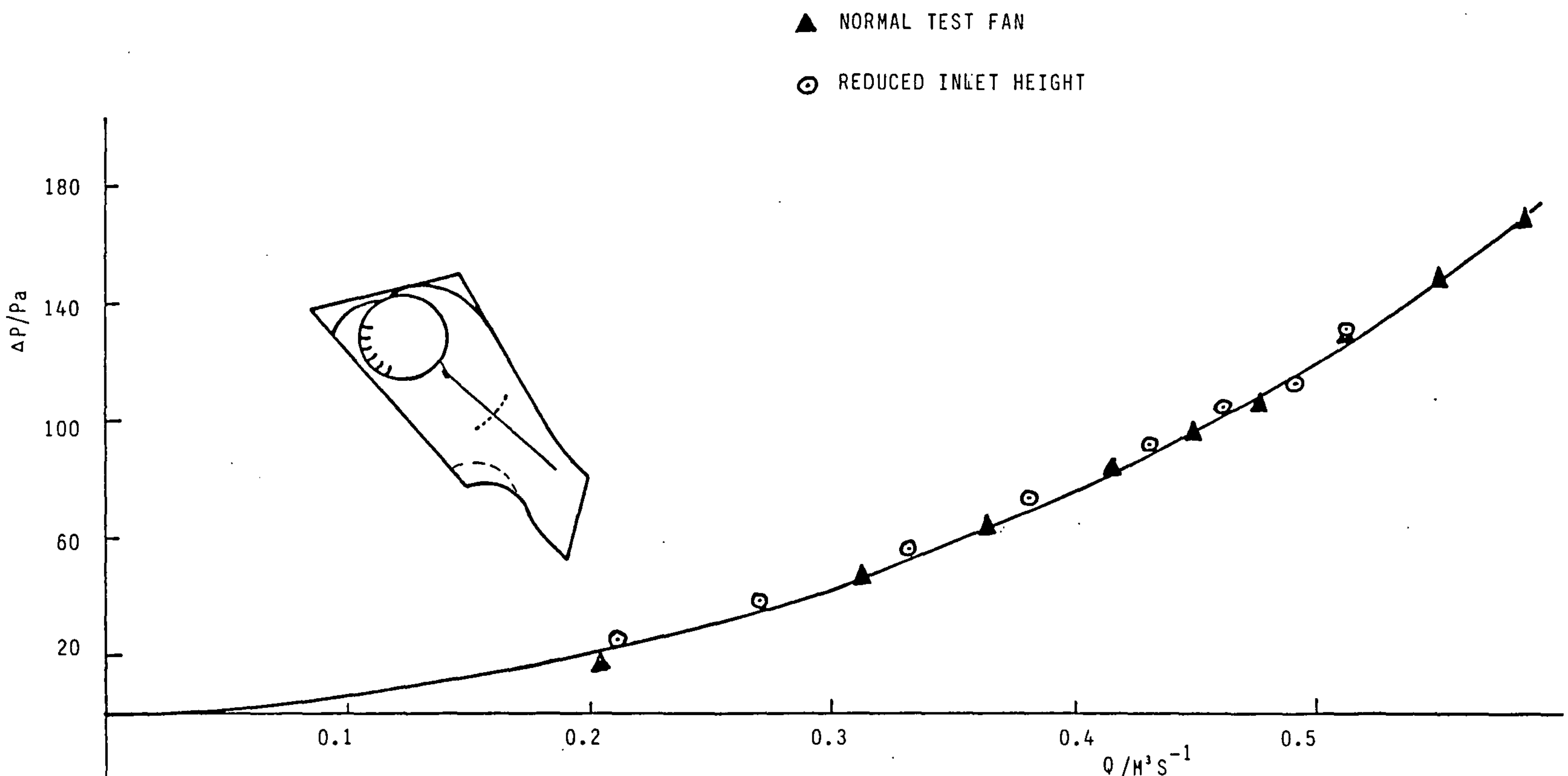


FIGURE 3.30 FAN CONFIGURATION 1-REDUCED INLET HEIGHT

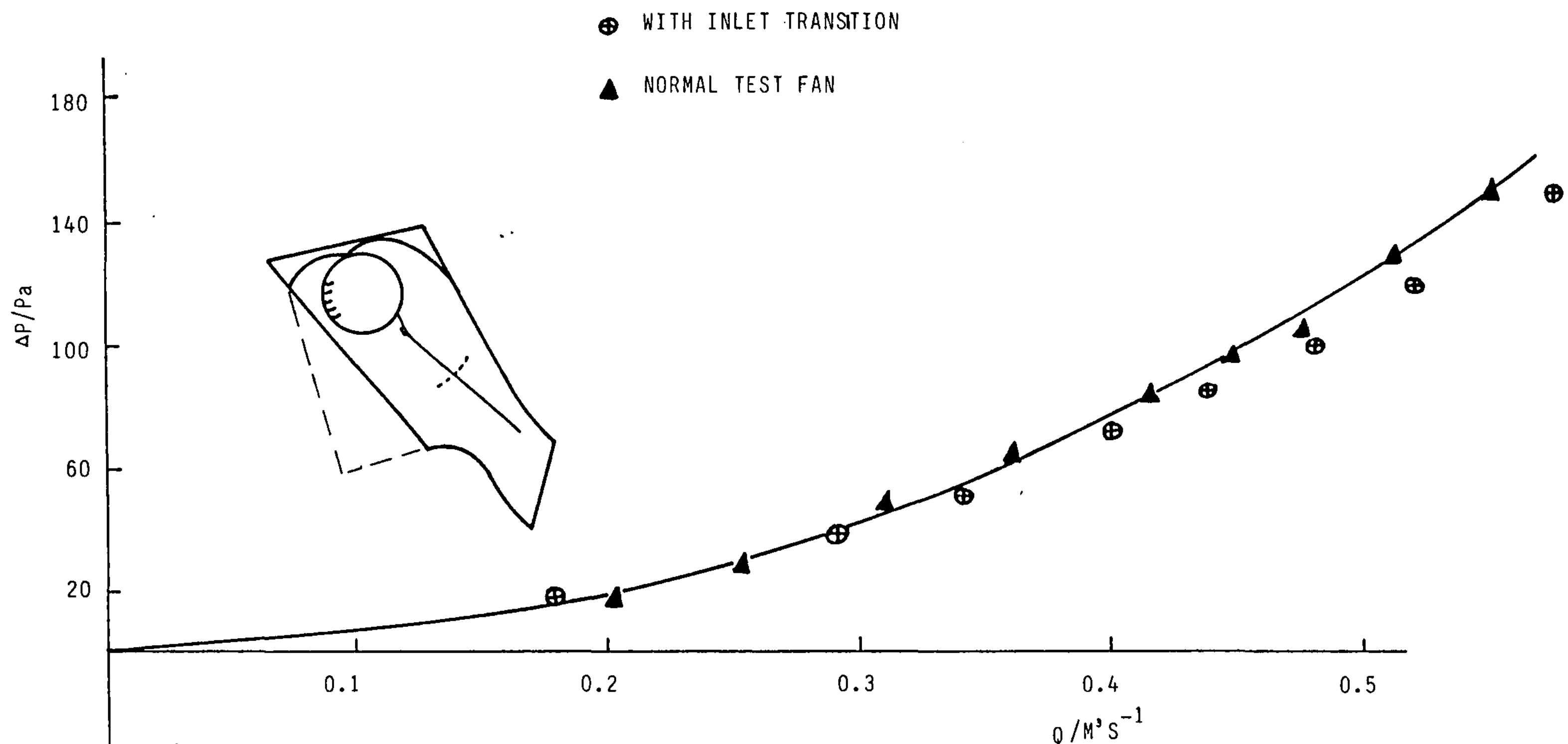


FIGURE 3.31 CONFIGURATION 1 - WITH INLET TRANSITION

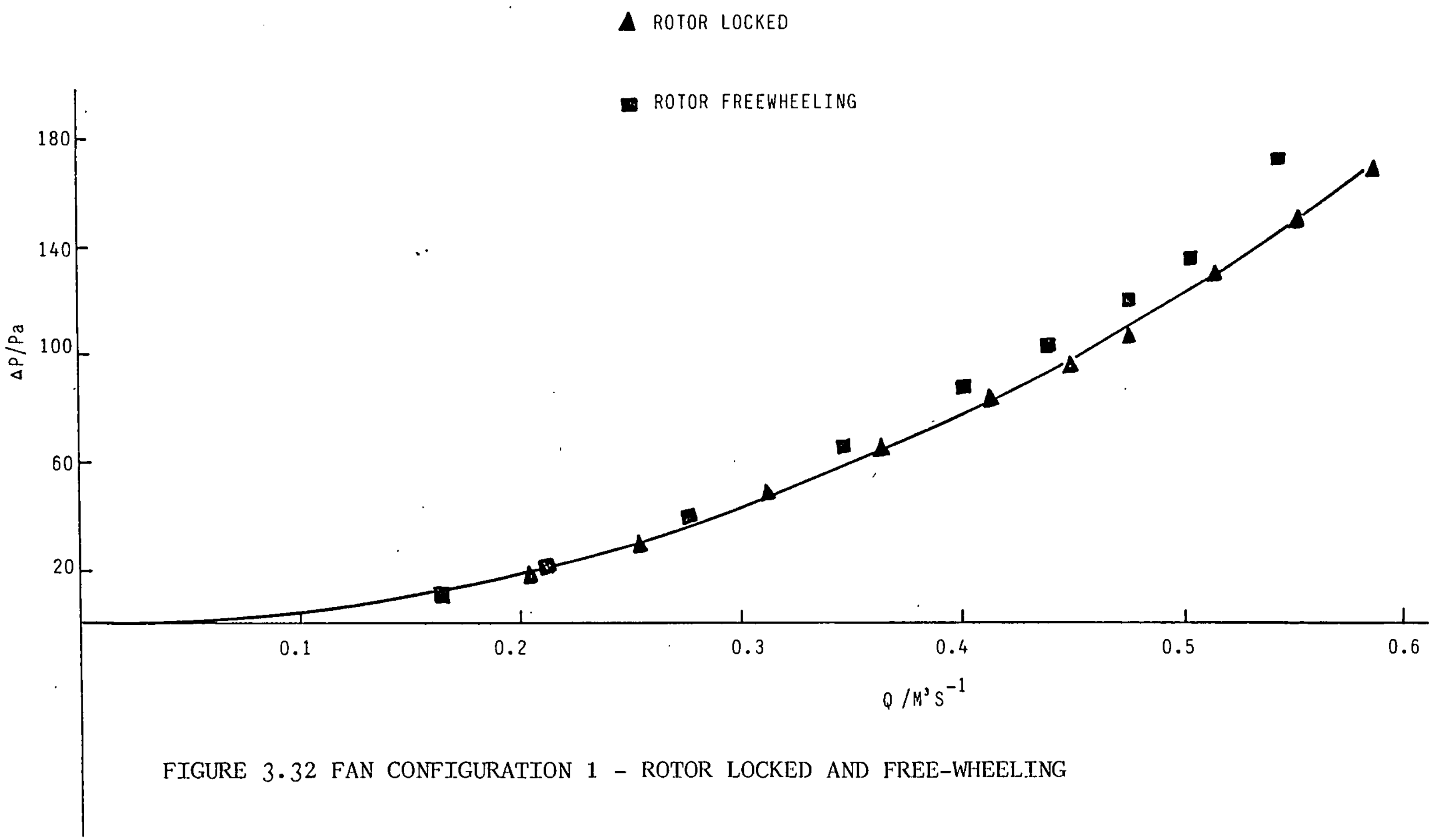


FIGURE 3.32 FAN CONFIGURATION 1 - ROTOR LOCKED AND FREE-WHEELING

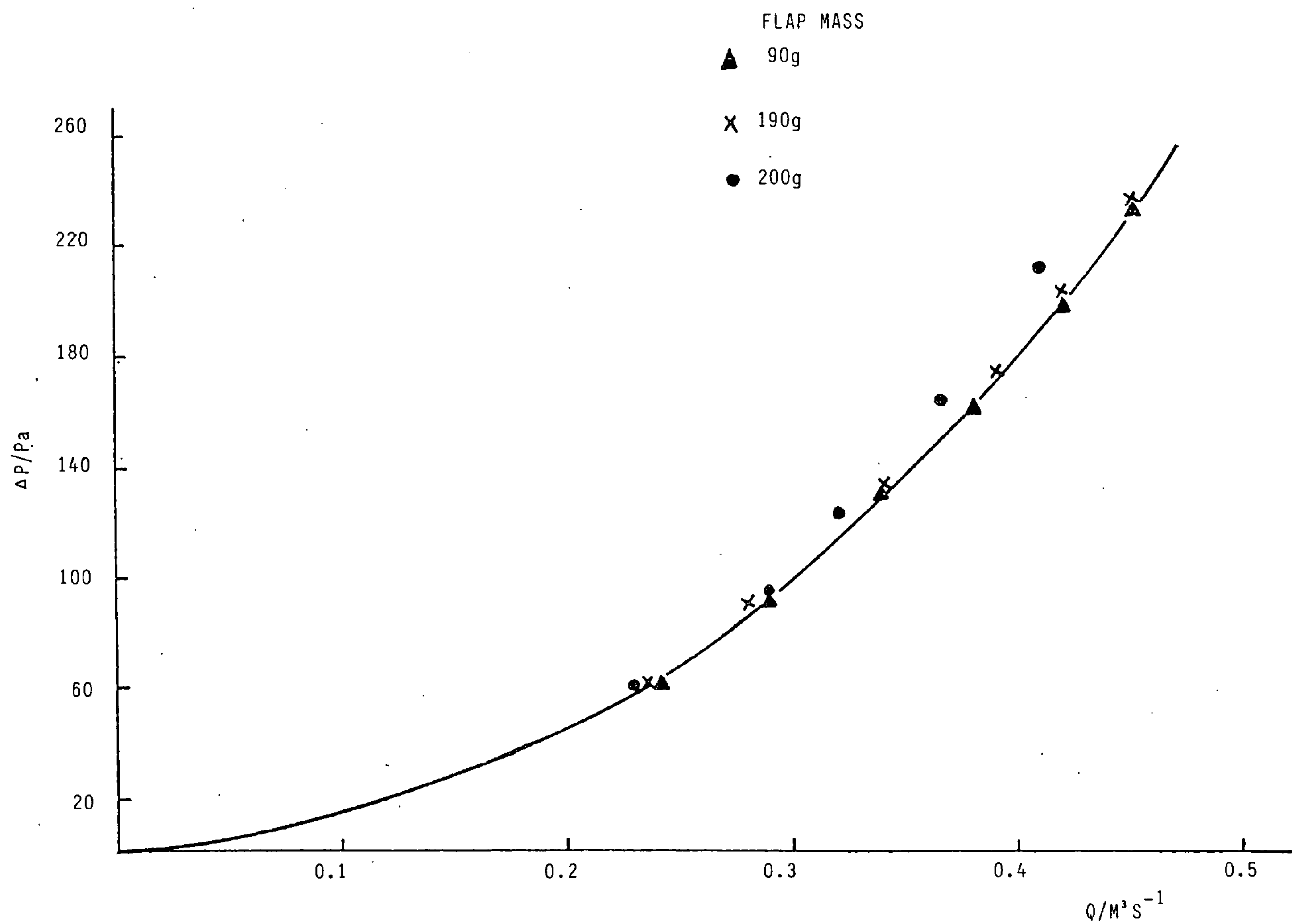


FIGURE 3.33 FAN CONFIGURATION 2 - EFFECT OF FLAP MASS

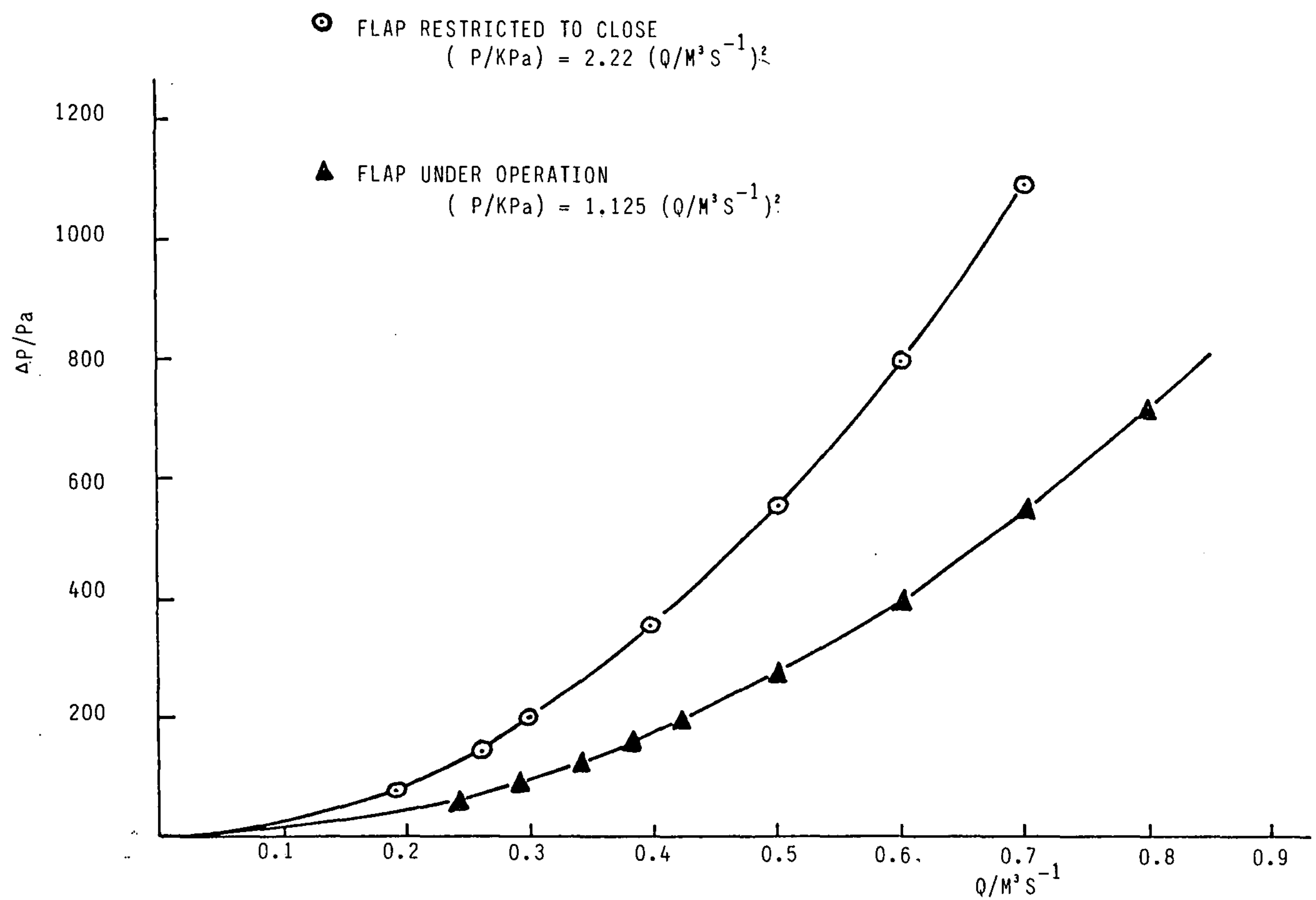


FIGURE 3.34 FAN CONFIGURATION 2 - GAINS DUE TO FLAP OPERATION

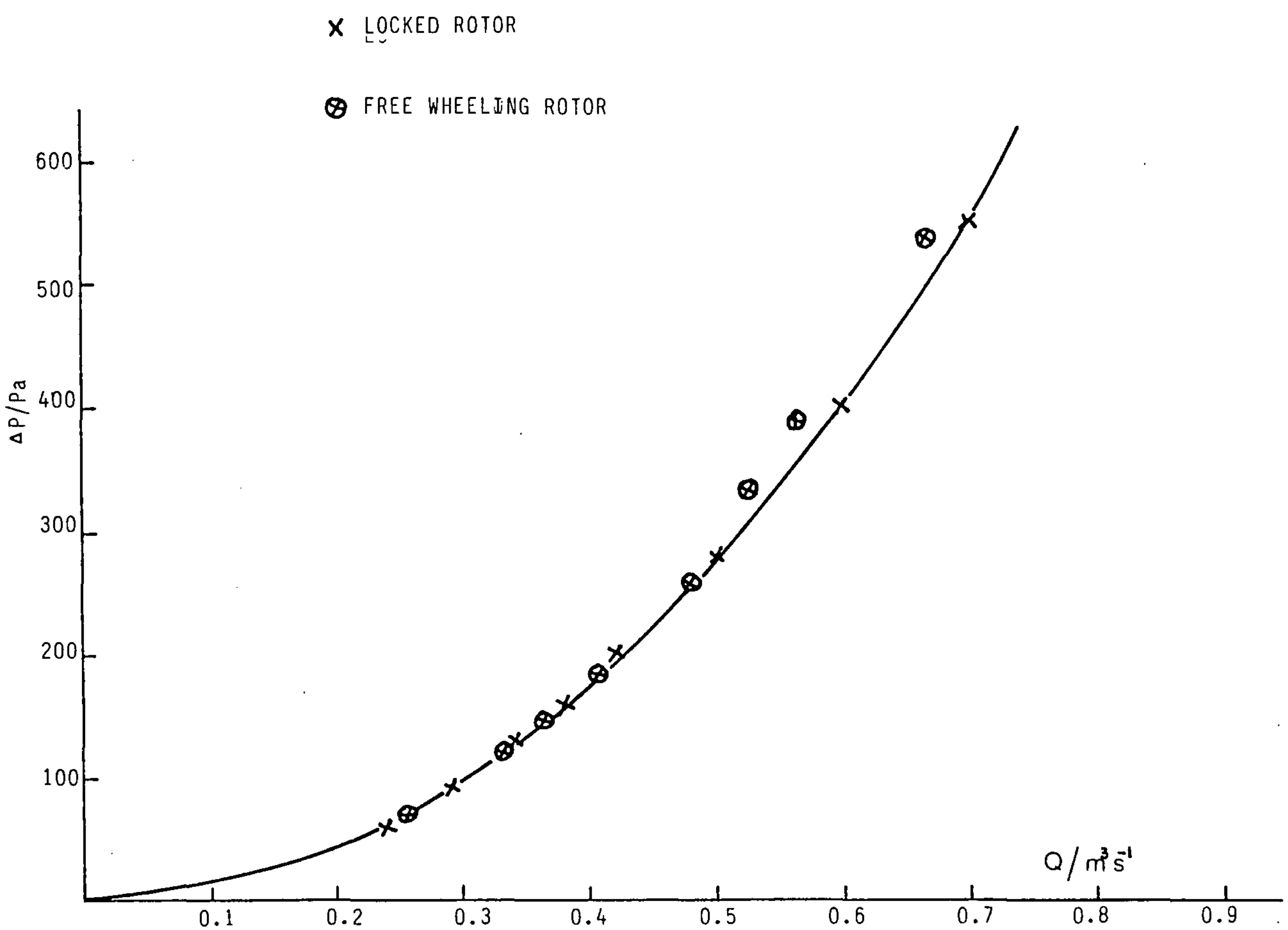


FIGURE 3.35 FAN CONFIGURATION 2 - FAN LOCKED AND FREE WHEELING

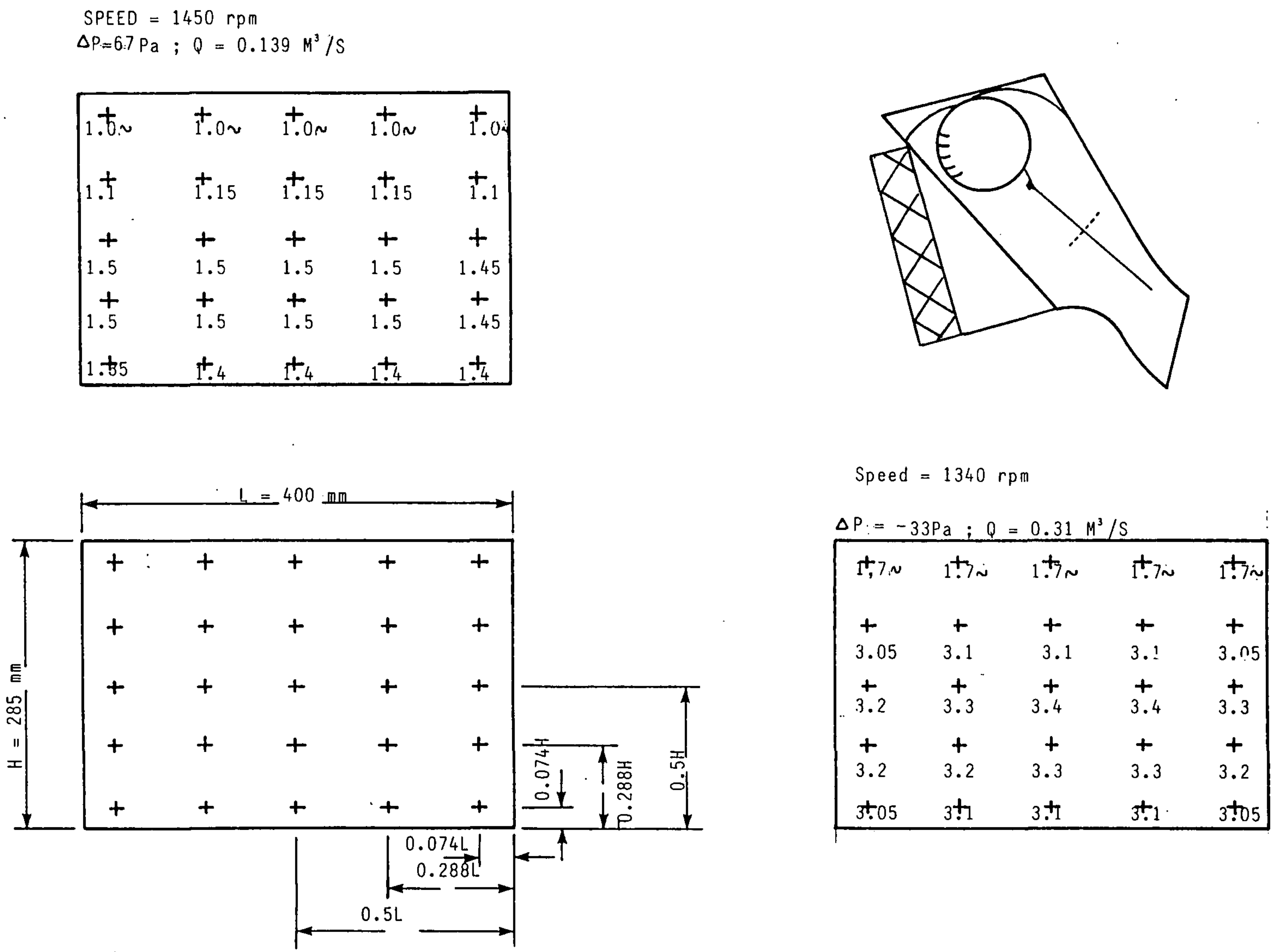
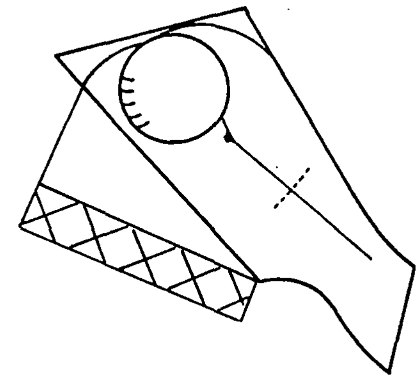
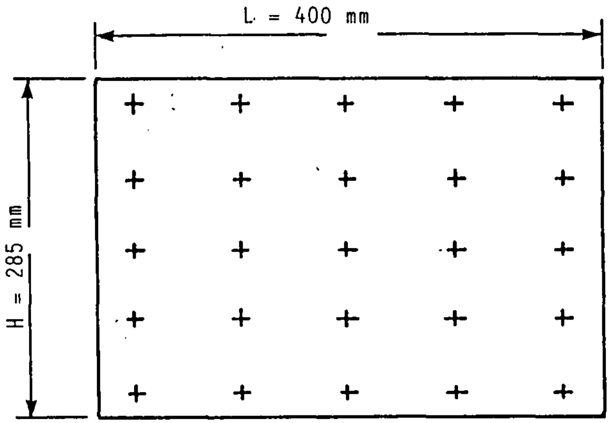


FIGURE 3.36 FAN CONFIGURATION 1 - VELOCITY DISTRIBUTION AT INLET

SPEED = 1391 rpm
ΔP = 27 Pa ; Q = 0237 M^3/S

1.0~	1.0~	1.0~	1.0~	1.0~
2.7	2.75	2.8	2.75	2.7
2.6	2.7	2.7	2.7	2.6
2.45	2.5	2.5	2.5	2.5
2.0	2.1	2.1	2.1	2.05





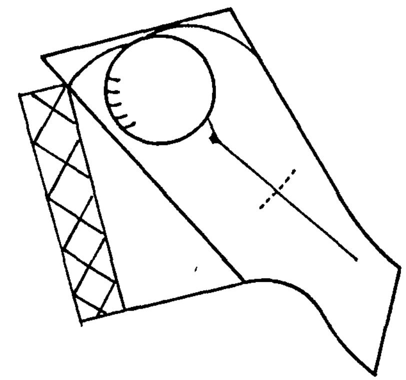
SPEED = 1366 rpm
ΔP = -14.5 Pa ; Q = 0.3 M^3/S

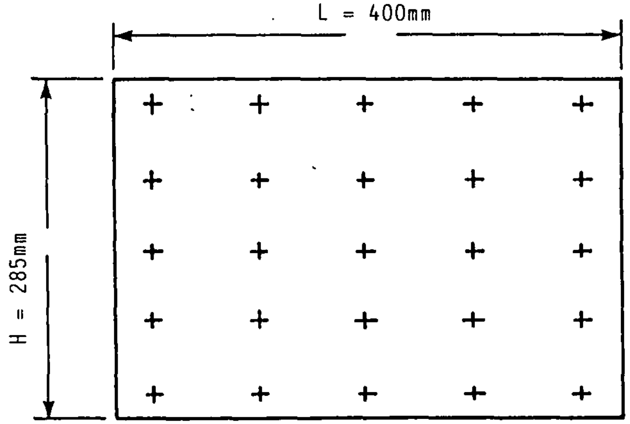
1.1~	1.1~	1.1~	1.1~	1.1~
3.3	3.4	3.4	3.4	3.3
3.1	3.25	3.4	3.3	3.15
2.9	3.0	3.0	3.0	2.9
2.75	2.8	2.8	2.8	2.75

FIGURE 3.37 FAN CONFIGURATION 1 - VELOCITY DISTRIBUTION AT INLET

SPEED = 0.0 rpm
ΔP = -102 Pa; Q = 0.3M^3/S

1.3~	1.3~	1.3~	1.3~	1.3~
2.7	3.0	3.1	3.1	2.7
2.7	3.1	3.3	3.2	2.8
2.7	3.1	3.2	3.2	2.8
3.05	3.0	3.1	3.0	3.0





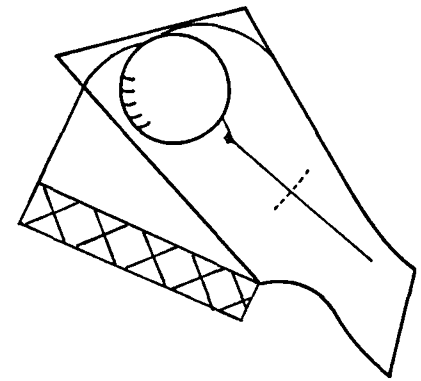
SPEED = 0.0 rpm
ΔP = -224 Pa; Q = 0.47 M^3/S

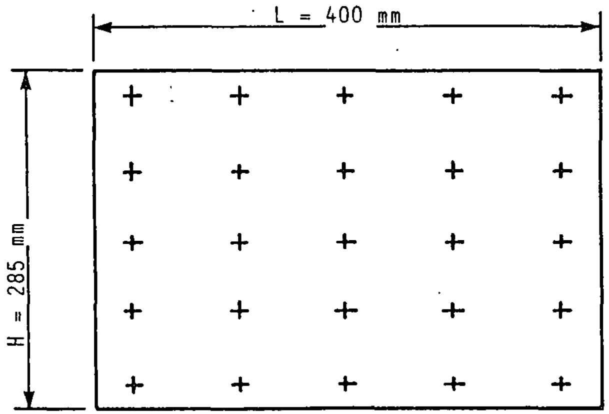
1.8~	1.8~	1.8~	1.8~	1.8~
4.0	4.6	4.8	4.6	4.0
3.9	4.7	4.8	4.7	3.9
3.9	4.6	4.8	4.6	3.9
3.8	4.6	4.8	4.5	4.5

FIGURE 3.38 FAN CONFIGURATION 1 - VELOCITY DISTRIBUTION AT INLET

SPEED = 0.0 rpm
$\Delta P = - 54$ Pa; $Q = 0.21\ M^3/S$

0.9~	0.9~	0.9~	0.9~	0.9~
1.8	2.1	2.2	2.2	1.8
2.0	2.2	2.3	2.1	2.0
2.0	2.2	2.2	2.1	1.9
2.0	2.1	2.15	2.0	2.0





SPEED = 0.0 rpm

$\Delta P = -170$Pa; $Q = 0.4\ M^3/S$

1.5~	1.5~	1.5~	1.5~	1.5~
3.9	4.0	4.1	4.1	4.0
4.2	4.2	4.3	4.2	4.2
4.0	4.0	4.2	4.1	4.0
3.8	3.9	4.1	4.0	3.8

FIGURE 3.39 FAN CONFIGURATION 1 - VELOCITY DISTRIBUTION AT INLET

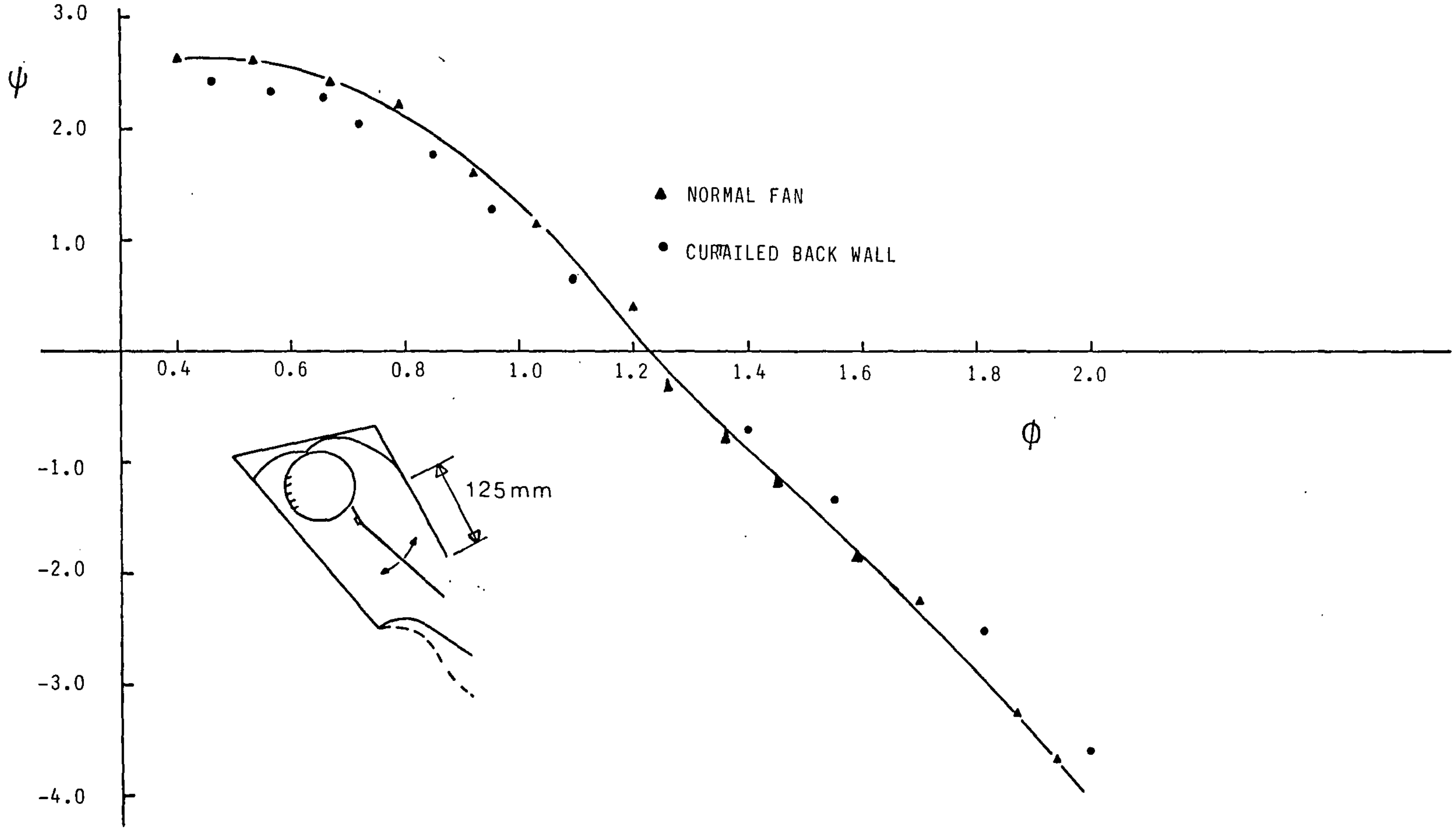


FIGURE 3.40 FAN CONFIGURATION - CURTAILED BACKWALL

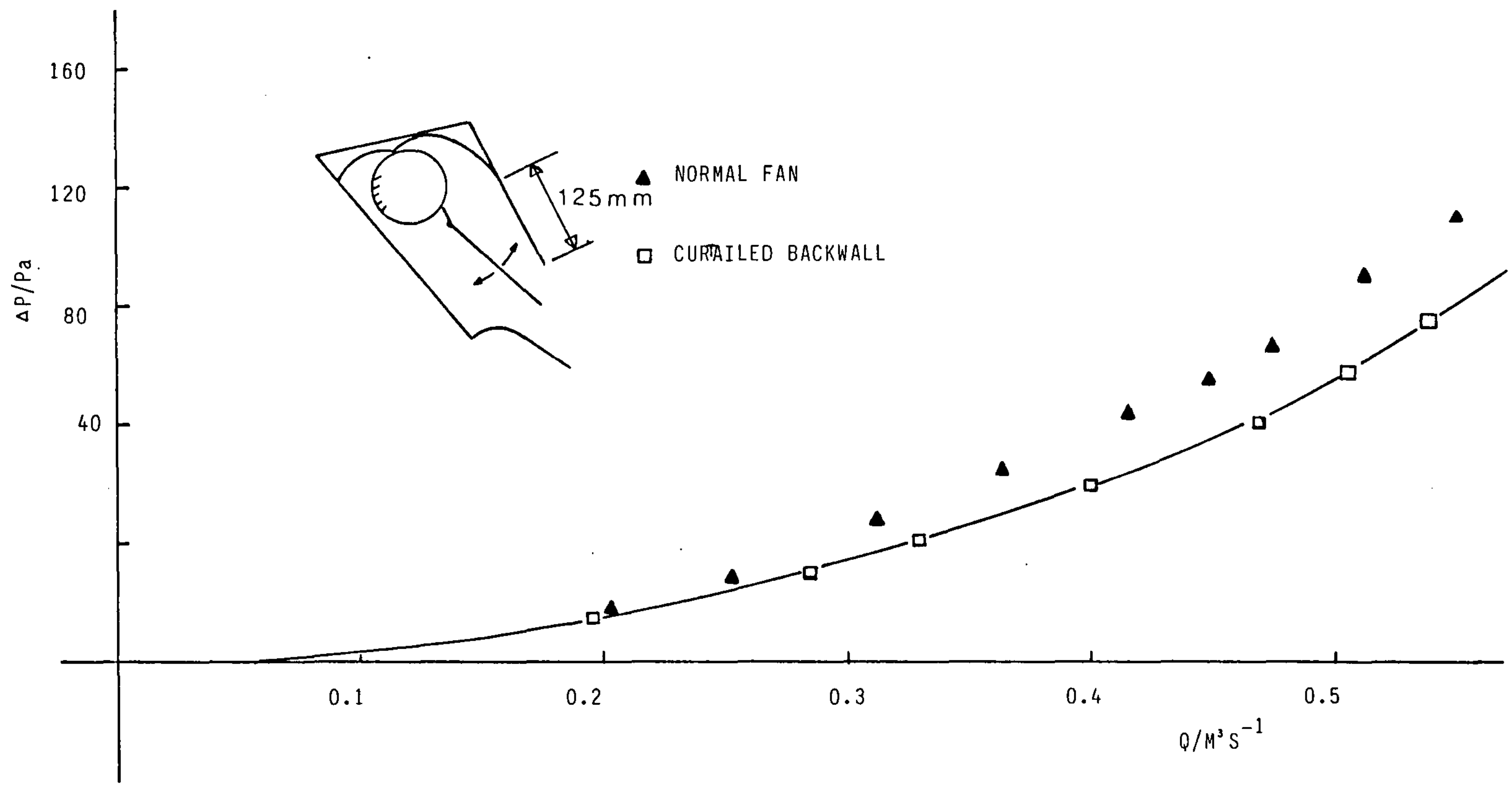


FIGURE 3.41 FAN CONFIGURATION 1 - CURTAILED BACKWALL

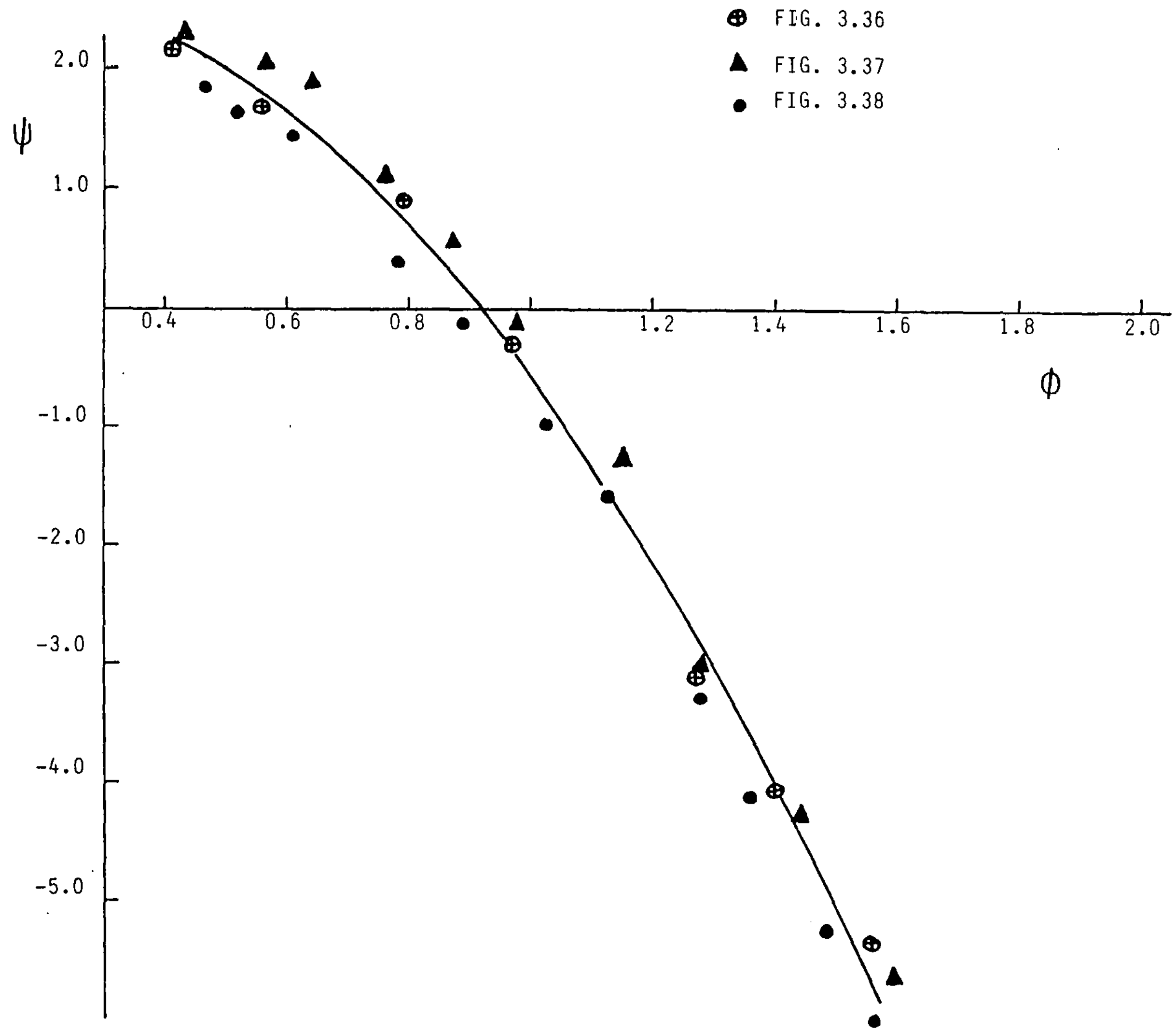


FIGURE 3.42 FAN CONFIGURATION 1 - WITH THE RADIATOR

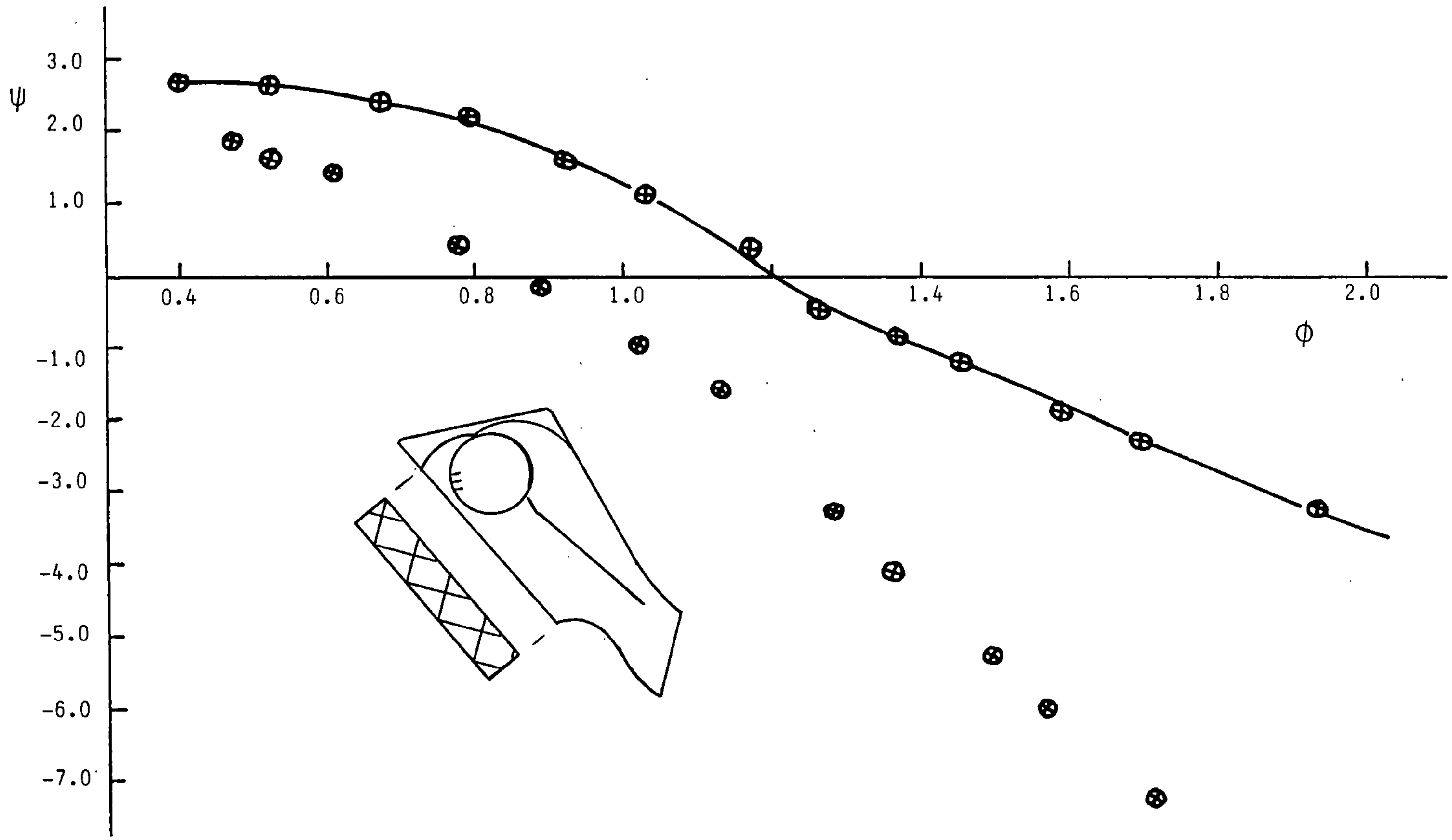


FIGURE 3.43 FAN CONFIGURATION 1 - WITH AND WITHOUT RADIATOR

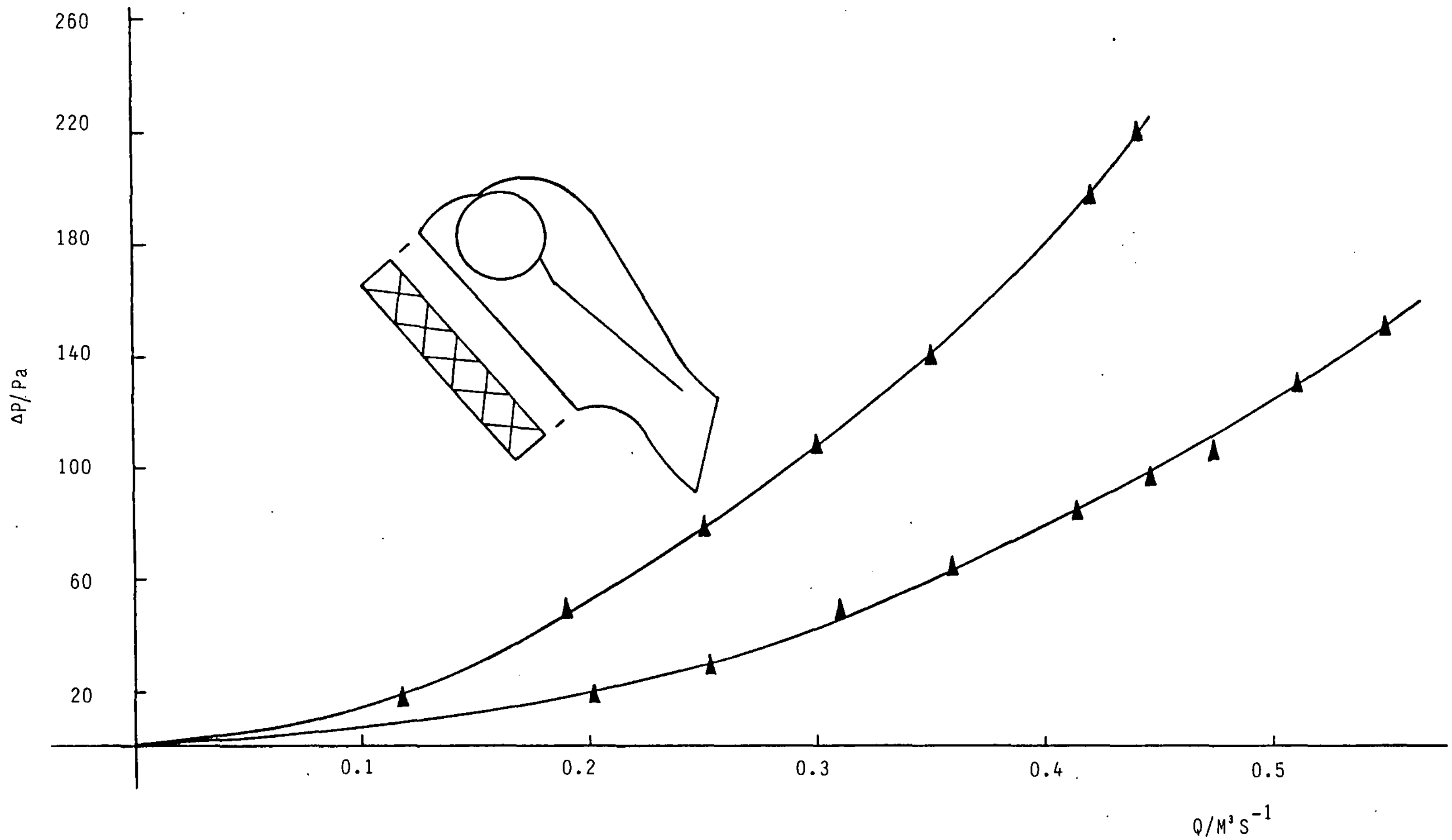


FIGURE 3.44 FAN CONFIGURATION 1 - WITH AND WITHOUT RADIATOR

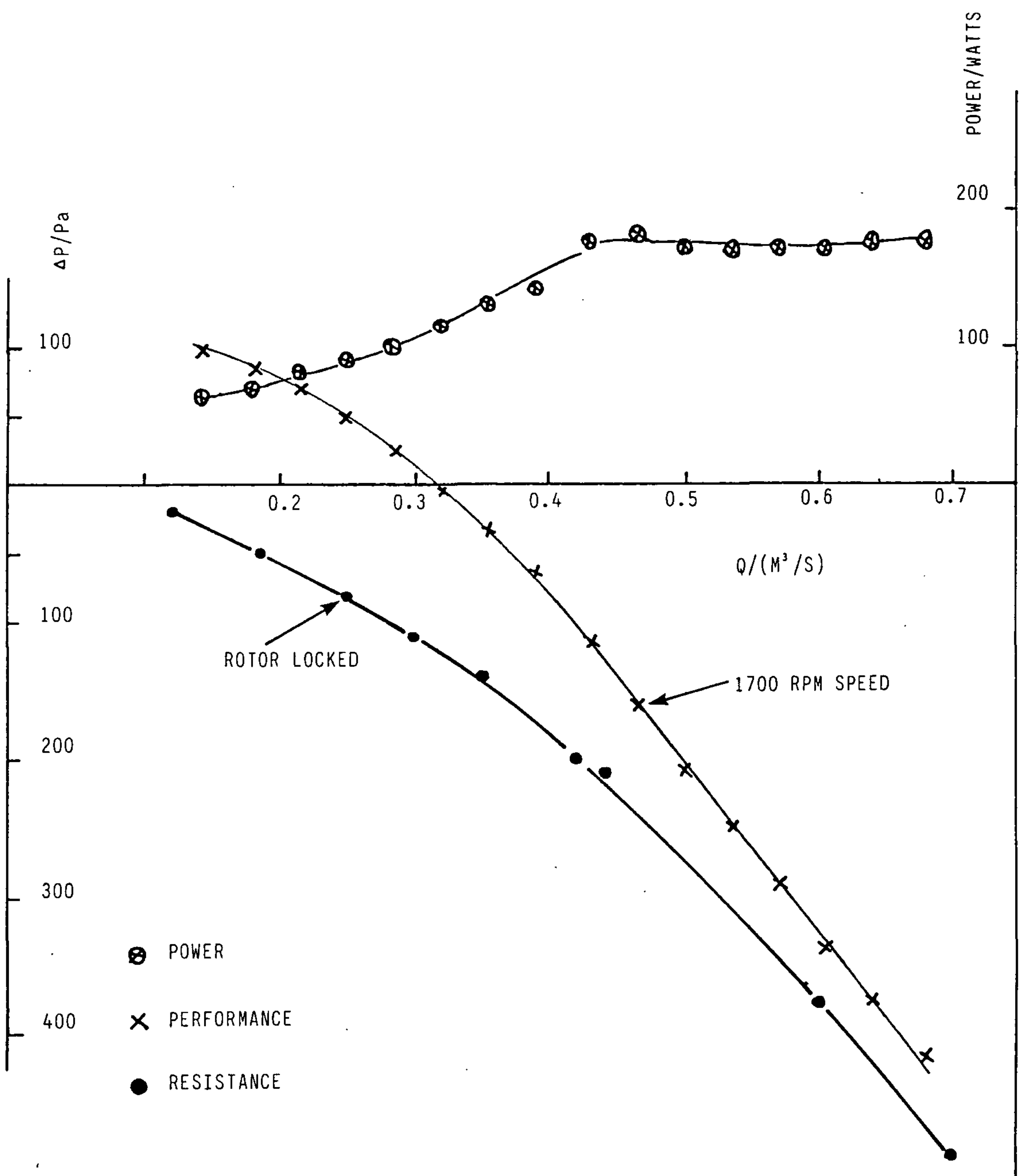


FIGURE 3.45 FAN CONFIGURATION 1 - OVERALL BEHAVIOUR WITH RADIATOR AT INLET

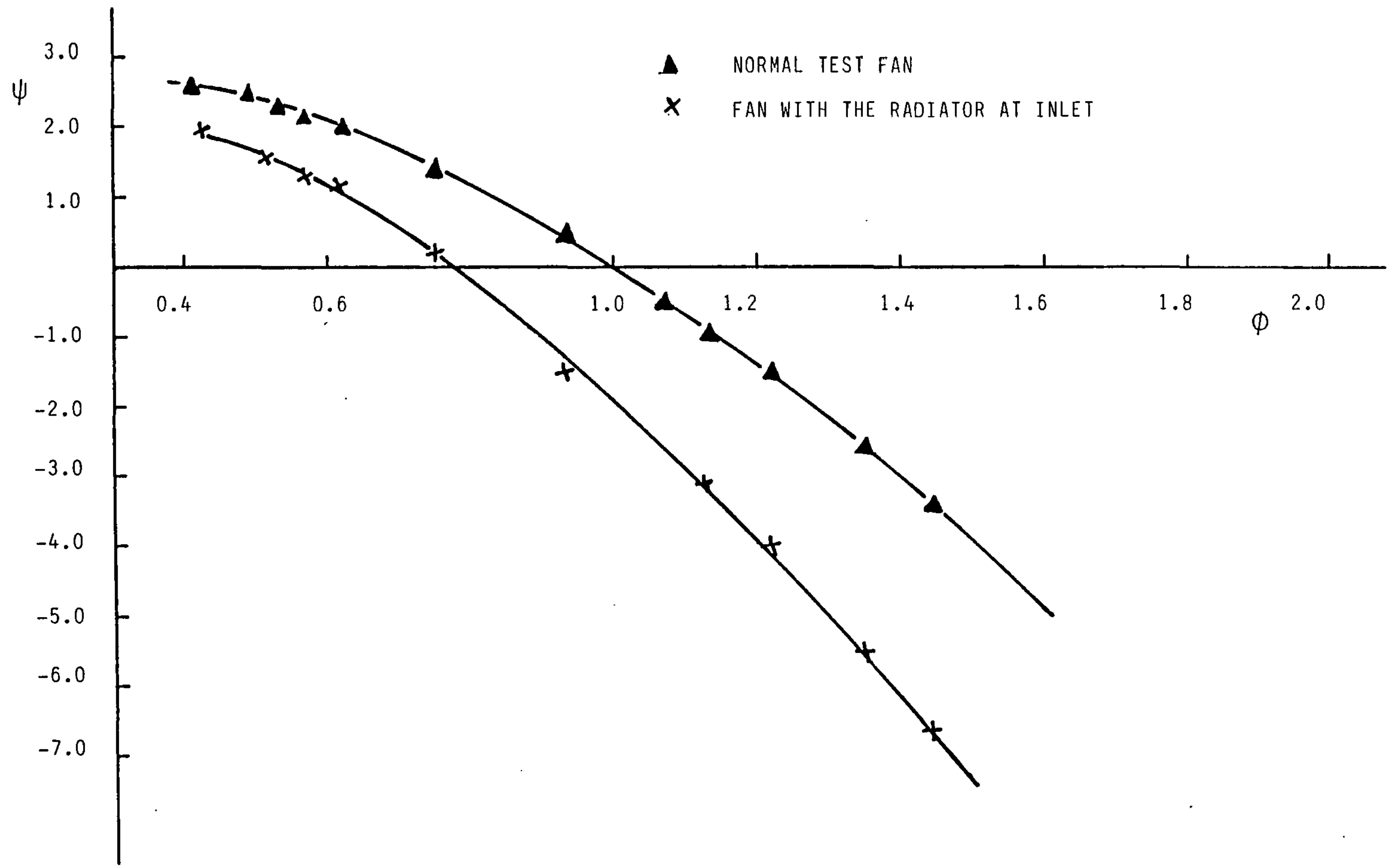


FIGURE 3.46 FAN CONFIGURATION 2

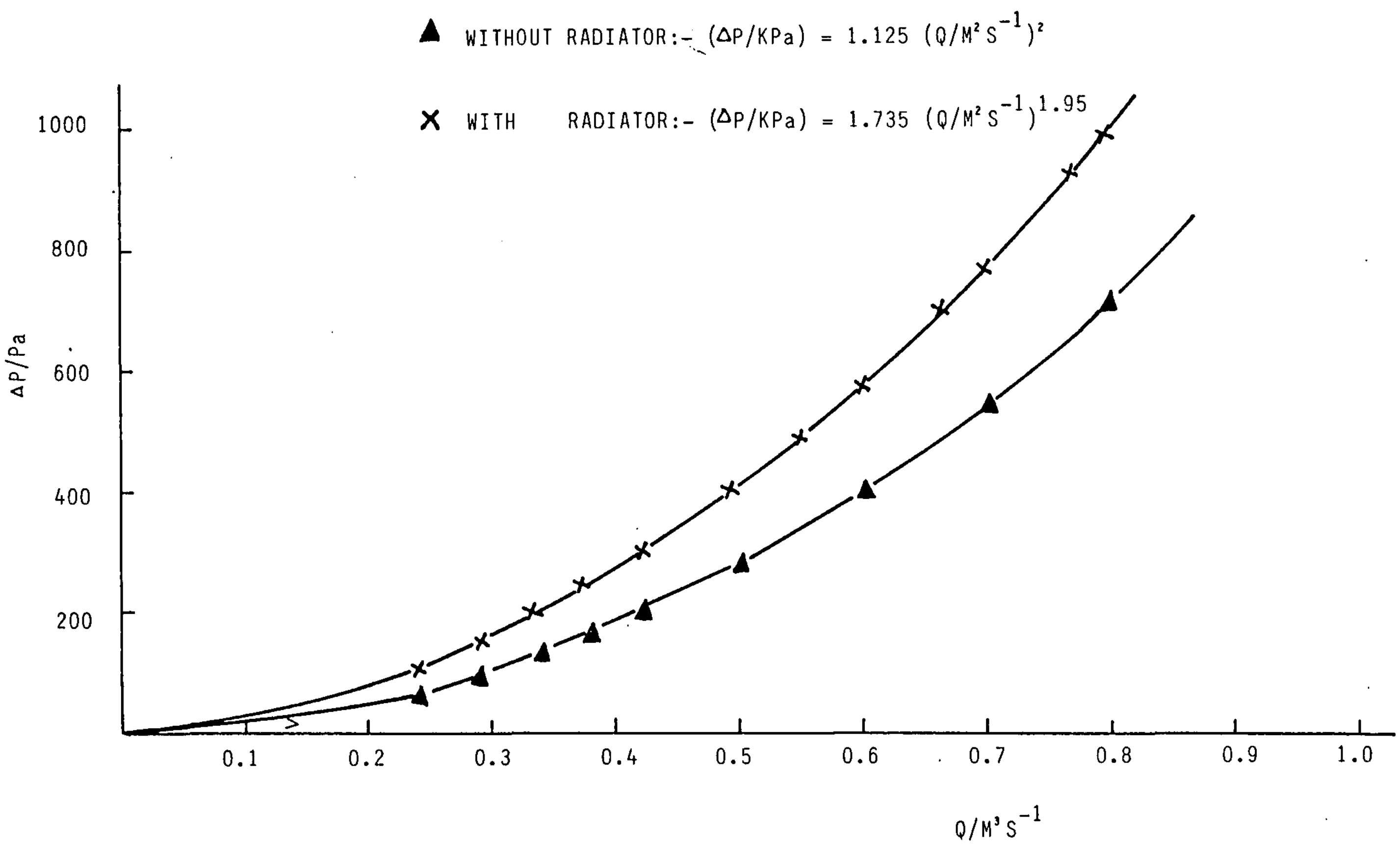


FIGURE 3.47 FAN CONFIGURATION 2

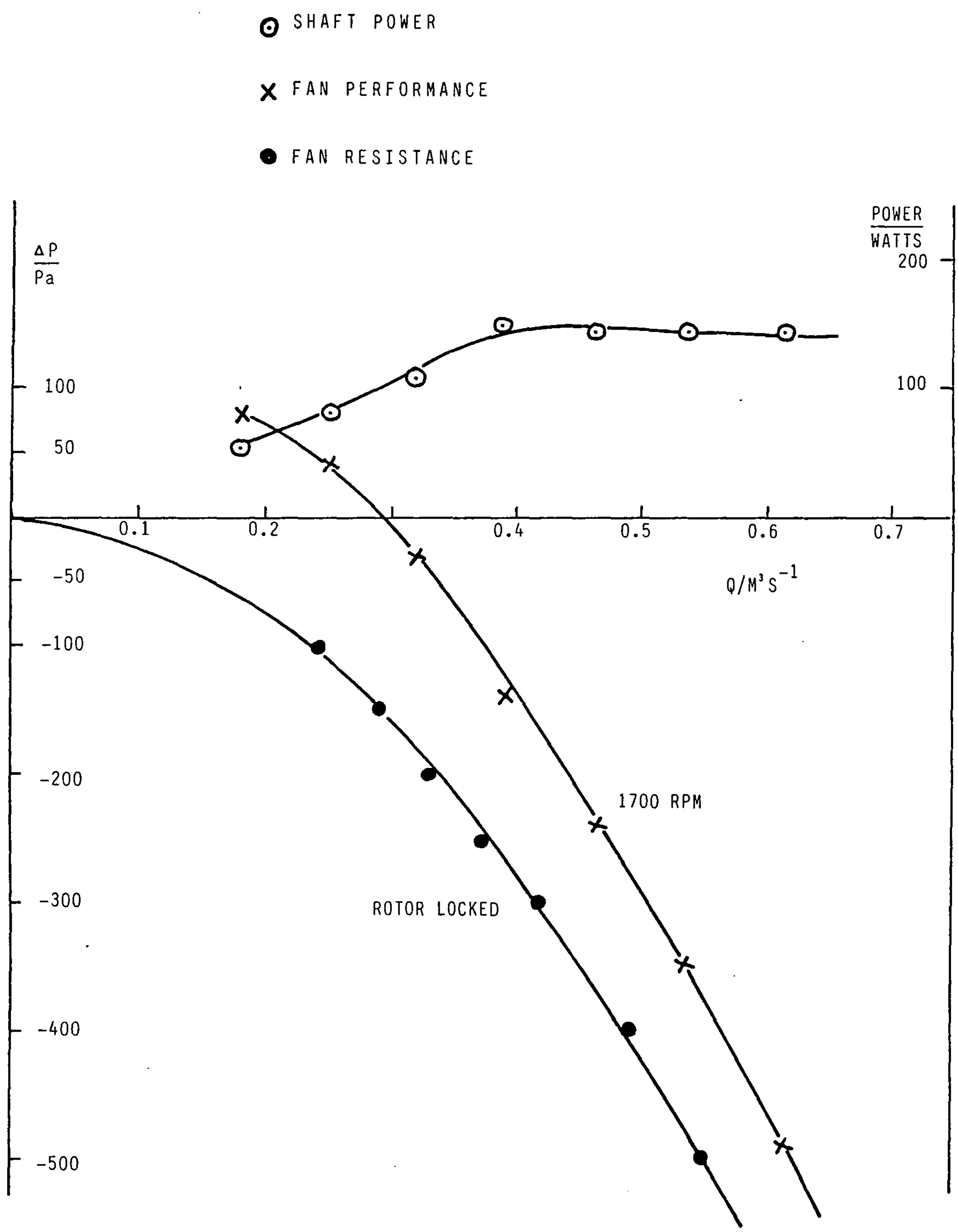


FIGURE 3.48 FAN CONFIGURATION 2 - OVERALL BEHAVIOUR WITH RADIATOR AT INLET

CHAPTER FOUR

COMPUTER AIDED SYSTEM MATCHING

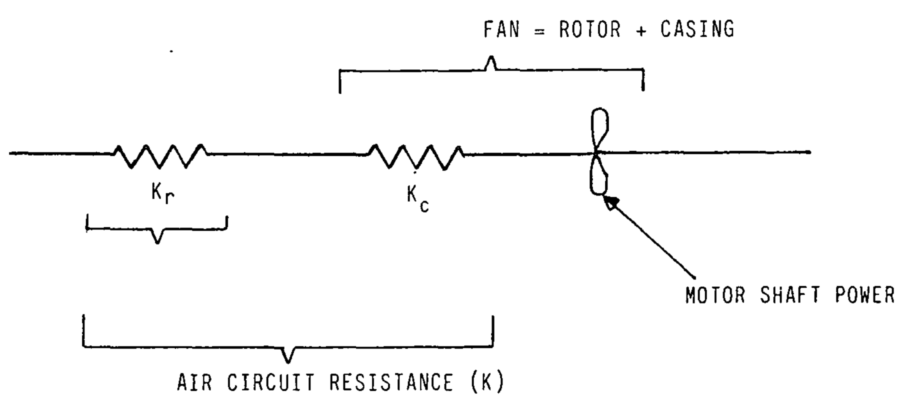
4. Computer Aided System Matching

4.1 Computer Model

4.1.1 Principle

The purpose of this system simulation is to match up components of the cooling air system, namely the fan , a 12v permanent magnet motor and the air circuit, to given input parameters which might be based on critical conditions. Also to predict the performance of the system throughout the operating range, when the system is operating in a steady state.

SYSTEM CIRCUIT



The characteristics of system components were represented as follows

i) Cross flow fan characteristic in dimensionless form: ie

$$\psi = f(\phi);\ \eta = g(\phi) \qquad - \qquad (1)$$

f and g would be known fan functions, experimentally determined

ii) 12vDC permanent magnet motor performance in the form of:

Torque = - M Speed + C

or $T = e(N)$ (2)

where e would be known as motor function.

Torque and speed are measured in Nm and RPM.

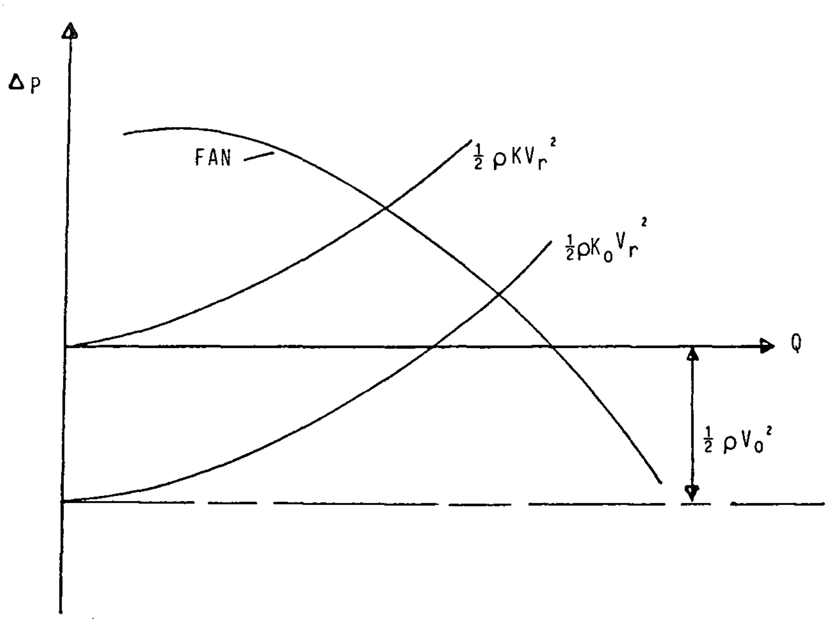
And M = 1/speed regulation constant of the motor

C = No load speed/speed regulation constant

= Torque at zero speed

iii) The air circuit characteristic was assumed to be as in turbulent flow; ie; total pressure drop is proportional to square of velocity ($\frac{1}{2}\rho K V_r^2$)

where K = Air circuit resistance coefficient or effective system resistance coefficient (K_o) which is equal to air circuit resistence coefficient inclusive of the reduction in resistance due to dynamic flow head, resulted from vehicle motion. Hence $\frac{1}{2}\rho K_o V_r^2 = \frac{1}{2}\rho K V_r^2 - \frac{1}{2}\rho V_o^2$ -(3)



And r where $\Delta P = r(V_r)$, would be known as air circuit function.

The combination of the fan, motor and system resistance coefficient can be expressed (as in Section 2.3.2).

$$Q = -\frac{120}{\pi}\left(\frac{\eta_b \eta_f . M}{\rho D^2 \psi}\right) + \frac{1}{2}\left[\left(\frac{240\eta_b\eta_f M}{\pi\rho D^2 \psi}\right)^2 + \left(\frac{16\ \eta_b\eta_f CL\phi}{\rho\psi}\right)\right]^{\frac{1}{2}} \quad - (4)$$

$$K = \frac{\psi}{\phi^2}\left[\frac{H}{D}\right]^2 \quad — \quad (5)$$

And if the effective system resistance resistance coefficients for vehicle, stationary and at speed are K_s and K_o respectively. Then from equation (3)

$$(K_s - K_o) = \left[\frac{V_o}{V_r}\right]^2 = \left[\frac{V_o\ H.L}{Q_o}\right]^2 \quad - (6)$$

where V_r and Q_o are radiator face velocity and flow rate of the system at 'V_o' vehicle speed.

Therefore the corresponding operating point, ϕ_o, ψ_o, η_o of the fan at V_o vehicle speed should satisfy the expression

$$K_o = \frac{\psi_o}{\phi_o^2}\left[\frac{H}{D}\right]^2 \quad - (7)$$

4.1 2 <u>Computer Simulation of Fan Dimensionless Characteristics</u>

The fan characteristic plot of ψ (and η) against ϕ was broken into 34 sub sets to have equal divisions of flow coefficient. The first and last values of flow coefficients were chosen to be 0.3 and 2.0. Then each sub-set was expressed by a continuous function of ϕ. For example the ψ of the i^{th} sub set was expressed by

$$= A1(i)\ \phi^2 + A2(i)\ \phi + A3(i) \quad - (8)$$

and $d\psi/d\phi = 2A1(i)\,\phi + A2(i) = 0$ represents the point of maximum value of pressure coefficient, which was $\phi = 0.4$ (for both configurations).

Therefore $0.8\,A1(i) + A2(i) = 0$ -- (9) for all values of i.

Since the corresponding values of pressure coefficient of the first and last values of the flow coefficient (ϕ_i, ϕ_{i+1}), of the i^{th} sub-set are known,

$$\psi_i = A1(i)\phi^2_i + A2(i)\phi_i + A3(i) \qquad - (10)$$

$$\psi_{i+1} = A1(i)\phi^2_{i+1} + A2(i)\,\phi_{i+1} + A3(i) \qquad - (12)$$

Therefore equations (9), (10) and (12) were evaluated for A1, A2 and A3. Hence the continuous expression $\psi = A1(i)\,\phi^2 + A2(i)\phi + A3(i)$ was determined for the i^{th} sub-set. Similarly all of the 34 sub-sets were computer simulated. Same method was adopted to simulate the characteristic plot of η against ϕ.

The stable fan operating values of ϕ in the i^{th} sub-set is given by (from eq.1).

$$\phi = \frac{-A2(i) - \left[A2(i)^2 - 4A1(i).(A3(i) - \psi)\right]^{\frac{1}{2}}}{2A1(i)} \qquad (13)$$

Or

$$\left[A1(i) - \frac{\psi}{\phi^2}\right]\phi^2 + A2(i)\phi + A3(i) = 0$$

Hence $$\psi = \frac{-A2(i) - \left[A2(i)^2 \quad 4.(A1(i) - \psi/\phi^2).\ A3(i)\right]^{\frac{1}{2}}}{2\left[A1(i) - \psi/\phi^2\right]} \qquad -(14)$$

where ψ/ϕ^2 is throttling coefficient.

When the value of ψ (or ψ/ϕ^2) is known, the 34 sub-sets would be first searched to find the corresponding step in which ψ (or ψ/ϕ^2) is a composition. Then the continuous function of ϕ, of that particular step would be determined as before. This function is then used to find ϕ from equation (13) (or 14). Therefore this simulation would now allow determination of ϕ,η and ψ/ϕ^2 correspond to any value of ϕ in between 0.3 and 2.0 and vice versa.

When the fan power is determined from these dimensionless characteristics, the coefficient of performance of the fan, defined as $\psi\phi/\eta$, is generally needed considering. But in the range, where ψ is very small, the η is also very small. This leads to an unpredictable error in the value of ψ/η . Therefore in this relatively small range a linear relationship (fig.4.5) of ϕ and ψ/η , satisfying $\psi/\eta = B\phi + G$, was assumed.

4.1.3 Input Data Structures

The system matching might be approached from four different input data structures. They are

1) Diameter, length, C and M
2) Shaft power, Speed, System Resistance Coefficient
3) Diameter, Length, Flowrate, System Resistance Coefficient
4) Speed, System Resistance Coefficient, C and M

4.1.3.1 Input Data Structure Type 1

The flow chart and the transition functions were shown in figures 4.1 and 4.6. This structure and output data were formed to illustrate the overall picture of the system performance for a given fan/motor combination. An example of an output was illustrated in table 4.9.

Outputs, flow rate, pressure rise, speed, shaft power, effective system resistance coefficient, fan inlet face velocity were related to one point of fan dimensionless characteristic (ϕ, ψ, η) by evaluating equations (1) to (5). And these outputs were tabulated for ϕ starting from 0.3 and increasing in 34 steps of 0.05 to the last value 2.0. Then the relationship of radiator face velocity (V_r) to vehicle speed (V_o) was tabulated in accordance with equation (6).

4.1.3.2 <u>Input Data Structure Type 2</u>

The flow chart and the transition functions were illustrated in figures 4.2 and 4.8. this structure would be used for a design approach based on available power constraints. First the throttling coefficient (ψ/ϕ^2) was evaluated from equation (5). The fan characteristics were searched for corresponding ϕ, ψ and η and then D^4L and QD^2 were evaluated from equations (1) and (2). An example of an output, of this type was shown in table 4.11. The input data of two critical requirements would determine two corresponding motor equations, $C = f_1(M)$ and $C = f_2(M)$. Then evaluating these equations for C and M would ascertain the characteristic equation of the motor.

4.1.3.3 <u>Input Data Structure Type 3</u>

The flow chart and the transition functions were illustrated in figures 4.3 and 4.7. This structure would be used for a design approach based on limited engine compartment space constraints. An example of an output, of this type was demonstrated in table 4.13. First the throtteling coefficient (ψ/ϕ^2) was evaluated from equation (5), then the fan characteristics were searched for corresponding ϕ, ψ and η . Hence the fan speed were determined from ϕ, then fan pressure rise and power from ψ and η. As in Section 4 1.3.2 above, two critical conditions would determine the characteristic equation of the motor.

4.1.3.4 Input Data Structure Type 4

The flow chart and the transition functions were illustrated in figures 4.4 and 4.8 This structure is similar to type 2, except the system would be matched to a particular, given motor. An example of an output data was demonstrated in table 4.12.

4.2 Matching to an Overall Requirement

The overall heat dissipation requirement was as indicated in Section 2.2.1. The radiator was type XB4FB - 8005 - K40 with the characteristic as illustrated in figures 2.4 and 2.5. The coolant flow rate was maintained at 2.0 L/s. The airflow pressure drop characteristics indicated in figure 2.4 relates to free inlet and outlet, which includes the entry loss and kinetic energy loss at outlet of the radiator. But, when the radiator is fitted to crossflow fan inlet, it represent a ducted exit. Therefore, the effective total pressure drop of the radiator for ducted outlet installations would be $\frac{1}{2}\rho V_r^2$ less than indicated in figure 2.4. Furthermore, experimentally obtained crossflow fan characteristics of this thesis, also includes the entry loss of its inlet. Therefore, when considering the performance or resistance of a fan/radiator combination, this entry loss should be excluded. For the cross flow inlet geometry an entry loss coefficient of 1 can be assumed.[2] Hence the total pressure drop of the fan/radiator systems air circuit resistance is

= (Indicated radiator ΔP) - $\frac{1}{2}\rho V_r^2$ + (pressure drop across fan)- $\frac{1}{2}\rho V_r^2$

= (Indicated radiator ΔP) - ρV_r^2 (pressure drop across fan)

The indicated radiator pressure drop satisfies the relation:

$17.06\ (V_r/ms^{-1})^{1.7}$

The effective radiator pressure drop, defined as:

$17.06\ (V_r/ms^{-1})^{1.7} - \rho V_r^2$

was plotted in figure 4.14. This plot v rified the effective radiator pressure drop to be:

$15.86\ (V_r/ms^{-1})^{1.7}$

When this resistance was approximated to one characterising the square law (turbulent flow resistance), with a resistance coefficient of 17.3, the extreme operating radiator velocities, 6m/s and 3m/s, deviated only by 5% (fig.4.14). If necessary, this approximation may be easily further improved by dividing the operating range to several sub-sets and then approximating each of these sub-set ranges to a corresponding system resistance coefficient.

The radiator heat dissipation characteristics of ref. 20 were copied to a large scale in figure 4.15.

4.2.1 Match 1

Cross flow fan = configuration 1

Radiator area = 0.155m²

Fan covered radiator area = 0.09m²

Fan diameter = 0.08m

Fan length = 0.4m

Fan resistance coefficient, when unenergised (fig.3.28) = 10.53

Fan motor type (as in fig.4.16) = MP 8140/C

Fan motor characteristic: T = -0.00118 N + 3.3

Therefore, fan/radiator system's air circuit resistance coefficient, when fan is unenergised = 10.53 + 17.3 = 27.83.

At V_o vehicle velocity: $\frac{1}{2} \rho V_o^2 = \frac{1}{2} \rho (27.83). V_r^2$

Hence, when fan is unenergised $V_o/V_r = 5.28$.

The cooling air circuit's resistance coefficient, when fan is driven (energised) = 17.3

The computer outputs of this particular fan characteristic (energised state) were illustrated first in table 4.9, then the V_r and V_o relationship, based on system resistance coefficient 17.3. These results and $V_o/ V_r = 5.28$, were combined for fan covered area of 0.09m² and uncovered area of 0.065m² to determine the overall flowrate. Then with the aid of figure 4.15 the corresponding radiator heat dissipation for 59°C extreme air/water temperature differential (ref. 20), was calculated and plotted in figure 4.17.

4.2.2 Match 2

Cross flow fan = configuration 1

Radiator area = 0.155m²

Fan covered radiator area = 0.077m²

Fan diameter = 0.09m

Fan length = 0.3m

Fan motor type = MP8140/C (fig.4.16)

As in match 1, the radiator heat dissipation was calculated and plotted in figure 4.17.

4.2.3 Match 3

Two critical conditions, of a design based on restricted space, were matched to find the appropriate electric motor.

As an example, the fan diameter and length was restricted to 0.09 and 0.35m, and the two critical conditions were taken as the cooling flow requirement, $0.3M^3/s$ and $0.37\ M^3/S$ at $V_O = 0$ and 17m/s. The system resistance coefficient (at stationary) was 17.3.

Therefore when vehicle is travelling at 17m/s (35mph): - the radiator face velocity = 0.37/(2.85 x 0.09 x 0.35)

= 4.12m/s

the effective system resistance coefficient at $V_o = 17$ m/s, is 2

The computer outputs of these two critical conditions were shown in table 4.13.

Hence, first critical condition gives: -2222M + C = 0.708

Second critical condition gives: -2079M+C = 0.762

By evaluating these two equations for M and C, determined the motor equation to be T/Nm = -0.00038 (N/rpm) + 1.55.

4.1.3.5 Transition Functions

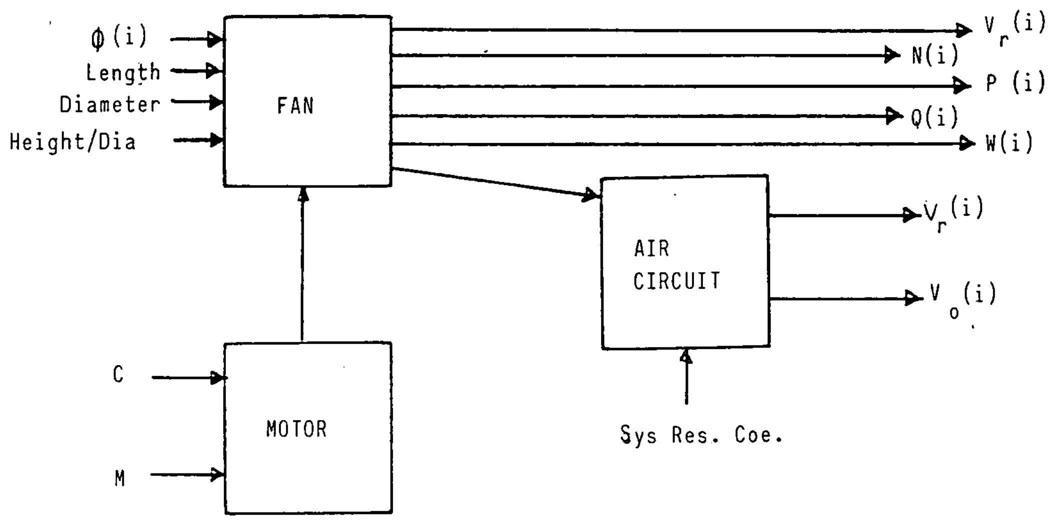


FIGURE 4.1 STUCTURE TYPE 1

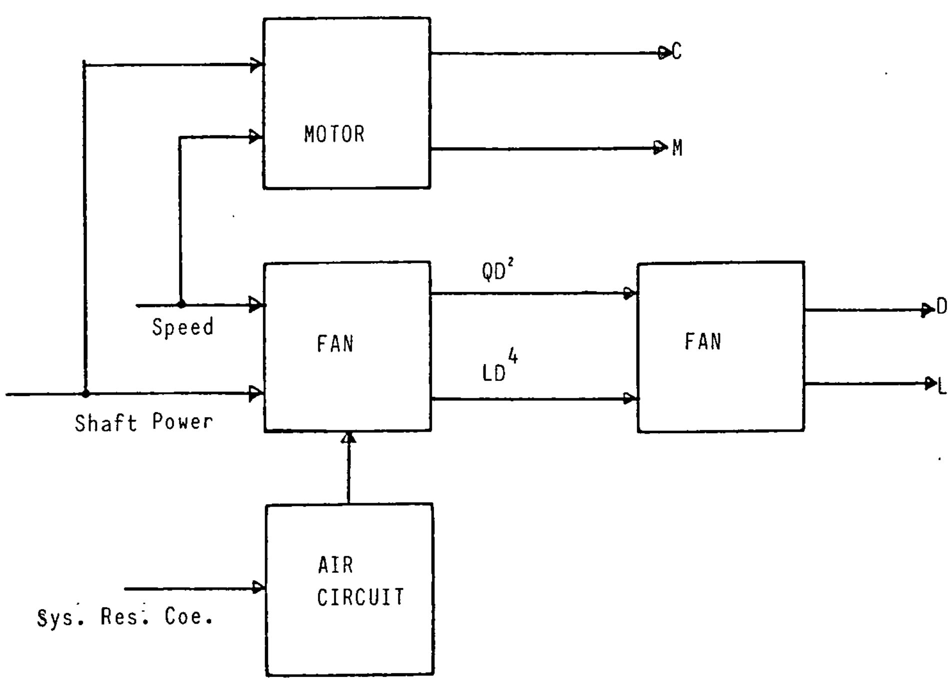


FIGURE 4.2 STRUCTURE TYPE 2

4.1.3.5 Transition Functions - Cont.

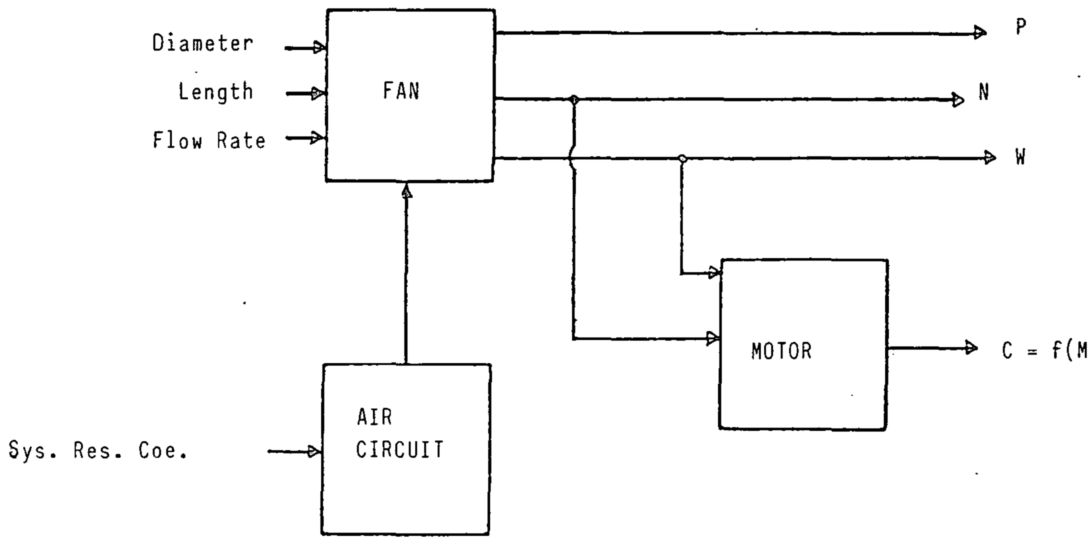


FIGURE 4.3 STRUCTURE TYPE 3

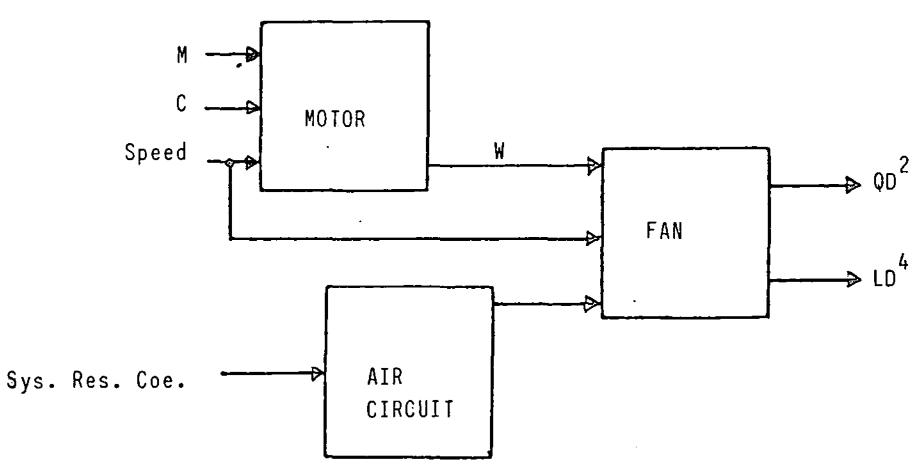


FIGURE 4.4 - STRUCTURE TYPE 4

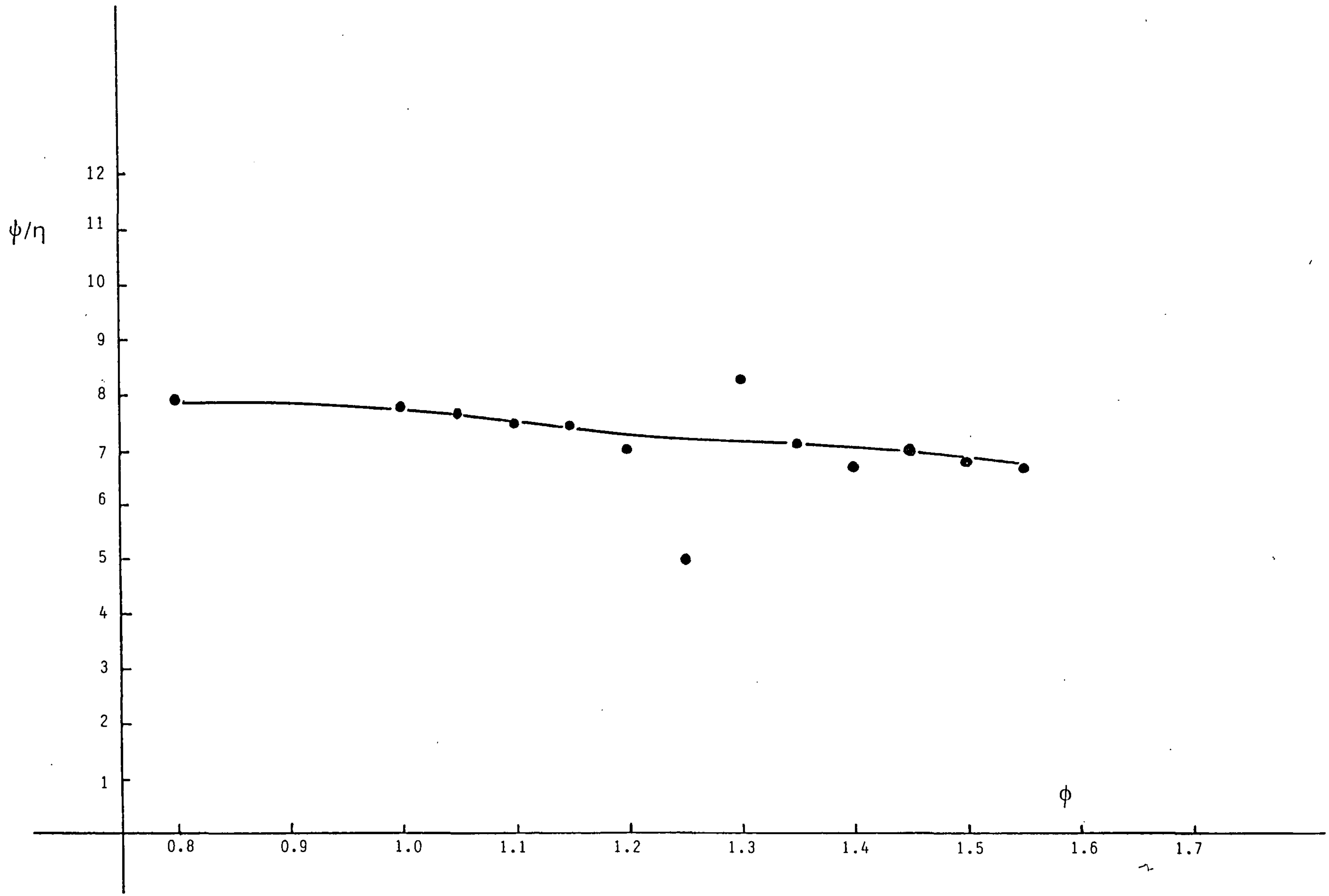


FIGURE 4.5 FAN CONFIGURATION 1 - RANGE OF UNPREDICTABLE ERROR

4.1.3.6 FLOW CHARTS

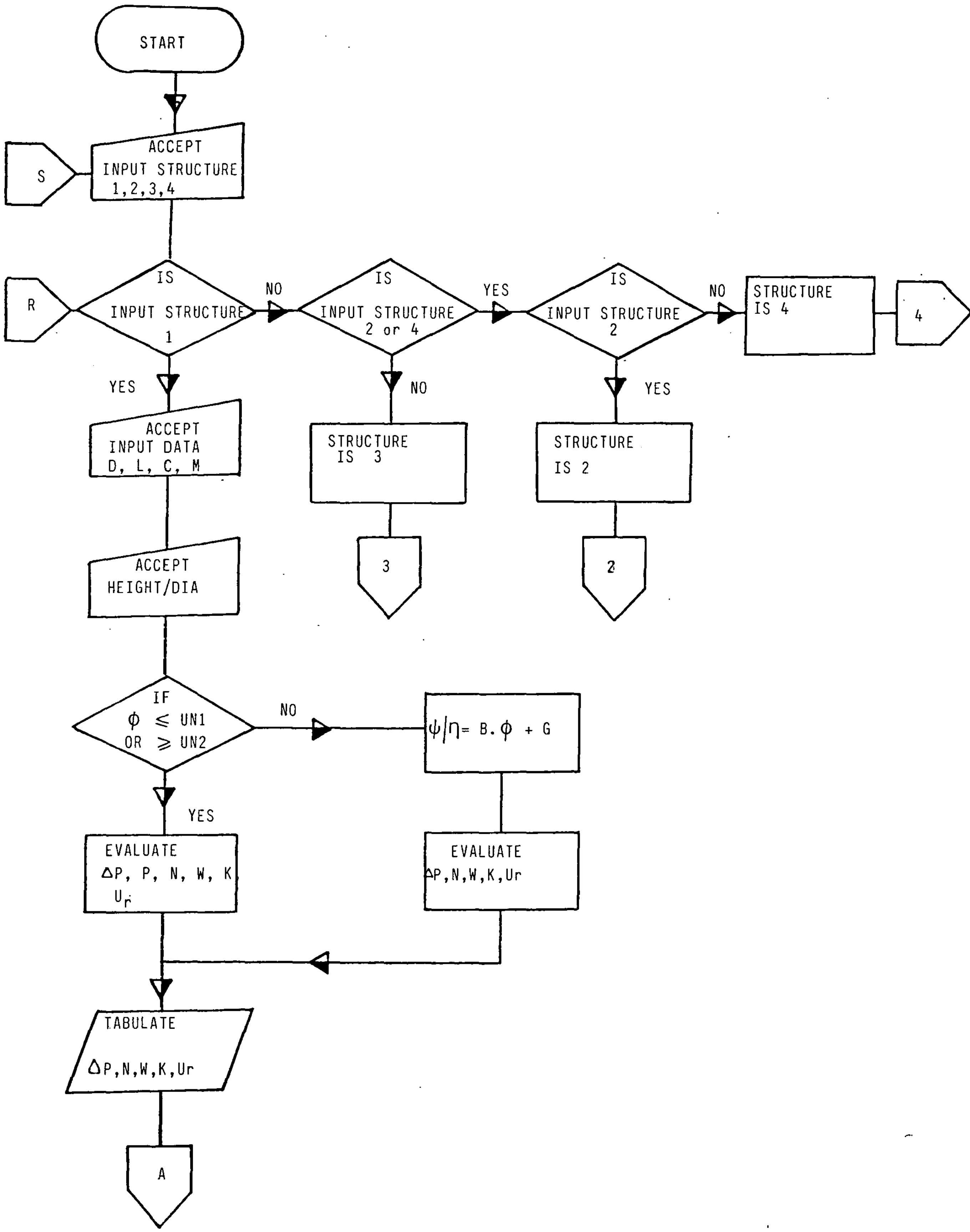


FIGURE 4.6 BEGINNING OF PROGRAMME

4.1.3.6 FLOW CHARTS - CONTINUED

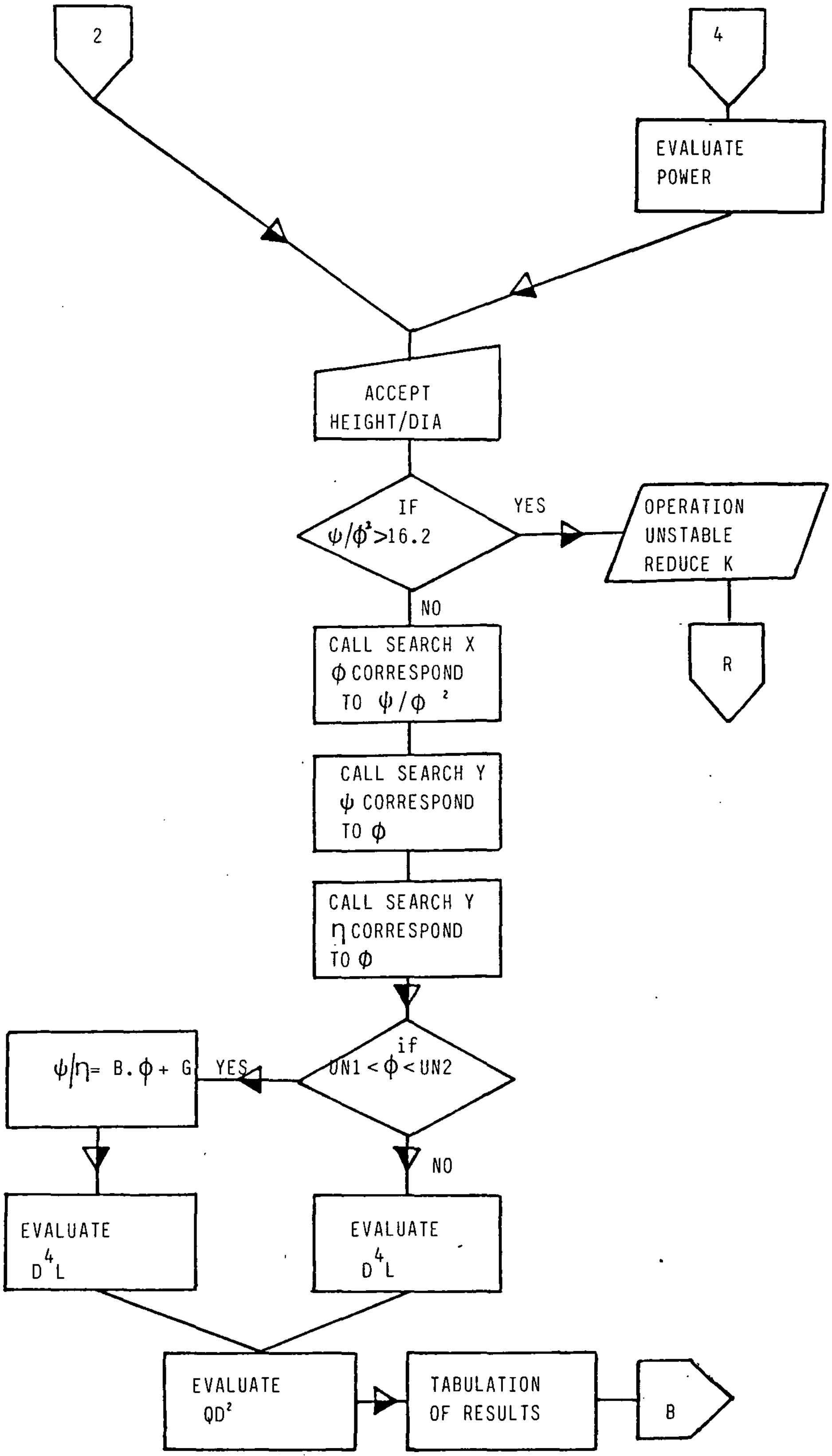


FIGURE 4.7 MID - PROGRAMME

4.1.3.6 FLOW CHARTS - CONTINUED

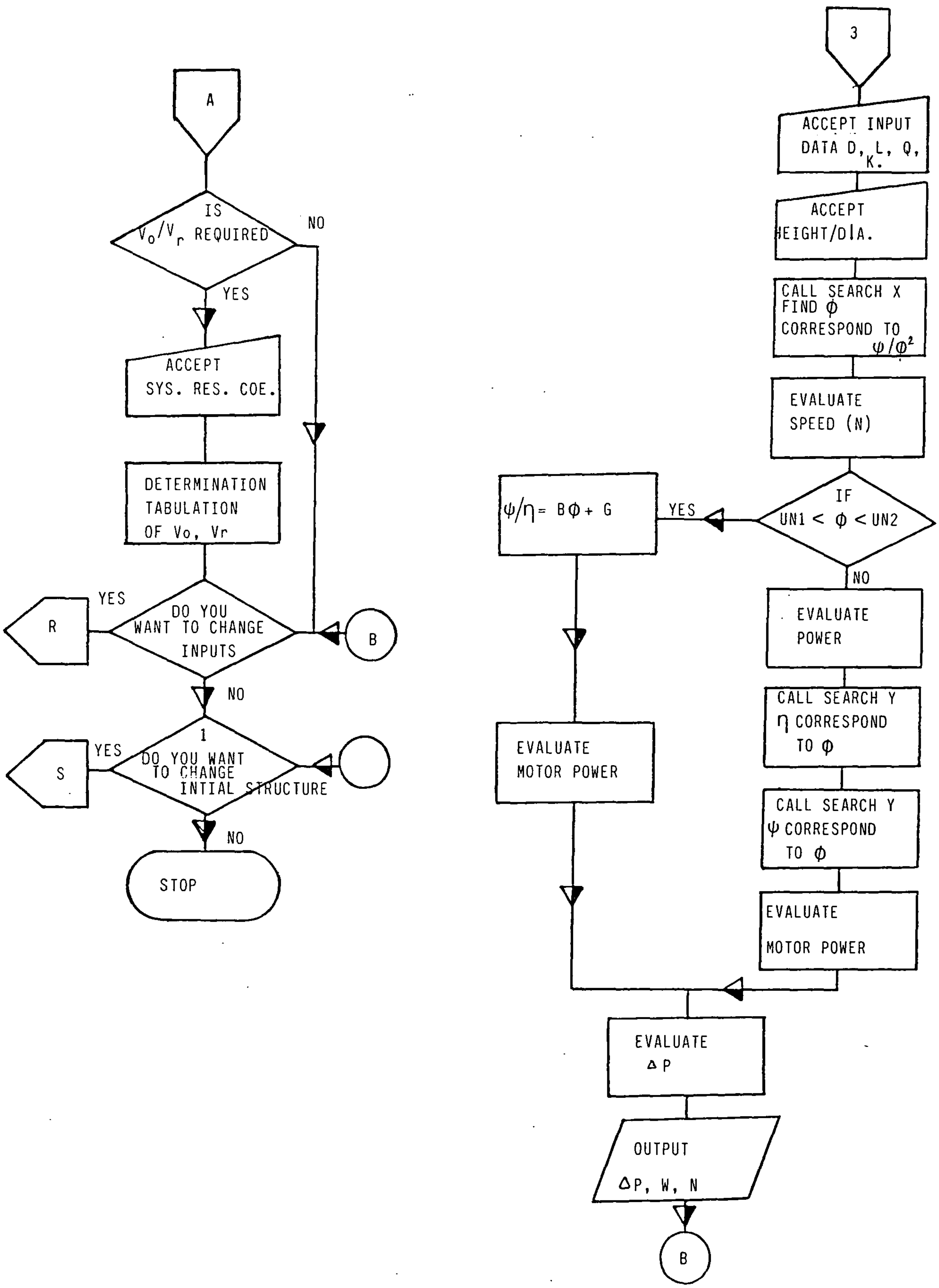


FIGURE 4.8 END OF PROGRAMME

TABLE 4.9 - MATCH 1

DIAMETER=0.080METERS;   LENGTH=0.400METERS
FAN TORQUE=-0.118E-02 .SPEED+3.30 N.METERS.
HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.10	176.7	2541	80.2	234.6	1.12
0.35	0.26	0.12	175.9	2511	88.6	175.7	1.29
0.40	0.29	0.13	175.1	2505	90.2	134.5	1.47
0.45	0.30	0.15	172.3	2485	95.7	106.3	1.64
0.50	0.32	0.17	168.3	2479	97.3	84.5	1.82
0.55	0.33	0.18	163.5	2468	100.2	68.5	2.00
0.60	0.33	0.20	158.0	2450	104.9	56.4	2.16
0.65	0.33	0.21	152.8	2433	109.3	47.1	2.33
0.70	0.32	0.23	144.4	2415	113.9	39.0	2.49
0.75	0.30	0.24	132.9	2396	118.6	31.8	2.64
0.80	0.28	0.25	124.1	2369	125.2	26.7	2.79
0.85	0.26	0.27	113.4	2349	129.9	21.9	2.94
0.90	0.24	0.28	103.0	2331	134.1	18.0	3.08
0.95	0.21	0.29	89.7	2307	139.5	14.4	3.22
1.00	0.18	0.31	76.8	2282	145.1	11.4	3.35
1.05	0.15	0.32	62.4	2270	147.7	8.5	3.50
1.10	0.12	0.33	48.4	2261	149.6	6.0	3.66
1.15	0.08	0.35	31.8	2245	153.0	3.7	3.79
1.20	0.05	0.36	18.7	2254	151.1	2.0	3.98
1.25	0.01	0.38	2.6	2239	154.2	0.3	4.11
1.30	-0.03	0.39	-13.0	2225	157.1	-1.2	4.25
1.35	-0.07	0.40	-25.5	2202	161.8	-2.2	4.37
1.40	-0.11	0.41	-40.1	2181	165.9	-3.3	4.49
1.45	-0.16	0.43	-53.9	2208	160.6	-4.1	4.71
1.50	-0.20	0.44	-67.8	2183	165.5	-4.9	4.81
1.55	-0.25	0.46	-80.9	2191	163.9	-5.4	4.99
1.60	-0.31	0.47	-96.5	2196	162.9	-6.0	5.16
1.65	-0.36	0.49	-109.0	2194	163.3	-6.4	5.32
1.70	-0.41	0.50	-122.9	2182	165.7	-6.9	5.45
1.75	-0.46	0.51	-134.9	2178	166.5	-7.2	5.60
1.80	-0.52	0.52	-149.2	2173	167.4	-7.5	5.75
1.85	-0.58	0.54	-161.8	2174	167.2	-7.7	5.91
1.90	-0.64	0.55	-175.7	2168	168.4	-8.0	6.05
1.95	-0.71	0.57	-190.3	2166	168.8	-8.2	6.21
2.00	-0.78	0.58	-203.1	2169	168.2	-8.3	6.38

SYSTEM RESISTANCE COEFFICIENT=   17.3

VR	VO	VR	VO	VR	VO	VR	VO	VR	VO	VR	VO
3.2	5.5	3.4	8.2	3.5	10.4	3.7	12.3	3.8	14.0	4.0	15.6
4.1	17.0	4.3	18.3	4.4	19.3	4.5	20.4	4.7	21.7	4.8	22.7
5.0	23.8	5.2	24.9	5.3	25.9	5.5	26.8	5.6	27.7	5.8	28.7

VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.

TABLE 4.10

DIAMETER=0.090METERS: LENGTH=0.300METERS
FAN TORQUE=-0.118E-02 .SPEED+3.30 N.METERS.
HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.10	216.4	2499	91.9	234.6	1.24
0.35	0.26	0.11	214.7	2465	101.0	175.7	1.43
0.40	0.29	0.13	213.6	2459	102.6	134.5	1.63
0.45	0.30	0.14	209.6	2436	108.5	106.3	1.81
0.50	0.32	0.15	204.7	2430	110.1	84.5	2.01
0.55	0.33	0.17	198.6	2417	113.4	68.5	2.20
0.60	0.33	0.18	191.5	2398	118.1	56.4	2.38
0.65	0.33	0.20	184.9	2379	122.7	47.1	2.56
0.70	0.32	0.21	174.4	2360	127.3	39.0	2.73
0.75	0.30	0.22	160.2	2338	132.5	31.8	2.90
0.80	0.28	0.24	149.2	2309	139.1	26.7	3.05
0.85	0.26	0.25	136.0	2287	144.0	21.9	3.22
0.90	0.24	0.26	123.3	2267	148.3	18.0	3.37
0.95	0.21	0.27	107.1	2242	153.6	14.4	3.52
1.00	0.18	0.28	91.5	2215	159.2	11.4	3.66
1.05	0.15	0.29	74.3	2201	162.0	8.5	3.82
1.10	0.12	0.31	57.6	2192	163.7	6.0	3.99
1.15	0.08	0.32	37.8	2174	167.2	3.7	4.14
1.20	0.05	0.33	22.3	2185	165.1	2.0	4.34
1.25	0.01	0.34	3.1	2169	168.2	0.3	4.48
1.30	-0.03	0.36	-15.4	2153	171.2	-1.2	4.63
1.35	-0.07	0.37	-30.2	2129	175.6	-2.2	4.75
1.40	-0.11	0.38	-47.3	2107	179.5	-3.3	4.88
1.45	-0.16	0.39	-63.8	2135	174.5	-4.1	5.12
1.50	-0.20	0.40	-80.1	2109	179.2	-4.9	5.23
1.55	-0.25	0.42	-95.6	2118	177.6	-5.4	5.43
1.60	-0.31	0.43	-114.1	2122	176.9	-6.0	5.62
1.65	-0.36	0.45	-128.8	2120	177.2	-6.4	5.79
1.70	-0.41	0.46	-145.1	2108	179.3	-6.9	5.93
1.75	-0.46	0.47	-159.2	2103	180.2	-7.2	6.09
1.80	-0.52	0.48	-176.1	2099	180.9	-7.5	6.25
1.85	-0.58	0.49	-190.9	2099	180.9	-7.7	6.42
1.90	-0.64	0.51	-207.2	2092	182.1	-8.0	6.57
1.95	-0.71	0.52	-224.4	2091	182.3	-8.2	6.74
2.00	-0.78	0.53	-239.5	2094	181.8	-8.3	6.92

SYSTEM RESISTANCE COEFFICIENT= 17.3

VR	VO	VR	VO	VR	VO	VR	VO	VR	VO	VR	VO
3.5	6.0	3.7	8.9	3.8	11.4	4.0	13.4	4.1	15.3	4.3	17.0
4.5	18.5	4.6	19.9	4.8	21.0	4.9	22.1	5.1	23.7	5.2	24.6
5.4	25.9	5.6	27.1	5.8	28.2	5.9	29.1	6.1	30.1	6.2	31.1

VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.

TABLE 4.11

SHAFT POWER=130.0WATTS   SPEED=2250.0RPM SYS.RESISTANCE COE.= 11.0

HEIGHT/DIAMETER=2.85

DIA. TO THE POWER 4, INTO LENGTH=0.152E-04 .

DIAMETER METERS	LENGTH METERS	FLOW RATE. M3/S
0.050	2.437	0.724
0.060	1.175	0.503
0.070	0.634	0.370
0.080	0.372	0.283
0.090	0.232	0.224
0.100	0.152	0.181
0.110	0.104	0.150
0.120	0.073	0.126
0.130	0.053	0.107

TABLE 4.12

SPEED=2250.0 RPM   SYS.RES.COE.= 11.0

FAN TORQUE=-0.118E-02 .SPEED+3.30

HEIGHT/DIAMETER=2.85

DIA. TO THE POWER 4, INTO LENGTH=0.178E-04 .

DIAMETER METERS	LENGTH METERS	FLOW RATE. M3/S
0.050	2.848	0.847
0.060	1.374	0.588
0.070	0.741	0.432
0.080	0.435	0.331
0.090	0.271	0.261
0.100	0.178	0.212
0.110	0.122	0.175
0.120	0.086	0.147
0.130	0.062	0.125

TABLE 4.13

```
DIAMETER=0.090 METERS   LENGTH=0.350 METERS
FLOWRATE=0.300 M3/S     SYS. RESISTANCE COE.=  17.3*

HEIGHT/DIAMETER=2.85

          FAN PRESSURE RISE= 115.9 PA.

          SPEED             = 2222. RPM.

          MOTOR SHAFT POWER=  164.8 WATTS.
```

```
DIAMETER=0.090 METERS   LENGTH=0.350 METERS
FLOWRATE=0.370 M3/S     SYS. RESISTANCE COE.=   2.0*

HEIGHT/DIAMETER=2.85

          FAN PRESSURE RISE=  20.4 PA.

          SPEED             = 2079. RPM.

          MOTOR SHAFT POWER=  166.0 WATTS.
```

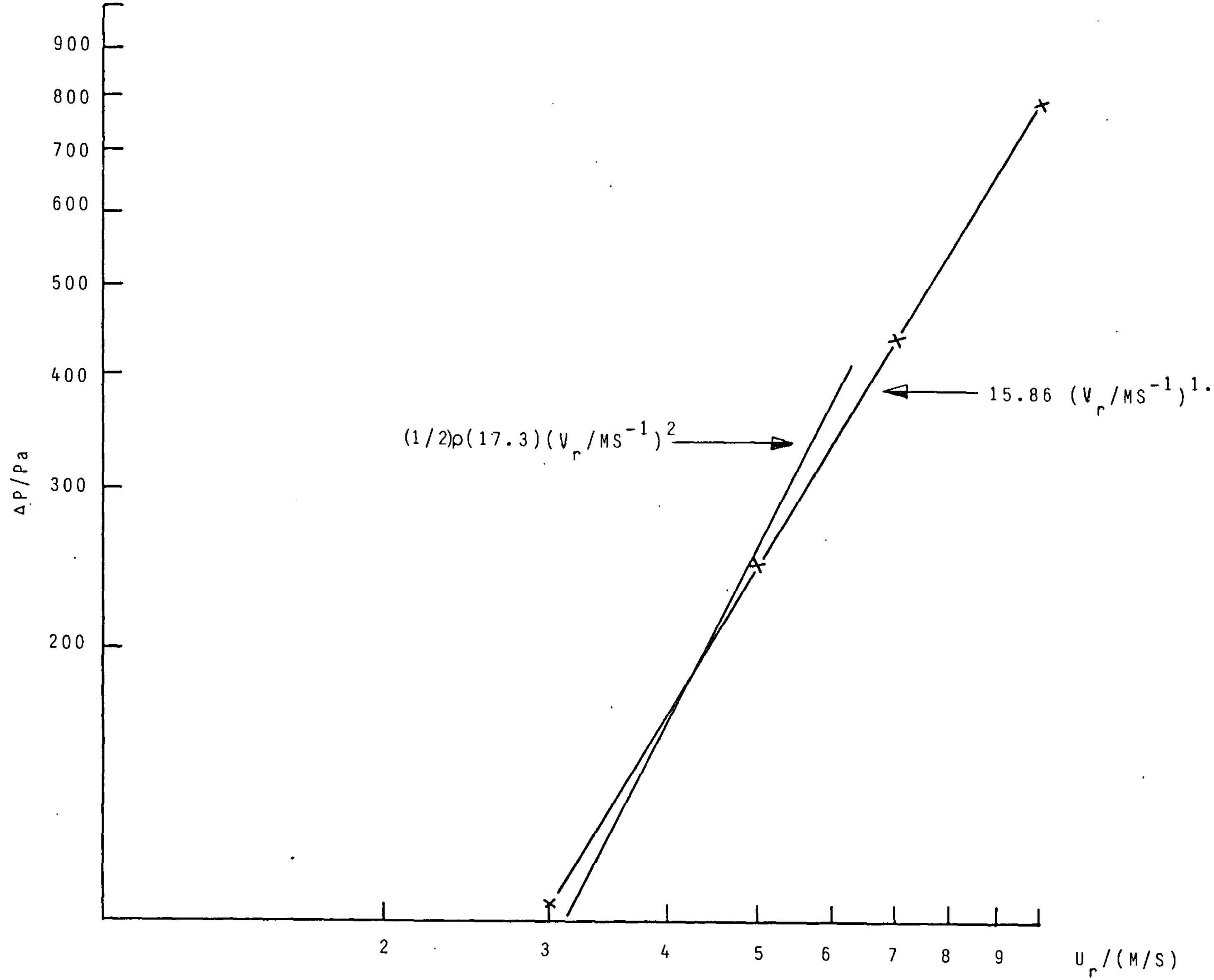


FIGURE 4.14 RADIATOR PRESSURE DROP CHARACTERISTIC

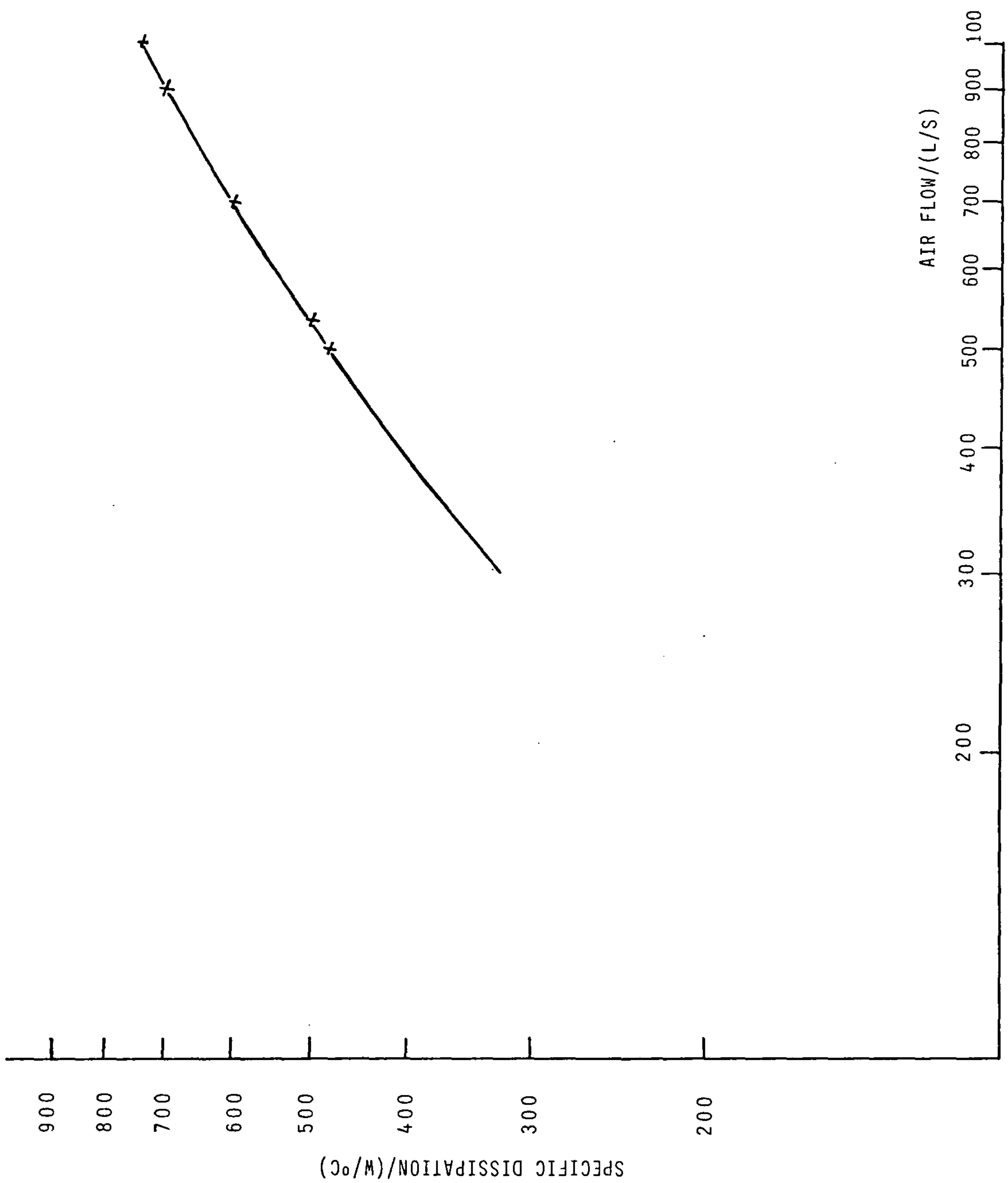
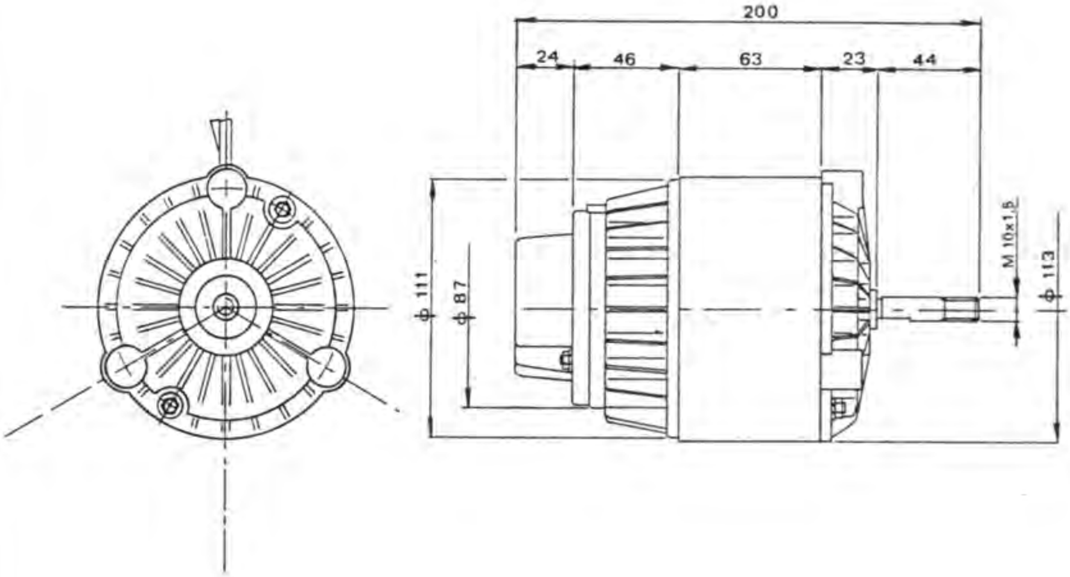


FIGURE 4.15 RADIATOR HEAT DISSIPATION

GATE
ASTI - ITALY

FIGURE 4.16

MP 8140/C



Caratteristiche

Tensione nominale 24 V
Giri nominali 2250
Corrente nominale 8 A
Potenza resa 150 W
Senso di rotazione reversibile
Servizio continuo
Temperatura di funzionamento
−40°C +80°C

Caractéristiques

Tension nominale 24 V
Tours minute 2250
Courant nominale 8 A
Puissance rendue 150 W
Sens de rotation réversible
Service continu
Temperature de travail
−40°C +80°C

Features

Rated voltage 24 V
R.P.M. 2250
Rated current 8 A
Effective Horsepower 150 W
Direction of rotation reversible
Service continuous
Working temperature
−40°C +80°C

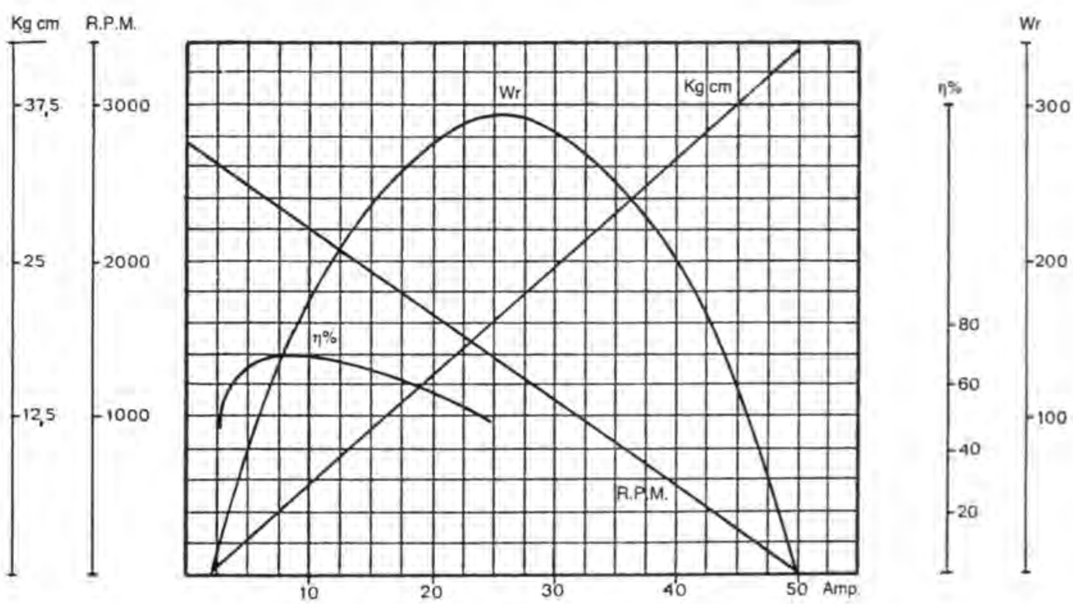


TABLE 4.18

DIAMETER=0.100METERS;   LENGTH=0.250METERS
FAN TORQUE=-0.118E-02 .SPEED+3.30 N.METERS.
HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.10	254.1	2437	108.3	234.6	1.34
0.35	0.26	0.11	250.9	2399	117.8	175.7	1.54
0.40	0.29	0.13	249.3	2391	119.8	134.5	1.76
0.45	0.30	0.14	244.0	2365	126.1	106.3	1.96
0.50	0.32	0.15	238.0	2358	127.8	84.5	2.17
0.55	0.33	0.17	230.5	2344	131.1	68.5	2.37
0.60	0.33	0.18	221.7	2322	136.2	56.4	2.56
0.65	0.33	0.20	213.5	2301	140.9	47.1	2.75
0.70	0.32	0.21	200.8	2279	145.7	39.0	2.93
0.75	0.30	0.22	184.0	2255	150.9	31.8	3.11
0.80	0.28	0.23	170.6	2222	157.7	26.7	3.27
0.85	0.26	0.24	155.2	2199	162.4	21.9	3.43
0.90	0.24	0.26	140.3	2177	166.7	18.0	3.60
0.95	0.21	0.27	121.6	2149	171.9	14.4	3.75
1.00	0.18	0.28	103.5	2119	177.4	11.4	3.89
1.05	0.15	0.29	83.9	2105	179.9	8.5	4.06
1.10	0.12	0.30	65.0	2095	181.6	6.0	4.23
1.15	0.08	0.31	42.6	2076	184.8	3.7	4.39
1.20	0.05	0.33	25.1	2087	183.0	2.0	4.60
1.25	0.01	0.34	3.5	2070	185.8	0.3	4.75
1.30	-0.03	0.35	-17.3	2053	188.6	-1.2	4.90
1.35	-0.07	0.36	-33.8	2028	192.6	-2.2	5.03
1.40	-0.11	0.37	-52.9	2004	196.2	-3.3	5.16
1.45	-0.16	0.39	-71.5	2034	191.6	-4.1	5.42
1.50	-0.20	0.39	-89.4	2007	195.8	-4.9	5.53
1.55	-0.25	0.41	-107.0	2016	194.4	-5.4	5.74
1.60	-0.31	0.42	-127.7	2021	193.7	-6.0	5.94
1.65	-0.36	0.44	-144.1	2019	194.0	-6.4	6.12
1.70	-0.41	0.45	-162.1	2005	196.1	-6.9	6.26
1.75	-0.46	0.46	-177.7	2000	196.8	-7.2	6.43
1.80	-0.52	0.47	-196.5	1995	197.6	-7.5	6.60
1.85	-0.58	0.48	-213.1	1996	197.4	-7.7	6.79
1.90	-0.64	0.49	-231.0	1989	198.5	-8.0	6.94
1.95	-0.71	0.51	-250.2	1987	198.7	-8.2	7.12
2.00	-0.78	0.52	-267.1	1990	198.3	-8.3	7.31

SYSTEM RESISTANCE COEFFICIENT=   17.3

VR	VO	VR	VO	VR	VO	VR	VO	VR	VO	VR	VO
3.8	6.4	3.9	9.5	4.1	12.1	4.2	14.2	4.4	16.2	4.6	18.0
4.8	19.6	4.9	21.1	5.0	22.2	5.2	23.4	5.4	25.0	5.5	26.0
5.7	27.4	5.9	28.7	6.1	29.8	6.3	30.8	6.4	31.8	6.6	32.9

VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.

TABLE 4.19

DIAMETER=0.100METERS;   LENGTH=0.330METERS

FAN TORQUE=-0.118E-02 .SPEED+3.30 N.METERS.

HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.12	237.1	2354	128.7	234.6	1.30
0.35	0.26	0.14	232.6	2310	138.9	175.7	1.49
0.40	0.29	0.16	230.9	2301	140.9	134.5	1.69
0.45	0.30	0.18	225.0	2272	147.3	106.3	1.88
0.50	0.32	0.20	219.3	2264	149.0	84.5	2.08
0.55	0.33	0.21	211.9	2247	152.6	68.5	2.27
0.60	0.33	0.23	203.1	2222	157.7	56.4	2.45
0.65	0.33	0.25	195.0	2199	162.4	47.1	2.63
0.70	0.32	0.26	182.9	2175	167.0	39.0	2.80
0.75	0.30	0.28	167.0	2148	172.1	31.8	2.96
0.80	0.28	0.29	154.1	2112	178.6	26.7	3.10
0.85	0.26	0.31	139.7	2087	183.0	21.9	3.26
0.90	0.24	0.32	126.0	2062	187.1	18.0	3.41
0.95	0.21	0.33	108.7	2032	192.0	14.4	3.55
1.00	0.18	0.35	92.2	2000	196.8	11.4	3.68
1.05	0.15	0.36	74.6	1985	199.0	8.5	3.83
1.10	0.12	0.38	57.7	1974	200.6	6.0	3.99
1.15	0.08	0.39	37.7	1954	203.4	3.7	4.13
1.20	0.05	0.41	22.3	1966	201.7	2.0	4.34
1.25	0.01	0.42	3.1	1948	204.2	0.3	4.47
1.30	-0.03	0.43	-15.3	1930	206.6	-1.2	4.61
1.35	-0.07	0.44	-29.8	1903	210.1	-2.2	4.72
1.40	-0.11	0.45	-46.4	1878	213.1	-3.3	4.83
1.45	-0.16	0.48	-63.0	1909	209.3	-4.1	5.09
1.50	-0.20	0.49	-78.6	1881	212.8	-4.9	5.18
1.55	-0.25	0.51	-94.1	1890	211.7	-5.4	5.38
1.60	-0.31	0.52	-112.3	1895	211.1	-6.0	5.57
1.65	-0.36	0.54	-126.8	1893	211.3	-6.4	5.74
1.70	-0.41	0.55	-142.4	1879	213.0	-6.9	5.87
1.75	-0.46	0.57	-156.0	1874	213.6	-7.2	6.03
1.80	-0.52	0.58	-172.4	1869	214.2	-7.5	6.18
1.85	-0.58	0.60	-186.9	1870	214.1	-7.7	6.36
1.90	-0.64	0.61	-202.5	1862	215.0	-8.0	6.50
1.95	-0.71	0.63	-219.3	1860	215.2	-8.2	6.67
2.00	-0.78	0.64	-234.2	1863	214.9	-8.3	6.85

SYSTEM RESISTANCE COEFFICIENT=   5.2

VR	VO	VR	VO	VR	VO	VR	VO	VR	VO	VR	VO
4.1	5.1	4.3	7.8	4.5	9.9	4.6	11.7	4.7	12.9	4.8	14.1
5.1	15.5	5.2	16.5	5.4	17.5	5.6	18.7	5.7	19.6	5.9	20.4

VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.

CHAPTER FIVE

DISCUSSION

5.0 Discussion

5.1 Discussion of Results

Previous researchers have tended to concentrate on fans with no inlet restriction and performance in the ranges of flow coefficient ϕ = 0.4 to 0.8. The purpose of this research was to relate the performance of cross flow fans to the requirements of automotive applications, therefore the fan performance under negative pressure rises for values of ϕ = 0.4 to 2.0 and its resistance, when un-energised, were also investigated. Preliminary investigations indicated that the undriven fan's resistance to ram air, mainly due to rotor obstruction, was relatively high. In addition to this, the power rising with flow rate, associated with cross flow machines, necessitated the incorporation of some means of a balancing valve to allow airflow to by-pass this obstruction, under these conditions.

The operation of this balancing valve ('by pass' flap) was ascertained to be stable and smooth, even under instantaneous heavy changes of load. In a real situation these instantaneous loadings might be caused by, vehicle acceleration and deceleration manoeuvre heavy wind effects, or passing vehicles. This 'by pass' flap was extremely beneficial to both the performance and in reducing resistance to airflow. Tests have indicated that the use of rubber for the whole flap was not viable for configuration 1.

Interestingly, increasing the flap moment so as to close it, improved the fan performance and efficiencies. But there were limitations to this. If overdone, the flap operation would become unstable with violent fluctuations. The opening of the flap causes the fan to discharge a jet of air at higher velocity than it would have done when the flap is closed, resulting in a venturi effect that would tend to open the flap more. This causes a reduction in the fan's contribution to airflow which could probably be the reason for the overall reduction indicated in airflow.

All of the quantitative data was obtained from the large rotor (D = 0.1m) and the small rotor (D = 0.06m) was only used for the scaling purposes. The results indicated that the novel cross flow fan does satisfy the similarity relationship aerodynamically and geometrically between fan Reynold's numbers 4.7×10^3 to 6.8×10^3 and flow coefficients of 0.4 to 2.0. It was not possible to determine performance below $\phi = 0.4$ due to heavy fluctuations making the measurement difficult. The test airway was also constructed to predict installed performance accurately. No flow instability was detected during testing except below $\phi = 0.4$. Fan configuration 1 demonstrated to perform better than 2.

The following features were found to be also desirable:-

i) Inlet transition to fan configuration 1 as in Figure 3.20.

ii) Outlet duct's back wall in fan configuration 1 curtailed as in Figure 3.40, if installation space allows for unobstructed discharge;

iii) Height to diameter ratio (H_i/D) between 2.45 and 2.85;

iv) The value of $D^2 Lgl/EI$ around $0.12 m^3/kg$ (for the plywood flap of $670 kg/M^3$ and 1.86mm density and thickness).

The pressure drop of the normal novel fan configuration 1, when rotor unenergised, satisfied the expression; $\Delta P/kPa = 0.486 (Q/m^3 s^{-1})^2$. Most of this was due to the kinetic energy loss at the exit. Exit to inlet velocity ratio was quite high ($V_e/V_r = 3.56$) due to the fan casing geometry. The unit could be suitably installed to utilize this high exit velocity to reduce the loss of air momentum to a minimum which would result in an equivalent drop (equation 3 of Section 2.3.1) in cooling drag. When the flap was restricted to closed position as a static vortex wall, the pressure drop increased to $\Delta P/kPa = 0.968 (Q/m^3 s^{-1})^2$.

The lowest resistance to airflow, when the rotor was unenergised, was indicated in fan configuration 1 for the geometry of curtailed outlet duct's back wall as $\Delta P/kPa = 0.380 (Q/m^3 s^{-1})^2$.

The pressure drop of the normal novel fan configuration 2, when rotor unenergised, satisfied the expression $\Delta P/kPa = 1.125 (Q/m^3 s^{-1})^2$, but when the flap was restricted to closed position, as a static vortex wall, this value increased to $\Delta P/kPa = 2.22 (Q/m^3 s^{-1})^2$.

The fan performance of configuration 1 was not adversely affected by the interaction of radiator core and fan air inflow geometry.

A detailed measurement of noise was not attempted. But that observed with the small fan, was found to be acceptable, except for a tendency to show high tone noise near $\phi = 0.4$ (ie; for highly throttled conditions).

The design point for the cooling system was chosen as the point of maximum bhp in the second gear. Though this can be considered as a good estimate, it would overdesign the system to some extent, since conditions of this nature would be hardly faced in real situations.

Therefore, fine tuning of a cooling system is only possible, when more information regarding the heat rejection characteristics of the engine such as 'Ratio of Heat Loss to Power', is known.

Analysis was conducted to determine the conditions involved in the appreciation of overall system performance and efficiency. The fan power is closely proportional to the cube of airflow velocity. Since the rate of heat removal by the radiator varies as the air velocity to the power between 0.65 to 0.8,[17,55] realization of cooling system performance purely by an increase in face velocity of a particular radiator is not desirable, but should be achieved through fin effectiveness to heat dissipation and airflow. The mathematical analysis also indicated:

i) Utilisation of ram air would be desirable in an electric fan system, but not in a system where the fan is driven by the engine;

ii) Fan/Motor interaction can be defined by an equation of the form $Q = f(\psi, \phi, C, M, \eta)$;

iii) Fan/Duct interaction can be well approximated by an equation of the form $\psi/\phi^2 = g(k, D, H)$

A simple mathematical model of a vehicle cooling system was produced to predict the system behaviour and to match its components via computer simulation, solved using the computer language Fortran 77. But thermal characteristics of radiator were excluded since it is beyond the main interests of these investigations. The computer program utilizes experimentally found performance characteristics for the fan and the air circuit. This model may be extended to include the heat balance as an integral sub-model.

The following combinations of components were found to satisfy the cooling target of second gear maximum bhp.

i) Fan: D = 0.08m, L = 0.4m, $N \simeq 2200$rpm

Radiator: x84FB - 8005-K40, area = $0.155m^2$

Motor: Gate MP8140/C; T = 0.00118N + 3.3;

ii) Fan: D = 0.09m, L = 0.3m $N \simeq 2100$ rpm

Radiator: x84CF-8005 - K40, area = $0.155m^2$

Motor: Gate MP8140/C; T = -0.00118N+3.3

Values of physical properties of atmospheric air, where applicable, were approximated to standard air of 20°C and 1 bar and the standard gravitational acceleration was taken as $9.8\ m/s^2$.

5.2 Constructional Features

Good manufacture is an important parameter for flow stability in cross flow fans. If this is not fulfilled, the cyclic variations in loading caused by these flow instabilities would induce large vibratory stresses in the blading, leading to impeller destruction. Availability of cheap cross flow fan units for domestic HVAC applications indicate, achieving cost effective, well manufactured units of these, as a mass produced item, are not difficult. These cross flow fan units were even able to compete successfully with its counterparts for these applications, especially for packaged compact units.

The novel fan configurations, as well, could easily adopt same simple constructional techniques for their casings. But application in question needs a strengthened rotor to withstand impact loadings associated with vehicle acceleration and deceleration manoeuvres, passing vehicles, and splashing of water when driven over a water puddle, or through a ford.

Therefore, especially for cross flow fans, where only one end is driven by a motor and the other end is forced to follow by the connecting blades, the rotor needs a high ratio of strength to moment of inertia.

Nowadays, the advent of inexpensive, strong materials for moulded plastic fans would satisfy this condition easily. And the initial tooling costs would be reduced to a minimum in applications such as these, where large quantities of units are involved. One of the indirect advantage associated with plastic fans are that they have very high noise damping properties compared with metal. In fact, use of plastics for automotive impellers are almost universal now in European vehicles. Stronger plastics capable of higher operating temperatures are also becoming available to meet more demanding requirements (eg, glass reinforced thermoplastic polyesters, polyphenylone sulphides and polyether sulphones[23]). Therefore, in this present age of plastics, there is no reason why a cost effective appropriate manufacturing technique for moulded plastic, for high strength and low moment of inertia, can not be adopted to form these rotors.

5.3 Design Summary

1) The casing can be made with simple constructional methods, keeping

also the geometry of the two configurations to their established design limits.

2) The fan configuration 1, which performs better than 2, should be preferred.

3) The curtailed back wall of fan configuration 1 can be used, if installation space allows for proper air discharge. But this would reduce the exit air momentum and increase the momentum loss.

4) Air inlet should be located in an over pressure area, while the outlets should open into zones, where low pressures exist.

5) The best engine cooling arrangement emerges as in Figure 5.1 which incorporates all ascertained advantageous aspects.

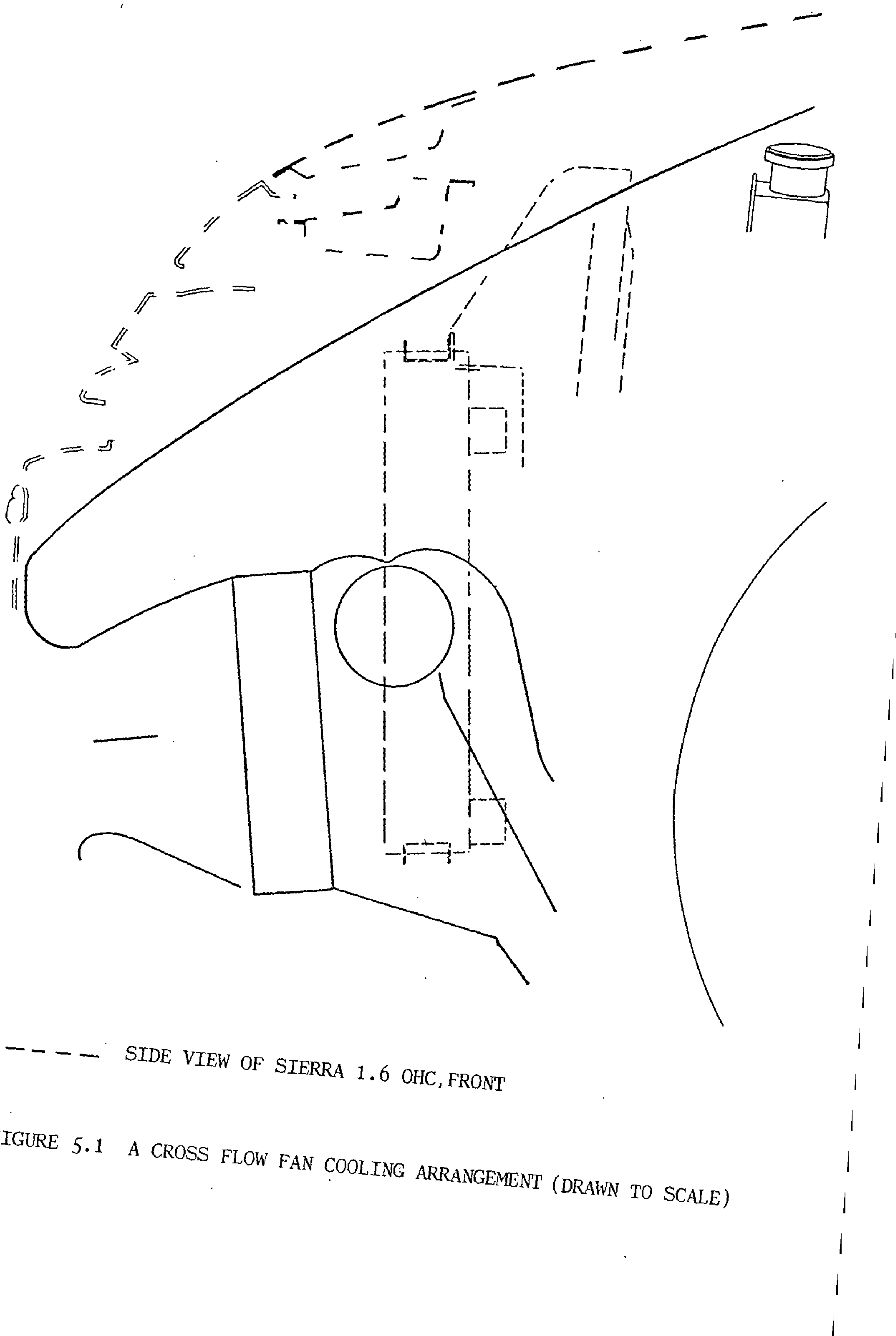
6) Inlet diffuser need not be long. Satisfactory inlet ducts need only be of a length equal to 0.6 times the radiator height and there is no benefit to be gained by exceeding a length of 1.2 times the radiator height.[47]

7) The inefficient conventional radiator grilles should be replaced by splitter vanes.

8) Fan would be best cut off under top gear conditions.

9) Fan 'bypass' flap can be thermostatically controlled to restrict the opening as necessary. This will avoid overcooling of the engine in some atmospheric and vehicle conditions.

10) Alternatively, the cross flow rotor can be also mounted vertical. Figure 5.2 demonstrates an example of this installation attitude.



– – – – SIDE VIEW OF SIERRA 1.6 OHC, FRONT

FIGURE 5.1 A CROSS FLOW FAN COOLING ARRANGEMENT (DRAWN TO SCALE)

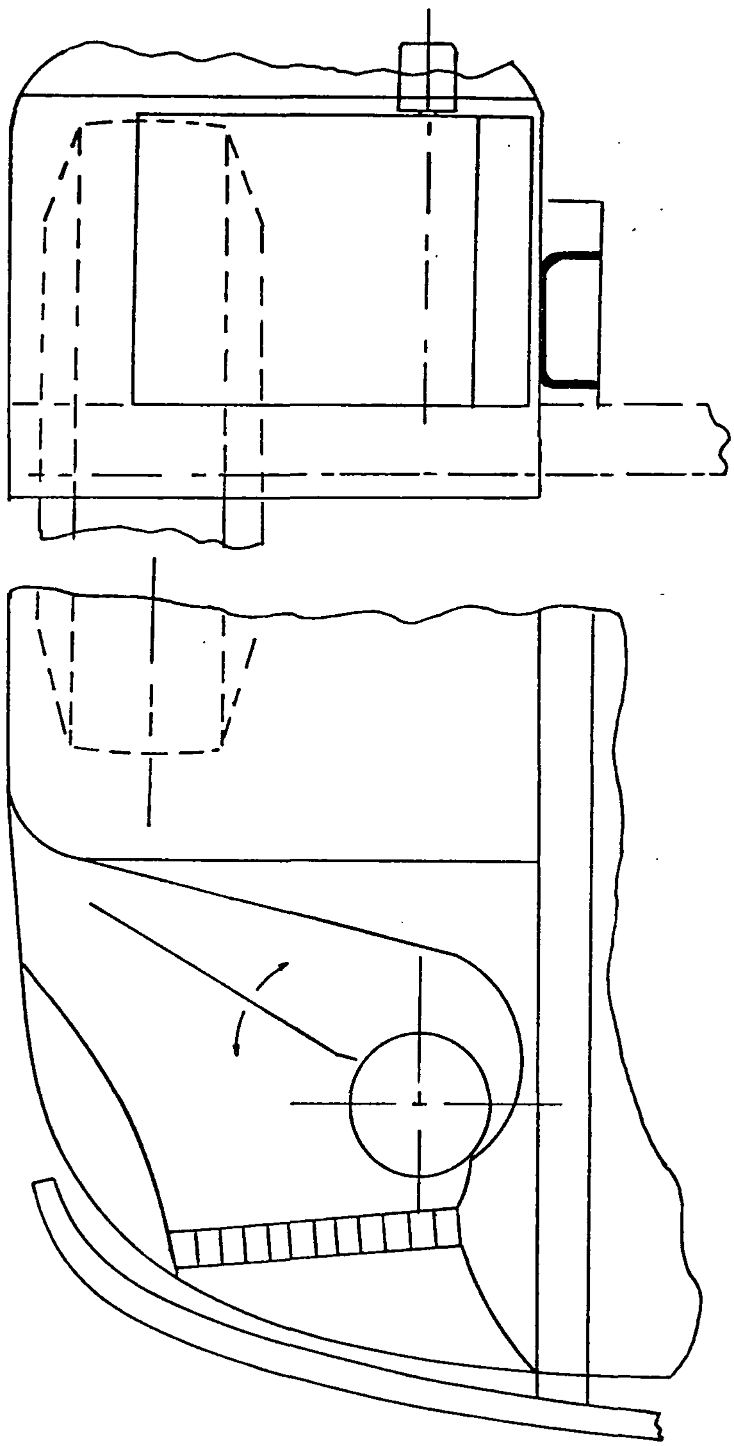


FIGURE 5.2 A CROSS FLOW FAN MOUNTED VERTICAL

CHAPTER SIX

CONCLUSIONS

6.0 Conclusions

The novel cross flow fan configurations should offer the following major advantages in terms of both installation and aerodynamics. A number of relatively minor disadvantages are also listed.

6.1 Installation Advantages

1) The novel cross flow fan configuration adds another degree of design flexibility, since its inlet and outlet are rectangular and it can be used in either 'S' or 'L' airflow configurations.
2) It incorporates ducting, that could direct airflow from and towards preferred locations.
3) The overall shape of the casing would reduce the ducting volume to a minumum.
4) Substantial simplifications to assembly of the engine cooling system to the motor car may be achieved by combining the components to form a pre-assembled package. This may be constructed in a suitable manner associated with a front spoiler or alternatively in a system with a relatively small face area with high density radiator, so that it can be installed in passenger vehicles at the side of an engine.
5) The compact and substantially low profile total package may take up to 50% less head room than conventional systems.
6) There is a self cleaning action of blades, because of airflow reversal across each blade during each revolution.
7) It allows application of a fully tailored smooth underpan over engine irregularities which would reflect beneficially on overall

drag, damp air-borne sound from the engine, improve the engine environment and also reduce the warming up phase of the internal combustion engine in winter.

6.2 Aerodynamic Advantages

1) They allow prediction of its installed performance accurately, which would reduce development work to a minimum.
2) They have no obstruction to airflow such as motor or fan rotor.
3) Leakage of air, a source of inefficiency in conventional layouts, is eliminated.
4) They provide a fairly uniform radiator face velocity distribution.
5) Engine does not behave as an obstruction to their airflow.
6) They allow use of a proper inlet diffuser and possibly an outlet diffuser.
7) They have no restrictions upstream to cause uneven velocity distribution at the face of radiator or to recirculate (stall) air around the radiator.
8) They enable better control of air through the radiator to avoid overheating or overcooling.
9) Their exit to inlet velocity ratio (V_e/V_r) is quite high due to the casing geometry. This reduces the overall air momentum loss to a minimum.
10) A ducted system as proposed allows the choice of inlet and outlet positions on the vehicle. Correct choice allows the use of existing high and low pressure regions on the vehicle surface to maximise the ram air effect. In addition the momentum loss of the cooling air may be eliminated thereby making a significant reduction to the overall vehicle drag.

6.3 Disadvantages

One of the disadvantages of this system is that there is no direct convective heat transfer from the engine to air, since direct interaction between cooling air and the engine is not possible. This may impose a relatively higher radiator duty.

Efficiencies associated with these novel cross flow fans should not be directly compared with other conventional automotive cooling axial fans. Indiscrimate comparisons can only be possible when the efficiency of the whole axial fan cooling air circuit is considered, which also includes so many undesirable effects such as engine blockage loss, recirculation loss, outlet kinetic energy loss etc. Investigations has demonstrated that the automotive axial fans, when these are considered, quite often have lower efficiencies.

While no direct costing has been undertaken, it is probable that the total cost of a system with its ducting will exceed that of an axial flow fan with no more than a simple shroud. The increase in cost would have to be weighed against the advantages listed above.

6.4 Future Work

The following further investigation is needed:

1) A thorough investigation into fan rotor construction for strength and low inertia.
2) Quantitative observation of noise would be useful.
3) In-vehicle testing to clarify its integrity under adverse atmospheric and vehicle conditions.

REFERENCES

1. ALLEN, D.J.: 'The Effect of Rotor and Casing Geometry on the Performance of Cross Flow Fans', Ph.D. Thesis 1981, University of Durham.

2. ASHRAE HANDBOOK: American Society of Heating, Refrigerating and Air Conditioning Engineers Fundementals, 1977.

3. BUCHHEIM, DEUTENBACH and LUCKOFF, 'Necessity and Premises for Reducing the Aerodynamic Drag of Future Passenger Cars', SAE paper, 810185.

4. CARR, G.W. (1983): 'Potential for Aerodynamic Drag Reduction to Car Design', Int. J. of Vehicle Design, Technological Advances in Vehicle Design Series, SP3, Impact of Aerodynamics on Vehicle Design, pp.44-56.

5. CHIOLI, J.P.: 'Thermal Performance Deterioration in Crossflow Heat Exchanges Due to the Flow Nonuniformity', Transactions of the ASME, Vol.100, Nov. 1978.

6. COSTELLI, GIABRIELE and GIORDANENGO, 'Experimental Analysis of Engine Cooling Systems', SAE paper 790397.

7. DALY, B.B.: 'Interaction Between the Fan and the System', C110184, I. Mech. E. Conference on Installation Effects in Ducted Fan Systems, May, 1984.

8. DEEPROSE, W.M. and SMITH, T.W.: 'The Usefulness of BS 848 Part I: 1980 in Establishing the Installed Performance of a Fan', C115/84, I. Mech. E. Conference on Installation Effects in Ducted Fan Systems, May, 1984.

9. DANIEL, C. and WOOD, S.F.: 'Fitting Equations to Data', 1971, New York, Wiley-Interscience.

10. ECK, B.: 'Fans', Oxford, Pergamon Press, 1973.

11. EMMENTHAL, K.D. and HUCHO, W.H.: 'A Rational Approach to Automotive Radiator Systems Design', SAE paper 740083.

12. EMMELMANN, HUCHO and JANSSEN: 'The Optimization of Body Details. A Method for Reducing the Aerodynamic Drag of Road Vehicles', SAE publication PT.79/18.

13. EVANS and WHITTLE: 'Improving the Fuel Consumption of Heavy Goods Vehicles', Journal of the Institue of Energy, March, 1983.

14. ELECTRO-CRAFT CORPORATION, USA: 'DC Motors, Speed Controls, Servo Systems. An Engineering Handbook', Pergamon Press Ltd, 1977.

15. FERNANDO, L.M. and HOLGATE, M.J., 'The Performance and Application of Cross Flow Fans for Automotive Engine Cooling Systems', A Feasibility Study, first annual report, Ford Contract WP 013554.

16. FLEGIL, H. and BEZ, U. (1983): 'Aerodynamics - Conflict or Compliance in Vehicle Layout', Int. J. of Vehicle Design SP3, pp.9-43.

17. PRIEDE, T. and ANDERON, D.: 'Likely Advances in Mechanics, Cooling, Vibration and Noise of Automotive Engines'.

18. FORD ENERGY REPORT - 1982.

19. FORD TEST REPORT - SCF 5593, May, 1982.

20. FORD TEST REPORT - SCF 6095.

21. FORD'S CARS CATALOGUE - May/June, 1985.

22. FOX, L.R.: 'Optimization Methods for Engineering Design'.

23. HAWES, S.P.: 'Improved Passenger Car Cooling Systems', SAE paper 760112.

24. HEVAC. ASSOCIATION: 'Fan Application Guide', Heating, Ventilating and Air Conditioning Manufacturers Association Ltd., 1975.

25. HIERETH, H. and CHARZINSKI, H-P.: 'Some Aspects Concerning Noise Reduction of Passenger Cars with Diesel and Gasoline Engines', CI33/79.

26. HILL, P.G. and PETERSON, C.R.: 'Mechanics and Thermodynamics of Propulsion', Addison-Wesley, INC., 1965.

27. HOLGATE, M.J. and HAINES, P.: 'Scaling of Cross Flow Fans - an Experimental Comparison', Proc. I. Mech. E. Conference on Scaling for Performance Production in Rotodynamic Machines, Stirling, 1977.

28. HOLGATE, M.J.: 'A Cross Flow Wind Turbine', International Symposium on Wind Energy Systems, Sept., 1976.

29. IHVE GUIDE.

30. IKEGAMI, H. and MURATA, S.: 'Experimental Study of the Cross Flow Fan', Science of Machine, 18, 3, 1966.

31. ILBERGI, H. and SADEH, W.Z., 'Flow Theory and Performance of Tangential Fans', Proc. Inst. Mech. Engrs., London, 180, 1965-66.

32. JAMES, SMITH and WOLFORD: 'Applied Numerical Methods for Digital Computation with Fortran', International Textbook Co., 1967.

33. KAKAC, BERGLES and MAYINGER: 'Heat Exchanges - Thermal/Hydraulic Fundamentals and Design', Hemisphere Publishing Corp, 1981.

34. KAYS, W.M. and LONDON, A.L.: 'Compact Heat Exchangers', McGraw-Hill, Inc., 1964.

35. LAISE, MELLIN and PRYJMAK: 'Electric Cooling Fan with High Ram Airflow - A Fuel Economy Improvement', SAE paper 790722.

36. MELLIN, C.R.: 'Noise and Performance of Automotive Cooling Fans', SAE paper 800031.

37. MILLER, D.S.: 'Internal Flow Systems', BHRA Fluid Engineering Series.

38. MOORE, A.: 'The Tangential Fan - Analysis and Design', Conference on Fan Technology and Practice, Institute of Mechanical Engineers, London, April, 1972.

39. MORTIER, P.: U.S. Patent No. 507 445 (1893).

40. MURATA, S. and NISHIHARA, K.: 'An Experimental Study of Cross Flow Fan, First Report - Effects of Housing Geometry on the Fan Performance', Bulletin of J.S.M.E., 19, March 1976.

41. MURATA, S. and NISHIHARA, K.: 'An Experimental Study of Cross Flow Fan, Second Report - Movements of Eccentric Vortex Inside Impeller', Bulletin of J.S.M.E., 19 March, 1976.

42. MARRAY, W.M.: 'The Effect of Blade Angle on Cross Flow Fan Performance', Final Year Project, University of Durham, 1982.

43. OLSON, M.E.: 'Aerodynamic Effect of Front End Design on Automobile Engine Cooling Systems', SAE paper 760188, Feb., 1976.

44. OSBORNE, W.C.: 'The Selection and Use of Fans', Engineering Design Guides (33), Oxford University Press - 1979.

45. OSBORNE, W.C.: 'Fans', pergamon Press, 1966.

46. PACE, I.S.: 'Cooling Systems Performance in Fighting Vehicles', C112/84, I. Mech. E. Conference on Installation Effects in Ducted Fan Systems, May, 1984.

47. PAISH, M.G. and STAPLEFORD, W.R.: 'A Study to Improve the Aerodynamics of Engine Cooling Systems', MIRA Reports 1966/15 and 1968/4.

48. PAISH, M.G. and STAPLEFORD, W.R.: 'Rational Approach to the Aerodynamics of Engine Cooling System Design', Proc. I. Mech. E., Vo/183, 1968/69.

49. PORTER, A.M. and MARKLAND, E.: 'A Study of the Cross Flow Fan', Journal of Mechanical Engineering Science, 12, 6, 1970.

50. POULIN, J.E.: 'Practical Considerations in DC Motor and Amplifier Selection', IEEE Transactions on Industry Applications, Vol. lA-20, No.5, 1984.

51. PRESZLER, L. and LAJOS, T.: 'Experiments for the Development of the Tangential Flow Fan', Proc. 4th Conf. on Fluid Machinery, (Budapest, 1972).

52. ROUSSILLON, G. (1983): 'Aerodynamic Optimization of the Peugeot V.E.R.A. - 01', Int. J. of Vehicle Design, Technological Advances in Vehicle Design Series, SP3, Impact of Aerodynamics on Vehicle Design, pp.114-131.

53. REYNOLDS, A.J.: 'Thermofluid Dynamics', John Wiley & Sons, 1971.

54. SCHAUB, V.W. and CHARLES, H.N.: 'Ram Air Effects on the Air Side Cooling System Performance of a Typical North American Passenger Car', SAE paper 800032.

55. SCIBOR-RYLSKI, A.J.: 'Road Vehicle Aerodynamics', Pentech Press, 1984.

56. SITKEI, G.: 'Heat Transfer and Thermal Loading in Internal Combustion Engines', 1974, Budapest.

57. STAPELFORD, W.R.: 'Factors Affecting the Selection of Fans for Automotive Engine Cooling Applications', International Conference on Fan Design and Applications, paper F2, Guildford, Sept., 1982.

58. STOECKER, W.F.: 'Design of Thermal Systems', McGraw-Hill, Inc., 1971.

59. STRATTON, STRINGER and TAYLOR: 'Engine Cooling System Design and Development', Proc. Inst. of Mech. Engineers 1965-66, Vol.180, pt 2A, No.8.

60. THIEN, G.E.: Low Noise Level Cooling Systems - Fan Cooled', meeting of the FVV in Filderstadt, April, 1979.

61. TRAMPOSCH, H.: 'Cross Flow Fan', Paper A.S.M.E., No.64-WAFE - 26, 1964.

62. TUCKEY, P.R.: 'Performance and Aerodynamics of a Cross Flow Fan', Ph.D. Thesis, 1983, University of Durham.

63. WHITE, R.G. and WALKER, J.G.: 'Noise and Vibration',

64. WILLIAMS, J.: 'An Automotive Front-End Design Approach for Improved Aerodynamics and Cooling', SAE paper 850281.

65. YAMAFUJI, K.: 'Studies on the Flow of Cross Flow Impellers - 1st Report, Experimental Study', Bull. Jap. Soc. Mech. Engrs., Sept, 1975.

APPENDICES

PPENDIX 1 - COMPUTER PROGRAMME

isting of file : FMMATCH1.FTN

```
C
C      PROGRAM FMMATCH1
$BATCH
C      FAN/MOTOR MATCH
C     ---------------
C
C      DATE:- MAY 1985
C
C     DESCRIPTION:-
C     -----------
C     THE PURPOSE IS TO MATCH UP CROSS FLOW FAN, MOTOR, SYSTEM AND
C     DUTY OF THE FLOW RATE IN A VEHICLE COOLING PACKAGE
C
C     CROSS FLOW ARE IN DIMENSIONLESS FORM
C
C     FAN MOTOR CHARACTERISTIC IS IN THE FORM OF :-
C                          TORQUE(T)=-M.SPEED(N)+C
C     WHERE M= ONE OVER SPEED REGULATION CONSTANT OF MOTOR
C                                 FOR SPEED IN RPM.
C     C=NO LOAD SPEED/SPEED REGULATION CONSTANT
C
C     SYSTEM CHARACTERISTICS- ASSUMED TURBULENT FLOW
C                             RESISTANCE IS PROPORTIONAL TO
C                             SQUARE OF VELOCITY
C
C     SYMBOLS:-
C            QCOE(I)=FLOW COEFFICIENT
C            PCOE(I)=PRESSURE COEFFICIENT
C            EIFF(I)=EIFFICIENCY(STATIC)
C            Q=FLOW RATE IN M3/S, P=STATIC PRESSURE IN PA
C            N=SPEED IN RPM, W=MOTOR POWER IN WATTS
C            D=FAN DIAMETER IN METERS, L=FAN LENGTH IN METERS
C            T=FAN TORQUE IN NEWTON METERS
C            RW=RATED MOTOR POWER
C            RN=RATED MOTOR SPEED
C            D4L=DIA. TO THE POWER FOUR, INTO LENGTH
C            R=DUCT HEIGHT TO DIAMETER RATIO
C            SD=PRESSURE COE. OVER SQUARE OF FLOW COE.
C              =SPECIFIC DIAMETER TO THE POWER OF FOUR
C
C
C     THIS PROGRAMME ONLY HAS FACILITIES TO EXECUTE ONE TYPE OF
C     GEOMETRICALLY SIMILAR CROSS FLOW FANS, AT A TIME
C     THEREFORE FOR DIFFERENT FAN TYPES, ALLOCATE APPROPRIATE VALUES
C     FOR THE FOLLOWING PARAMETERS-
C         PCOE(I),EIFF(I),R, EXPRESSIONS D4L & VR/VO
C
C      NOTE:- THE VALUE OF SHI/ETA BECOMES UNPREDICTABLE, WHEN
C      SHI AND ETA ARE BOTH VERY SMALL. THEREFORE A LINEAR
```

```
C      RELATIONSHIP IS ASSUMED BETWEEN THIS RANGE FOR SHI/ETA
C      VS. PHI AS  SHI/ETA=B.PHI+G
C          LET THIS RANGE BE UNI<SHI/ETA<UN2
C
       DIMENSION QCOE(35),PCOE(35), EIFF(35),P(35),Q(35)
      :,N(35),W(35),K(35)
      :,VR(35),VO(35)
      :,S(35),DCOE(35)
C
       COMMON QCOE,PCOE,EIFF
C
       REAL D,L,C,M,G,SD,R,K
      :,RW,RN,A
      :,PHI,SHI,ETA
      :,UN1,UN2,B
C
       CHARACTER CDATA*1,SYSRES*1,RATIO*1
      :  ,VEH*1
      :,CRES*1
      :,ADATA*1
C
C
05     CONTINUE
C      GLOBAL INITIAL VALUES OF THIS FAN GEOMETRY ARE AS FOLLOWS
C      FAN TYPE-
       R=2.6
       UN1=1.2
       UN2=1.35
       G=7.0
       B=0.0
       WRITE(5,6)
06     FORMAT(1X,79('*'),//)
C      READS INPUT DATA TYPE
C      ---------------------
       WRITE(5,8)
8      FORMAT(1X,'THE ORDER OF INPUT DATA MAY BE ONE OF THE FOLLOWING:-'
      :/1X,'TYPE NUMBER.'/1X,'1). DIA.,LENGTH,C AND M.'//
      :1X,'2). SHAFT POWER, SPEED, SYS. RESISTANCE COE..'//
      :1X,'3). DIA., LENGTH, FLOW RATE, SYS. RESISTANCE COE..'//
      :1X,'4). SPEED, SYS. RESISTANCE COE., C AND M..'
      ://5X,'- UNITS ARE IN SI -')
       READ *,KDATA
10     CONTINUE
C      -----------------
C      WHEN INPUT TYPE IS 1 EI: DIA, LENGTH,C & M
C      -----------------------------------------
       IF(KDATA.EQ.1)THEN
       WRITE(5,12)
12     FORMAT(1X,'INPUT DATA:<D,L,C,M>
      :.'//8X,'WHERE D=DIAMETER, L=FAN LENGTH AND C & M ARE,'
      ://8X,'MOTOR CHARACTERISTICS SATISFING TORQUE=-M.SPEED+C.')
C
```

```
         READ *,D,L,C,M
C        CALL TO CHANGE 'R', IF NECESSARY
         CALL DHRATIO(R)
C
         WRITE(5,20)D,L,M,C,R
 20      FORMAT(1X,'DIAMETER=',F5.3,'METERS;',3X,'LENGTH=',F5.3,'METERS '
        :/1X,'FAN TORQUE=-',E.3,'.SPEED+',F4.2,2X,'N.METERS.'/
        :1X,'HEIGHT/DIAMETER=',F4.2,
        :1X,'SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).',/
        :1X,'*',76X,'*')
C
         WRITE(5,25)
 25      FORMAT(4X,'QCOE',3X,'EIFF',3X,'FLOW RATE',3X,'PRESSURE',
        :4X,'SPEED',5X,'POWER',4X,'SYSRES',8X,'V(R).'//
        :19X,'IN M3/S',6X,'PA',8X,'RPM',6X,
        :'WATTS',/1X,'*',78X,'*')
C
         DO 50 I=1,35
C
C       COMBINED FAN PERFORMANCE & MOTOR CHARACTERISTICS MAY BE WRITTEN AS
C       FOLLOWS:-
C
         IF(QCOE(I).LE.UN1.OR.QCOE(I).GE.UN2)THEN
         Q(I)=-28.65*EIFF(I)*M/(PCOE(I)*D**2)+0.5*((57.3*EIFF(I)*M/(PCOE(I
        :)*D**2))**2
        :+12*C*L*EIFF(I)*QCOE(I)/PCOE(I))**0.5
C
         ELSE
C
C        EI: BETWEEN THE UNPREDICTABLE RANGE OF SHI/ETA
C
         Q(I)=-28.65*M/((B*PHI+G)*D**2)+0.5*((57.3*M/((B*PHI+G)*D**2))**2
        :+12*C*L*QCOE(I)/(B*PHI+G))**0.5
         ENDIF
 50      CONTINUE
C
C        CALCULATION OF OTHER PARAMETERS
C
         DO 62 I=1,35
C
         P(I)=0.6*PCOE(I)*(Q(I)/(QCOE(I)*D*L))**2
         N(I)=19.1*Q(I)/(L*QCOE(I)*D**2)
         W(I)=0.1047*(C*N(I)-M*N(I)**2)
         K(I)=PCOE(I)*(R/QCOE(I))**2
         H=R*D
         VR(I)=Q(I)/(L*H)
C
         WRITE(5,60)QCOE(I),EIFF(I),Q(I),P(I),N(I),W(I),K(I),VR(I)
 60      FORMAT(4X,F4.2,4X,F5.2,5X,F4.2,5X,F6.1,5X,I4,4X,F8.1,4X,F6.1,
        :6X,F5.2)
C
 62      CONTINUE
```

```
C      -------------------------------------------
C
         ELSEIF(KDATA.EQ.2.OR.KDATA.EQ.4)THEN
 270     CONTINUE
              IF(KDATA.EQ.2)THEN
              WRITE(5,300)
 300          FORMAT(1X,'TYPE INPUT DATA:- SHAFT POWER,SPEED,SYS.RES.COE..
     :')
              READ *,RW,RN,RCOE
              CALL DHRATIO(R)
              WRITE(5,320)RW,RN,RCOE,R
 320          FORMAT(1X,'SHAFT POWER=',F5.1,'WATTS',3X,'SPEED=',F6.1,
     :'RPM'01X,'SYS.RESISTANCE COE.=',F6.1//
     :1X,'HEIGHT/DIAMETER=',F4.2//)
C
              ELSE
              WRITE(5,350)
 350          FORMAT(1X,'INSERT INPUT DATA:- SPEED, SYS.RES.COE., C & M.')
              READ *,RN,RCOE,C,M
              CALL DHRATIO(R)
              WRITE(5,380)RN,RCOE,M,C,R
 380          FORMAT(1X,'SPEED=',F6.1,' RPM',4X,'SYS.RES.COE.=',F6.1//
     :1X,'FAN TORQUE=-',E.3,'.SPEED+',F4.2//
     :1X,'HEIGHT/DIAMETER=',F4.2//)
              RW=0.1047*(C*RN-M*RN**2)
                 IF(RW.LT.0.0)THEN
                 WRITE(5,382)C/M
  382            FORMAT(1X,'VALUES CHOSEN ARE UNREALISTIC!'/
     :           1X,'THE ZERO TOQUE SPEED IS=',F6.1,'RPM.'//)
                 GOTO 270
                 ELSE
                 ENDIF
              ENDIF
C
        SD=RCOE/R**2
C     WHEN S. DIAMETER IS GREATER THEN 16.2, FLOW COE. IS LESS THEN 0.4
C     THEREFORE:-
          IF(SD.GT.16.2)THEN
          A=RCOE/(16.2*R**2)
          WRITE(5,385)A
 385      FORMAT(1X,'OPERATION UNSTABLE'/
     :1X,'FOR STABILITY REDUCE SYS. RES. COE. AT LEAST BY A FACTOR OF'/
     :1X,    F5.2)
              IF(KDATA.EQ.2)GOTO 270
          ENDIF
        DO 390 I=1,35
        DCOE(I)=PCOE(I)/QCOE(I)**2
 390    CONTINUE
C
C       SEARCH PHI , CORRESPOND TO SD
        CALL SEARCHX(DCOE,SD,PHI)
 398    CONTINUE
```

```
      READ *,D,L,C,M
C     CALL TO CHANGE 'R', IF NECESSARY
      CALL DHRATIO(R)
C
      WRITE(5,20)D,L,M,C,R
20    FORMAT(1X,'DIAMETER=',F5.3,'METERS;',3X,'LENGTH=',F5.3,'METERS '
     :/1X,'FAN TORQUE=-',E.3,'.SPEED+',F4.2,2X,'N.METERS.'/
     :1X,'HEIGHT/DIAMETER=',F4.2,
     :1X,'SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).',/
     :1X,'*',76X,'*')
C
      WRITE(5,25)
25    FORMAT(4X,'QCOE',3X,'EIFF',3X,'FLOW RATE',3X,'PRESSURE',
     :4X,'SPEED',5X,'POWER',4X,'SYSRES',8X,'V(R).'//
     :19X,'IN M3/S',6X,'PA',8X,'RPM',6X,
     :'WATTS',/1X,'*',78X,'*')
C
      DO 50 I=1,35
C
C    COMBINED FAN PERFORMANCE & MOTOR CHARACTERISTICS MAY BE WRITTEN AS
C    FOLLOWS:-
C
      IF(QCOE(I).LE.UN1.OR.QCOE(I).GE.UN2)THEN
      Q(I)=-28.65*EIFF(I)*M/(PCOE(I)*D**2)+0.5*((57.3*EIFF(I)*M/(PCOE(I
     :)*D**2))**2
     :+12*C*L*EIFF(I)*QCOE(I)/PCOE(I))**0.5
C
      ELSE
C
C     EI: BETWEEN THE UNPREDICTABLE RANGE OF SHI/ETA
C
      Q(I)=-28.65*M/((B*PHI+G)*D**2)+0.5*((57.3*M/((B*PHI+G)*D**2))**2
     :+12*C*L*QCOE(I)/(B*PHI+G))**0.5
      ENDIF
50    CONTINUE
C
C     CALCULATION OF OTHER PARAMETERS
C
      DO 62 I=1,35
C
      P(I)=0.6*PCOE(I)*(Q(I)/(QCOE(I)*D*L))**2
      N(I)=19.1*Q(I)/(L*QCOE(I)*D**2)
      W(I)=0.1047*(C*N(I)-M*N(I)**2)
      K(I)=PCOE(I)*(R/QCOE(I))**2
      H=R*D
      VR(I)=Q(I)/(L*H)
C
      WRITE(5,60)QCOE(I),EIFF(I),Q(I),P(I),N(I),W(I),K(I),VR(I)
60    FORMAT(4X,F4.2,4X,F5.2,5X,F4.2,5X,F6.1,5X,I4,4X,F8.1,4X,F6.1,
     :6X,F5.2)
C
62    CONTINUE
```

```
C
C     CALCULATION OF 'VO' CORRESPOND TO 'VR'
C
      WRITE(5,80)
 80   FORMAT(1X,'IS INLET VELOCITY(VR) REQUIRED, AS A FUNCTION OF
     :',/1X,'VEHICLE VELOCITY? TYPE (Y OR N).')
      READ'(A)',VEH
      IF(VEH.EQ.'Y')THEN
 82   CONTINUE
      WRITE(5,85)
 85   FORMAT(1X,'THEN THE SYSTEM RESISTANCE COE. IS REQUIRED:--',/
     :1X,'INPUT A SYSTEM RESISTANCE COEFFICIENT**')
      READ *,RES1
      DO 90 I=1,35
      IF(RES1.GT.K(I))GOTO 95
 90   CONTINUE
 95   CONTINUE
C
C     DETERMINATION OF 'VO' VS. 'VR' FOLLOWS
      DO 100 J=I,35
      S(J)=(RES1-K(J))**0.5
      VO(J)=S(J)*Q(J)/(H*L)
      VR(J)=Q(j)/(H*L)
 100  CONTINUE
      WRITE(5,110)RES1
 110   FORMAT(1X,65('-'),//1X,'SYSTEM RESISTANCE COEFFICIENT= ',F6.1,/
     :1X,25('-'))
      WRITE(5,120)
 120   FORMAT(1X,6(5X,'VR',4X,'VO'))
C
C        -TABULATION OF RESULTS
      DO 200 J=I,35,6
      KP=INT((34-I)/6)
      IF((J-I).GE.(KP*6))GOTO 230
      WRITE(5,130)VR(J),VO(J),VR(J+1),VO(J+1),VR(J+2),VO(J+2),VR(J+3)
     :,VO(J+3),VR(J+4),VO(J+4),VR(J+5),VO(J+5)
 0130  FORMAT(1X,6(3X,F4.1,1X,F5.1))
 200  CONTINUE
 230  CONTINUE
      WRITE(5,250)
 250  FORMAT(1X,'VR & VO ARE INLET & VEHICLE VELOCITIES.',
     :/1X,'IN M/S.',/1X,'*',78X,'*')
C
      WRITE(5,*)'DO YOU WANT TO CHANGE SYSRES.? TYPE(Y OR N)'
      READ'(A)',CRES
      IF(CRES.EQ.'Y')THEN
      GOTO 82
      ENDIF
      ENDIF
C     ----------------
C     WHEN INPUT TYPE IS 2 OR 4 EI: SHAFT POWER, SPEED,
C     SYS.RES.COE., OR SPEED,SYS.RES.COE.,C & M
```

```
C
C     CALCULATION OF 'VO' CORRESPOND TO 'VR'
C
      WRITE(5,80)
80    FORMAT(1X,'IS INLET VELOCITY(VR) REQUIRED, AS A FUNCTION OF
     :',/1X,'VEHICLE VELOCITY? TYPE (Y OR N).')
      READ'(A)',VEH
      IF(VEH.EQ.'Y')THEN
82    CONTINUE
      WRITE(5,85)
85    FORMAT(1X,'THEN THE SYSTEM RESISTANCE COE. IS REQUIRED:--',/
     :1X,'INPUT A SYSTEM RESISTANCE COEFFICIENT**')
      READ *,RES1
      DO 90 I=1,35
      IF(RES1.GT.K(I))GOTO 95
90    CONTINUE
95    CONTINUE
C
C     DETERMINATION OF 'VO' VS. 'VR' FOLLOWS
      DO 100 J=I,35
      S(J)=(RES1-K(J))**0.5
      VO(J)=S(J)*Q(J)/(H*L)
      VR(J)=Q(j)/(H*L)
100   CONTINUE
      WRITE(5,110)RES1
110    FORMAT(1X,65('-'),//1X,'SYSTEM RESISTANCE COEFFICIENT= ',F6.1,/
     :1X,25('-'))
      WRITE(5,120)
120    FORMAT(1X,6(5X,'VR',4X,'VO'))
C
C        -TABULATION OF RESULTS
      DO 200 J=I,35,6
      KP=INT((34-I)/6)
      IF((J-I).GE.(KP*6))GOTO 230
      WRITE(5,130)VR(J),VO(J),VR(J+1),VO(J+1),VR(J+2),VO(J+2),VR(J+3)
     :,VO(J+3),VR(J+4),VO(J+4),VR(J+5),VO(J+5)
0130   FORMAT(1X,6(3X,F4.1,1X,F5.1))
200   CONTINUE
230   CONTINUE
      WRITE(5,250)
250   FORMAT(1X,'VR & VO ARE INLET & VEHICLE VELOCITIES.',
     :/1X,'IN M/S.',/1X,'*',78X,'*')
C
      WRITE(5,*)'DO YOU WANT TO CHANGE SYSRES.? TYPE(Y OR N)'
      READ'(A)',CRES
      IF(CRES.EQ.'Y')THEN
      GOTO 82
      ENDIF
      ENDIF
C     ----------------
C     WHEN INPUT TYPE IS 2 OR 4 EI: SHAFT POWER, SPEED,
C     SYS.RES.COE., OR SPEED,SYS.RES.COE.,C & M
```

```
C      ------------------------------------------
C
        ELSEIF(KDATA.EQ.2.OR.KDATA.EQ.4)THEN
 270    CONTINUE
             IF(KDATA.EQ.2)THEN
             WRITE(5,300)
 300         FORMAT(1X,'TYPE INPUT DATA:- SHAFT POWER,SPEED,SYS.RES.COE..
     :')
             READ *,RW,RN,RCOE
             CALL DHRATIO(R)
             WRITE(5,320)RW,RN,RCOE,R
 320         FORMAT(1X,'SHAFT POWER=',F5.1,'WATTS',3X,'SPEED=',F6.1,
     :'RPM'01X,'SYS.RESISTANCE COE.=',F6.1//
     :1X,'HEIGHT/DIAMETER=',F4.2//)
C
             ELSE
             WRITE(5,350)
 350         FORMAT(1X,'INSERT INPUT DATA:- SPEED, SYS.RES.COE., C & M.')
             READ *,RN,RCOE,C,M
             CALL DHRATIO(R)
             WRITE(5,380)RN,RCOE,M,C,R
 380         FORMAT(1X,'SPEED=',F6.1,' RPM',4X,'SYS.RES.COE.=',F6.1//
     :1X,'FAN TORQUE=-',E.3,'.SPEED+',F4.2//
     :1X,'HEIGHT/DIAMETER=',F4.2//)
             RW=0.1047*(C*RN-M*RN**2)
                IF(RW.LT.0.0)THEN
                WRITE(5,382)C/M
  382           FORMAT(1X,'VALUES CHOSEN ARE UNREALISTIC!'/
     :          1X,'THE ZERO TOQUE SPEED IS=',F6.1,'RPM.'//)
                GOTO 270
                ELSE
                ENDIF
             ENDIF
C
        SD=RCOE/R**2
C     WHEN S. DIAMETER IS GREATER THEN 16.2, FLOW COE. IS LESS THEN 0.4
C     THEREFORE:-
         IF(SD.GT.16.2)THEN
         A=RCOE/(16.2*R**2)
         WRITE(5,385)A
 385     FORMAT(1X,'OPERATION UNSTABLE'/
     :1X,'FOR STABILITY REDUCE SYS. RES. COE. AT LEAST BY A FACTOR OF'/
     :1X,    F5.2)
              IF(KDATA.EQ.2)GOTO 270
         ENDIF
        DO 390 I=1,35
        DCOE(I)=PCOE(I)/QCOE(I)**2
 390    CONTINUE
C
C       SEARCH PHI , CORRESPOND TO SD
        CALL SEARCHX(DCOE,SD,PHI)
 398    CONTINUE
```

```
C
C      SEARCH SHI, CORRESPOND TO PHI
       CALL SEARCHY(PHI,SHI,0.8,PCOE)
C
C      SEARCH ETA CORRESPOND TO PHI
       CALL SEARCHY(PHI,ETA,1.2,EIFF)
         IF(PHI.GT.UN1.AND.PHI.LE.UN2)THEN
C        EI: UNPREDICTABLE RANGE OF SHI/ETA
         D4L=10449.5*RW/((PHI*RN**3)*(B*PHI+G))
         ELSE
         D4L=10449.5*RW*ETA/(PHI*SHI*RN**3)
         ENDIF
       QD2=RN*PHI*D4L/19.1
C
C      TABULATION OF RESULTS
C
       WRITE(5,400)D4L,(0.05+0.01*I,D4L/(0.05+0.01*I)**4,
      :QD2/(0.05+0.01*I)**2,I=0,8)
  400  FORMAT(1X,'DIA. TO THE POWER 4, INTO LENGTH=',E.3,'.'/
      :4X,'DIAMETER',4X,'LENGTH',4X,'FLOW RATE.'/
      :4X,' METERS ',4X,'METERS',4X,'  M3/S   '/
      :(5X,F5.3,6X,F5.3,6X,F5.3))
C      ------------------
C
C      WHEN INPUT DATA TYPE IS 3, EI: DIA. LENGTH,FLOW RATE,
C      SYS.RES.COE.
C      ----------------------------------------------------
       ELSEIF(KDATA.EQ.3)THEN
       WRITE(5,420)
  420  FORMAT(1X,'INSERT INPUT DATA:- DIA.,LENGTH,FLOW RATE,SYS.RES.COE.
      :')
       READ *,D,L,Q1,RCOE
       CALL DHRATIO(R)
       SD=RCOE/R**2
       DO 430 I=1,35
       DCOE(I)=PCOE(I)/QCOE(I)**2
  430  CONTINUE
C
C      SEARCH PHI CORRESPOND TO SD
       CALL SEARCHX(DCOE,SD,PHI)
       RN=19.1*Q1/(L*PHI*D**2)
        IF(PHI.GT.UN1.AND.PHI.LE.UN2)THEN
        RW=0.0000957*L*D**4*PHI*(B*PHI+G)*RN**3
        ELSE
C       SEARCH ETA, CORRESPOND TO PHI
        CALL SEARCHY(PHI,ETA,1.2,EIFF)
C
C       SEARCH SHI, CORRESPOND TO PHI
        CALL SEARCHY(PHI,SHI,0.8,PCOE)
        RW=0.0000957*L*D**4*PHI*SHI*RN**3/ETA
        ENDIF
       DELP=0.6*SD*(Q1/(L*D))**2
```

```
      WRITE(5,450)D,L,Q1,RCOE,R
  450 FORMAT(1X,'DIAMETER=',F5.3,' METERS',3X,'LENGTH=',F5.3,' METERS'/
     :1X,'FLOWRATE=',F5.3,' M3/S',5X,'SYS. RESISTANCE COE.=',F6.1,'*'//
     :1X,'HEIGHT/DIAMETER=',F4.2//)
      WRITE(5,480)DELP,RN,RW
  480 FORMAT(10X,'FAN PRESSURE RISE=',F6.1,' PA.'//
     :10X,'SPEED            =',F6.0,' RPM.'//
     :10X,'MOTOR SHAFT POWER=',F7.1,' WATTS.'//)
C     ---------------------
      ELSE
C     EI: WHEN TYPED, AN ERRORONES INPUT DATA TYPE NUMBER
      WRITE(5,900)
  900 FORMAT(1X,'TRY AGAIN ')
      GOTO 05
      ENDIF
C     ---------------------------
C
      WRITE(5,*)'DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)'
      READ'(A)',ADATA
      IF(ADATA.EQ.'Y')GOTO 10
      WRITE(5,*)'DO YOU WANT TO CHANGE INITIAL INPUT DATA? TYPE(Y OR N)
     :'
      READ'(A)',CDATA
      IF(CDATA.EQ.'Y')THEN
      GOTO 05
      ENDIF
C
      STOP
      END
C
      BLOCK DATA
C     ----------
      COMMON QCOE(35),PCOE(35),EIFF(35)
C     VALUES OF FLOW COEFFICIENT
      DATA(QCOE(I),I=1,35)/0.3,0.35,0.4,0.45,0.5,0.55,0.6,0.65,0.7,0.75
     :,0.8,0.85,0.9,0.95,1.0,1.05,1.1,1.15,1.2,1.25,1.3,1.35,1.4,1.45,
     :1.5,1.55,1.6,1.65,1.7,1.75,1.8,1.85,1.9,1.95,2.0/
C     CORRESPONDING VALUES OF PRESSURE COEFFFICIENT
      DATA(PCOE(I),I=1,35)/2.6,2.65,2.65,2.65,2.6,2.55,2.5,2.45,2.35,
     :2.2,2.1,1.95,1.8,1.6,1.4,1.15,0.9,0.6,0.35,0.05,-0.25,-0.5,
     :-0.8,-1.05,-1.35,-1.6,-1.9,-2.15,-2.45,-2.7,-3.0,-3.25,
     :-3.55,-3.85,-4.1/
C
C     CORRESPONDING VALUES OF STATIC EIFFICIENCY
      DATA(EIFF(I),I=1,35)/0.25,0.26,0.29,0.3,0.32,0.33,0.33,0.33,
     :0.32,0.3,0.28,0.26,0.24,0.21,0.18,0.15,0.12,0.08,0.05,0.01,
     :-0.03,-0.07,-0.11,-0.16,-0.2,-0.25,-0.31,-0.36,-0.41,-0.46,
     :-0.52,-0.58,-0.64,-0.71,-0.78/
      END
C
      SUBROUTINE DHRATIO(R)
C     --------------------
```

```
      WRITE(5,15)
 15   FORMAT(1X,'DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
     :',/1X,'THE DEFAULT VALUE IS 2.6.
     :',/45X,'TYPE (Y OR N)')
      READ'(A)',RATIO
      IF(RATIO.EQ.'Y')THEN
      WRITE(5,16)
 16   FORMAT(1X,'CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .',/
     :1X,'INPUT <HEIGHT/DIAMETER> .',//)
C
      READ *,R
      ELSE
      R=2.6
      ENDIF
      WRITE(5,18)
 18   FORMAT(1X,65('-'),/)
      RETURN
      END
C
      SUBROUTINE INTERPOL(A1,A2,A3,GR)
C     -------------------------------
      COMMON/RANGE/X1,X2,Y1,Y2
      A1=(Y2-Y1)/(X2**2-X1**2+GR*(X1-X2))
      A2=-GR*A1
      A3=Y1-A1*X1**2-A2*X1
      RETURN
      END
C
      SUBROUTINE SEARCHY(PHI,COMY,GR,YCOE)
C     -----------------------------------
      DIMENSION YCOE(35)
      COMMON QCOE(35),PCOE(35),EIFF(35)
      COMMON/RANGE/X1,X2,Y1,Y2
      DO 100 I=1,35
      IF(QCOE(I).LE.PHI.AND.QCOE(I+1).GE.PHI)GOTO 150
 100  CONTINUE
 150  CONTINUE
      X1=QCOE(I)
      X2=QCOE(I+1)
      Y1=YCOE(I)
      Y2=YCOE(I+1)
      CALL INTERPOL(A1,A2,A3,GR)
      COMY=A1*PHI**2+A2*PHI+A3
      RETURN
      END
C
      SUBROUTINE SEARCHX(YCOE,COMY,PHI)
C     ---------------------------------
      COMMON QCOE(35),PCOE(35),EIFF(35)
      COMMON/RANGE/X1,X2,Y1,Y2
      DIMENSION YCOE(35)
      DO 200 I=1,35
```

```
      IF(YCOE(I).GE.COMY.AND.YCOE(I+1).LE.COMY)GOTO 250
200   CONTINUE
250   CONTINUE
      X1=QCOE(I)
      X2=QCOE(I+1)
      Y1=PCOE(I)
      Y2=PCOE(I+1)
      CALL INTERPOL(A1,A2,A3,0.8)
          IF(YCOE(I).NE.PCOE(I))THEN
          A1=A1-COMY
          A3=A3+COMY
          ENDIF
      PHI=(-A2-SQRT(A2**2-4*A1*(A3-COMY)))/(2*A1)
      RETURN
      END
$BEND
```

APPENDIX 2 - CONTEXT OF AN EXECUTION

```
********************************************************************************

THE ORDER OF INPUT DATA MAY BE ONE OF THE FOLLOWING:-
TYPE NUMBER.
1). DIA.,LENGTH,C AND M.

2). SHAFT POWER, SPEED, SYS. RESISTANCE COE..

3). DIA., LENGTH, FLOW RATE, SYS. RESISTANCE COE..

4). SPEED, SYS. RESISTANCE COE., C AND M..

    - UNITS ARE IN SI -
1
INPUT DATA:<D,L,C,M>

       WHERE D=DIAMETER, L=FAN LENGTH AND C & M ARE,

       MOTOR CHARACTERISTICS SATISFING TORQUE=-M.SPEED+C.
.08,.4,3.3,.00118
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                              TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .


2.85
----------------------------------------------------------------

DIAMETER=0.080METERS;    LENGTH=0.400METERS
FAN TORQUE=-0.118E-02 .SPEED+3.30  N.METERS.
HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).
```

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.10	176.7	2541	80.2	234.6	1.12
0.35	0.26	0.12	175.9	2511	88.6	175.7	1.29
0.40	0.29	0.13	175.1	2505	90.2	134.5	1.47
0.45	0.30	0.15	172.3	2485	95.7	106.3	1.64
0.50	0.32	0.17	168.3	2479	97.3	84.5	1.82
0.55	0.33	0.18	163.5	2468	100.2	68.5	2.00
0.60	0.33	0.20	158.0	2450	104.9	56.4	2.16

0.65	0.33	0.21	152.8	2433	109.3	47.1	2.33
0.70	0.32	0.23	144.4	2415	113.9	39.0	2.49
0.75	0.30	0.24	132.9	2396	118.6	31.8	2.64
0.80	0.28	0.25	124.1	2369	125.2	26.7	2.79
0.85	0.26	0.27	113.4	2349	129.9	21.9	2.94
0.90	0.24	0.28	103.0	2331	134.1	18.0	3.08
0.95	0.21	0.29	89.7	2307	139.5	14.4	3.22
1.00	0.18	0.31	76.8	2282	145.1	11.4	3.35
1.05	0.15	0.32	62.4	2270	147.7	8.5	3.50
1.10	0.12	0.33	48.4	2261	149.6	6.0	3.66
1.15	0.08	0.35	31.8	2245	153.0	3.7	3.79
1.20	0.05	0.36	18.7	2254	151.1	2.0	3.98
1.25	0.01	0.38	2.6	2239	154.2	0.3	4.11
1.30	-0.03	0.39	-13.0	2225	157.1	-1.2	4.25
1.35	-0.07	0.40	-25.5	2202	161.8	-2.2	4.37
1.40	-0.11	0.41	-40.1	2181	165.9	-3.3	4.49
1.45	-0.16	0.43	-53.9	2208	160.6	-4.1	4.71
1.50	-0.20	0.44	-67.8	2183	165.5	-4.9	4.81
1.55	-0.25	0.46	-80.9	2191	163.9	-5.4	4.99
1.60	-0.31	0.47	-96.5	2196	162.9	-6.0	5.16
1.65	-0.36	0.49	-109.0	2194	163.3	-6.4	5.32
1.70	-0.41	0.50	-122.9	2182	165.7	-6.9	5.45
1.75	-0.46	0.51	-134.9	2178	166.5	-7.2	5.60
1.80	-0.52	0.52	-149.2	2173	167.4	-7.5	5.75
1.85	-0.58	0.54	-161.8	2174	167.2	-7.7	5.91
1.90	-0.64	0.55	-175.7	2168	168.4	-8.0	6.05
1.95	-0.71	0.57	-190.3	2166	168.8	-8.2	6.21
2.00	-0.78	0.58	-203.1	2169	168.2	-8.3	6.38

```
IS INLET VELOCITY(VR) REQUIRED, AS A FUNCTION OF
VEHICLE VELOCITY? TYPE (Y OR N).
Y
THEN THE SYSTEM RESISTANCE COE. IS REQUIRED:--
INPUT A SYSTEM RESISTANCE COEFFICIENT**
17.3
-----------------------------------------------------------------

SYSTEM RESISTANCE COEFFICIENT=   17.3
-------------------------
```

VR	VO	VR	VO	VR	VO	VR	VO	VR	VO	VR	VO
3.2	5.5	3.4	8.2	3.5	10.4	3.7	12.3	3.8	14.0	4.0	15.6
4.1	17.0	4.3	18.3	4.4	19.3	4.5	20.4	4.7	21.7	4.8	22.7
5.0	23.8	5.2	24.9	5.3	25.9	5.5	26.8	5.6	27.7	5.8	28.7

```
VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.
*                                                              *
DO YOU WANT TO CHANGE SYSRES.? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
Y
INPUT DATA:<D,L,C,M>

       WHERE D=DIAMETER, L=FAN LENGTH AND C & M ARE,
```

```
       MOTOR CHARACTERISTICS SATISFING TORQUE=-M.SPEED+C.
.1,.25,3.3,.00118
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                                                 TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .


2.85
---------------------------------------------------------------

DIAMETER=0.100METERS;   LENGTH=0.250METERS
FAN TORQUE=-0.118E-02 .SPEED+3.30  N.METERS.
HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).
*                                                                                                   *
```

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.10	254.1	2437	108.3	234.6	1.34
0.35	0.26	0.11	250.9	2399	117.8	175.7	1.54
0.40	0.29	0.13	249.3	2391	119.8	134.5	1.76
0.45	0.30	0.14	244.0	2365	126.1	106.3	1.96
0.50	0.32	0.15	238.0	2358	127.8	84.5	2.17
0.55	0.33	0.17	230.5	2344	131.1	68.5	2.37
0.60	0.33	0.18	221.7	2322	136.2	56.4	2.56
0.65	0.33	0.20	213.5	2301	140.9	47.1	2.75
0.70	0.32	0.21	200.8	2279	145.7	39.0	2.93
0.75	0.30	0.22	184.0	2255	150.9	31.8	3.11
0.80	0.28	0.23	170.6	2222	157.7	26.7	3.27
0.85	0.26	0.24	155.2	2199	162.4	21.9	3.43
0.90	0.24	0.26	140.3	2177	166.7	18.0	3.60
0.95	0.21	0.27	121.6	2149	171.9	14.4	3.75
1.00	0.18	0.28	103.5	2119	177.4	11.4	3.89
1.05	0.15	0.29	83.9	2105	179.9	8.5	4.06
1.10	0.12	0.30	65.0	2095	181.6	6.0	4.23
1.15	0.08	0.31	42.6	2076	184.8	3.7	4.39
1.20	0.05	0.33	25.1	2087	183.0	2.0	4.60
1.25	0.01	0.34	3.5	2070	185.8	0.3	4.75
1.30	-0.03	0.35	-17.3	2053	188.6	-1.2	4.90
1.35	-0.07	0.36	-33.8	2028	192.6	-2.2	5.03
1.40	-0.11	0.37	-52.9	2004	196.2	-3.3	5.16
1.45	-0.16	0.39	-71.5	2034	191.6	-4.1	5.42
1.50	-0.20	0.39	-89.4	2007	195.8	-4.9	5.53
1.55	-0.25	0.41	-107.0	2016	194.4	-5.4	5.74
1.60	-0.31	0.42	-127.7	2021	193.7	-6.0	5.94
1.65	-0.36	0.44	-144.1	2019	194.0	-6.4	6.12
1.70	-0.41	0.45	-162.1	2005	196.1	-6.9	6.26
1.75	-0.46	0.46	-177.7	2000	196.8	-7.2	6.43

```
   1.80     -0.52     0.47     -196.5     1995       197.6      -7.5       6.60
   1.85     -0.58     0.48     -213.1     1996       197.4      -7.7       6.79
   1.90     -0.64     0.49     -231.0     1989       198.5      -8.0       6.94
   1.95     -0.71     0.51     -250.2     1987       198.7      -8.2       7.12
   2.00     -0.78     0.52     -267.1     1990       198.3      -8.3       7.31
IS INLET VELOCITY(VR) REQUIRED, AS A FUNCTION OF
VEHICLE VELOCITY? TYPE (Y OR N).
Y
THEN THE SYSTEM RESISTANCE COE. IS REQUIRED:--
INPUT A SYSTEM RESISTANCE COEFFICIENT**
17.3
-----------------------------------------------------------------

SYSTEM RESISTANCE COEFFICIENT=   17.3
-------------------------
     VR     VO      VR     VO      VR     VO      VR     VO      VR     VO      VR     VO
    3.8    6.4     3.9    9.5     4.1   12.1     4.2   14.2     4.4   16.2     4.6   18.0
    4.8   19.6     4.9   21.1     5.0   22.2     5.2   23.4     5.4   25.0     5.5   26.0
    5.7   27.4     5.9   28.7     6.1   29.8     6.3   30.8     6.4   31.8     6.6   32.9
VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.
*                                                                                *
DO YOU WANT TO CHANGE SYSRES.? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE INITIAL INPUT DATA? TYPE(Y OR N)
Y
*******************************************************************************


THE ORDER OF INPUT DATA MAY BE ONE OF THE FOLLOWING:-
TYPE NUMBER.
1). DIA.,LENGTH,C AND M.

2). SHAFT POWER, SPEED, SYS. RESISTANCE COE..

3). DIA., LENGTH, FLOW RATE, SYS. RESISTANCE COE..

4). SPEED, SYS. RESISTANCE COE., C AND M..

    - UNITS ARE IN SI -
3
INSERT INPUT DATA:- DIA.,LENGTH,FLOW RATE,SYS.RES.COE.
.09,.35,.3,17.3
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                             TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .
```

```
2.85
-----------------------------------------------------------------

DIAMETER=0.090 METERS   LENGTH=0.350 METERS
FLOWRATE=0.300 M3/S     SYS. RESISTANCE COE.=  17.3*

HEIGHT/DIAMETER=2.85


         FAN PRESSURE RISE= 115.9 PA.

         SPEED              = 2222. RPM.

         MOTOR SHAFT POWER=  164.8 WATTS.


DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
Y
INSERT INPUT DATA:- DIA.,LENGTH,FLOW RATE,SYS.RES.COE.
.09,.35,.37,2
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                             TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .


2.85
-----------------------------------------------------------------

DIAMETER=0.090 METERS   LENGTH=0.350 METERS
FLOWRATE=0.370 M3/S     SYS. RESISTANCE COE.=   2.0*

HEIGHT/DIAMETER=2.85


         FAN PRESSURE RISE=  20.4 PA.

         SPEED              = 2079. RPM.

         MOTOR SHAFT POWER=  166.0 WATTS.


DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE INITIAL INPUT DATA? TYPE(Y OR N)
Y
*******************************************************************************
```

```
THE ORDER OF INPUT DATA MAY BE ONE OF THE FOLLOWING:-
TYPE NUMBER.
1). DIA.,LENGTH,C AND M.

2). SHAFT POWER, SPEED, SYS. RESISTANCE COE..

3). DIA., LENGTH, FLOW RATE, SYS. RESISTANCE COE..

4). SPEED, SYS. RESISTANCE COE., C AND M..

    - UNITS ARE IN SI -
2
TYPE INPUT DATA:- SHAFT POWER,SPEED,SYS.RES.COE..
130,2250,11
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                              TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .


2.85
------------------------------------------------------------------

SHAFT POWER=130.0WATTS   SPEED=2250.0RPM SYS.RESISTANCE COE.=  11.0

HEIGHT/DIAMETER=2.85


DIA. TO THE POWER 4, INTO LENGTH=0.152E-04 .
   DIAMETER     LENGTH     FLOW RATE.
    METERS      METERS       M3/S
    0.050       2.437       0.724
    0.060       1.175       0.503
    0.070       0.634       0.370
    0.080       0.372       0.283
    0.090       0.232       0.224
    0.100       0.152       0.181
    0.110       0.104       0.150
    0.120       0.073       0.126
    0.130       0.053       0.107
DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE INITIAL INPUT DATA? TYPE(Y OR N)
Y
********************************************************************************


THE ORDER OF INPUT DATA MAY BE ONE OF THE FOLLOWING:-
TYPE NUMBER.
1). DIA.,LENGTH,C AND M.
```

```
2). SHAFT POWER, SPEED, SYS. RESISTANCE COE..

3). DIA., LENGTH, FLOW RATE, SYS. RESISTANCE COE..

4). SPEED, SYS. RESISTANCE COE., C AND M..

    - UNITS ARE IN SI -
4
INSERT INPUT DATA:- SPEED, SYS.RES.COE., C & M.
2250,11,3.3,.00118
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                             TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .


2.85
-----------------------------------------------------------------

SPEED=2250.0 RPM    SYS.RES.COE.=  11.0

FAN TORQUE=-0.118E-02 .SPEED+3.30

HEIGHT/DIAMETER=2.85


DIA. TO THE POWER 4, INTO LENGTH=0.178E-04 .
   DIAMETER    LENGTH    FLOW RATE.
    METERS     METERS      M3/S
    0.050      2.848      0.847
    0.060      1.374      0.588
    0.070      0.741      0.432
    0.080      0.435      0.331
    0.090      0.271      0.261
    0.100      0.178      0.212
    0.110      0.122      0.175
    0.120      0.086      0.147
    0.130      0.062      0.125
DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE INITIAL INPUT DATA? TYPE(Y OR N)
Y
******************************************************************************


THE ORDER OF INPUT DATA MAY BE ONE OF THE FOLLOWING:-
TYPE NUMBER.
1). DIA.,LENGTH,C AND M.
```

```
2). SHAFT POWER, SPEED, SYS. RESISTANCE COE..

3). DIA., LENGTH, FLOW RATE, SYS. RESISTANCE COE..

4). SPEED, SYS. RESISTANCE COE., C AND M..

    - UNITS ARE IN SI -
1
INPUT DATA:<D,L,C,M>

       WHERE D=DIAMETER, L=FAN LENGTH AND C & M ARE,

       MOTOR CHARACTERISTICS SATISFING TORQUE=-M.SPEED+C.
.08,.4,3.3,.00118
DO YOU WANT TO CHANGE THE INLET HEIGHT/DIAMETER ?
THE DEFAULT VALUE IS 2.6.
                                            TYPE (Y OR N)
Y
CHANGE THE DEFAULT VALUE(2.6) OF HEIGHT/DIA. .
INPUT <HEIGHT/DIAMETER> .


2.85
-----------------------------------------------------------------

DIAMETER=0.080METERS;   LENGTH=0.400METERS
FAN TORQUE=-0.118E-02 .SPEED+3.30  N.METERS.
HEIGHT/DIAMETER=2.85 SYSRES=SYSTEM RESISTANCE BASED ON INLET VELOCITY(VR).
*                                                                             *
```

QCOE	EIFF	FLOW RATE IN M3/S	PRESSURE PA	SPEED RPM	POWER WATTS	SYSRES	V(R).
0.30	0.25	0.10	176.7	2541	80.2	234.6	1.12
0.35	0.26	0.12	175.9	2511	88.6	175.7	1.29
0.40	0.29	0.13	175.1	2505	90.2	134.5	1.47
0.45	0.30	0.15	172.3	2485	95.7	106.3	1.64
0.50	0.32	0.17	168.3	2479	97.3	84.5	1.82
0.55	0.33	0.18	163.5	2468	100.2	68.5	2.00
0.60	0.33	0.20	158.0	2450	104.9	56.4	2.16
0.65	0.33	0.21	152.8	2433	109.3	47.1	2.33
0.70	0.32	0.23	144.4	2415	113.9	39.0	2.49
0.75	0.30	0.24	132.9	2396	118.6	31.8	2.64
0.80	0.28	0.25	124.1	2369	125.2	26.7	2.79
0.85	0.26	0.27	113.4	2349	129.9	21.9	2.94
0.90	0.24	0.28	103.0	2331	134.1	18.0	3.08
0.95	0.21	0.29	89.7	2307	139.5	14.4	3.22
1.00	0.18	0.31	76.8	2282	145.1	11.4	3.35
1.05	0.15	0.32	62.4	2270	147.7	8.5	3.50
1.10	0.12	0.33	48.4	2261	149.6	6.0	3.66
1.15	0.08	0.35	31.8	2245	153.0	3.7	3.79
1.20	0.05	0.36	18.7	2254	151.1	2.0	3.98

1.25	0.01	0.38	2.6	2239	154.2	0.3	4.11
1.30	-0.03	0.39	-13.0	2225	157.1	-1.2	4.25
1.35	-0.07	0.40	-25.5	2202	161.8	-2.2	4.37
1.40	-0.11	0.41	-40.1	2181	165.9	-3.3	4.49
1.45	-0.16	0.43	-53.9	2208	160.6	-4.1	4.71
1.50	-0.20	0.44	-67.8	2183	165.5	-4.9	4.81
1.55	-0.25	0.46	-80.9	2191	163.9	-5.4	4.99
1.60	-0.31	0.47	-96.5	2196	162.9	-6.0	5.16
1.65	-0.36	0.49	-109.0	2194	163.3	-6.4	5.32
1.70	-0.41	0.50	-122.9	2182	165.7	-6.9	5.45
1.75	-0.46	0.51	-134.9	2178	166.5	-7.2	5.60
1.80	-0.52	0.52	-149.2	2173	167.4	-7.5	5.75
1.85	-0.58	0.54	-161.8	2174	167.2	-7.7	5.91
1.90	-0.64	0.55	-175.7	2168	168.4	-8.0	6.05
1.95	-0.71	0.57	-190.3	2166	168.8	-8.2	6.21
2.00	-0.78	0.58	-203.1	2169	168.2	-8.3	6.38

IS INLET VELOCITY(VR) REQUIRED, AS A FUNCTION OF
VEHICLE VELOCITY? TYPE (Y OR N).
Y
THEN THE SYSTEM RESISTANCE COE. IS REQUIRED:--
INPUT A SYSTEM RESISTANCE COEFFICIENT**
17.3

SYSTEM RESISTANCE COEFFICIENT= 17.3

VR	VO	VR	VO	VR	VO	VR	VO	VR	VO	VR	VO
3.2	5.5	3.4	8.2	3.5	10.4	3.7	12.3	3.8	14.0	4.0	15.6
4.1	17.0	4.3	18.3	4.4	19.3	4.5	20.4	4.7	21.7	4.8	22.7
5.0	23.8	5.2	24.9	5.3	25.9	5.5	26.8	5.6	27.7	5.8	28.7

VR & VO ARE INLET & VEHICLE VELOCITIES.
IN M/S.
* *
DO YOU WANT TO CHANGE SYSRES.? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE ABOVE INPUT DATA? TYPE(Y OR N)
N
DO YOU WANT TO CHANGE INITIAL INPUT DATA? TYPE(Y OR N)
N
STOP
LMF -END OF TASK CODE= 0 PROCESSOR=1.553 TSK-ELAPSED=8:57
LOG CON:

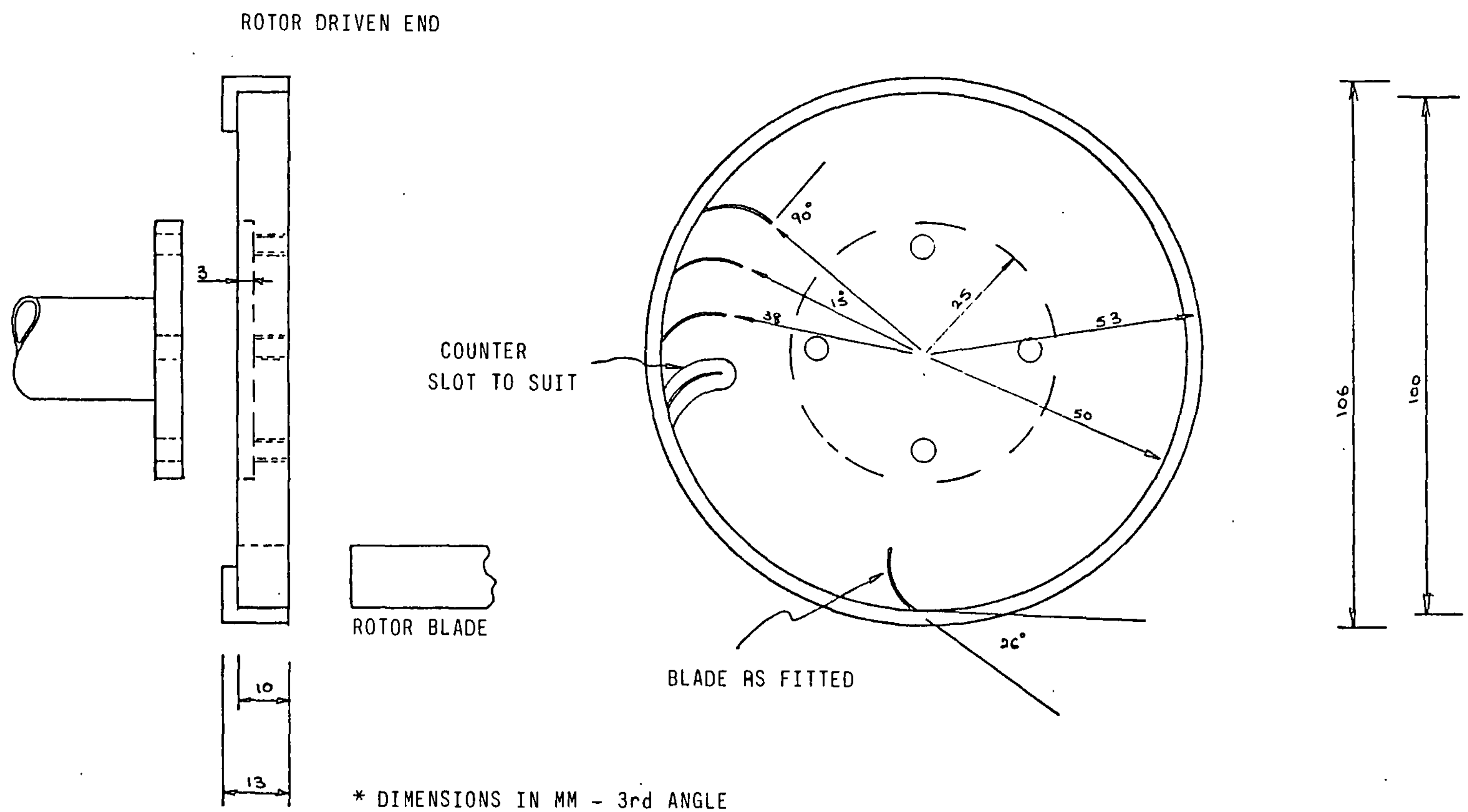


FIGURE A1.1 ROTOR END PLATE

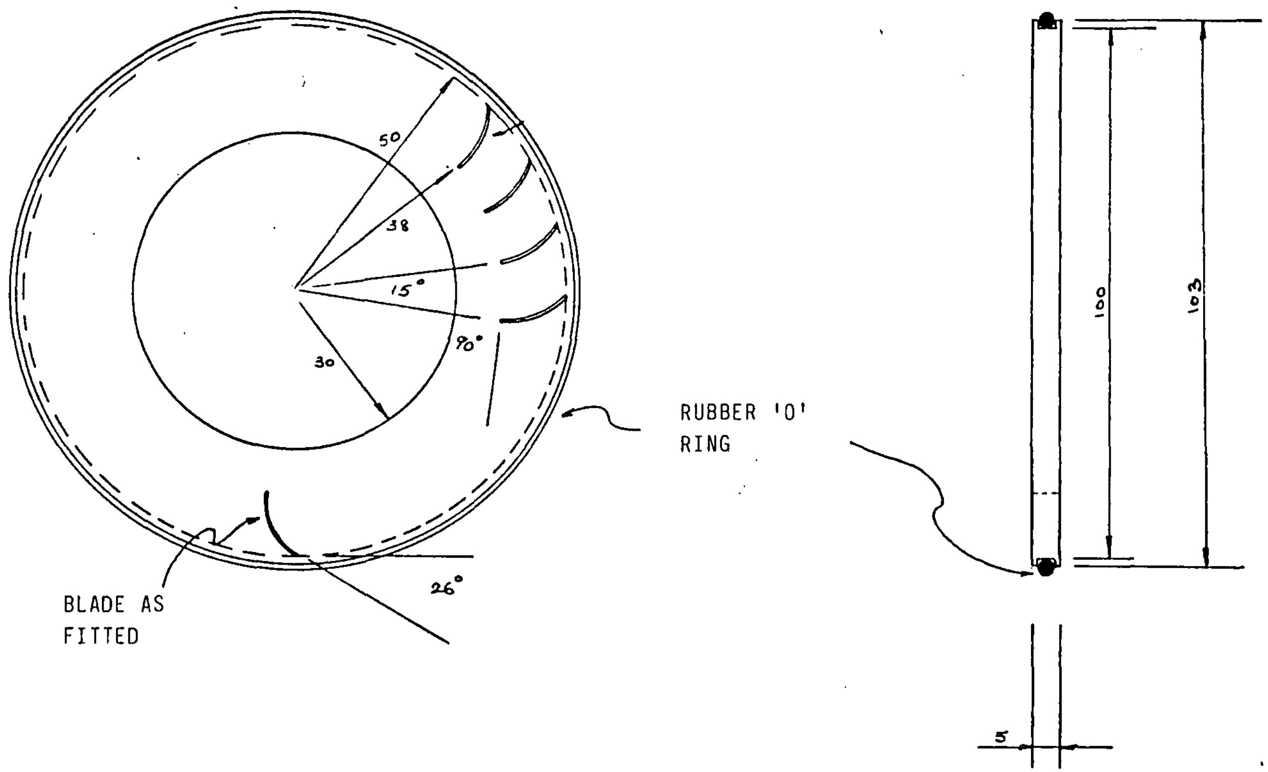


FIGURE A1.2 ROTOR MID-PLATE

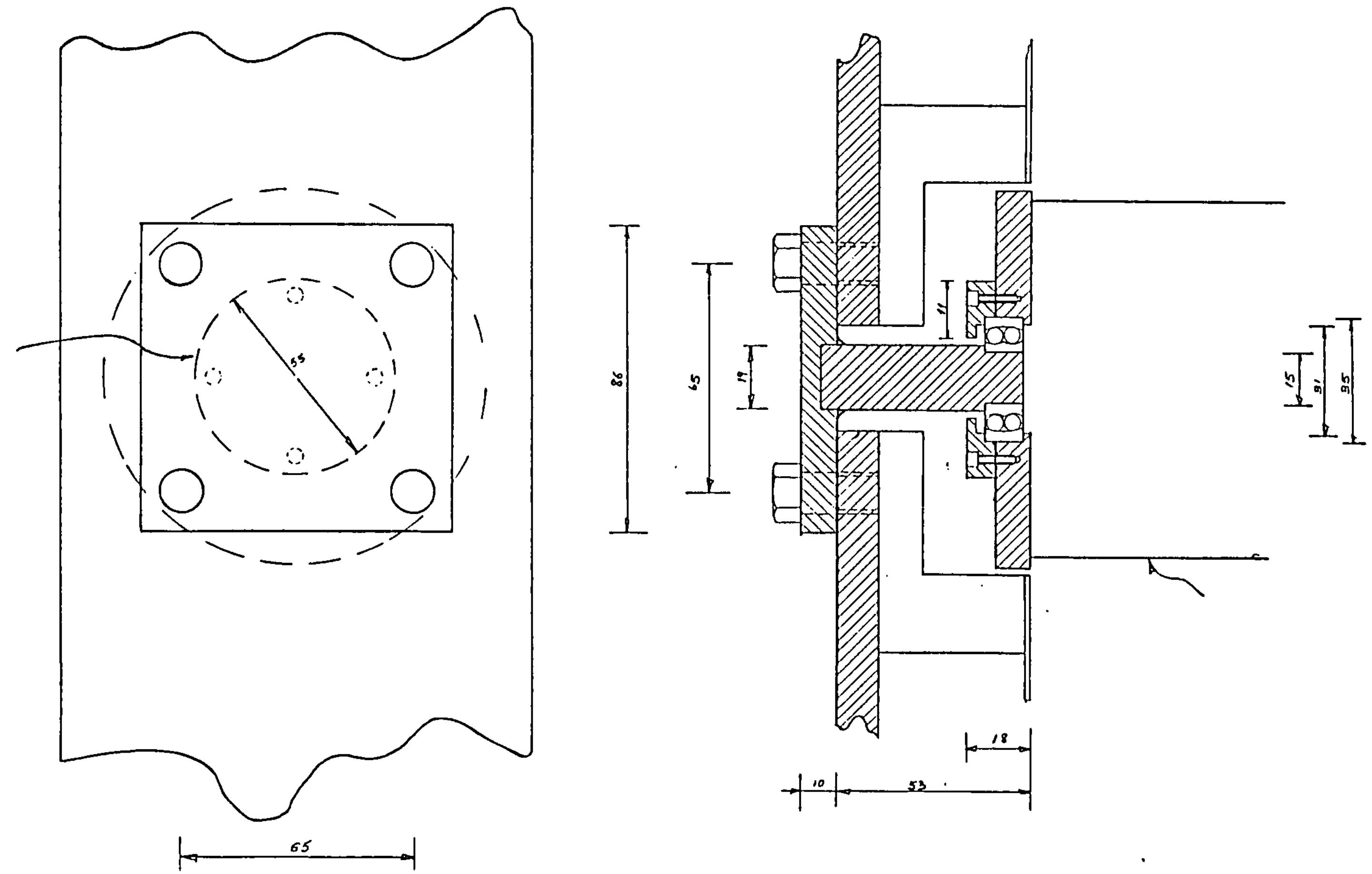


FIGURE A2 - ROTOR NON-DRIVEN END

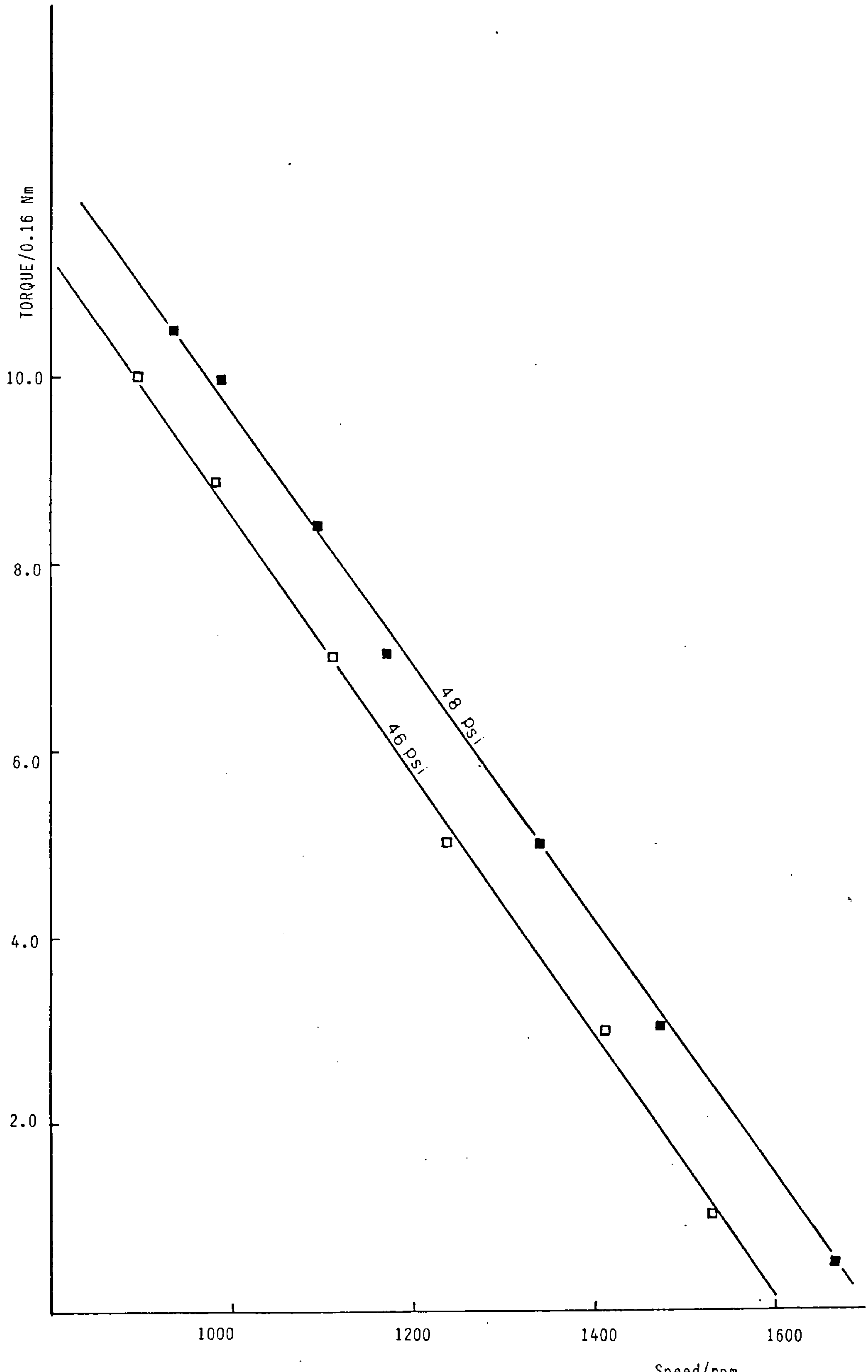


FIGURE A3 CALIBRATION CHART

* * * * *