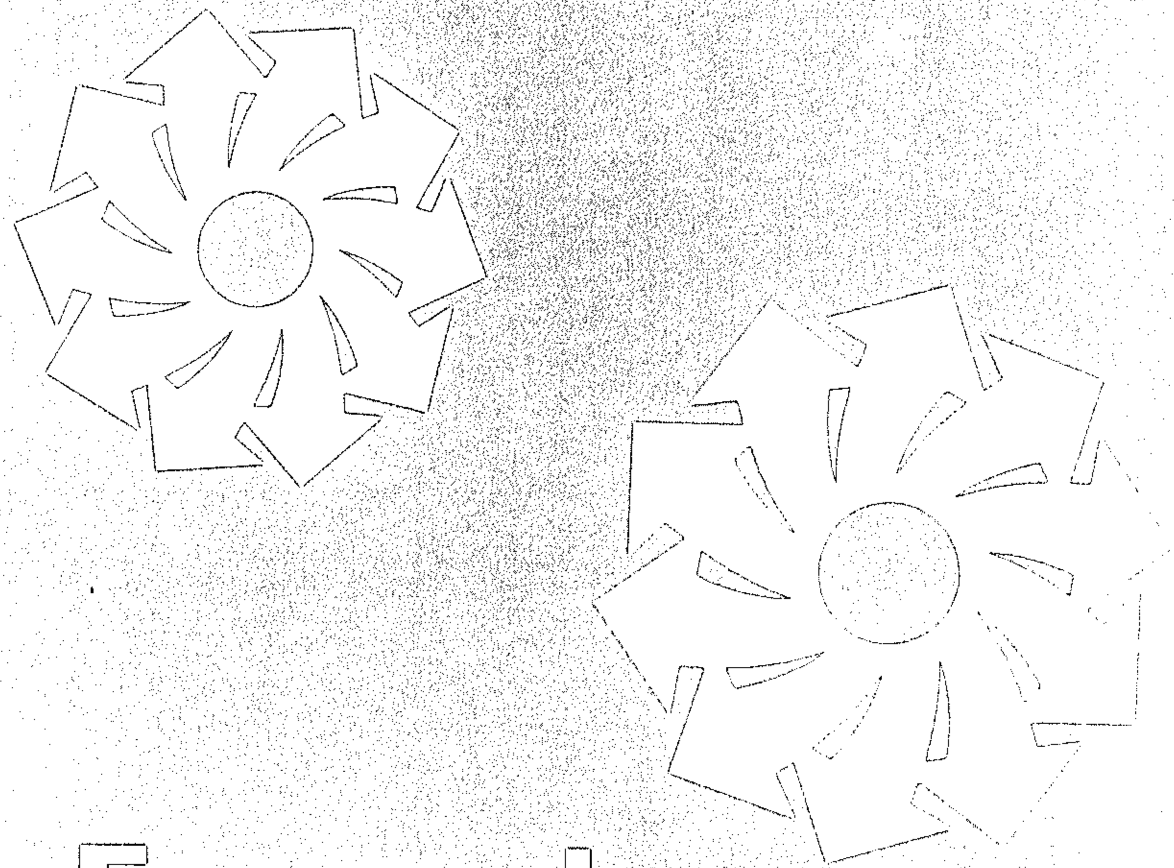
The Fan Manufacturer's Association



Fan and Ductwork Installation Guide

First Edition

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PREFACE

This guide has been prepared by the Fan Manufacturers' Association and was sponsored by the following companies and organisations:

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FAN AND DUCTWORK INSTALLATION GUIDE

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FOREWORD

Good practice in terms of the ductwork components installed adjacent to a fan, is the starting point for providing an efficient and reliable air distribution system. The information summarised in this publication will be of particular benefit to air system designers, equipment specifiers and building services contractors. Additionally, fan manufacturers will be better placed to advise on the fan installation and the connecting duct system. The publication provides the air movement industry with a significant increase in the available knowledge on the effects of ductwork components fitted close to a fan. Furthermore, it is based on a very thorough test and analysis programme covering many different types of duct fittings and all the fan types commonly used in clean air applications.

1. INTRODUCTION

1.1 Background

For some time it has been known that the installation of ductwork close to a fan has a significant effect on the performance of both the fan and ductwork fitting itself. The magnitude of the effect is, generally, not known and as a result, air distribution system designers and fan manufacturers often add contingencies to the calculated or specified performance of a fan to allow for shortfalls in performance caused by installation effects. This brings a consequential increase in capital and running costs and the system does not operate at its design condition. With strong competition, both nationally and internationally, there are significant economic advantages to suppliers and purchasers in being able to select the type and size of fan which will meet the performance specification in a given installation.

The influence of straight ductwork on fan aerodynamic performance was recognised in the publication of BS 848: Part 1 in 1980 by the inclusion of four basic fan installation configurations corresponding to:

Type A free inlet: free outlet
Type B free inlet: ducted outlet
Type C ducted inlet: free outlet
Type D ducted inlet: ducted outlet

A conference on "Installation Effects in Ducted Fan Systems" was held at the Institution of Mechanical Engineers in May 1984. The level of attendance at the conference and the discussion engendered showed that there was considerable interest and concern about the topic of "Installation Effects". One of the outcomes was the agreement of the necessity to set up a series of tests to establish the effect of the most common ductwork fittings on the aerodynamic performance of several types of fan. The HEVAC Association convened meetings of representatives of various organisations serving the interests of the air movement industry. Out of these discussions, a series of aims and objectives regarding installation effects was defined and a programme of research was agreed.

The aims of the research work were:

1. To improve the international competitiveness of the British air movement industry.

2. To enable manufacturers, air distribution system designers and specifiers to have greater assurance that a fan specified for a particular installation will achieve the stated performance.

3. To improve energy efficiency by establishing more accurate data on the effect on fan performance of duct fittings closely connected to a fan.

4. The presentation of this data in a manner offering simple and convenient use.

Short term and long term objectives were defined. These were:

A. Short term objectives:

1. To establish, by experimental measurement, the effect of commonly-used, fan-connected ductwork fittings on fan aerodynamic performance when installed in conjunction with a variety of general purpose fans.

   and

2. To analyse and collate the test data with the specific aim of making the information available in a simple and convenient manner to users such as consultants, contractors, plant air distribution system designers and fan manufacturers. It was proposed that this was done through HEVAC publications.

B. Long term objectives.

In the longer term it is intended that the work should be extended to include acoustic effects and make the aerodynamic work applicable to any fan in any installation. This will require more detailed experimental analysis of flow patterns with a restricted number of fans. Ultimately, it is intended to develop computer modelling of systems and fans incorporating three-dimensional flow analysis.

The following long term objectives were agreed:

1. To extend the work to cover the acoustic performance of the fan.

2. To extend the work so that an air distribution system designer will be able to predict the aerodynamic performance of any fan in any installation.

1.2 The Research Programme

The work which is reported in this publication meets the short term objectives. Proposals for the work were prepared jointly by representatives from NEL, HEVAC, FMA, HVCA and CIBSE and were presented to the DTI in August 1987 with a request that the DTI provided part of the project funding. In January 1988, DTI approved a grant covering 40 per cent of the project costs and by July 1988 FETA had secured sufficient funding from HEVAC, FMA, INTERBUILD, HVCA and CIBSE to instruct NEL to commence work.

The experimental study involved tests on eight separate fans embracing axial flow, mixed flow and centrifugal flow machines and a range of commonly used ductwork fittings including various types of bend, transitions and a silencer. Photographs, presented in Appendix A, show the eight fans tested; three were axial in type with various hub to tip ratios and with or without guide vanes; one was a mixed flow machine with outlet guide vanes; the remaining four were centrifugal units with either forward or backward curved blades and either single or double inlet.

Photographs of a selection of the fittings used in the study are also presented in Appendix A. All were made of galvanised steel with welded flanges.

This publication summarises and collates all the test data, presenting it in a user-friendly format and compares it with published data widely used in industry.

NOTE:

HEVAC — *The Heating Ventilating and Air-Conditioning Manufacturers' Association. It is the Building Services Division of FETA.*

FETA — *The Federation of Environmental Trade Associations.*

FMA — *The Fan Manufacturers' Association.*

HVCA — *The Heating and Ventilating Contractors' Association.*

CIBSE — *The Chartered Institution of Building Services Engineers.*

NEL — *National Engineering Laboratory. It is an executive agency of the Department of Trade and Industry.*

2. NOTATION AND DEFINITIONS

K	Loss Coefficient Factor (see Equation (2))	
N_s	Specific Speed (see Equation (1))	
p_t	Fan Total Pressure	Pa
Q	Volumetric Flowrate	m^3/s
Q0	Volumetric Flowrate at zero P_t	m^3/s
Q1	Volumetric Flowrate at point 1 (see Fig 1 and B.1.1.)	m^3/s
Q3	Volumetric Flowrate at point 3 (see Fig 1 and B.1.1.)	m^3/s
Q5	Volumetric Flowrate at point 5 (see Fig B.1.1.)	m^3/s
SEF	System Flow Effect Factor: NEL — Q3/Q1 or Q5/Q1 see below	
	'System Effect' Factor: AMCA — Q3/Q1 or see Fig 1	
ρ	Air Density	kg/m^3
Ω	Fan Rotational Speed	rad/s

Loss Coefficient Factor: This factor relates the pressure drop across a component to the inlet kinetic energy (sometimes called velocity or dynamic pressure) e.g. a component loss coefficient of 3 means that the pressure drop across that component is three times the kinetic energy (expressed in pressure units) of the fluid entering that component.

Specific Speed: The specific speed is an indication of the shape of a fan best suited to meet a specific duty. This term is explained in greater detail within the text.

'System Effect' Factor and System Flow Effect Factor: Fitting ductwork to a fan affects the performance of that fan. As well as the ductwork's associated pressure drop due to friction, there is an effect due to the presence of the fitting 'being there', the so-called system effect.

There are two ways of considering this effect; one in terms of system pressure loss (the AMCA approach — termed the 'System Effect' Factor) or in terms of fan flowrate variations (the NEL approach — termed the System Flow Effect Factor). Whilst either approach is valid, the NEL approach is more suited to the needs of the system designer who wants to gauge the consequences of a particular fan-system combination on volumetric throughput.

3. SUMMARY OF TEST AND ANALYSIS PROCEDURE

All the tests on single inlet fans were carried out in accordance with installation type D of British Standard BS 848: part 1: 1980[1], Fans for General Purposes, Methods of Testing Performance. For the double-inlet machines installation type B was used. The only variation from the standard test procedure was the use of spiral wound ductwork because of its wide application in the construction industry for heating, air conditioning and ventilation plant. Each fan and fitting combination was treated as a fan unit for the purposes of the test.

The pressure loss versus flowrate characteristic for each fitting and separation spacer duct was obtained experimentally by installing the fitting in the test rig in place of a test fan. The pressure loss of the fitting was determined by measuring pressures upstream of the fitting and downstream of the etoile flow straightener and making the appropriate allowances for the etoile pressure loss and duct wall friction.

The test instrumentation and the procedure for deriving the aerodynamic performance data follows BS 848: Part 1: 1980 (ref 1) and hence is not reported here. Instead, a series of photographs and schematic line diagrams, which summarise the test installation, are presented in Appendix A.

Appendix B gives a brief résumé of the analysis procedures which enabled the performance installation effects to be evaluated and presents the concept of the System Flow Factors which are used to quantify the magnitude of the installation effect. The data is presented in a non-dimensionalised format so that it is applicable to fans and installations for which the tests were representative.

4. SYSTEM FLOW EFFECT DATA AND ITS APPLICATION

4.1 Presentation of the Data

A summary of the duties of all of the fans used in the test series is given in Table 1. Tables 2 and 3 list information on which fan, fitting and spacer combination was tested at inlet and at outlet. Data is presented for the following fittings:

- ★ rectangular/circular transition
- ★ short square bend
- ★ square mitred bend
- ★ segmented bend
- ★ rectangular splitter silencer
- ★ rectangular/rectangular box connection and
- ★ banjo connection

4.2 Application of the System Flow Effect Data

The air distribution system designer of an installation will have four basic aspects to consider:

TABLE 1
DETAILS OF TEST FANS

FAN No.	1	2	3	4	5	6	7
Type	AXIAL						MIXED
Setting	24°	30°	24°	32°	24°	32°	
Guide Vanes	NONE	NONE	FITTED	FITTED	NONE	NONE	FITTED
Diameter mm	630	630	630	630	630	630	630
Efficiency (bep)	52%	50%	50%	49%	34%	47%	54%
Flow (bep) m^3/s	4.1	4.9	3.1	3.7	5.2	5.75	2.2
Total Pressure (bep) Pa	240	245	310	370	950	1090	590
Input Power (bep) kw	1.85	2.50	1.95	2.80	14.3	22.8	2.50
Speed r/min	1440	1440	1440	1440	2900	2900	1470
Qbep/Qo	0.74	0.73	0.75	0.69	0.73	0.61	0.66
Spec Speed	5.74	6.18	4.12	3.94	4.64	4.40	2.19

note: (bep) refers to "best efficiency point"

FAN No.	8	9	10	11
Type	CENTRIFUGAL			
Blading	FORWARD CURVED	BACKWARD CURVED	FORWARD CURVED	BACKWARD CURVED
Entry	SINGLE	SINGLE	DOUBLE	DOUBLE
Diameter mm	630	610	630	510
Efficiency (bep)	53%	62%	48%	53%
Flow (bep) m^3/s	4.25	4.85	5.60	3.45
Total Pressure (bep) Pa	1120	2850	1040	1090
Input Power (bep) kw	8.90	22.4	11.6	7.15
Speed r/min	850	2600	850	1800
Qbep/Qo	0.69	0.61	0.49	0.46
Spec Speed	1.09	1.76	0.93	1.50

* what type of fan should be used
* whether the ductwork needs to be located at the inlet or outlet or both
* what type of duct fitting is required and
* what spacing can be accommodated between the fan and the duct fitting

Normally, the air distribution system designer will know the flowrate required and, having estimated the system pressure losses, can thereby define the duty required of the fan. He can then use fan manufacturers' catalogues of pressure/volume performance characteristics to make an appropriate fan selection on the assumption that the installation configuration will have no effect on the aerodynamic performance.

With this information and data given in Table 1, the air distribution system designer can select which of the eleven different fans tested in this study is most representative of his selected fan. The choice of which of the tested fans is most representative may be clear and will be based largely on the type of fan he has selected, eg. axial, mixed or radial flow. Alternatively, a convenient parameter to assist in selecting an appropriate fan may be specific speed.

Specific speed is a non-dimensional parameter which combines the impeller rotational speed, fan flowrate and total pressure rise at best efficiency into a non-dimensional quality which effectively defines the optimum type of fan.

$$N_s = \frac{\Omega (Q)^{0.5}}{(p_t/\varrho)^{0.75}} \qquad \text{Eq (1)}$$

A specific speed of 3 or greater normally suggests an axial machine type of fan would be the best selection whilst a

TABLE 2
DETAILS OF TESTS AT INLET

Fitting & Spacing		Fan 1	2	3	4	5	6	7	8	9
Rect/Circ Transition	0 D	x	x	x	x	x	x	x	x	x
	0.5 D						x			
	1 D	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x
Short Square Bend	0 D	x	x	x	x	x	x	x	x	x
	0.5 D	x	x	x	x					
	1 D	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x
Square Mitred Bend	0 D	x	x	x	x	x	x	x	x	x
	0.5 D		x	x	x					
	1 D	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x
Segmented Bend	0 D	x	x	x	x	x	x	x	x	x
	0.5 D	x	x	x	x					x
	1 D	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x
Rect Splitter Silencer	0 D							x		
	0.5 D									
	1 D							x		
	2 D							x		
Banjo Connection	0 D									x
	0.5 D									
	1 D								x	
	2 D								x	x

X denotes combination tested
no entry denotes no test was undertaken

TABLE 3
DETAILS OF TESTS AT OUTLET

Fitting & Spacing		Fan 1	2	3	4	5	6	7	8	9	10	11
Rect/Circ Transition	0 D	x	x	x	x	x	x	x		x	x	x
	0.5 D											
	1 D	x	x	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x	x	x
Short Square Bend	0 D	x	x	x	x	x	x	x	x	x	x	x
	0.5 D		x		x	x	x					
	1 D	x	x	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x	x	x
Square Mitred Bend	0 D	x	x	x	x	x	x	x	x	x	x	x
	0.5 D			x	x	x	x					
	1 D	x	x	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x	x	x
Segmented Bend	0 D	x	x	x	x	x	x	x	x	x	x	x
	0.5 D					x	x					
	1 D	x	x	x	x	x	x	x	x	x	x	x
	2 D	x	x	x	x	x	x	x	x	x	x	x
Rect Splitter Silencer	0 D					x	x	x				x
	0.5 D											
	1 D					x	x	x				x
	2 D					x	x	x				x
Rect/Rect Box	0 D								x	x	x	x
	0.5 D								x			
	1 D								x	x	x	x
	2 D								x	x	x	x

X denotes combination tested
no entry denotes no test was undertaken

value of 1.8 or less suggests a centrifugal machine. Values between these two extremes tend to suggest that a mixed flow would be most appropriate. It is possible to design machines outwith these ranges but there is usually some design compromise in terms of size, efficiency or blading to be made.

Having selected a fan, the air distribution system designer has to decide whether the installation requires a fitting located close to the inlet or outlet of the fan and what type of duct fitting is required. Placing a fitting close to the fan can have an effect on the fan aerodynamic performance and the magnitude of this effect needs to be calculated.

To assist the air distribution system designer in making an initial fitting selection, Tables 4 and 5 give the range of the

TABLE 4
SURVEY OF SYSTEM FLOW EFFECT FACTORS AT INLET (SEF)

Fitting	Axial & Mixed Flow Fans							Centrifugal	
	1	2	3	4	5	6	7	8	9
Rect/Circ Transition	0.988 - 1.002	0.974 - 0.997	0.988 - 0.998	0.990 - 1.012	0.979 - 1.002	0.974 - 1.012	0.990 - 0.995	0.996 - 1.002	0.965 - 0.999
Short Square Bend	0.919 - 0.958	0.847 - 0.918	0.966 - 0.986	0.941 - 0.991	0.968 - 0.984	0.960 - 0.980	0.976 - 0.997	0.962 - 1.019	0.942 - 0.987
Square Mitred Bend	0.935 - 0.968	0.874 - 0.916	0.948 - 0.983	0.965 - 0.990	0.960 - 0.980	0.950 - 0.971	0.975 - 0.984	0.948 - 0.974	0.948 - 0.987
Segmented Bend	0.930 - 0.975	0.912 - 0.955	0.975 - 0.996	0.975 - 1.002	0.978 - 0.994	0.949 - 0.987	0.982 - 0.993	0.972 - 0.988	0.996 - 0.996
Rect Splitter Silencer						0.943	- 0.968		
Banjo Connection								0.803 - 0.904	0.732 - 0.867

Note: This table applies to values of Q1:Q0 between 0.6 and 0.8 (i.e. typical commercial values)

TABLE 5
SURVEY OF SYSTEM FLOW EFFECT FACTORS AT OUTLET (SEF)

Fitting	Axial & Mixed Flow Fans							Centrifugal Fans			
	1	2	3	4	5	6	7	8	9	10	11
Rect/Circ Transition	0.995 - 1.010	0.986 - 0.992	0.989 - 1.001	0.987 - 1.008	0.990 - 1.020	0.976 - 0.985	0.986 - 0.995	1.006 - 1.009	0.971 - 1.001	1.034 - 1.055	1.000 - 1.058
Short Square Bend	0.964 - 0.997	0.939 - 0.973	0.977 - 1.004	0.969 - 1.004	0.987 - 1.060	0.989 - 1.081	0.978 - 0.994	0.983 - 0.996	0.964 - 0.992	1.014 - 1.033	0.977 - 1.005
Square Mitred Bend	0.945 - 0.971	0.902 - 0.936	0.947 - 0.989	0.954 - 0.987	0.954 - 1.020	0.969 - 1.011	0.976 - 0.987	0.972 - 0.987	0.954 - 0.987	0.978 - 0.994	0.971 - 0.989
Segmented Bend	0.985 - 1.010	0.963 - 0.991	0.990 - 1.010	0.990 - 1.012	0.939 - 1.020	0.980 - 1.038	0.975 - 1.001	0.994 - 0.996	0.962 - 0.992	0.967 - 0.975	0.984 - 1.022
Rect Splitter Silencer					0.936 - 1.000	0.865 - 0.976	0.934 - 0.971				0.851 - 0.908
Rect/Rect Box Connection								1.002 - 1.007	0.963 - 0.994	1.046 - 1.056	0.999 - 1.075

Note: This table applies to values of Q1:Q0 between 0.6 and 0.8 (i.e. typical commercial values)

TABLE 6
QUALITATIVE SURVEY OF FITTINGS AT INLET
(Q1:Q0 = 0.7)
SYSTEM FLOW EFFECT

Fitting & Spacing		Axial & Mixed Flow							Cent	
		1	2	3	4	5	6	7	8	9
Rect/Circ Transition	0 D	a	a	a	a	a	a	a	a	a
	0.5 D						a			
	1 D	a	a	a	a	a	a	a	a	a
	2 D	a	a	a	a	a	a	a	a	a
Short Square Bend	0 D	b	c	a	b	a	a	a	a	a
	0.5 D	b	c	a	a					
	1 D	b	c	a	a	a	a	a	a	a
	2 D	b	b	a	a	a	a	a	a	a
Square Mitred Bend	0 D	b	c	a	a	a	a	a	a	a
	0.5 D		c	a	a					
	1 D	b	c	a	a	a	a	a	a	a
	2 D	b	b	a	a	a	a	a	a	a
Segmented Bend	0 D	b	b	a	a	a	a	a	a	a
	0.5 D	b	b	a						
	1 D	a	a	a	a	a	a	a	a	a
	2 D	a	b	a	a	a	a	a	a	a
Rect Splitter Silencer	0 D							a		
	0.5 D									
	1 D							a		
	2 D							a		
Banjo Connection	0 D									c
	0.5 D									
	1 D								c	
	2 D								c	c

a (0-5%) could be considered insignificant
b (5-10%) could be considered significant
c (over 10%) could be considered large or excessive

measured System Flow Effect Factor for each fitting over the flow range Q1/Q0 = 0.6 to 0.8, ie approximately ±10 per cent of best efficiency flow which covers most commercial selections. It should be borne in mind that axial machines tend to exhibit stall characteristics at around Q1/Q0 ratios of 0.5. From these tables the most suitable fitting can be analysed in more detail.

Information on the magnitude of the installation effect is presented in simplified, qualitative form in Tables 6 and 7 for the different fan and fitting combinations tested. The data is given for a flowrate ratio of Q1/Q0 = 0.7 (ie near best efficiency point). The data are classified into three basic categories depending on the magnitude of the effect, namely

a. (0 to 5 per cent) could be considered insignificant

b. (5 to 10 per cent) could be considered significant

c. (over 10 per cent) could be considered large or excessive.

The air distribution system designer will have to decide whether the predicted loss of performance is acceptable or consider whether the installation can be modified. The air distribution system designer should also check the effect of the fitting on the fan performance over a wider flow range, if the fan duty is likely to be variable. Appendix C gives a worked example to clarify the procedure.

TABLE 7
QUALITATIVE SURVEY OF FITTINGS AT OUTLET
(Q1:Q0 = 0.7)
SYSTEM FLOW EFFECT

Fitting & Spacing		Axial & Mixed Flow							Centrifugal			
		1	2	3	4	5	6	7	8	9	10	11
Rect/Circ Transition	0 D	a	a	a	a	a	a	a		a	a	a
	0.5 D											
	1 D	a	a	a	a	a	a	a	a	a	b	a
	2 D	a	a	a	a	a	a	a	a	a	b	a
Short Square Bend	0 D	a	a	a	a	a	a	a	a	a	a	a
	0.5 D		a		a	a	a					
	1 D	a	a	a	a	a	a	a	a	a	a	a
	2 D	a	b	a	a	a	a	a	a	a	a	a
Square Mitred Bend	0 D	a	b	a	a	a	a	a	a	a	a	a
	0.5 D			a	a	a	a					
	1 D	a	b	a	a	a	a	a	a	a	a	a
	2 D	a	b	a	a	a	a	a	a	a	a	a
Segmented Bend	0 D	a	a	a	a	a	a	a	a	a	a	a
	0.5 D					a						
	1 D	a	a	a	a	a	a	a	a	a	a	a
	2 D	a	a	a	a	a	a	a	a	a	a	a
Rect Splitter Silencer	0 D					a	b	a				c
	0.5 D											
	1 D					a	b	a				c
	2 D					a	b	b				c
Rect/Rect Box	0 D								a	a	b	a
	0.5 D								a			
	1 D								a	a	b	a
	2 D								a	a	a	a

a (0-5%) could be considered insignificant
b (5-10%) could be considered significant
c (over 10%) could be considered large or excessive

5. COMPARISON WITH OTHER PUBLISHED DATA

There are a number of publications giving estimates of 'system effects' and losses in fan duct system although most are based on limited data or theoretical considerations. The authors of this guide are, however, not aware of any publication which incorporates both aspects in a similar manner to that reported here. Hence, comparisons are made separately, ie firstly with 'system effect' and secondly with loss factors.

5.1 Installation Effect Factors

The only known document dealing with 'system effect' is the AMCA guide (ref 2) which relates the 'system effect' for various fan types, fittings and air velocities as a series of curves. The findings from this study have been compared with the information in the AMCA guide.

Fig. 1, taken from the information given in Reference 2, shows the AMCA approach to defining the 'system effect' factor. The starting point is point 1, the intersection of the estimated system line and the fan characteristic. In the AMCA analysis, the system pressure losses are taken to

6.12 THE SYSTEM EFFECT

Figure 6-7 illustrates deficient fan/system performance resulting from one or more of the undesirable flow conditions listed in Section 6.10. It is assumed that the system pressure losses, shown in system curve A, have been accurately determined, and a suitable fan selected for operation at Point 1. However, no allowance has been made for the effect of the system connections on the fan's performance. To compensate for this System Effect it will be necessary to add a System Effect Factor to the calculated system pressure losses to determine the actual system curve. The factor for any given configuration is velocity dependent and will, therefore, vary across the range of flow volumes of the fan (see Figure 7-1).

In Figure 6-7 the point of intersection between the fan performance curve and the actual system curve B is Point 4. The actual flow volume will, therefore, be deficient by the difference from 1-4. To achieve design flow volume an SEF equal to the pressure difference between Point 1 and 2 should have been added to the calculated system pressure losses and the fan selected to operate at Point 2. Note that because the System Effect is velocity related, the difference represented between Points 1 and 2 is greater than the difference between Points 3 and 4.

The System Effect Factor includes only the effect of the system configuration on the fan's performance.

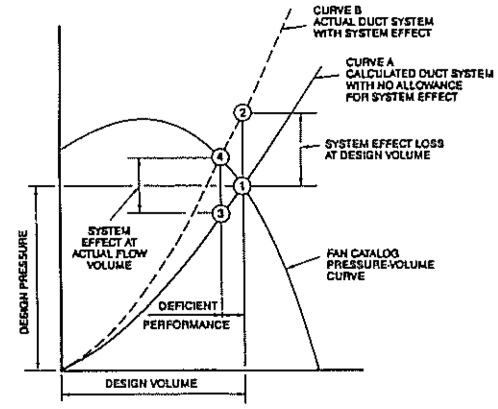

Figure 6-7 Deficient Fan/Duct System Performance System Effect Ignored

Figure 7-1 System Effect Curves

Enter the chart at the appropriate air velocity (on the abcissa) read up to the applicable curve, then read across from the curve (to the ordinate) to find the System Effect Factor at standard air density.

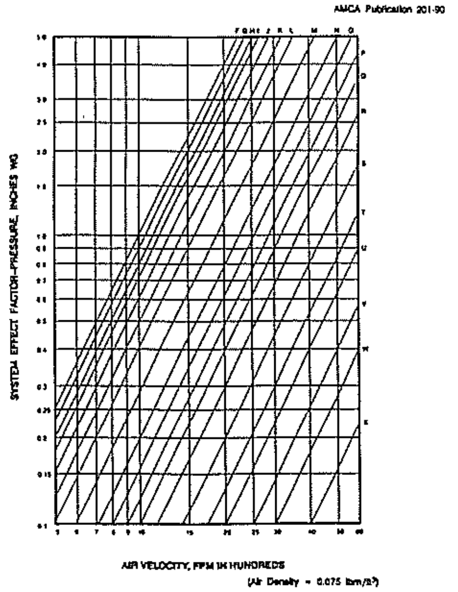

Figure 7-1 System Effect Curves

Fig. 1 AMCA Definition of 'System Effect' (Based on Reference 2) include the pressure loss of the fittings being placed close to the fan. AMCA then use their chart to give a 'system effect' factor — a pressure loss which is to be added to the calculated system resistance. The 'system effect' factor is added to the estimated system loss given at point 2.

From this point AMCA move down the new system line drawn through point 2 to point 4. The flowrate at point 4 is the same as that at point 3 on the original system curve. In this way it is possible to derive a Q3/Q1 ratio similar to that developed in the study.

To enable a direct comparison of the appropriate AMCA data with the findings from the study, the test data from the current series was used to derive AMCA type 'system effect' factors. Having selected a combination of fan, fitting and system line the procedure used was as follows:

The system line cuts the fan + fittings characteristic at Q3. At that flowrate the pressure due to the component loss was calculated and a new system line obtained, which cuts the original fan curve at a flow equivalent to point 1 in the AMCA diagram. The AMCA procedure was then used in reverse to calculate the system effect factor and compared with the AMCA published data. Three fan types and three fittings were used for comparison purposes, the vane axial, the tube axial and the backward curved single inlet centrifugal with the short square bend, the square mitred bend and the segmented bend.

The calculated factors are presented in Table 8 along with values derived from the AMCA publication for the appropriate installation.

The factors show fairly good agreement for the axials but the AMCA losses for the centrifugal were about double that obtained from the NEL tests.

5.2 Component Pressure Loss

There are a number of publications recording values of loss factors for various bends and fittings. A few of the most widely used, namely, Woods (3), CIBSE (4), Miller (5) and Buffalo Forge (6) were studied to compare with the test values.

For convenience, the experimentally determined pressure drop was calculated from the quadratic fit of the curve of the measured pressure losses versus flowrate for a flowrate of 4.7 m^3/sec (the averaged best efficiency flow for the fans on test) and the loss factor, K, derived from the equation

pressure drop = K × inlet dynamic pressure ... Eq (2)

Table 9 compares the loss factors determined from the current tests with published data. The values agree fairly well though it is difficult to obtain exact comparisons because the test arrangements in which the pressure losses were measured are not indicated and are liable to be quite different; in this study measurements were all made in circular ducting and transition sections were used.

TABLE 8
COMPARISON OF 'SYSTEM EFFECT' FACTORS

			'System Effect' Factor (Pa)		
Location	Fitting	NEL/AMCA	Tube Axial	Vane Axial	Centrifugal
Inlet	Short Square Bend	NEL	27 → 112	0 → 45	87 → 187
		AMCA	11 → 100	12 → 75	75 → 374
	Segmented Bend	NEL	4 → 21	0 → 17	0 → 95
		AMCA	7 → 22	0 → 12	75 → 274
	Mitred Bend	NEL	0 → 59	12 → 45	0 → 219
		AMCA	-	-	125 → 498
Outlet	Short Square Bend	NEL	-32 → 0	-25 → 0	0 → 87
		AMCA	0	15 → 57	75 → 249
	Segmented Bend	NEL	-47 → 0	-37 → 0	0 → 87
		AMCA	0	7 → 14	-

TABLE 9
COMPARISON OF LOSS COEFFICIENT

Fitting	Loss Coefficient K					
	NEL TEST		Daly	CIBSE	Miller	Buffalo Forge
	Inlet	Outlet				
Rect/Circ Transition	0.07	0.07	-	-	-	-
Short Square Bend	0.36	0.36	0.70	0.23	0.7	-
Square Mitred Bend	0.40	0.35	0.30	0.35	0.3	0.35
Segmented Bend	0.32	0.31	0.38	0.3 - 0.36	0.3 (approx)	0.38
Rect/Rect Box	-	0.12	-	-	-	-
Rect Splitter Silencer	2.42	2.42	-	-	-	-
Banjo Connection	4.16	-	-	-	-	-

6. CONCLUSIONS

The experimental study of the effect of closely connected ductwork fittings on the aerodynamic performance of fans has shown that care must be taken when designing fan and ductwork systems to minimise the influence of the installation in adversely affecting aerodynamic performance.

Most of the fittings tested had a relatively minor effect (ie less than 5%) on fan performance but there were occasions when the fitting had a marked effect on the volumetric through flow of the system.

The publication shows that there are no clear trends and that both the type of fan, the fitting and its location can have an unpredictable effect. However, the publication provides a useful data base of information to enable the air distribution system designer (possibly with the aid of the fan manufacturer) to anticipate difficulties and to select the necessary solution to minimise the risks.

The results of comparison between the NEL tests and the AMCA method show a close similarity for axials but with centrifugal fans where the flow is possibly influenced by the volute design, the results were not so close and suggest that a detailed knowledge of the flow regime may be necessary to explain the differences.

The study has shown that the process of component and fan interaction is a complex one. A more detailed flow study may be necessary to explain the process more fully and eventually be used as the basis of a computation procedure to eliminate the need for extensive testing.

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LIST OF TABLES

LIST OF FIGURES

APPENDIX A

Pictorial Details of Fans and Fittings Under Test

This appendix presents a number of photographs and schematics to enable the reader to obtain an overall view of the project.

Fig. A.1

The ductwork used in the study. All the spacers and transitions are shown including the spiral wound ductwork and the attached silencer pieces.

Fig. A.2

Fig. A.2 shows Fig A.1. in schematic form.

Fig. A.3

The fans used in the study. In the front from left to right, they are respectively the mixed flow (Fan 7), the 0.589 hub/tip ratio tube axial (Fans 5 and 6), the 0.389 hub/tip ratio fan with guide vanes (Fans 3 and 4) and the 0.233 hub/tip ratio tube axial (Fans 1 and 2). In the background are the centrifugal fans; the double inlet backward curved machine (Fan 11), the single and double inlet forward curve units (Fans 8 and 10) and finally the single inlet backward curved machine (Fan 9). The fans were selected to be representative of a wide range of general purpose fans.

Fig. A.4

The test arrangement used to determine the performance of the axial and mixed flow machines. The inlet bellmouth shown in this figure was only temporary and was changed to a standard bellmouth before the test series began. From inlet to outlet the photograph shows; the inlet bellmouth, the inlet pressure and temperature measurement locations, the fan, the etoile flow straightener, the outlet pressure and temperature measurement locations, the flow venturi, the AMCA flow straightener, the silencer, the booster fan and outlet control damper.

Fig. A.5

Fig. A.5 shows Fig. A.4 in schematic form.

Fig. A.6

Some of the test instrumentation, namely the Hewlett Packard computer which, together with an analogue/digital interface card, takes voltages from the pressure and temperature sensors and records and processes the test data. Also shown are nine pressure transducers, power unit and switching box which cover the complete pressure range of all the test fans to measure the inlet, outlet and venturi inlet and differential pressures.

The Betz manometer as used for setting up purposes only. Not shown are the Norma power analyser or the frequency analyser which were used to measure motor input power and fan rotational speed respectively.

Fig. A.7

The bends tested. They comprise a square mitred bend, a five piece segmented bend and a short square radiused bend. The square bends required transition pieces to match the test ductwork.

Fig. A.8

The banjo connection, the silencer, the rectangular to circular transition and the rectangular to rectangular box connection.

Fig. A.9

The square mitred bend being tested at zero diameters from the inlet of an axial fan. The necessary transition pieces are clearly visible.

Fig. A.10

A schematic of the test arrangement used for testing a fan with the fitting located at the inlet side of the fan.

Fig. A.11

A schematic of the test arrangement used for testing a fan with the fitting located at the discharge side of the fan.

Fig. A.12

The test configuration for a test on a centrifugal fan with a fitting at the outlet.

Fig. A.13

The test configuration for a test on a centrifugal fan with the banjo connection at inlet.

Fig. A.1 Ductwork used in Study

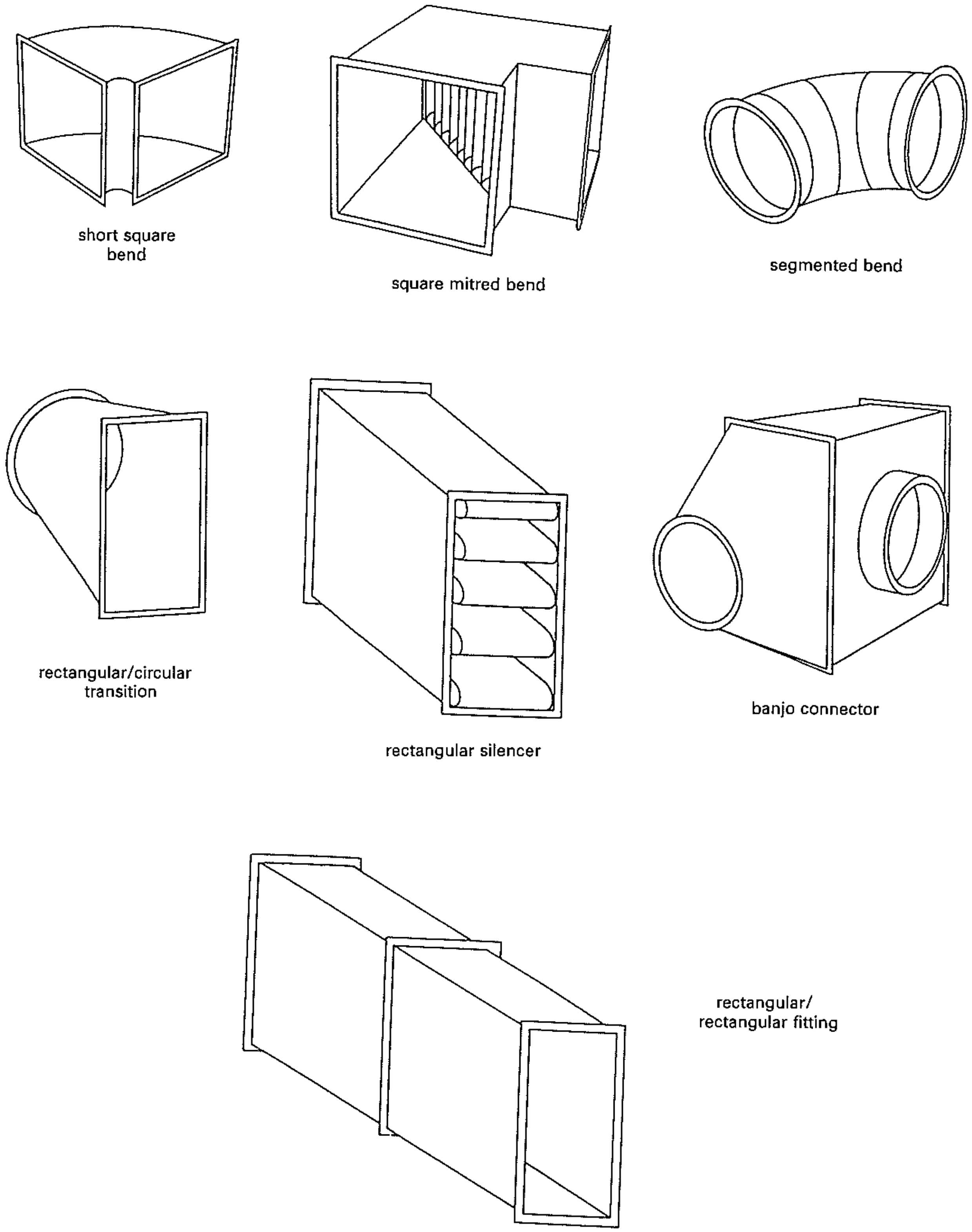

Fig. A.2 Ductwork used in Study

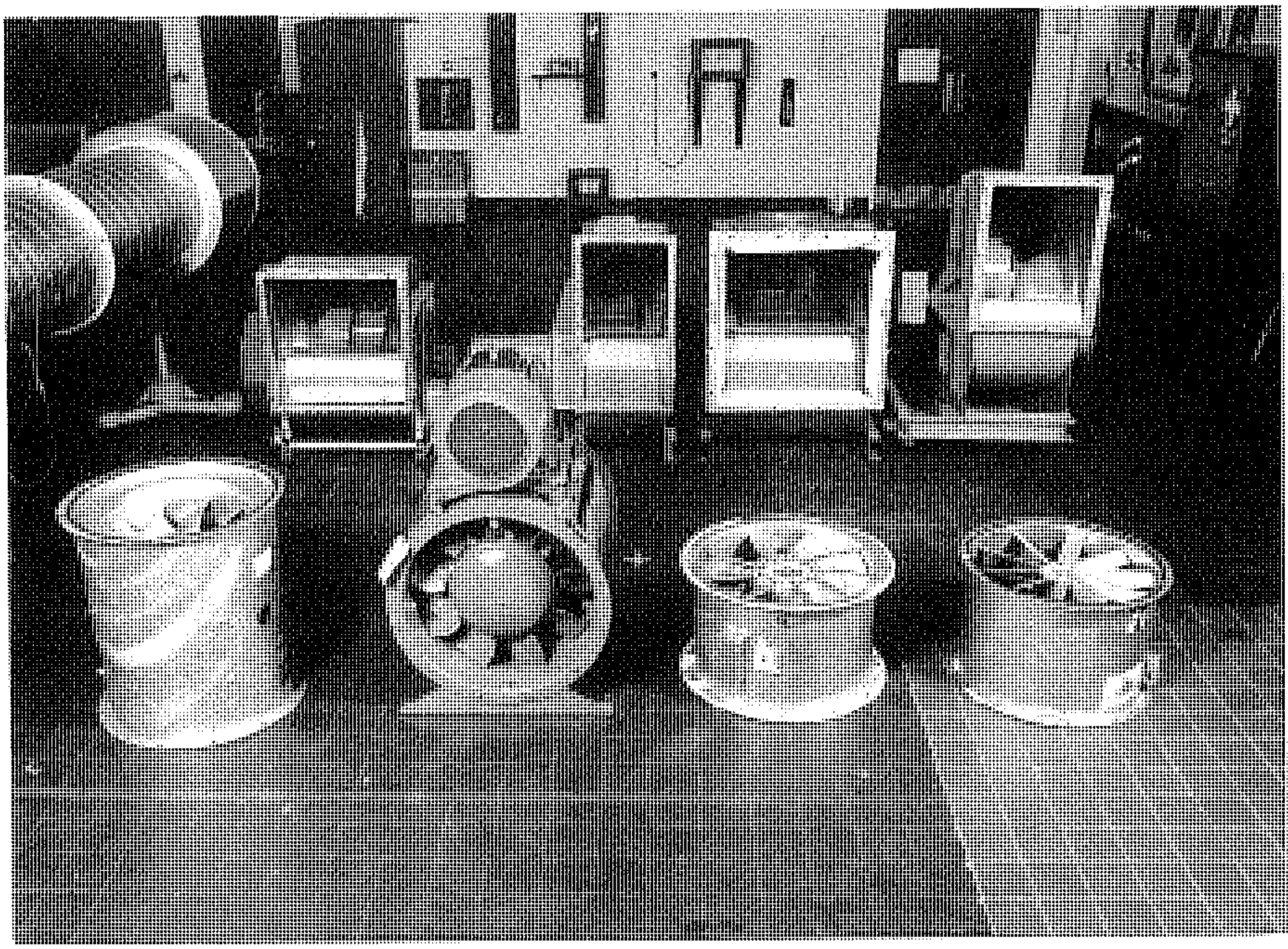

Fig. A.3 Fans used in Study

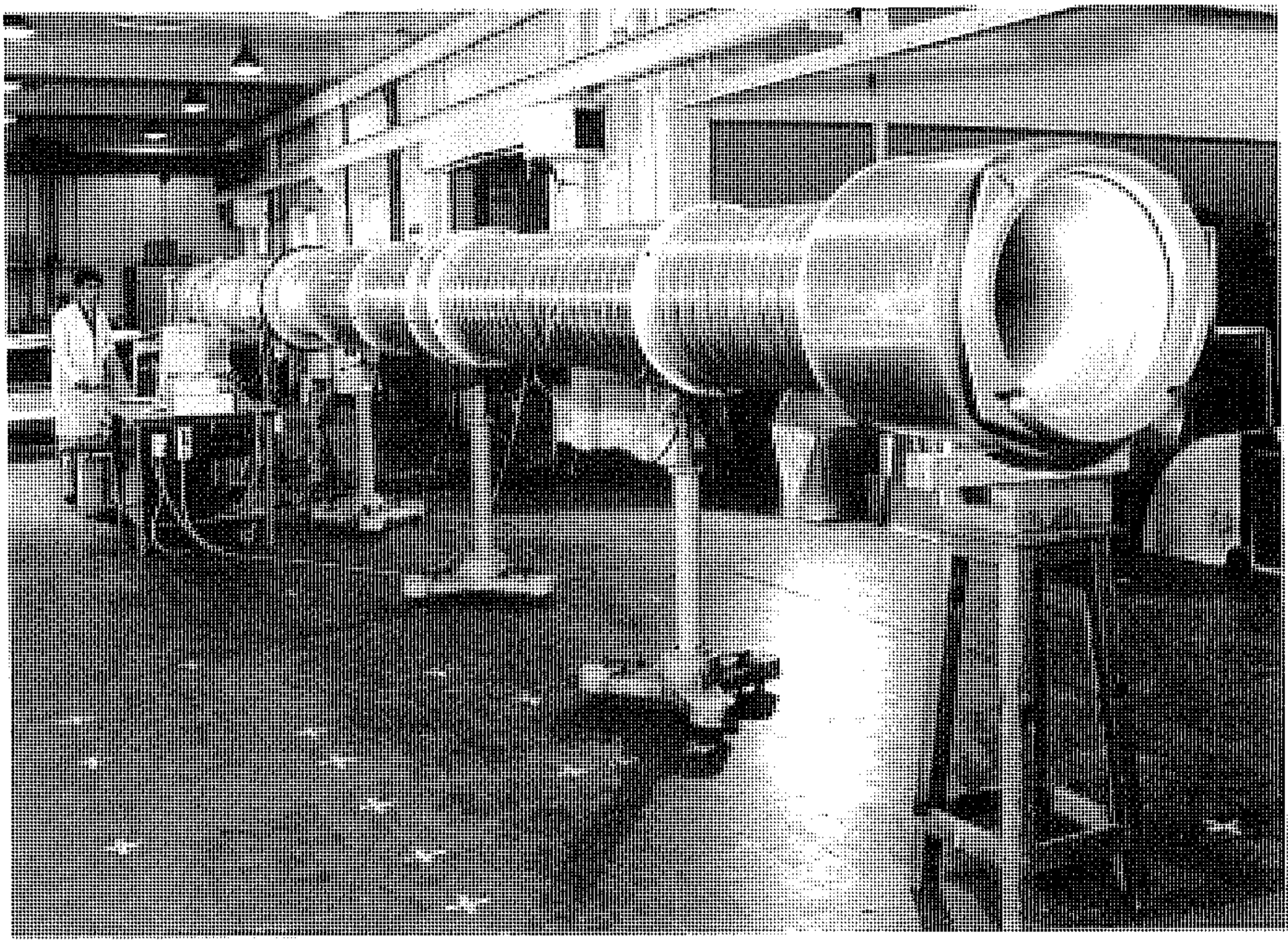

Fig. A.4 Fans under Test

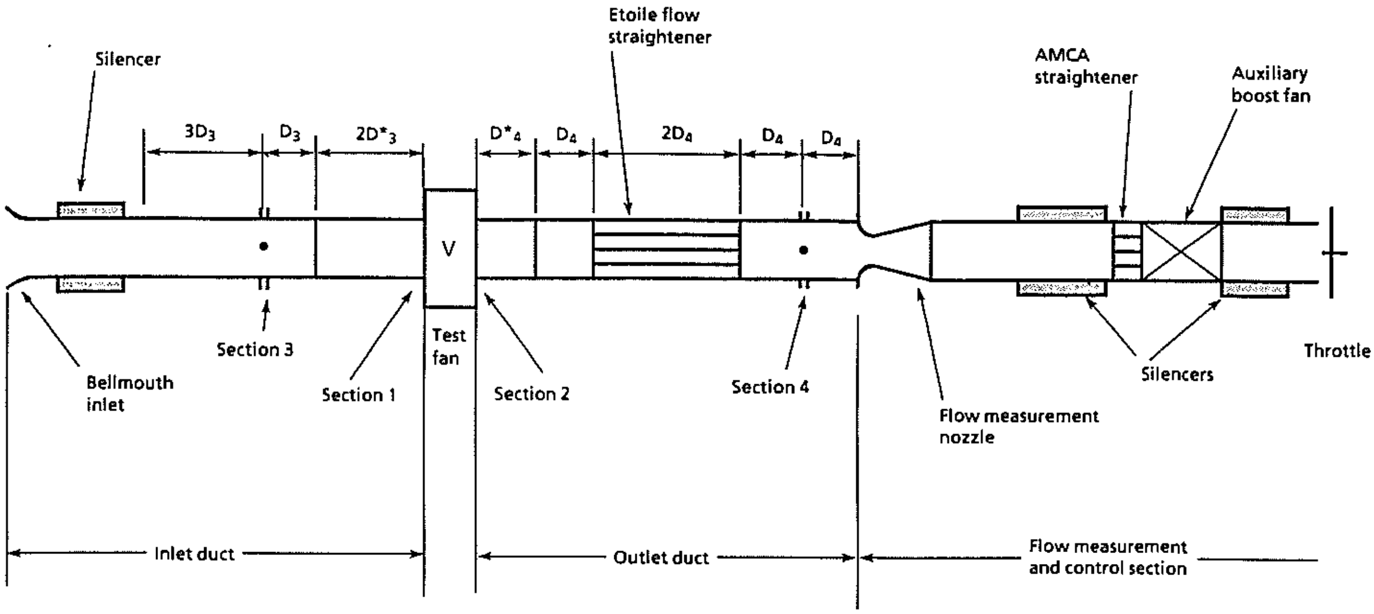

Fig. A.5 Standard Fan Aerodynamic Test Rig in Accordance with BS 848 : Part 1 : 1980

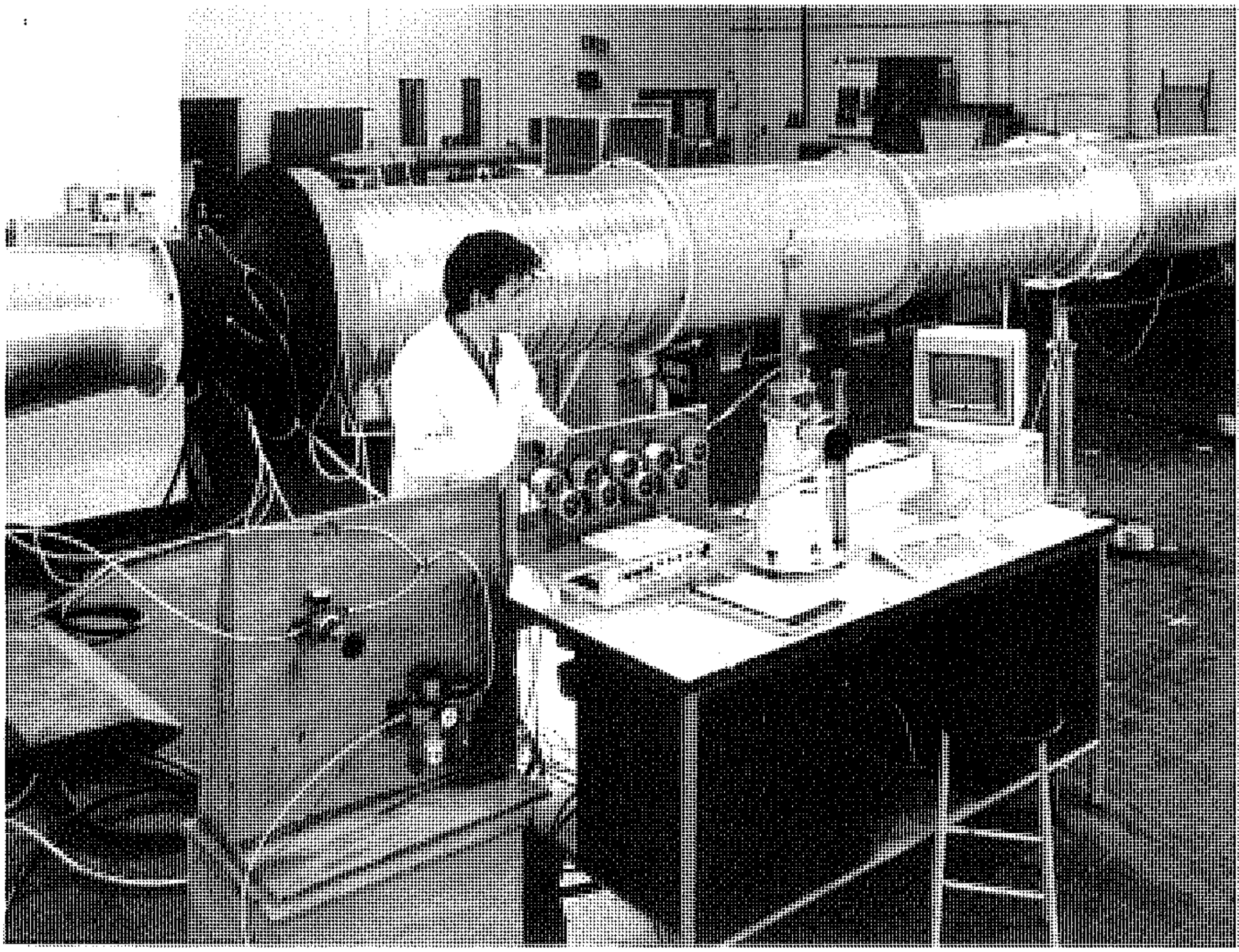

Fig. A.6 Instrumentation and Control Bench

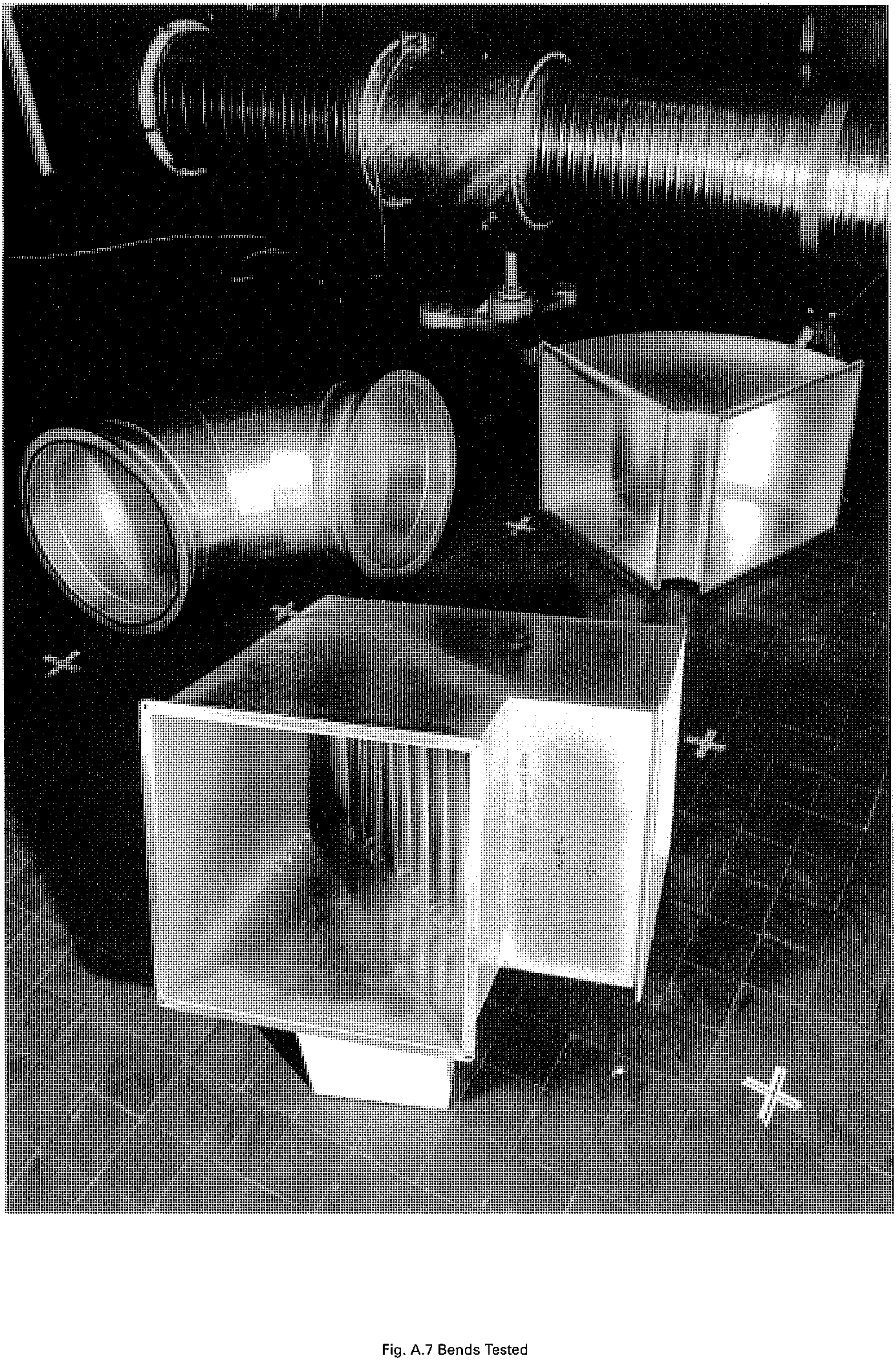

Fig. A.7 Bends Tested

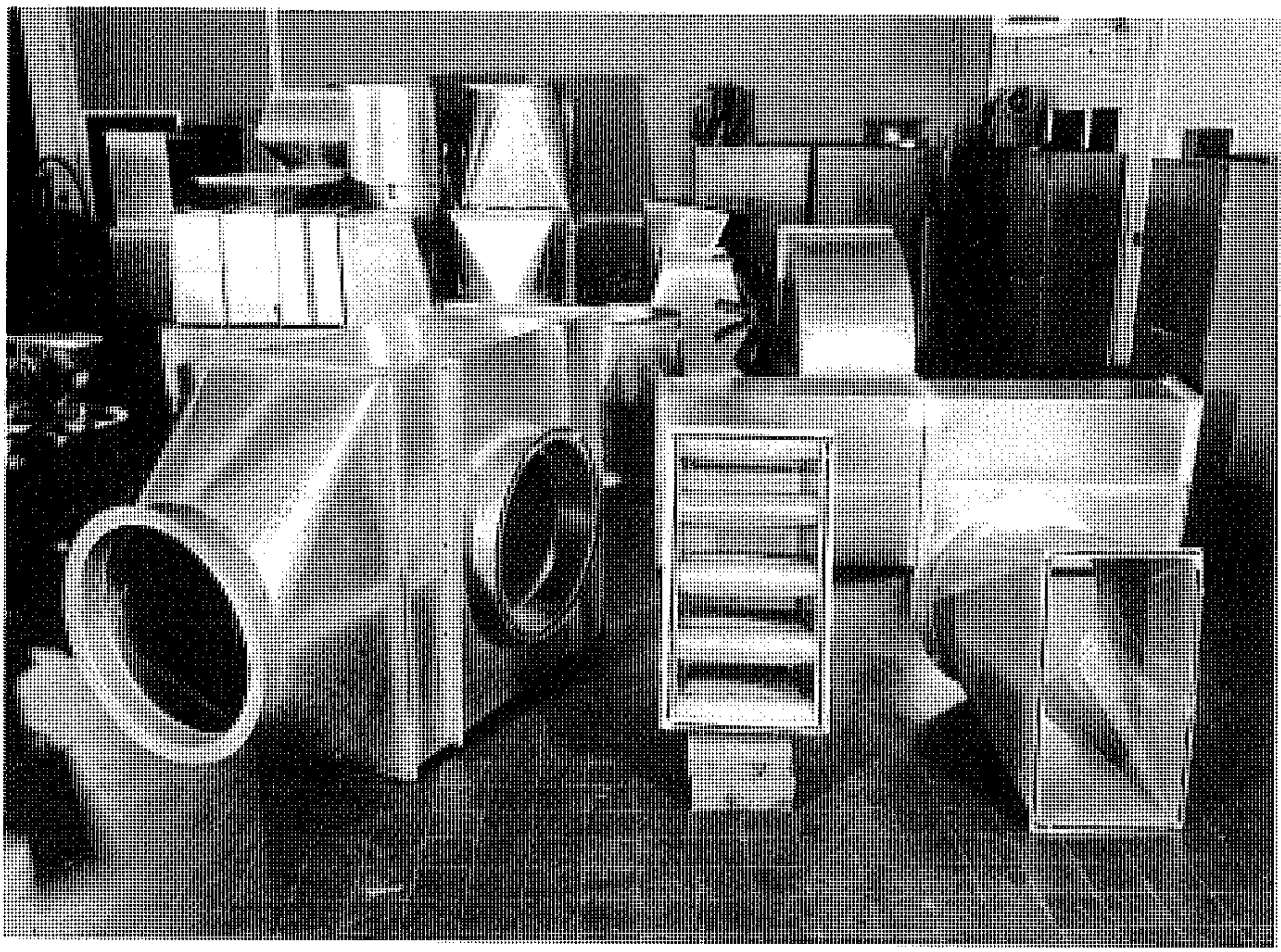

Fig. A.8 Banjo Connection, Splitter Silencer and Transition

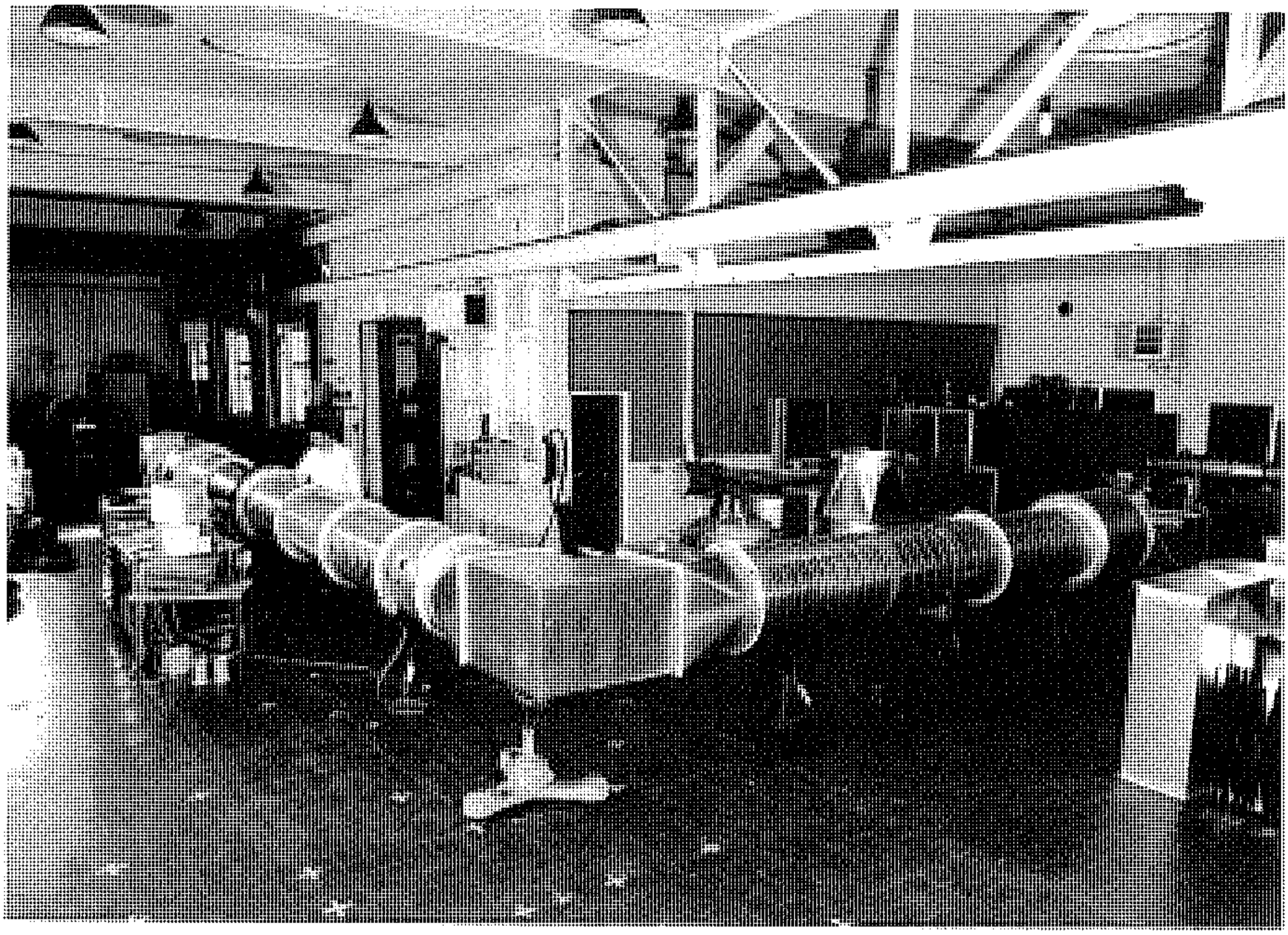

Fig. A.9 Test of Fan and Fitting at Inlet

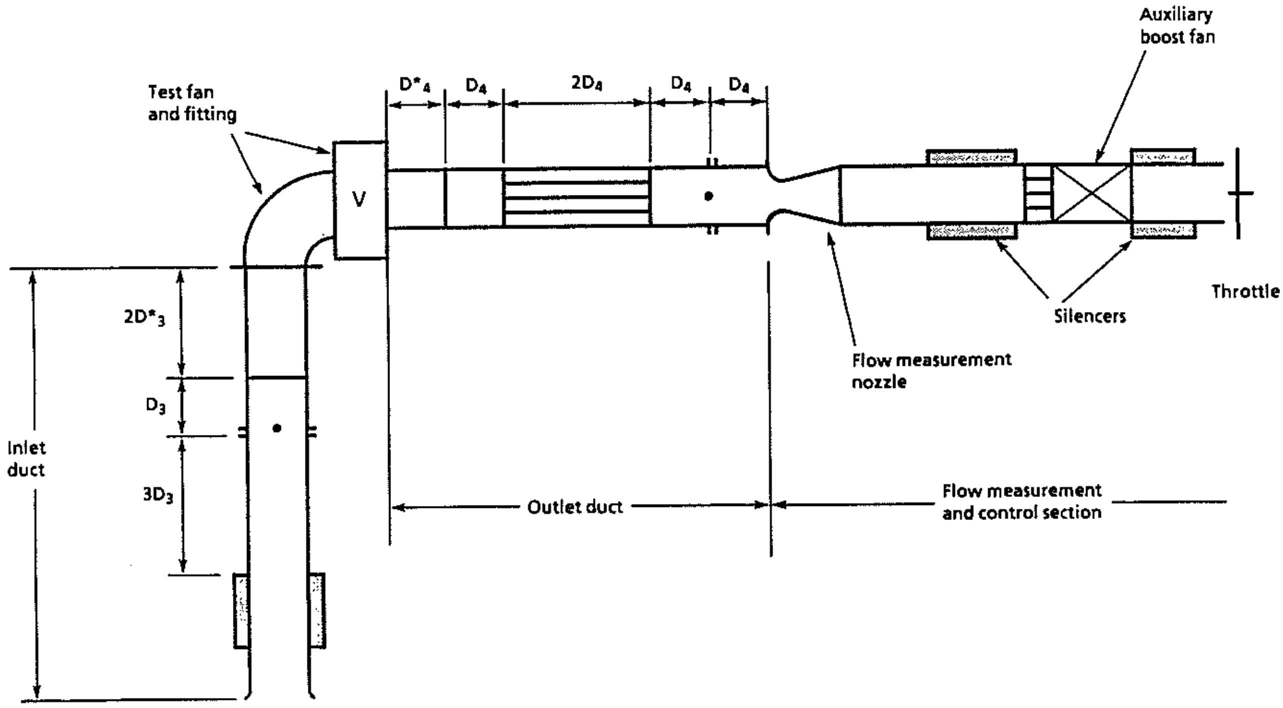

Fig. A.10 Test Rig for Determination of Installation Effect — Fitting at Fan Inlet

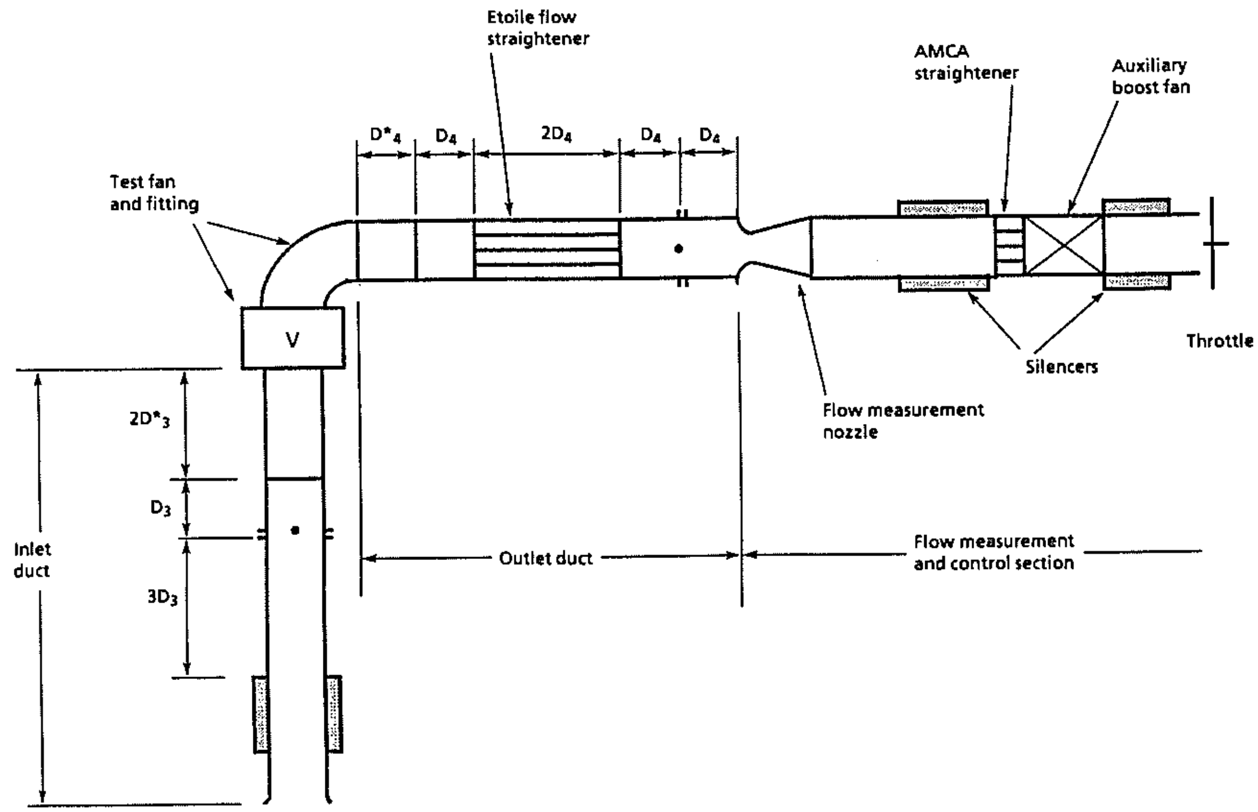

Fig. A.11 Test Rig for Determination of Installation Effect — Fitting at Fan Discharge

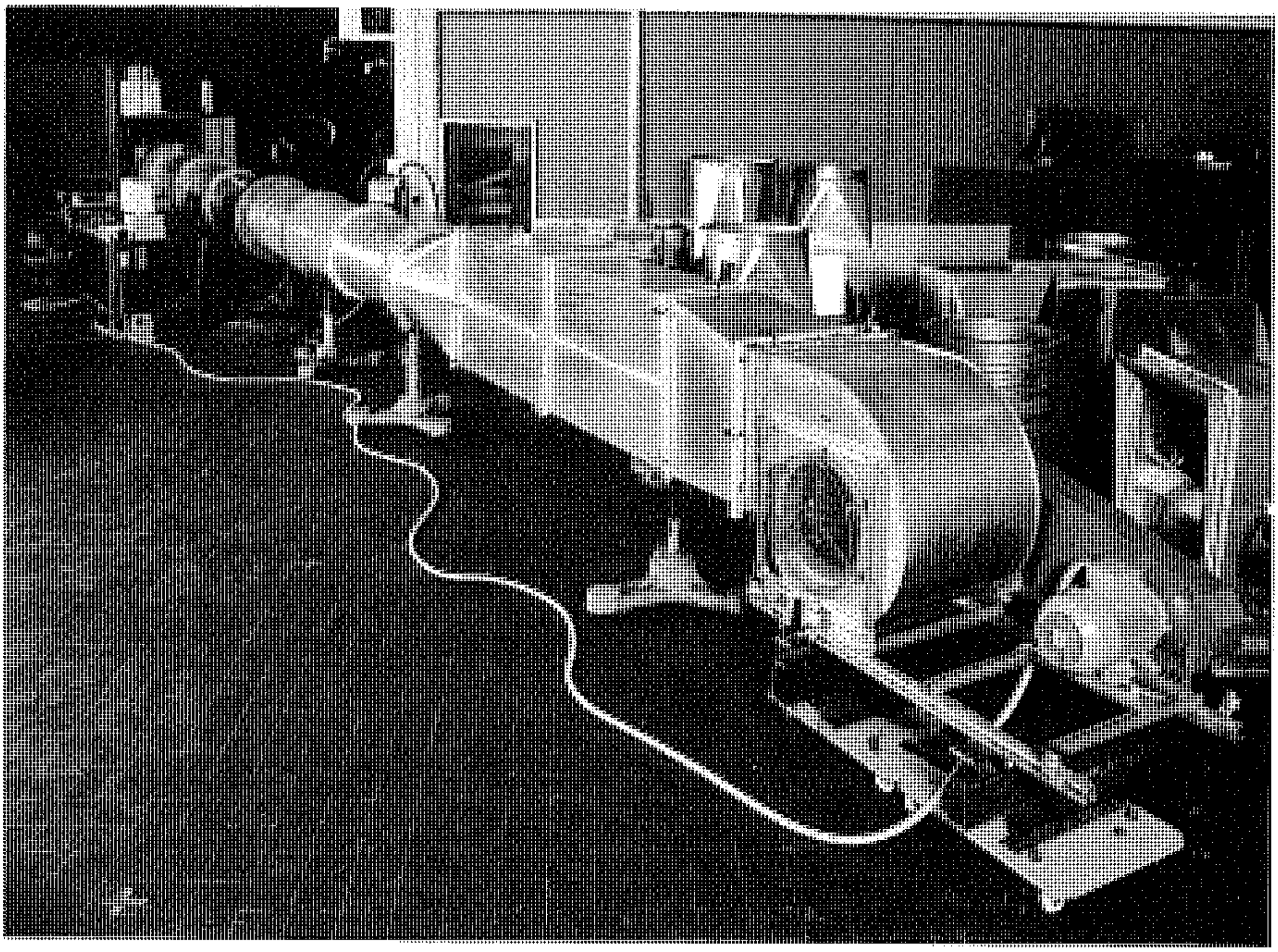

Fig. A.12 Test on Centrifugal with Fitting at Discharge

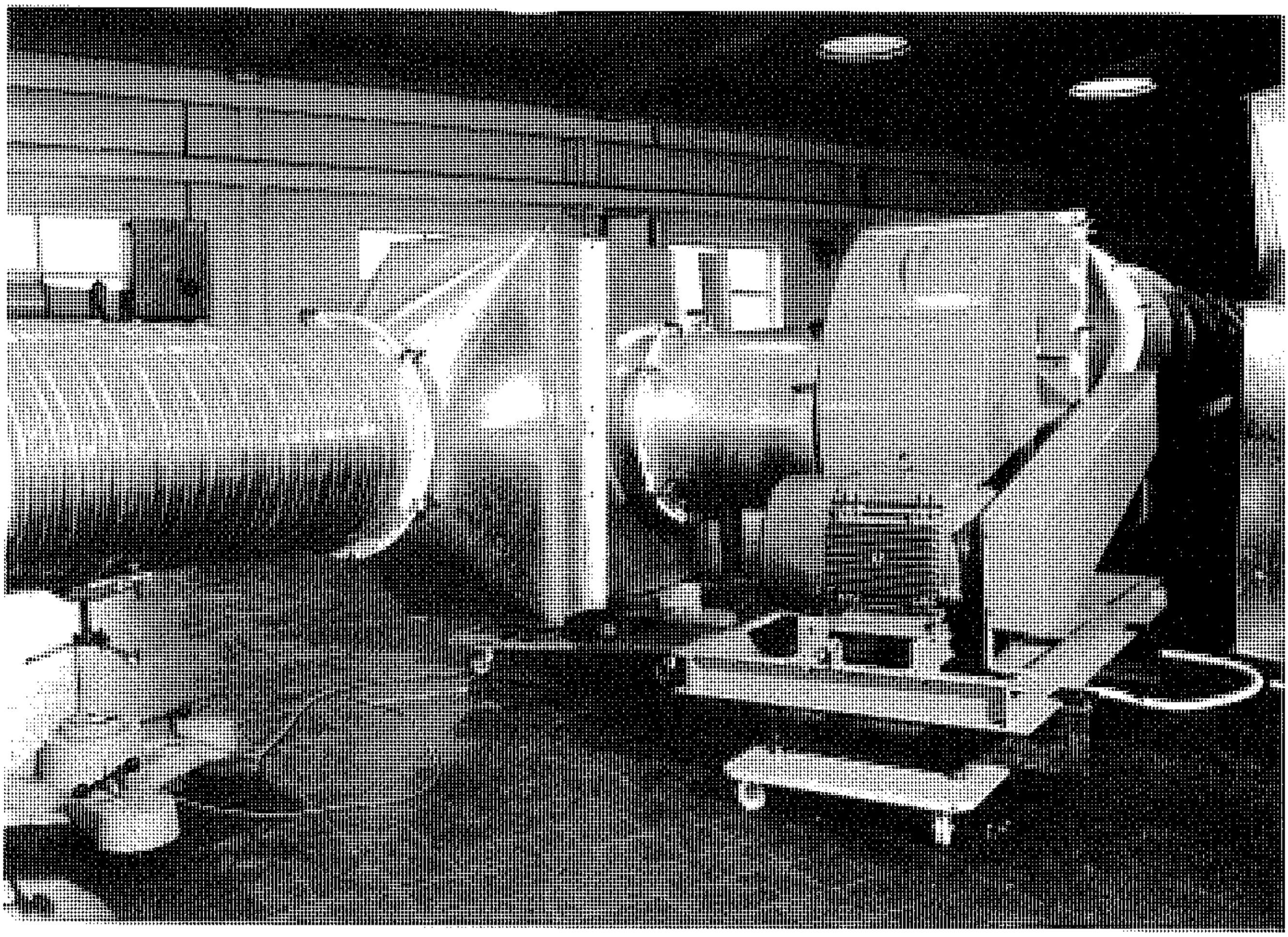

Fig. A.13 Test on Centrifugal with Banjo Connection at Inlet

APPENDIX B

Analysis Procedure

The aim of the data analysis was to compare the test performance for the fan plus fitting combination with the performance which would have been estimated using only the basic pressure loss data for the fittings and the isolated fan performance information.

The analysis procedure is most easily explained by reference to Fig. B.1.1. Firstly, the standardised data from each test was fitted by a least squares quadratic curve of the form

Total Pressure = a × (Flowrate)^2 + b × (Flowrate) + c

where a, b and c are constants.

The fan-only characteristic was then extrapolated to give the flowrate, Q0 at zero total pressure. The data obtained from the fitting-only study was also analysed in a similar form to derive loss curves of the form shown in Fig. B.1.2.

The fan-only characteristics and the fitting-only loss characteristics were then combined to derive a predicted fan plus fitting curve, see Fig. B.1.1. This predicted fan plus fitting curve can be compared with the measured characteristic of the fan plus fitting combination.

The next stage of the analysis involves the plotting of system resistance lines which intersected the fan-only characteristic at flowrates of 50, 60, 70, 80 and 90 per cent Q0. The intersection points are designated Q1. These system lines intersected the measured fan plus fitting characteristic at flowrates designated by Q3 and intersected the predicted fan plus fitting characteristic at flowrates designated by Q3 and intersected the predicted fan plus fitting characteristic at flowrates designated Q5.

If the fitting caused only a pressure loss effect and had no influence on the aerodynamic performance of the fan, then the flowrates Q3 and Q5 would be identical. If Q3 and Q5 are different, then the fitting is said to have caused an 'installation effect'. Q3:Q1 is termed the test System Flow Effect Factor and Q5:Q1 the predicted System Flow Effect Factor.

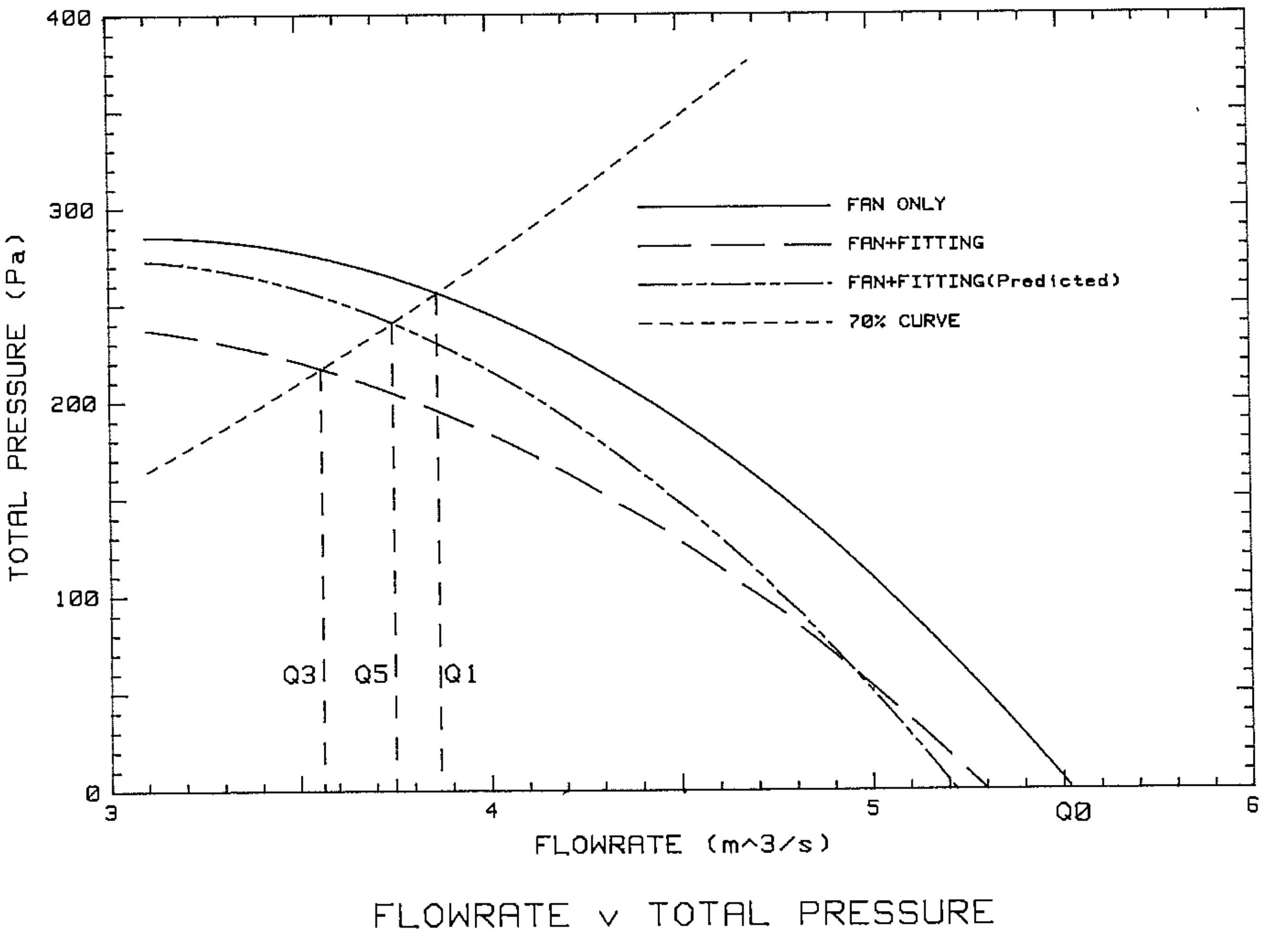

Fig. B.1.1 Analysis of Data

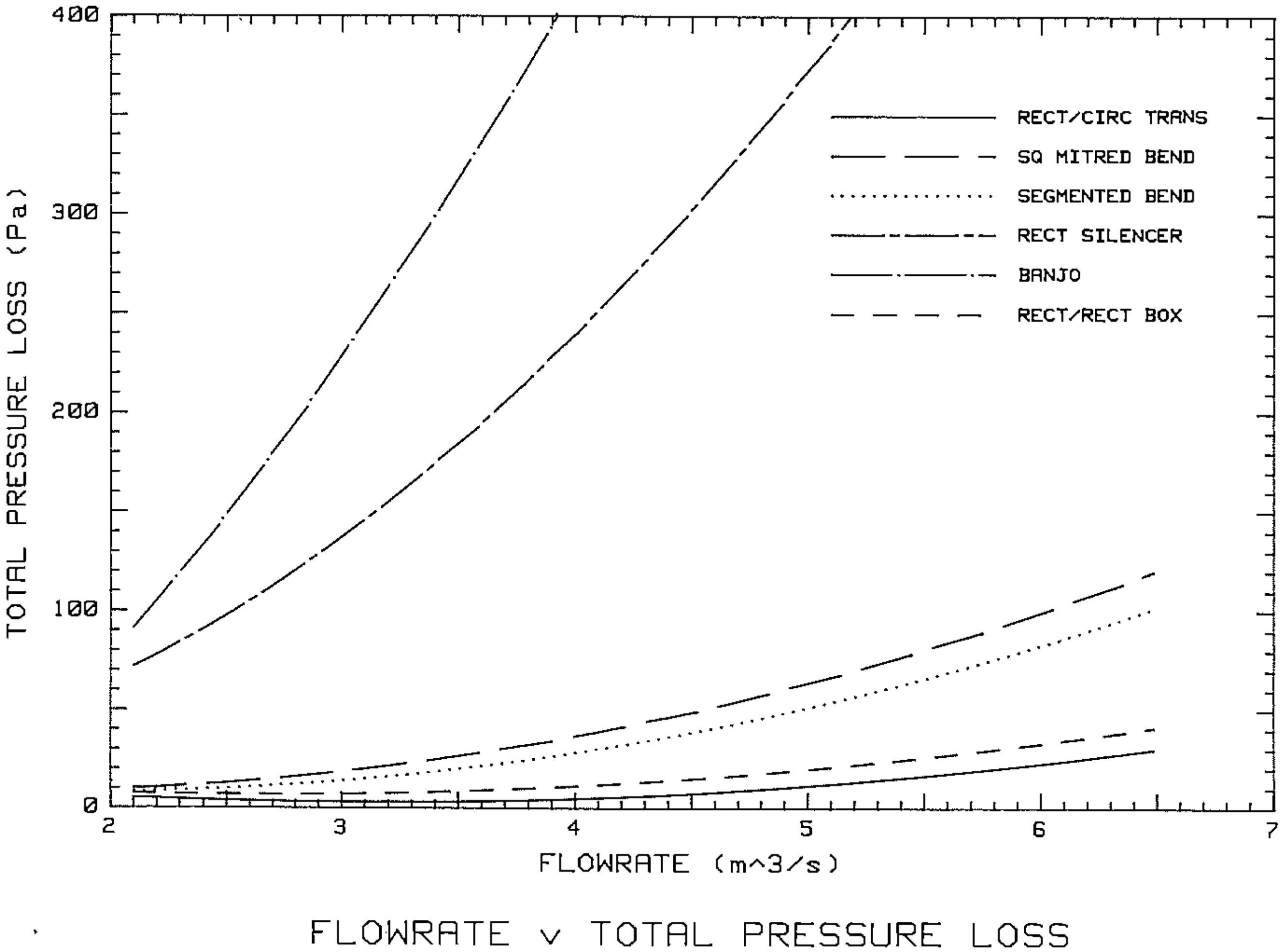

Fig. B.1.2 Example of Loss Characteristics

APPENDIX C

Worked Example

In this appendix a simplified example is given to illustrate the use of the tables in this guide.

Assuming a system has been designed to meet a duty requirement of 4.0 m^3/s of air at a standard density of 1.2 kg/m^3 and the losses in the system are estimated to be 275 Pa. ie the initial system characteristic ignoring system effect due to fan connected ductwork passes through the point — flow rate = 4.0 m^3/sec and total pressure = 275 Pa.

A fan catalogue is available from which the following fan performance is given

speed — 1440 rpm
fan diameter — 630mm

flowrate (m^3/s)	2.5	3.0	3.5	4.0	4.5	5.0
static pressure (Pa)	230	240	220	170	80	-20
input motor power (W)	1575	1800	1990	1940	1780	1430

Converting to a fan total pressure basis;

fan total pressure = fan static pressure + fan outlet dynamic pressure

where,

outlet dynamic pressure = 0.5 × air density × mean velocity^2

and

mean velocity = flowrate/gross fan outlet area.

Hence, for a flowrate of 4.0 m^3/s

mean velocity $= 4/(\pi/4 \times \text{diameter}^2)$
$= 12.83 m/s$

outlet dynamic pressure $= 0.5 \times 1.2 \times 12.83^2$
$= 98.79$ Pa

and

fan total pressure $= 170 + 98.79$
$= 268.79$ Pa

This is a little over 2 per cent down on the required pressure and some 1.5% down on the desired flow rate.

The fan overall efficiency is given by

overall efficiency = fan total pressure × flowrate/input power,
$= 268.79 \times 4/1940$
$= 0.554$

Calculating the machine's specific speed; from equation 1.

$(N_s) = (2\pi \times 1440/60) \times 4^{0.5}/(268.79/1.2)^{0.75}$
$= 5.21$

Completing the table for the other flowrates

flowrate (m^3/s)	2.5	3.0	3.5	4.0	4.5	5.0
total pressure (Pa)	268.6	295.6	295.6	268.8	205.0	134.4
overall efficiency (%)	42.6	49.3	52.0	55.4	51.8	47.0
specific speed (N_s)	4.1	4.2	4.5	5.2	6.8	9.8

The value of flow at zero total pressure, Q0, is a useful check on the validity of the data in this guide. Q0 can be calculated by either mathematical or graphical means. For this example Q0 = 5.84m^3/s and the ratio of duty flow to Q0 is 4/5.84 or .68, ie within the .6 to .8 application range. The data in this guide should be used outside of this range only with caution or when other data is not available.

Hence, the fan is an axial with best efficiency flow near the required duty point and, without considering system effect, is quite suitable for the task in hand. However, the design installation may necessitate a right angle bend near the inlet or exit of the fan.

Studying Table 1 it can be seen that fans 1 to 6 are axial with fans 1 and 5 being nearest to the specific speed value of 5.2. Thus, the designer should concentrate on data pertaining to these fans. However, a study of the data for the other four fans should not be neglected. Studying Tables 2 and 3 it can be seen that data is available for three possible bend types, ie the short square bend,, the segmented bend and the square mitred bend.

Tables 6 and 7 show that these bends have more effect on the inlet side of the fan than the discharge. If possible therefore, the bend should be fitted at the outlet. If the bend has to be situated at the inlet, re-examining Table 6 and surveying the data for all the fans, the least hazardous choice of bend would be a segmented bend preferably with 1D of straight duct between fan and bend.

Examining Table 4 it is seen that even for the segmented bend fitted to fan 1, a system flow effect factor ranging from .93 to .975 results. Thus, instead of the required duty of 4 m^3/s, a flowrate of the order of 3.72 — 3.9 m^3/s will be achieved. If the flowrate is critical a fan with a duty flow of 4.3 m^3/s (ie 4/.93) and total pressure of 320 Pa (ie $275/.93^2$) would have to be selected to ensure meeting the required 4 m^3/s throughput.