A design philosophy for centrifugal fans

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A DESIGN PHILOSOPHY
FOR CENTRIFUGAL FANS

by
John Stephen Hunter
(B.Eng. C.Eng. MIMech E.)

A masters thesis submitted in partial fulfilment of the requirements for the award of Master of Philosophy of the Loughborough University.

December 1996

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ABSTRACT

This thesis outlines the areas that need to be considered during the design and development of centrifugal fans, and in particular noise reduction techniques.

The main subjects covered are:

(a) Fan performance and noise level assessment.
(b) A description of the various fan types available and their uses.
(c) An explanation of how specific speed and performance coefficients can be used to determine the optimum fan type for a given application.
(d) Fan design concepts and their effect on noise and performance.
(e) Performance and noise level prediction methods.
(f) Experimental methods used for the design and selection of fans.
(g) Vibration measurement and balancing techniques.
(h) Stress calculations and tests to confirm the mechanical integrity of fans.

During this research a new range of fans for use in industrial dust collectors has been developed which offer the following improvements over the existing fan range;

(1) Increased efficiency and associated power savings.
(2) Reduced noise levels.
(3) Cost savings.

Key-words: Air
Blade
Centrifugal
Efficiency
Fan
Fluid
Impeller
Noise
Performance
ACKNOWLEDGEMENTS

I would like to thank the following people for their help and assistance during this research programme:

DCE Group Ltd. Personnel, in particular Mr P.D. Collins (Development Manager).

Loughborough University Personnel, in particular Professor M. Preston.

Professor G. Chapman of DeMontfort University (Executive head of Engineering & Manufacture, formerly Senior Lecturer at Loughborough University).
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<td>AD</td>
<td>Acoustic Diffuser</td>
</tr>
<tr>
<td>BTR</td>
<td>British Tyre &amp; Rubber</td>
</tr>
<tr>
<td>Dal.</td>
<td>Dalamatic</td>
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<tr>
<td>DCE</td>
<td>Dust Control Equipment</td>
</tr>
<tr>
<td>DI</td>
<td>Directivity Index</td>
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<tr>
<td>DLMV</td>
<td>Dalamatic Venting Unit</td>
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<tr>
<td>DTI</td>
<td>Department of Trade and Industry</td>
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<td>DU</td>
<td>Dalamatic Unit</td>
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<td>F-Range</td>
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<td>FED</td>
<td>Factorial Experimental Design</td>
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<td>FFT</td>
<td>Fast Fourier Transform</td>
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<td>G-Range</td>
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<tr>
<td>K-Range</td>
<td>New DCE fan range</td>
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<td>LEP,d</td>
<td>Daily Personal Exposure to noise</td>
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<td>Leq</td>
<td>Equivalent continuous sound level</td>
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<td>NC</td>
<td>Noise Criteria</td>
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<td>NEL</td>
<td>National Engineering Laboratory</td>
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<tr>
<td>NR</td>
<td>Noise Rating</td>
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<tr>
<td>SERC</td>
<td>Science and Engineering Research Council</td>
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<tr>
<td>SI</td>
<td>Sintamatic Insertable</td>
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<td>Sint.</td>
<td>Sintamatic</td>
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<td>SPL</td>
<td>Sound Pressure Level</td>
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<td>SU</td>
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NOMENCLATURE

A  Area
b  Width
C_d Discharge coefficient
d  Diameter
e_{pr} Permissible residual specific unbalance
E_{if} Efficiency
E_{if_m} Motor efficiency
F_c Centrifugal force
k  Loss coefficient
k_r Pressure coefficient
k_{pw} Power coefficient
K_{PW} Fan Power coefficient
k_q Flow coefficient
K_Q Volume coefficient
K_S Fan static pressure coefficient
K_T Fan total pressure coefficient
M  Mach number
m  Mass
n  Speed of rotation
n_s Specific speed
P  Pressure
P_A Air power
P_o Atmospheric pressure
P_S Static pressure
P_{SF} Fan static pressure
P_{SI} Static pressure at fan inlet
P_{SO} Static pressure at fan outlet
P_T Total pressure
P_{TF} Fan total pressure
P_{TI} Total pressure at fan inlet
\( P_{to} \)  Total pressure at fan outlet  
\( P_v \)  Velocity pressure  
\( P_{vi} \)  Velocity pressure at fan inlet  
\( P_{vo} \)  Velocity pressure at fan outlet  
\( P_w \)  Power  
\( P_{wa} \)  Absorbed power  
\( P_{ws} \)  Shaft power  
\( Q \)  Volume flow rate  
\( r \)  Radius  
\( Re \)  Reynolds number  
\( T \)  Temperature  
\( T_s \)  Shaft torque  
\( u \)  Impeller peripheral velocity  
\( U_{per} \)  Permissible residual unbalance  
\( v \)  Absolute velocity  
\( v_d \)  Duct velocity  
\( v_r \)  Relative velocity  
\( v_u \)  Tangential velocity  
\( v_m \)  Radial (or meridional) velocity  
\( v_e \)  Impeller eye velocity  
\( v_w \)  Swirl velocity  
\( z \)  Number of blades  
\( \beta \)  Blade angle  
\( \beta_i \)  Pre-swirl angle  
\( \delta \)  Density or diameter coefficient (non dimensional)  
\( \delta_o \)  Standard air density  
\( w \)  Angular velocity  
\( \sigma \)  Poissons ratio or speed coefficient (non dimensional)  
\( \sigma_r \)  Radial stress  
\( \sigma_t \)  Tangential stress  
\( \alpha \)  Clearance  
\( \mu \)  Viscosity
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<tr>
<td>( \phi )</td>
<td>Volume coefficient (non dimensional)</td>
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<tr>
<td>( \phi_a )</td>
<td>Apparent flow coefficient</td>
</tr>
<tr>
<td>( \psi )</td>
<td>Total pressure coefficient (non dimensional)</td>
</tr>
<tr>
<td>( \psi_a )</td>
<td>Apparent pressure coefficient</td>
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<tr>
<td>( \psi_s )</td>
<td>Static pressure coefficient (non dimensional)</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>Power coefficient (non dimensional)</td>
</tr>
<tr>
<td>( \tau )</td>
<td>Throttling coefficient (non dimensional)</td>
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1.0 INTRODUCTION

1.1 The Company and Products

1.1.1 The company

DCE Group Ltd. are market leaders in the design and manufacture of dust control equipment worldwide. It is a wholly owned subsidiary of BTR Plc. and DCE companies are located in Australia, France, Germany, Japan, Netherlands, Scandinavia, South Africa, Spain, USA, as well as the UK. It employs approximately 600 people, over half of whom are employed in the UK by the headquarters company at Thurcaston, Leicester. DCE are a medium sized company with an annual turnover of approximately £50 million worldwide.

DCE's products cover a range of industrial dust collectors which are recognised as the most technically advanced available. DCE have considerable expertise in the field of dust control with over 75 years experience and with more than a quarter of a million dust filters controlling dust in industrial environments all over the world.

DCE’s main production facility at Leicester represents the state of the art for batch sheet metal fabrication. Large component volumes, generated by high worldwide unit sales, have justified investment in the latest and most accurate manufacturing methods. This includes the use of laser cutting technology, CNC production equipment, machine robots and automated paint line. An essential element in group strategy is the intent to maintain a lead in terms of product technology, whilst continuing to embrace the very latest technology in design, manufacture and business systems.
DCE's expertise was recently recognized with the award - Le prix de promotion internationale de l'industrie - defense de l'environnement. We are the only dust control company ever to have received this award for environmental engineering.

As part of the annual budgeting requirements of BTR, DCE group Ltd. produce a 3 year strategic plan. This attempts to look forward to where the company is heading and to determine appropriate management actions to achieve its goals. The company development programme was highlighted in the last plan and high priority was given to:

(a) Upgrading the existing products in term of technical design and automated manufacture.
(b) Reducing unit costs by better design.
(c) Widening the product base into new markets.

1.1.2 Products

DCE makes the world's largest selling range of intermittent and continuous dust filters. These compact designs are engineered to pack the maximum effective filtration area into the minimum space. Six main dust collector ranges are manufactured, these being:

(a) Unimaster  
(b) Dalamatic  
(c) Sintamatic  
(d) Cased Units  
(e) Unit Concept  
(f) Silo-air

During the writing of this thesis the following additional products have been introduced:

(a) Excell  
(b) Unicell
The Unimaster is applied to single or small dust generation points and assures the user of low capital, operating and energy costs. They are simple to maintain and easy to relocate when necessary. The development of the 'unit concept' is one of DCE's major contributions to better dust control. To match the users application precisely, the versatile Unimaster has more than 500 combinations. This dust collector range is designed for intermittent duty and the filter bags are automatically cleaned when the unit is shut down to remove dependence on the operator.
(b) Dalamatic

The Dalamatic dust collector range are designed for continuous operation with minimum attention. The filter bags are continually subjected to the action of a reverse-jet cleaning system. At regular intervals each filter bag in turn receives a brief burst of compressed air which dislodges the dust from the bags. Dimensional versatility is imperative and the Dalamatic has a large choice of configurations, sizes and types, plus a range of regular and specialist filter fabrics. Within the dust chamber the filter-pocket internal support-frame or 'inserts' have been specially redesigned to last substantially longer than traditional designs.

Fig 2. Dalamatic Dust Collector
This product uses DCE's most advanced filtration media. This major breakthrough incorporates a dust filter medium of high-grade porous composites, with a microporous coating of PTFE which unlike traditional fabric media, produces rigid self supporting elements. The dust collection efficiency is unmatched by fabric, cartridge, or single-stage filters, and it has a high resistance to liquids, acids, and alkalis. The corrugated surface increases filtration area to three times that of conventional fabric filters, ensuring that the maximum filtration area can be accommodated into the minimum of space. Continuous operation of the filter is ensured by the electronically controlled automatic reverse jet cleaning system.

Fig.3, Sintamatic Dust Collector
(d) Cased Units

Fig. 4, Cased dust collector

(i) Dalamatic Cased Units
Each unit is built up in banks and tiers to match the capacity and configuration required by the customer. Each filter cell contains ten Dalamatic filter bags and like other Dalamatic units they are designed for continuous operation. A different fan is required for each application, which is normally supplied by a fan manufacturer on a one off basis.

(ii) Sintamatic Cased Units
The capacity of these units can be matched to the customer requirements by changing the number of banks, however they are always one tier high. Each filter cell contains ten Sintamatic filter elements and like other Sintamatic units it is designed for continuous operation at high collection efficiencies. Like the Dalamatic cased unit a different fan is required for each application, which is supplied by a fan manufacturer on a one off basis.
(e) Unit Concept
The marketing department at DCE found that there was a requirement for a self contained unit that is suitable for medium to large installations. For many applications this is an alternative to a Dalamatic or Sintamatic cased unit. The standard housing can be fitted with either Dalamatic filter bags or Sintamatic elements depending on the customers requirements, and is designed for continuous operation.

Fig.5 Unit Concept Dust Collector
The Unit Concept has the following main advantages and disadvantages when compared with cased units;

**Advantages**

(i) Because there are a limited number of unit variations it has been possible to select a standard fan range. These are supplied by a fan manufacturer, but because they are a standard range and not one off orders there are significant cost savings.

(ii) The fan is enclosed in an acoustically lined fan chamber as standard which reduces noise levels.

(iii) Ducting is not required from the dust collector to the fan which reduces costs and simplifies installation.

**Disadvantages**

(i) The Unit Concept is not of a modular construction, therefore the available sizes are limited to a standard range.
(f) Silo-air

The Silo-air is designed for continuous operation on silo venting applications. The cartridge filters which are manufactured from spun bonded polyester material are continually subjected to the action of a reverse-jet cleaning system like the Dalamatic and Sintamatic dust collectors. The unit has excellent noise reduction features, comprising of built-in fan attenuation using a perforated fan scroll and acoustically lined volute. An additional acoustic diffuser is also available for use in areas where noise levels are particularly sensitive.

![Fig.6 Silo-air Dust Collector](image_url)

Note.

This unit was developed and launched during 1992 and incorporates the perforated fan scroll concept detailed in section 6.2.7
1.1.3 DCE fan ranges.

DCE use a variety of fans in their dust collectors. Most are manufactured in house but some are bought in from various fan manufacturers. The table below shows the different fan ranges that are used for the various products.

<table>
<thead>
<tr>
<th>Fan Range</th>
<th>DCE Manufacture</th>
<th>UNIMASTER</th>
<th>DALAMATIC</th>
<th>SINTAMATIC</th>
<th>CASED</th>
<th>CONCEPT</th>
<th>SILO-AIR</th>
</tr>
</thead>
<tbody>
<tr>
<td>G-RANGE</td>
<td>YES</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>F-RANGE</td>
<td>YES</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>K-RANGE</td>
<td>YES</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>KS-RANGE</td>
<td>YES</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>SOLYVENT VENTEC LTD</td>
<td>NO</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>STOCKBRIDGE AIRCO LTD</td>
<td>NO</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>OTHER FAN SUPPLIER</td>
<td>NO</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 1, DCE fan ranges.

Notes.

G-Range    - Existing fan range primarily designed for fitment to Unimaster range of dust collectors, (see section 10.1).

F-Range    - Existing fan range primarily designed for fitment to Dalamatic range of dust collectors, (see section 10.1).

K-Range    - New fan range designed to be interchangeable and are therefore not unit specific, (see section 10.1).

KS-Range   - New fan range same as K range but with perforated scroll design (see section 6.2.7).

(a) Shaded boxes indicate that this fan range is used for the product above.
(b) The above table does not include new products that have been introduced since 1992 (see section 1.1.2 for new products).
1.2 International Noise Legislation

At least 10,000 people in the UK suffer from noise induced deafness and 1.7 million workers are estimated to be at risk from noise hazards in this country alone.

All employers have a general duty under the Health and Safety at Work Act 1974 to reduce risks to the lowest level reasonably practicable, which will include situations where noise may threaten health. The Health and Safety Commission has for some time thought the law should be reinforced. In 1981 it published draft regulations and guidance, but they were not developed because shortly afterwards the European Commission proposed a directive to establish a common European basis for law on noise at work. The directive was adopted in 1986, setting a 1990 deadline for any new legislation needed. In December 1987, the Health and Safety Commission published a fresh consultation document on how the directive could be satisfied in Great Britain. After receiving final proposals from the Health and Safety Commission the Government introduced the Noise at Work Regulations 1989, which came into force on January 1st, 1990.

The regulations deal only with hearing damage risks. Any other health, safety or welfare problems the noise might create would continue to be dealt with under the Health and Safety at Work Act. Regulation 6 reiterates the general principle of the Act that for any level of exposure, risks must be reduced to the lowest level reasonably practicable. Other regulations specify measures which must be taken where people are likely to be exposed to noise at or above three action levels. The levels of daily personal exposure to noise (designated LEP,d) can be regarded as the noise dose over a whole working day, measured without taking account of any ear protectors the workers are using. It is the same as the equivalent continuous sound level (designated Leq) of the old code, only the name is changed.
The three action levels laid down in the regulations are;

The first action level ........... where there is a daily personal noise exposure of 85 dB(A) or over.

The second action level ........... where there is a daily personal noise exposure of 90 dB(A) or over.

The peak action level ........... where a peak of 200 Pascals (140 dB re 20x10^-6 Pa) is reached or exceeded.

Taking a long term view, it is worth recognising that many members of the European Community favour a lower minimum level than that laid down in the regulations. The European Commission directive is due for review and there will undoubtedly be pressure to reduce the present action levels.

BS4142 "Method of rating industrial noise affecting mixed residential and industrial areas" [9] is intended to meet the need for rating various noises of an industrial nature affecting persons living in the vicinity. It gives a method of determining a noise level, together with procedures for assessing whether the noise in question is likely to give rise to complaints.

In general, a noise is liable to provoke complaints whenever its level exceeds the background noise by a certain margin, or when it attains a certain absolute level. The method of rating in this standard depends on comparing the specific noise level, corrected to take account of its character, with the background noise level.
Machine makers and suppliers

It is a requirement of the Health and Safety at Work Act that designers, manufacturers, importers and suppliers of articles and substances used at work must take reasonably practicable action to deal with risks their products create and to provide information about the products they supply. This will mean, for example, that where machines are capable of producing hazardous noise, the emission will need to be reduced to the lowest value reasonably practicable.

The new regulations supplement this duty by specifying that if machines are likely to cause employees to be exposed at or over the first action level adequate information in sales and operating literature must be provided on the noise likely to be generated. Where the second action level is likely to be equalled or exceeded, signs must also be attached to the equipment to warn of the danger to hearing.

It is considered that the most effective way of reducing noise levels at work is to incorporate noise prevention measures into the design of installations and the priority aim must be to achieve such noise reduction at source. The use of hearing protection is a complementary measure where exposure cannot reasonably be avoided by other means.
1.3 Summary & Conclusions

DCE manufacture a large range of dust collectors most of which are fitted with DCE manufactured fans.

Because of the Noise at Work Regulations 1989, which came into force on January 1990, DCE felt that it was necessary to reduce the dust collector noise levels.

Previously this problem had been addressed by improving the design of the acoustic diffusers which attenuate the fan outlet. However, it was decided that in order to reduce noise levels further the noise problem would have to be tackled at source, i.e. a new range of quieter fans was required. Initially it was intended to purchase a new range of quieter fans from a fan supplier. However, the increased cost and difficulty in finding fans that would fit within DCE’s standard range of dust collectors (see 4.0 Market Survey) meant that a new approach had to be considered.

It was decided to try and develop a range of quieter fans that could be manufactured in-house by DCE. In order to do this a Teaching Company Scheme between DCE & Loughborough University of Technology and sponsored by the DTI & SERC was set-up. The author was employed to carry out this project and this thesis is based on the work undertaken during the development of these fans.
2.0 FAN PERFORMANCE & NOISE LEVEL ASSESSMENT

2.1 Performance Evaluation

The main purpose of a fan is to achieve and maintain an air volume flow rate against the resistance of the system.

Fan performance is usually presented for selection purposes in the form of a characteristic curve of Air Volume Flow against the Fan Static Pressure developed by the fan.

2.1.1 Fan Static Pressure

For a fluid flowing through a pipe, Bernoulli’s equation states;

\[ P_1 + \frac{1}{2} \gamma v_1^2 = P_2 + \frac{1}{2} \gamma v_2^2 + \Delta P \]  \[ \text{[36, page 8]} \]

Where;

\[ P = \text{Static pressure (} P_s \text{)} \]
\[ \frac{1}{2} \gamma v^2 = \text{Velocity pressure (} P_v \text{)} \]

\[ P_{s1} + P_{v1} = P_{s2} + P_{v2} + \Delta P \]

Pressure loss along pipe, \( \Delta P \) = \((P_{s1} + P_{v1}) - (P_{s2} + P_{v2})\)

Although in most practical cases the air density remains constant, this will not be the case where the height between parts of the system are considerable, or there is a temperature gradient.

In the case of flow through a fan the pressure supplied by the fan is analogous to the pressure loss along a pipe but \( \Delta P \) is negative.

(i.e. Total pressure supplied by fan = - pressure loss along pipe).

The total pressure developed by a fan is the difference between the total pressure at the fan outlet and the total pressure at the fan inlet.

Fan total pressure, \( P_{TF} = (P_{s2} + P_{v2}) - (P_{s1} + P_{v1}) \)
BS848:Part1:1980 [4, page 26, clause 17] specifies that fans are classified according to one of four installation types. The method used for test purposes should be selected which most closely represents the configuration of the installed equipment. For DCE fans this is normally installation type C which is ducted inlet and free outlet.

Clause 29 of BS848 [4, page 60] details standard test methods with inlet side test ducts (type C installation). It also states that for this installation type the following expression applies which can be used to calculate the Fan Static Pressure;

\[
\text{Fan Static Pressure, } P_{SF} = - P_{TI} = -(P_{SI} + P_{VI})
\]

Where:
- \( P_{TI} \) = Total pressure at fan inlet
- \( P_{SI} \) = Static pressure at inlet
- \( P_{VI} \) = Velocity pressure at inlet

On the inlet side of the fan the static pressure is below atmospheric pressure, (ie. \( P_{SI} \) is negative).

This Fan Static Pressure, usually corrected to a standard air density at 20°C and ambient pressure, is plotted against Air Volume Flow to produce the fan performance curve (see section 2.1.3).

2.1.2 Motor Power

Motor Shaft Power is also commonly plotted against the Air Volume Flow Rate. The Motor Shaft Power is calculated from the Motor Absorbed Power which is normally measured using a Power Analyzer.

\[
\text{Motor Shaft Power} = \text{Motor Absorbed Power} \times \text{Motor Efficiency}
\]

The Motor Shaft Power is also usually corrected to a standard air density at 20°C.
2.1.3 Correction for Temperature

Fan performance curves are normally corrected to a standard air pressure of 101.325 kPa at 20°C. A change of air density will change both the fan performance curve and the system resistance curve. For a given volume flow rate fan pressure and system resistance are both proportional to the air density. High temperature or high altitude will change the air density. Although the volume flow is unchanged the mass flow is changed (being proportional to density). It is important to consider this in heat exchange calculations and for fans operating at high altitude. The fan laws can be used to correct for density changes (see chapter 7.1.1), however, generally for most applications which are at ambient conditions it is not necessary to consider the effect of temperature.

Note.
Temperatures in degrees Kelvin must be used for all calculations.
Example.
Reference Temperature = 20°C
Required Temperature = 100°C

(a) The air volume flow is not effected by density changes.
\[ Q_2 = Q_1 \]

(b) For a given flow rate the pressure is proportional to density.
\[ \frac{P_2}{P_1} = \frac{T_1}{T_2} \]

A reduction in density will occur as temperature increases (T \( \propto \) 1/\( \delta \))
\[ \frac{P_2}{P_1} = \frac{(T_1+273)}{(T_2+273)} \cdot \frac{P_1}{P_2} \]
\[ P_2 = \frac{(20+273)}{(100+273)} \cdot P_1 \]

(c) For a given flow rate the power is also proportional to density.
\[ \frac{P_{w2}}{P_{w1}} = \frac{T_1}{T_2} \]

\[ \frac{P_{w2}}{P_{w1}} = \frac{(T_1+273)}{(T_2+273)} \cdot \frac{P_{w1}}{P_{w2}} \]
\[ P_{w2} = \frac{(20+273)}{(100+273)} \cdot P_{w1} \]
\[ P_{w2} = (0.786) \cdot P_{w1} \]

Where;
\[ P_1 = \text{Pressure at Reference conditions, (mmWg).} \]
\[ P_2 = \text{Pressure at Required conditions, (mmWg).} \]
\[ P_{w1} = \text{Power at Reference conditions, (kW).} \]
\[ P_{w2} = \text{Power at Required conditions, (kW).} \]
\[ T_1 = \text{Temperature at Reference conditions, (°C).} \]
\[ T_2 = \text{Temperature at Required conditions, (°C).} \]
2.1.4 Fan selection.

To select a fan for an exhaust system it is necessary to establish two main factors, these being:
(i) The quantity of air required to achieve effective extraction.
(ii) The resistance or pressure loss in the system.

(a) Determination of required air flow.

The air flow required depends on the hood design used for the exhaust enclosure. The starting point for a hood design is determining the emission rate or velocity of the liberated dust. From this a capture velocity may be decided upon which will be influenced by the type of dust. Various formulae are available for calculating the required capture velocity but this is not relevant to this thesis. The face velocity of the air entering the hood must be greater than the required capture velocity for efficient dust extraction. The size of the hood depends on the area of the dust cloud that is to be extracted.

\[
\text{Air flow required} = \text{Hood area} \times \text{Hood face velocity}
\]
(b) Determination of resistance or pressure loss in system.

The main purpose of a fan is to achieve and maintain an air flow rate against the resistance of the system. The system has losses caused by the friction of the air against the surface of the ducting. The velocity of the air and the length of the duct being the main components. The smaller the diameter the higher the velocity then higher the friction losses. Dynamic or turbulence losses are caused whenever the duct varies in cross sectional area or it changes direction. Air flow through extraction systems can generally be regarded as turbulent so that each individual pressure loss is proportional to the square of the velocity. The air volume flow being a function of the velocity and cross sectional area.

\[ \text{Air volume flow} = \text{Duct velocity} \times \text{Duct area} \]

The sum of the system losses will therefore be proportional to the square or the air volume flow rate and can be shown as a parabola, (see fig.9).

**Analysis of fan performance**

The following procedure should be followed when assessing the suitability of a particular fan for a given application;

(i) Plot the specification point on fan performance curve (fig.9).

\[ \text{X-axis : Air volume flow required.} \]
\[ \text{Y-axis : Pressure required.} \]

(ii) Plot system resistance curve through the specification point and back to the origin of the graph such that the pressure varies as the square of the air volume.
(iii) Plot fan performance curve on same axis.

(iv) The point of intersection of the fan performance curve and the system resistance curve indicates the performance obtained from the fan when working on a system corresponding to the specified conditions.

(v) Comparison is then reported as a percentage increase or decrease on the required air volume flow rate required. On this basis the suitability of a fan for operation on a given system can be assessed.

Fig.9, Fan Performance & system curves
2.2 Noise Level Evaluation

2.2.1 Sound Pressure Levels

Noises or sounds are pressure waves or fluctuations. The units used for pressure are the \( \text{N/m}^2 \) or more commonly the Pascal, \( \text{Pa} \) (1 \( \text{N/m}^2 = 1 \text{ Pa} \)). Noise covers a very wide range of pressures, from the lowest audible pressure of \( 2 \times 10^{-5} \text{ Pa} \) to typically a Saturn rocket at 200,000 Pa.

![Diagram showing sound pressures](image)

**Fig.10, Examples of typical sound pressures**

Taking the lowest audible sound pressure as a reference, a ratio can be taken to any other sound pressure. This involves an enormous range of numbers so a logarithmic scale is used. Therefore a relationship for the sound pressure level (SPL) can be found.

\[
\text{Sound Pressure Level (SPL)} = 10 \log_{10} \left( \frac{\text{Sound Pressure}^2}{\text{Reference Pressure}^2} \right)
\]

Where, Reference Pressure = \( 2 \times 10^{-5} \text{ Pa} \)

These levels are measured in a unit called a decibel (dB).
2.2.2 The dB(A) Scale

The ear has an audible frequency range or spectrum from 31.5 Hz to 16 kHz. To allow for the differing sensitivity of the ear within this spectrum a weighted curve is used called 'A' weighting as shown below;

![Graph of A Weighting Scale]

**Fig.11, 'A' Weighting Scale**

This curve assigns a weighting to each frequency dependent on the ears sensitivity at this frequency. The lower frequencies where the ear is less sensitive are given a negative weighting but the mid to high frequencies where the ear is most sensitive are given a positive weighting. By taking a spectrum of noise and applying this weighting a dB(A) reading can be found for each frequency. An overall perceived noise level can also be found by combining these dB(A) readings logarithmically.

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>31.5</th>
<th>63</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1K</th>
<th>2K</th>
<th>4K</th>
<th>8K</th>
<th>16K</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Sound Pressure Level [dB]</strong></td>
<td>74</td>
<td>78</td>
<td>76</td>
<td>72</td>
<td>73</td>
<td>69</td>
<td>63</td>
<td>52</td>
<td>44</td>
<td>38</td>
</tr>
<tr>
<td><strong>A-Weighting</strong></td>
<td>-40</td>
<td>-27</td>
<td>-17</td>
<td>-9</td>
<td>-3</td>
<td>0</td>
<td>+1</td>
<td>+1</td>
<td>-1</td>
<td>-7</td>
</tr>
<tr>
<td><strong>Sound Pressure Level [dB(A)]</strong></td>
<td>34</td>
<td>51</td>
<td>59</td>
<td>63</td>
<td>70</td>
<td>69</td>
<td>64</td>
<td>53</td>
<td>43</td>
<td>31</td>
</tr>
</tbody>
</table>

**Table 2, Example of 'A' weighting (DU-K5-AD)**
2.2.3 NC and NR Curves

To help identify the frequencies that are causing the most annoyance from a particular noise source, two sets of curves are sometimes used, these being:

(a) Noise Criteria (NC) curves
(b) Noise Rating (NR) curves

By plotting the frequency spectrum of the noise source in dB on either of the charts it can be seen which frequencies are causing the most annoyance.

(a) Noise Criteria (NC) curves

![Noise Criteria (NC) curves](image)

*Fig.12, Noise Criteria (NC) curves*

These curves were originally developed in the United States to predict acceptable noise levels for offices, and are widely used in ventilating and air conditioning work.
(b) Noise Rating (NR) curves

These curves were developed in Europe and are intended for more general use than NC curves. They are often used for predicting community reaction to noise. They are currently being adopted by the International Standards Organisation (ISO), and tend to be widely used in Europe.
Example: DU-K5-AD

Fig. 14, Example of NR Curves

NR rating without acoustic diffuser = NR85
NR rating with acoustic box = NR70
2.2.4 Sound Power Levels (SWL)

Almost all noise calculations use the sound power level of the source. The reason that so much emphasis is placed on sound power level is that this is the only datum quantity of the source that does not change with environment.

If we were to measure the sound pressure level of a machine in one environment, and then moved it to a quite different one (say from a large open space to inside a small factory building) we would find that the sound pressure level at the same distance from the machine had changed. The sound power level of the machine on the other hand would not have changed. It is this fact which enables us to predict before the move takes place, what the new sound pressure level will be.

\[
\text{Sound Power Level (SWL)} = 10 \log_{10} \left( \frac{\text{Sound Power}}{\text{Reference Power}} \right)
\]

Where, Reference Power = \(10^{-12}\) watts

These levels are measured in decibels (dB).

\[
\text{Sound Pressure Level (SPL)} = 10 \log_{10} \left( \frac{(\text{Sound Pressure})^2}{(\text{Reference Pressure})^2} \right)
\]

Where, Reference Pressure = \(2 \times 10^{-5}\) N/m\(^2\)

These levels are measured in decibels (dB).
Spherical Radiation
Assuming that sound pressure measurements have been taken in free field conditions (i.e. in free space) and the source is radiating noise equally in all directions the relationship between SWL and SPL at a distance \( r \) from the noise source for spherical radiation is given by the following equation:

\[
SPL = SWL - 20 \log_{10} r - 11 \text{ dB}
\]

Hemispherical Radiation
While the model of a noise source located in free space and radiating in all directions is representative of some practical situations (overflying aircraft, noise from the top of a tall building or industrial stack), by far the most common arrangement is when the source is near the ground. If the area around the source is substantially free of reflecting surfaces or barriers, we have the important case of hemispherical propagation.

All the energy that would have been radiated downwards had the source been in free space, is now reflected by the ground and constrained to radiate into the hemispherical region above the ground. Since the area of a hemisphere is half the area of a sphere the local acoustic intensity at a distance \( r \) from the source is doubled, and the sound pressure level is therefore increased by 3 dB. The relationship between SPL and SWL at distance \( r \) from the noise source for hemispherical radiation is therefore given by the following equation:

\[
SPL = SWL - 20 \log_{10} r - 8 \text{ dB}
\]
2.2.5 Directivity Index

Not all sources radiate uniformly in all directions. If the source has a marked directional characteristic, some factor needs to be added to the equation to account for directivity. In any measurement of sound power the information necessary to evaluate directivity index in any angular direction, \( \phi \) [normally written \( DI(\phi) \)], will have been measured as a matter of course and is given by the following equation:

\[
DI(\phi) = SPL(\phi) - SPL(ave)
\]

Where:

- \( SPL(\phi) \) : Sound pressure level measured at a distance \( r \) from the source and angle \( \phi \) to the axis
- \( SPL(ave) \) : Average sound pressure level at a distance \( r \) from the source

Equations for spherical and hemispherical radiation can therefore be derived which account for enhanced radiation in a particular direction as follows:

**Spherical Radiation**

\[
SPL(ave) = SWL - 20 \log_{10} r - 11 \quad dB
\]

\[
DI(\phi) = SPL(\phi) - SPL(ave)
\]

Therefore:

\[
SPL(\phi) - DI(\phi) = SWL - 20 \log_{10} r - 11 \quad dB
\]

\[
SPL(\phi) = SWL - 20 \log_{10} r - 11 + DI(\phi) \quad dB
\]

**Hemispherical Radiation**

\[
SPL(ave) = SWL - 20 \log_{10} r - 8 \quad dB
\]

\[
DI(\phi) = SPL(\phi) - SPL(ave)
\]

Therefore:

\[
SPL(\phi) - DI(\phi) = SWL - 20 \log_{10} r - 8 \quad dB
\]

\[
SPL(\phi) = SWL - 20 \log_{10} r - 8 + DI(\phi) \quad dB
\]
2.3 Summary & Conclusions

2.3.1 Performance evaluation

For fan selection purposes the following parameters are usually plotted against air volume flow rate;

(a) Fan static pressure
(b) Motor shaft power

The above characteristic curves should be corrected to a standard air density of 1.2 kg/m³ at 20°C, however, there is generally no need to consider changes for normal ambient atmospheric conditions.

The suitability of a particular fan is determined by plotting a system resistance curve through the specification point. The point of intersection of the fan performance curve and the system resistance curve indicates the performance obtained from the fan when operating on a system corresponding to the specified conditions.

2.3.2 Noise level evaluation

Noises or sounds are pressure fluctuations which are measured as sound pressure levels (SPL) in decibels (dB). The human ear is sensitive to different frequency ranges and to give a perceived sound pressure level the 'A' weighted scale is used dB(A).

To identify frequencies that are causing the most annoyance from a particular noise source the following curves are often used;

(a) Noise criteria (NC) curves
(b) Noise rating (NR) curves

Sound pressure levels are often converted to sound power levels (SWL) for calculation purposes. Sound power levels are used because they do not change when the environment around the sound source changes, unlike sound pressure levels which do.

Not all noise sources radiate sound uniformly in all directions. The directivity index (DI) is used to account for this directional characteristic.
3.0 FAN TYPES

A fan is a rotary bladed machine which continuously supplies energy to the air or gas passing through it. There are three main components in a fan; the impeller (sometimes referred to as the wheel or rotor), the means of driving it, and the casing.

In order to cover a very wide range of applications fans are manufactured in a variety of types. These can be broken down into five main categories as follows:

(1) Axial
(2) Centrifugal
(3) Cross flow
(4) Mixed flow
(5) Propeller

3.1 Axial fans

In this type air enters and leaves the fan axially giving a straight through configuration. The tips of the impeller blades, which are commonly of aerofoil section, run with as fine a clearance as is practicable (consistent with cost of manufacture) in a cylindrical casing. Duties are usually high to medium volume flow at medium to low pressures. In its simplest form there is an impeller only with its driving motor mounted within a cylindrical casing and the discharge flow usually contains a fairly pronounced element of rotation or swirl, due to the work done by the impeller torque.
Thus the absolute velocity of the air leaving is higher than the axial velocity with the result that some of the total pressure developed by the impeller does not appear as useful fan total pressure. More sophisticated designs have guide vanes downstream of the impeller which remove the rotational component, thus slowing down the air and converting some of the excess velocity pressure to more useful static pressure. Another way to achieve the maximum amount of useful pressure is to have pre-rotational vanes upstream of the impeller. These rotate the air in a direction opposite to that of the impeller rotation and with careful design the air will leave the fan in an axial direction. A third way to achieve the maximum useful pressure is to dispense with guide vanes and to have two impellers driven in opposite directions within the same fan casing. The second impeller then acts in a manner similar to that of an upstream guide vane unit, taking its pre-rotated air from the discharge of the first impeller. Such an assembly is known as a contra-rotating fan. True axial discharge from any of these units will be possible only for a single operating condition.
Although not developing as much pressure as centrifugal fans of the same impeller diameter and speed, axial flow fans may be used for applications involving the movement of air uncontaminated with solids and at moderate temperatures, depending on the maximum permitted ambient temperature of the electric motor drive where this is in the air stream. All these types have non-overloading power characteristics with performance variations as illustrated below.

![Typical curves for an axial fan](image_url)

**Fig.16,** Typical curves for an axial fan
3.2 Centrifugal fans

Air enters the impeller axially and is typically discharged radially into a volute casing, the impeller rotation being towards the casing outlet.

![Diagram of centrifugal fan](image)

**Fig. 17, Centrifugal fan**

Air duties covered are generally medium to low volume flow rates (typically 750m$^3$/hr to 30,000m$^3$/hr) at medium to high pressures (typically 100mmwg to 500mmwg). The amount of work done on the air, evident in the pressure development of the fan, depends primarily on the size of the impeller and the angle of the fan blades. The main varieties are characterised by the type and angle of the impeller blades as follows.

![Diagram of blade forms](image)

**Fig. 18, Typical blade forms**
3.2.1 Backward Bladed \( (\beta_2 < 90^\circ) \)

In this type the blades slope or curve backwards from the heel (inner edge) to the tip (outer edge) relative to the direction of rotation and may be of plate or aerofoil form.

![Fig. 19, Backward Bladed Centrifugal fan](image)

This type combines a fairly steep pressure characteristic over the normal working range with good efficiency. The maximum power required is generally little in excess of the power absorbed by the fan when working at maximum efficiency. Consequently, a motor able to cope with all operating conditions may economically be selected. This type of power characteristic is often referred to as 'non-overloading'.

![Fig. 20, Typical curves for a backward bladed centrifugal fan](image)
3.2.2 Radial Bladed \( (\beta_2 = 90^\circ) \)

Radial bladed fans have impellers whose blade tips or even the whole blades are radial.

**Radial Tipped**

Radial tipped impellers have blades with radial tips.

![Fig. 21, Radial Tipped](image)

In this type the power characteristic is almost a straight line rising from a minimum at zero flow to a maximum, at maximum flow but not so steeply as in the case of the forward curved type. Peak efficiencies are usually intermediate between those of backward bladed and forward curved fans.

![Fig. 22, Typical curves for a radial tipped centrifugal fan](image)
Radial Blades

In this type the blade has no curved heel to give good air entry conditions as in the case of the radial tipped. The impeller may have a conventional backplate and shroud type or may be of the paddle bladed type with flat blades attached directly to a hub.

These fans and particularly the latter variation are capable of handling fairly high concentrations and quite large pieces of gas borne solids. Such fans are used mostly for moving air containing solids since they remain reasonably free from blockage and can withstand considerable wear before failure. Performance characteristics are similar to those for radial tipped fans but efficiencies are lower.
3.2.3 Forward Curved (β2 > 90°)

Characterised by a relatively large number of shallow blades curved forward in the direction of rotation. As a result of the shallower blades, the number of them has to be increased in order to have the necessary influence on the air during its passage through the impeller.

As the power characteristic curves steeply upwards from zero flow to maximum flow at zero static pressure this fan is liable to overload its driving unit if operated significantly above its rated air volume duty. It can be used to handle relatively large air volumes at low running speeds and therefore tends to provide a compact installation. The peak efficiency tends to be somewhat lower than that of backward bladed fans.
3.3 Cross Flow fans

The cross flow, or tangential fan as it is sometimes called, has an impeller with blades shaped somewhat like those of a forward curved centrifugal fan impeller. However, both ends of the impeller are sealed and it is fitted into a casing in which air enters at the periphery on one side, passes through the impeller, and leaves from the periphery at the other side.

![Cross Flow fan](image)

Fig. 26, Cross Flow fan

Little development of large size units appears to have taken place, probably due to the fact that most of the pressure development is in the form of velocity pressure and also the fan efficiency is low. However, volume flow is almost unlimited since the impeller may be made of any practicable length as throttling at the inlet is not a problem. This type of fan is commonly used in small domestic heaters.

![Typical curves for a cross flow fan](image)

Fig. 27, Typical curves for a cross flow fan
3.4 Mixed Flow fans

A fan in which the air path through the impeller is intermediate between the axial and centrifugal types, see below;

Fig. 28, Mixed flow fan

This type gives the benefit of increased pressure when compared with axial fans, but is capable of being constructed to provide either axial or radial discharge.

Fig. 29, Typical curves for a mixed flow fan
3.5 Propeller fans

Probably the simplest form of fan, this type comprises a motor driving directly an impeller, which runs with quite a large clearance in an orifice. Sometimes it may run open as a desk or ceiling fan merely to circulate air.

![Propeller fan diagram](image)

Fig. 30, Propeller fan

Duties covered are high volume and low pressure. Mounting of the fan can effect the performance substantially and power can increase markedly towards closed discharge. Applications usually involve moving air through a partition from one open space to another.

![Typical curves for a propeller fan](image)

Fig. 31, Typical curves for a propeller fan
3.6 Fan Selection

3.6.1 Fan Laws [36]

The performance of a fan in terms of pressure, volume flow and power absorbed depends on a number of factors, the most important being:

(a) The design and type of fan.
(b) The point of operation on the volume flow / pressure characteristic.
(c) The size of fan.
(d) The speed of rotation of the impeller.
(e) The condition of the air or gas passing through the fan.

It is customary for a manufacturer to make a range of fans of varying sizes to a single design, thus producing a series of geometrically similar fans (homologous series). It is convenient to be able to compute the performance of each fan from the minimum test data. The pressure/volume flow relationship is not generally capable of being expressed as a simple mathematical function. However, by considering any single point of operation on the characteristic curve it is possible to derive some simple relationships known as the fan laws shown below:

\[ Q = k_q d^a u^b \delta^c \mu^d \]  \hspace{1cm} (1)

\[ P = k_p d^e u^f \delta^g \mu^h \]  \hspace{1cm} (2)

Where:

- \(d\) = Impeller diameter
- \(u\) = Impeller peripheral speed
- \(\delta\) = Air density
- \(\mu\) = Coefficient of viscosity
- \(k_q\) = Flow coefficient
- \(k_p\) = Pressure coefficient
Equating the dimensions of equation (1):

\[ l^b t^c = (l^b t^c)^d (m l^3 t^{-1})^d \]

\[ = l^{b+c-3} t^{d-1} m^{c+d} \]

Equating indices:

\[ \begin{align*}
  m & : 0 = c + d \\
  t & : -1 = -b - d \\
  l & : 3 = a + b - 3c - d
\end{align*} \]

Therefore:

\[ \begin{align*}
  a &= 2 - d \\
  b &= 1 - d \\
  c &= -d
\end{align*} \]

Equation (1) becomes:

\[ Q = k_q d^2 u^d \delta^d \mu^d \]

Equating the dimensions of equation (2):

\[ m l^2 t^2 = (l^e t^f)^g (m l^3 t^{-1})^h \]

\[ = l^{e+f-3} t^{g-h} m^{e+h} \]

Equating indices:

\[ \begin{align*}
  m & : 1 = g + h \\
  t & : -2 = -f - h \\
  l & : -1 = e + f - 3g - h
\end{align*} \]

Therefore:

\[ \begin{align*}
  e &= -h \\
  f &= 2 - h \\
  g &= 1 - h
\end{align*} \]

Equation (2) becomes:

\[ P = k_p d^h u^h \delta^h \mu^h \]

\[ P = k_p u^2 \delta^2 (du\delta/\mu)^h \] 

.............. (4)
In equations (3) & (4), the term \( \frac{du}{d\mu} \) is seen to be of the same form as Reynolds Number. It may be taken as the Reynolds number of the fan \((Re)\) based on the impeller diameter and peripheral velocity and equations (3) & (4) become:

\[
\begin{align*}
Q &= k_q d^2 u f_1(Re) \\
P &= k_p u^2 \delta f_2(Re)
\end{align*}
\]

Where \( f_1(Re) \) and \( f_2(Re) \) are variable factors based on Reynolds number.

If the speed of rotation of the impeller is \( n \), then \( u = \pi d n \) and equations (5) & (6) can be written as:

\[
\begin{align*}
Q &= k_q d^3 n f_1(Re) \\
P &= k_p d^2 n^2 \delta f_2(Re)
\end{align*}
\]

In practice it is found that Reynolds number has little effect over quite wide ranges of impeller diameters, speed and air density. The above equations are therefore normally simplified as follows:

\[
\begin{align*}
Q &= k_q d^3 n \\
P &= k_p d^2 n^2 \delta
\end{align*}
\]

The fan power can also be derived:

\[
\begin{align*}
P_w &= P Q \\
P_w &= k_{pw} d^5 n^3 \delta
\end{align*}
\]

Where \( k_{pw} \) is the Power coefficient.

The coefficients \( k_q, k_p, \) and \( k_{pw} \), will be constant for a range of geometrically similar fans, and for a particular point of operation on the pressure/volume characteristic.
3.6.2 Performance coefficients [36]

Since the fan laws are valid for any particular point on the fan pressure/volume characteristic, similar laws will be valid for every other point of operation, the only difference being the numerical values of the coefficients. Thus a plot of $k_p$ against $k_q$ will have exactly the same form as the pressure/volume characteristic of each fan in the homologous series. It may be used, therefore, to represent the performance of the series design with that of another series design. A system of performance coefficients based on this principle is often used which are detailed as follows;

Volume coefficient, $K_Q = \frac{Q}{d^3 (n/1000)}$

Where:

$Q = \text{Volume flow rate} \ (\text{ft}^3/\text{min})$  
$n = \text{Speed of rotation} \ (\text{rev/min})$  
$d = \text{Impeller diameter} \ (\text{feet})$

Pressure coefficient, $K_T = \frac{P_{TF}}{d^2 (n/1000)^2}$

Where:

$P_{TF} = \text{Fan total pressure} \ (\text{inch Wg})$  
$n = \text{Speed of rotation} \ (\text{rev/min})$  
$d = \text{Impeller diameter} \ (\text{feet})$

The coefficient has the subscript T when referring to fan total pressure and subscript S when referring to fan static pressure.

Fan power coefficient, $K_{FW} = \frac{P_{WA}}{d^5 (n/1000)^3}$

Where:

$P_{WA} = \text{Fan absorbed power} \ (\text{h.p.})$  
$n = \text{Speed of rotation} \ (\text{rev/min})$  
$d = \text{Impeller diameter} \ (\text{feet})$

The coefficients are not strictly non-dimensional in these expressions.

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Commonly used non-dimensional coefficients based on equations (3) and (4) are as follows:

Volume coefficient, $\phi = \frac{Q}{(\pi d^2/4)u}$

Total pressure coefficient, $\psi = \frac{P_{TF}}{(1/2 \delta u^2)}$

Static pressure coefficient, $\psi_s = \frac{P_{SF}}{(1/2 \delta u^2)}$

Power coefficient, $\lambda = \frac{\phi \cdot \psi}{E_{fr}}$

Where:
- $Q =$ Volume flow (ft$^3$/min) or (m$^3$/s)
- $P_{TF} =$ Fan total pressure (inch Wg) or (Pa)
- $P_{SF} =$ Fan static pressure (inch Wg) or (Pa)
- $d =$ Impeller diameter (feet) or (metres)
- $u =$ Impeller peripheral velocity (ft/min) or (m/s)
- $\delta =$ Air density (lb/ft$^3$) or kg/m$^3$
- $E_{fr} =$ Efficiency

3.6.3 Specific speed [36]

In practice, both the pressure coefficient ($\psi$) and volume coefficient ($\phi$) have been found inadequate for numerical evaluation of the important characteristics of fans. A given volume and pressure can be produced by various fans which are widely different in their dimensions. Often a fan is required with a specified speed so coefficients relating fan dimensions and speed are required.
Specific speed is the speed at which a fan would run to give unit volume flow and unit pressure. It applies to all fans in a homologous series, and is derived from the fan laws by eliminating the diameter term as follows:

Using equation (10)

\[ P = k_v d^2 n^2 \delta \]
\[ P \propto d^2 n^2 \delta \]
\[ d \propto P^{0.5}/(n^{0.5}) \]

Using equation (9)

\[ Q = k_v d^3 n \]
\[ Q \propto d^3 n \]
\[ Q \propto d^3 n \]

Therefore

\[ Q \propto (P^{1.5} n)/(n^3 \delta^{3.5}) \]
\[ Q \propto (P^{1.5})/(n^2 \delta^{1.5}) \]
\[ n \propto (P^{0.75})/(Q^{0.5} \delta^{0.75}) \]

Substituting \( n_s \), when \( Q=1 \) and \( P=1 \)

\[ n_s \propto 1/\delta^{0.75} \]
\[ n_s/n = 1/\delta^{0.75} (Q^{0.5} \delta^{0.75})/(P^{0.75}) \]
\[ n_s/n = (Q^{0.5})/(P^{0.75}) \]

Specific speed, \( n_s \) = \( (n Q^{0.5})/(P^{0.75}) \)

Where:

\[ n_s \quad \text{Specific speed (rev/min)} \]
\[ n \quad \text{Impeller speed (rev/min)} \]
\[ Q \quad \text{Volume flow rate (cfm)} \]
\[ P \quad \text{Static pressure (inch wg)} \]
The specific speed is almost always based on fan performance under standard air conditions. However, when changes in air density are significant variations in air density need to be allowed for by using the following equations [45]:

\[
\text{Specific speed, } n_s = \left(\frac{n \cdot Q^{0.8}}{(P \cdot \delta_o/\delta)^{0.75}}\right)
\]

Where:

- \(\delta_o\) = Standard air density \((\text{lb/cu ft})\)
- \(\delta\) = Local air density \((\text{lb/cu ft})\)

The efficiency of a particular type of fan package depends on the specific speed, as shown in the fig below;

![Fan efficiency vs Specific speed for various fan types](image)

**Fig. 32, Fan efficiency vs Specific speed for various fan types**

It can be seen that the centrifugal fans cover the lower specific speeds and axial fans cover the higher specific speeds.
3.6.4 Dimensionless coefficients [24]

The value of the specific speed is dependant on the system of units, however, there are a series of dimensionless coefficients which are not dependant on the units used.

**Speed coefficient ($\sigma$)**

The speed coefficient ($\sigma$) is a non-dimensional specific speed which is defined as;

$$\sigma = \frac{\phi^{0.5}}{\psi^{0.75}}$$

**Diameter coefficient ($\delta$)**

The diameter coefficient ($\delta$) is defined as;

$$\delta = \frac{\psi^{0.25}}{\phi^{0.5}}$$

The coefficients $\sigma$ and $\delta$ can be determined by using the following graph with the aid of the values of $\phi$ and $\psi$;

Fig. 33, Evaluation of the dimensionless coefficients
These coefficients can then be used to determine the dimensions and rotating speed of an impeller using the following graph;

![Graph showing determination of dimensions & rotating speed](image)

Fig. 34, Determination of dimensions & rotating speed
The speed coefficient ($\sigma$) and diameter coefficient ($\delta$) can be used to determine the most suitable impeller for a given application. In the following graph, impellers are represented in terms of $\sigma$ and $\delta$. The best impeller of each type is inserted in the appropriate position on the $\delta - \sigma$ graph so that all impellers are confined within a narrow band.

![Graph showing $\sigma - \delta$ diagram with maximum efficiencies and peripheral velocities.](image)

**Fig. 35, $\sigma - \delta$ diagram with maximum efficiencies and peripheral velocities.**

To determine the most suitable impeller for a definite value of $\sigma$ it is only required to read the corresponding value of $\delta$ plotted on the $\sigma - \delta$ curve. If the design indicated at the point of intersection on the curve is selected an optimum efficiency will be obtained providing the construction of the blades is also satisfactory. However, if the design is varied significantly from the design indicated on the curve, then a relatively high efficiency is impossible, irrespective of how well the overall impeller has been constructed.
Throttling coefficient (τ)
There is an interaction between a fan and a duct to which it may be connected because of the retarding or throttling action of the resistance set up by the duct upon the air handled by the fan. In general, the resistance is proportional to the pressure developed divided by the square of the volume.

\[ \text{Throttling coefficient, } \tau = \frac{\psi}{\phi^2} \]

Coefficient of performance (λ)
For the driving power, another coefficient can be defined as follows;

\[ \text{Coefficient of performance, } = \frac{\phi \cdot \psi}{E_{\text{\textit{\Pi}}}} \]

\[ E_{\text{\textit{\Pi}}}, \text{ Static efficiency } \% \]

(Approx. value from fig. 32)

3.6.5 Calculating fan assembly dimensions [45]

Major fan dimension
The major dimension of the fan case depends on the pressure coefficient, which is the ratio of the static pressure to the kinetic energy of the air (i.e. pressure equivalent of blade tip velocity). The relationship between impeller and housing dimensions can be factored into the defining equation to yield "apparent pressure coefficient, \( \psi_a \). Like efficiency, the apparent pressure coefficient is a function of specific speed and can be defined as (see fig.36 & fig.37);

\[ \text{Apparent pressure coefficient, } \psi_a = \frac{2.35 \times 10^6 \cdot P_s \cdot \delta_s}{N^2 \cdot A^2} \]

Where;

- \( P_s \) = Static pressure \ (\text{inch wg})
- \( N \) = Impeller speed \ (\text{rev/min})
- \( A \) = Major fan dimension \ (\text{inch})
- \( \delta_s \) = Standard air density \ (\text{lb/cu ft})
- \( \delta \) = Local air density \ (\text{lb/cu ft})
Apparent pressure coefficients for centrifugal fans range from 0.1 to 0.7. Apparent pressure coefficients for axial fans range from 0.05 to 0.5.

Thus the fan major fan case dimension can be calculated as follows:

\[
\text{Major dimension} = \sqrt{\frac{2.35 \times 10^5 \cdot P_s \cdot \delta}{N^2 \cdot \Psi_a \cdot \delta}}
\]

Fig. 36, Apparent pressure coefficient for Centrifugal fans
Housing width

For axial fans the housing width is equal to the major dimension. For centrifugal fans the process is very similar to that for finding the major dimension, but it makes use of the apparent flow coefficient $\phi_a$ rather than the apparent pressure coefficient, see below;

Fig. 37, Apparent pressure coefficient for axial fans

Fig. 38, Apparent flow coefficients
Thus the housing width can be calculated as follows:

\[
\text{Housing width} = \frac{175 \cdot Q}{N \cdot A^2 \cdot \phi_a}
\]

Where:
- \( Q \) = Volume flow rate (cfm)
- \( N \) = Impeller speed (rev/min)
- \( A \) = Major fan dimension (inch)
- \( \phi_a \) = Apparent flow coefficient density

To find the total fan assembly width for centrifugal fans add the motor length.

**Fan Power**

The fan power depends on the static efficiency of the fan \((E_{\text{eff}})\) at the chosen specific speed, see Fig 32. The mechanical power that must be delivered to the fan is the absorbed power \((P_{WA})\) which can be expressed as:

\[
P_{WA} = \frac{Q \cdot P_s}{8.52 \cdot E_{\text{eff}}}
\]

Where:
- \( Q \) = Volume flow (cfm)
- \( P_s \) = Static pressure (inch wg)
- \( E_{\text{eff}} \) = Static efficiency (%)

The shaft power \((P_{Ws})\) can be found by dividing the absorbed power by the motor efficiency:

\[
P_{Ws} = \frac{P_{WA}}{Eff_m}
\]

Where:
- \( P_{WA} \) = Absorbed power (watts)
- \( Eff_m \) = Motor efficiency (%)
3.6.6 Fan Design Selection

When selecting the most suitable fan design for a given application it is inevitable that it will sometimes be necessary to sacrifice efficiency to meet other requirements. The following list of desirable features will help to quantify the main fan parameters:

(a) **Maximum efficiency** - This requirement is satisfied by a design with a suitable value of the speed coefficient (\(\sigma\)) or alternatively specific speed (\(n_s\)).

(b) **Minimal noise generated** - Maximum value of the pressure coefficient (\(\psi\)) and low blade tip speed.

(c) **Flow path characteristics** - Intake and discharge paths that are in line with each other indicate that an axial fan will be most suitable, whereas flow paths angled at 90° to each other suggest a centrifugal fan.

(d) **Minimal wear in operation with dust laden gases** - Maximum value of the pressure coefficient (\(\psi\)).

(e) **Large capacity** - Maximum value of volume coefficient (\(\phi\)).

(f) **Maximum capacity with minimum size (normally cheapest design)** - The product of the pressure coefficient (\(\psi\)) and volume coefficient (\(\phi\)) must be as large as possible.

(g) **Pressure characteristic requirements** - Type of fan curve required, ie. steep, flat, or peakless.

(h) **Power characteristic requirements** - Type of power curve required, ie. non-overloading and maximum power limit.

(i) **Regulation of capacity requirements** - eg. regulation at constant pressure or a definite change of pressure with the volume.

(j) **Low weight**.

(k) **Definite installation requirements**.

(l) **Requirements of the inlet and outlet connections** - a circular cross section is a better match for an axial fan, whilst a rectangular cross section better suites a centrifugal fan.

(m) **Impeller with the least moment of inertia**.

(n) **Impeller with maximum strength**.
The table below shows the main fan types and approximate values for the various fan coefficients which can be used to make an assessment of the most suitable fan for a particular application.

<table>
<thead>
<tr>
<th>Type</th>
<th>$\varphi$</th>
<th>$\psi$</th>
<th>$\psi-\eta$</th>
<th>$c$</th>
<th>$\delta$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.0</td>
<td>2.4</td>
<td>2.4</td>
<td>0.35-0.6</td>
<td>1.14-1.19</td>
</tr>
<tr>
<td>2</td>
<td>1.0</td>
<td>2.3</td>
<td>2.3</td>
<td>0.438-0.592</td>
<td>1.19-1.22</td>
</tr>
<tr>
<td>3</td>
<td>0.3</td>
<td>0.75</td>
<td>0.225</td>
<td>0.68</td>
<td>1.7</td>
</tr>
<tr>
<td>4</td>
<td>0.2</td>
<td>0.6</td>
<td>0.12</td>
<td>0.657</td>
<td>1.965</td>
</tr>
<tr>
<td>5</td>
<td>0.13</td>
<td>1.0</td>
<td>0.13</td>
<td>0.381</td>
<td>2.72</td>
</tr>
<tr>
<td>6</td>
<td>0.03</td>
<td>1.1</td>
<td>0.033</td>
<td>0.182</td>
<td>1.62</td>
</tr>
<tr>
<td>7</td>
<td>0.00185</td>
<td>1.1</td>
<td>0.00233</td>
<td>0.04</td>
<td>24.4</td>
</tr>
<tr>
<td>8</td>
<td>0.2-0.2</td>
<td>0.05-0.01</td>
<td>0.005-0.02</td>
<td>1.6-0.8</td>
<td>1.0-1.78</td>
</tr>
<tr>
<td>9</td>
<td>0.3</td>
<td>0.5</td>
<td>0.15</td>
<td>0.824</td>
<td>1.535</td>
</tr>
<tr>
<td>10</td>
<td>0.3</td>
<td>0.7</td>
<td>0.21</td>
<td>0.715</td>
<td>1.62</td>
</tr>
</tbody>
</table>

Table 3, Tabulation of coefficients for different fan types
3.6.7 Worked Example - K21 fan

The following duty is required with a fixed speed direct drive fan:

Volume Flow, $Q = 11000$ m$^3$/hr = 6475 CFM = 3.06 m$^3$/s
Impeller speed, $n$ = 3000 rpm
Fan static pressure, $P$ = 300 mmWg = 11.8 " wg

Using the specific speed to determine most efficient fan type

Specific speed, $n_s = (n.Q^{0.5})/(P^{0.75})$

$$n_s = \frac{3000 \times (6475)^{0.5}}{(11.8)^{0.75}}$$

$$= \frac{3000 \times 80.467}{6.367}$$

$$= 37916$$

From the graph it is apparent that a backward curved centrifugal impeller with a high width to diameter ratio will have the highest efficiency.
Using table of coefficients for different fan types (table 3).
The values of $\phi$ and $\Psi$ for impellers with a high width to diameter ratio are as follows:

\[
\begin{align*}
\phi &= 0.13 \text{ to } 0.2 = \text{ approx } 0.165 \\
\Psi &= 0.6 \text{ to } 1.0 = 0.8
\end{align*}
\]

Using fig 40 the required impeller diameter and speed can be determined to give the target volume flow of 3.06 m$^3$/s and pressure of 300 mmWG.

therefore;

\[
\begin{align*}
\text{Diameter, } D_2 &= 0.57 \text{m} = 570 \text{mm} \\
\text{Impeller speed, } n &= 2600 \text{ rpm}
\end{align*}
\]

(see fig.40)

The previous values are ideal values, however, if the fan is direct drive the motor speed will be fixed. DCE use 2 pole motors for all its fan which are direct drive so the impeller speed must be 3000 rpm.

To give the required volume flow and pressure one or more of the coefficients must be adjusted to give a new impeller diameter.

If the volume coefficient ($\phi$) remains at the ideal value of 0.165 and the pressure coefficient ($\Psi$) is changed from the ideal value of 0.8 to 0.7, the required impeller diameter is found to be 510mm.
If the pressure coefficient ($\Psi$) remains at the ideal value 0.8, and the volume coefficient ($\phi$) is changed from the ideal value of 0.165 to 0.25 the required impeller diameter is found to be 450mm.

Fig. 40, using the graph in fig.34 to determine impeller diameter and speed
Using fig. 33 to evaluate the dimensionless coefficients $\delta$ and $\sigma$.

Fig. 41. Using fig. 33 to determine coefficients $\delta$ and $\sigma$

Using the ideal coefficients for a 470mm diameter at 2600 rpm;

ie. $\phi = 0.165$ and $\psi = 0.8$

then $\delta = 2.3$ and $\sigma = 0.48$

Using the coefficients for a 510mm diameter impeller at 3000 rpm;

ie. $\phi = 0.165$ and $\psi = 0.7$

then $\delta = 2.3$ and $\sigma = 0.53$

Using the coefficients for a 450mm diameter impeller at 3000 rpm;

ie. $\phi = 0.25$ and $\psi = 0.8$

then $\delta = 1.95$ and $\sigma = 0.58$
Using fig.35 to ensure that impeller is an efficient design

Fig. 42, Using fig.35 to ensure impeller design is efficient

It can be seen that all the above impellers are very near to the maximum efficiency curve and are therefore of the optimum design.
Overall fan dimensions

The overall fan dimensions can be calculated using the apparent flow coefficient and the apparent pressure coefficient.

Using fig.36 to determine apparent pressure coefficient, $\Psi_a$.

![Graph showing apparent pressure coefficient vs specific speed](image)

**Fig. 43, using fig.36 to determine $\Psi_a$.**

When $n_s = 37916$ then $\Psi_a = 0.14$

Major dimension, $A$ = $\sqrt{\frac{2.35 \times 10^6 \cdot P_s \cdot \delta_a}{N^2 \cdot \delta}}$

= $\sqrt{\frac{(2.35 \times 10^6) \cdot (11.8)}{(3000^2) \cdot (0.14)}}$

= $\sqrt{2201}$

= 47 inch = 1192 mm

82
Using fig. 38 to determine apparent flow coefficient, $\phi_a$

![Diagram showing apparent flow coefficient versus specific speed]

Fig. 44, Using fig.38 to determine $\phi_a$

When $n_s = 37916$ then $\phi_a = 0.021$

Housing width, $W = \frac{175 \cdot Q}{N \cdot A^2 \cdot \phi_a}$

\[= \frac{(175)(6475)}{(3000)(47^2)(0.021)}\]

\[= 8.14 \text{ inch} = 207 \text{ mm}\]
Calculation of Shaft Power

Using fig.39 to determine static efficiency, $E_{fr}$

When $n_s = 37916$ then $E_{fr} = 66\%$

$$P_{wA} = \frac{Q \cdot P_s}{8.52 \cdot E_{fr}}$$

$$P_{wA} = \frac{(6475) \cdot (11.8)}{(8.52) \cdot (0.66)} = 13587 \text{ watts}$$

The shaft power ($P_{ws}$) can be found by dividing the absorbed power by the motor efficiency:

$$P_{ws} = \frac{P_{wA}}{\text{Motor } E_{fr}}$$

$$= \frac{13587}{0.90} = 15097 \text{ watts}$$

$$= 15.1 \text{ kW}$$

If factors such as space are critical the coefficients can be modified to reduce fan dimensions but this will change the specific speed and hence the fan efficiency and power requirements will be effected.
3.7 Summary & Conclusions

The following five main categories of fan are manufactured for a variety of application:

(1) Axial  
(2) Centrifugal  
(3) Cross flow  
(4) Mixed flow  
(5) Propeller

The specific speed and other dimensionless coefficients can be used to determine the most efficient fan type and to determine the key design parameters.

By calculating the specific speeds the most efficient fan type can be selected to meet DCE's requirements.

Smallest DCE fan is F1/G1 with following duty requirement:

Volume Flow, $Q = 1000 \text{ m}^3/\text{hr} = 589 \text{ CFM}$

Impeller speed, $n = 3000 \text{ rpm}$

Fan static pressure, $P = 150 \text{ mmWg} = 5.91 " \text{ wg}$

Specific speed, $n_s = (n \cdot Q^{0.5})/(P^{0.75})$

\[
ns = \frac{3000 \times (589)^{0.5}}{(5.91)^{0.75}}
\]

\[
ns = \frac{3000 \times 24.27}{3.79}
\]

\[
ns = 19219
\]
Largest DCE fan is K21 with following duty requirement

Volume Flow, $Q = 11000 \text{ m}^3/\text{hr} = 6475 \text{ CFM}$
Impeller speed, $n = 3000 \text{ rpm}$
Fan static pressure, $P = 300 \text{ mmWg} = 11.8" \text{ wg}$

Specific speed, $n_s = (n \cdot Q^{0.5})/(P^{0.75})$

$$n_s = \frac{3000 \times (6475)^{0.5}}{(11.8)^{0.75}} = \frac{3000 \times 80.467}{6.367} = 37916$$

![Graph showing specific speed for DCE fans](image)

Fig. 45, Specific speed for DCE fans

From the above curves it is apparent that backward curved centrifugal fans with high width to diameter ratios have the highest efficiency for DCE's requirements. This thesis will therefore focus on backward bladed centrifugal fan design.
4.0 MARKET SURVEY

DCE originally intended to purchase a new range of fans from an outside supplier. A number of companies were contacted in order to satisfy their requirements for lower noise levels. The findings of this market survey are summarised in this chapter.

The criteria that had to be considered were as follows (see 5.0 Define problem for more detail);

(a) Fan assembly dimensions.
(b) Noise levels.
(c) Fan performance.
(d) Cost.
(e) Power consumption.

This market survey compared bought out fans with the F-Range of fans which are primarily used for the Dalamatic range of dust collectors. The F-range of fans consists of seven fans from the smallest F1 (0.75 kW motor) to the largest F12 (11 kW motor).

The tables on the following pages summarize the quotations received from the six most favourable fan manufacturers.
### 4.1 Fan assembly dimensions

<table>
<thead>
<tr>
<th></th>
<th>DCE</th>
<th>ENGART</th>
<th>FAN SYSTEMS</th>
<th>HAMPSON ALAMEIN</th>
<th>PUNKER</th>
<th>STANDARD &amp; POCHIN</th>
<th>UTILE</th>
</tr>
</thead>
<tbody>
<tr>
<td>F1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>H</td>
<td>391</td>
<td>581</td>
<td>636</td>
<td>515</td>
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<tr>
<td>L</td>
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<td>658</td>
<td>572</td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
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<td></td>
<td></td>
</tr>
<tr>
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<td></td>
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<td>930</td>
<td>907</td>
<td>724</td>
<td>826</td>
<td></td>
</tr>
</tbody>
</table>

Table 4, Fan case dimensional comparison, (mm)

88
It can be seen from the table that all the fans are larger than the present range of in-house manufactured fans, with the exception of Utile who proposed to use DCE fan cases. These large cases are unacceptably large as this would involve significant modifications to the existing range of DCE dust collectors to allow fitment of these larger fans.
4.2 Noise levels

From the above table it would appear that the bought out fans are considerably quieter than the in-house manufactured DCE fans. However when the F6 Punker fan was tested at DCE a sound pressure level of 101 dB(A) was measured rather than the 82-87 dB(A) quoted by Punker. The large difference between the DCE fan measurements and the values quoted by the fan suppliers is due to the measuring method. DCE measure the noise level of the fan at a distance of 1 metre from the fan inlet. The fan manufacturers measure at the same distance from the fan, however, they duct the fan inlet and outlet so that only a measure of the fan break out is recorded and not the total fan noise level.

Table 5, Sound pressure levels, dB(A)
4.3 Performance

<table>
<thead>
<tr>
<th></th>
<th>DCE</th>
<th>ENGART</th>
<th>FAN SYSTEMS</th>
<th>HAMPSON ALAMEIN</th>
<th>PUNKER</th>
<th>STANDARD &amp; POCHIN</th>
<th>UTILE</th>
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<tr>
<td>F1</td>
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<td>500</td>
<td>500</td>
<td>510</td>
<td>430</td>
<td>450</td>
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<td>930</td>
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<td>930</td>
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<td>940</td>
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<td>3050</td>
<td>2680</td>
<td>2600</td>
<td>2740</td>
<td>3025</td>
</tr>
</tbody>
</table>

Table 6, Performance at typical operating point, (m³/hr)

The fan performance suitability for the various companies based on the above table can be summarized as follows;

Engart - Adequate match
Fan Systems - Good match
Hampson Alamein - Too flat
Punker - Too flat
Standard & Pochin - Too flat
Utile - Good match
### 4.4 Cost

<table>
<thead>
<tr>
<th></th>
<th>DCE</th>
<th>ENGART</th>
<th><strong>FAN SYSTEMS</strong></th>
<th>HAMPSON ALAMEIN</th>
<th>PUNKER</th>
<th>STANDARD &amp; POCHIN</th>
<th>UTILE</th>
</tr>
</thead>
<tbody>
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<td>538</td>
<td>would not give until after testing</td>
<td>430</td>
<td>143</td>
<td>281</td>
<td>73</td>
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<tr>
<td><strong>F3</strong></td>
<td>77</td>
<td>571</td>
<td>-</td>
<td>465</td>
<td>374</td>
<td>289</td>
<td>134</td>
</tr>
<tr>
<td><strong>F5</strong></td>
<td>91</td>
<td>614</td>
<td>-</td>
<td>500</td>
<td>374</td>
<td>308</td>
<td>128</td>
</tr>
<tr>
<td><strong>F6</strong></td>
<td>108</td>
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<td>-</td>
<td>500</td>
<td>407</td>
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<td>-</td>
<td>510</td>
<td>432</td>
<td>338</td>
<td>157</td>
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<td>-</td>
<td>575</td>
<td>496</td>
<td>403</td>
<td>195</td>
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<td>939</td>
<td>-</td>
<td>680</td>
<td>581</td>
<td>456</td>
<td>245</td>
</tr>
</tbody>
</table>

**Table 7**, Fan assembly costs, March 1988 (£)

It can be seen from the above table that cost is a significant problem, as all the bought out fans are considerably more expensive than the in-house manufactured fans. The Utile fans are more attractively priced, however this quotation was based on supplying impellers and inlet cones only, which would then be fitted to DCE's standard fan assemblies.
### 4.5 Motor Size

<table>
<thead>
<tr>
<th></th>
<th>DCE</th>
<th>ENGART</th>
<th>FAN SYSTEMS</th>
<th>HAMPSON ALAMEIN</th>
<th>PUNKER</th>
<th>STANDARD &amp; POCHIN</th>
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<tr>
<td>F1</td>
<td>0.75</td>
<td>1.1</td>
<td>0.75</td>
<td>1.1</td>
<td>0.55</td>
<td>0.75</td>
<td>0.75</td>
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<tr>
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<td>2.2</td>
<td>3.0</td>
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<td>1.1</td>
<td>1.5</td>
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<td>3.0</td>
<td>3.0</td>
<td>1.5</td>
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<td>3.0</td>
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<td>2.2</td>
<td>3.0</td>
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</tr>
<tr>
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<td>3.0</td>
<td>3.0</td>
<td>3.0</td>
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<td>5.5</td>
<td>4.0</td>
<td>4.0</td>
<td>7.5</td>
</tr>
<tr>
<td>F12</td>
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<td>11.0</td>
<td>7.5</td>
<td>7.5</td>
<td>5.5</td>
<td>5.5</td>
<td>11.0</td>
</tr>
</tbody>
</table>

**Table 8, Motor size, (kW)**

From the above table it would appear that many of the bought out fans offer significant benefits over the existing in-house manufactured fans from a motor size point of view. However, when comparing motor size requirements it is important to compare the fan performance curves as well. A fan with a smaller motor may appear attractive but if the fans performance is not similar it is not a realistic comparison. There is also the possibility that the motor has not been sized correctly which may cause the fan motor to overload.
4.6 Summary & Conclusions

In order to satisfy DCE requirements for a range of quieter fans seventeen fan manufacturers where contacted. Based on the following criteria six quotations were analyzed in detail;

(a) Fan assembly dimensions.
(b) Noise levels.
(c) Fan performance.
(d) Cost.
(e) Power consumption.

After analysis of the quotations it soon became apparent that because of the high costs and large overall fan dimensions of bought out fans the only reasonable approach would be to initiate an in-house exercise to try and develop a range of quieter fans to meet DCE's specific requirements.

This project was therefore initiated to satisfy these requirements (see section 1.3).
5.0 DEFINE PROBLEM

5.1 Specify Objectives and Variables

5.1.1 Objectives
The following are the main objectives in order of priority:

(1) Maintain or reduce fan assembly external dimensions.
(2) Reduce fan noise levels.
(3) Maintain performance (over typical operating range).
(4) Minimise costs.
(5) Reduce power consumption.
(6) Standardize on parts to reduce stock levels.
(7) Completely interchangeable with existing fans.
(8) Utilize current production practises.

5.1.2 Variables
In order to satisfy the above objectives the following main variables need to be assessed for each prototype fan:

(a) Fan assembly dimensions.
(b) Fan Sound Pressure level.
(c) Fan Performance Curve (Fan Static Pressure vs Volume flow).
(d) Costs
(e) Power Curve (Motor Shaft Power vs Volume flow).
5.2 Sales Survey

A survey of past applications established that the F5/G5 and F3/G3 fans were the most popular fans manufactured by DCE during 1989. Initially efforts will be concentrated on these two fan sizes, with any benefits offering maximum return.

Fig.47, DCE fan sales figures for 1989
5.3 Operating Point Analysis

Any one fan can be used on a number of different DCE products, and is required to provide a range of air volumes against different system resistances. Therefore the first exercise of this project was to research past applications and identify typical operating ranges, that should be satisfied by each fan. This work also identified the less important regions of fan performance, where compromise can be made.

This work was carried out using the following filtration velocities to calculate the air volume flow rates;

\[
\begin{align*}
\text{DLMV & UMA units} & = 2.0 \text{ m/min} \\
\text{DU units} & = 2.6 \text{ m/min}
\end{align*}
\]

The following two types of graph were produced for each fan size to determine the primary and secondary ranges;

(a) The number of sales for each fan size against air volume flow. These graphs can be used to determine primary and secondary operating ranges, and also identify the more popular units for each fan. The figure below shows an example of this type of graph for the F10/G10 fan;

![Sales vs Air Volume (F10/G10)](image)

Fig.48, Sales vs Air Volume (F10/G10)
(b) The cumulative sales for each fan against air volume flow rate. These graphs can be used to determine the number and/or percentage of sales that are included in the primary and secondary ranges. The figure below shows an example of this type of graph for the F10/G10 fan;

![Cumulative Sales vs Air Volume (F10/G10)](image)

Fig.49, Cumulative Sales vs Air Volume (F10/G10)

The graph below is a summary graph which shows the primary and secondary ranges when marked on the performance curves of all the fan sizes;

![Performance curves with primary & secondary ranges](image)

Fig.50, Performance curves with primary & secondary ranges
5.4 Summary & Conclusions

The following fans are presently manufactured by DCE:

- **G range**: Used for Unimaster range of dust collectors (UMA).
  - G1, G3, G5, G8, G10, G11, G12
  - F1, F3, F5, F6, F10, F11, F12

- **F range**: Used for Dalamatic (DLMV & DU) and Sintamatic insertable (SI) ranges.

The following are the main objectives in order of priority:

1. Maintain or reduce fan assembly external dimensions.
2. Reduce fan noise levels.
3. Maintain performance (over typical operating range).
4. Minimise costs.
5. Reduce power consumption.
6. Standardize on parts to reduce stock levels.
7. Completely interchangeable with existing fans.
8. Utilize current production practices.

The typical operating ranges for each of the fans were calculated using historic sales information for the various dust collector sizes that had been supplied with a particular fan size.

In order to satisfy the above objectives the following main variables need to be assessed for each prototype fan:

- (a) Fan assembly dimensions.
- (b) Fan Sound Pressure level.
- (c) Fan Performance Curve (Fan Static Pressure vs Volume flow).
- (d) Costs
- (e) Power Curve (Motor Shaft Power vs Volume flow).

After analysis of past sales figures for the various fan sizes it has been determined that the F5/G5 and F3/G3 fans are the most popular so initial efforts will be concentrated on these.

The new DCE range of quieter/increased efficiency fans will be designated K range.
6.0 FAN DESIGN CONCEPTS

To satisfy the objectives listed in section 5.1 various fan design concepts need to be investigated. The main objective is to reduce the noise levels of DCE's standard fan range whilst not increasing the fan assembly dimensions. The aerodynamic noise from a centrifugal fan can be divided into two main components;

(1) **Harmonic part** - A tone is produced by the interaction of the air flow leaving the impeller with the cut-off of the fan casing. This tone is known as the blade passing tone and is at the following frequency and associated harmonics. If this harmonic part is reduced the overall sound pressure level is also likely to be reduced.

\[
\text{Blade passing frequency} = \text{Blade number} \times \text{Speed of rotation} \\
(hz) \quad (rev/sec)
\]

(2) **Broad band part** - Broad band noise is caused by turbulent flow in both the impeller and the casing, also vortex shedding at the blade trailing edges, and it is of a random nature. If this turbulence is reduced the fan noise level should reduce, and also more efficient operation often results in reduced power consumption requirements.

6.1 Impeller Design Features

Backward bladed impellers will be used because of their relatively high efficiency and non-overloading power characteristics which are the requirements of DCE (see section 3.7).
6.1.1 Blade Design

The flow of air through the blade passage of a centrifugal fan impeller is often far from ideal. The object of blade design should be to provide the minimum of flow separation. The inlet blade angle can be more smoothly joined to the outlet blade angle using a curved blade which produces less flow separation than for straight blades and is therefore more efficient.

6.1.2 Impeller Diameter

The impeller diameter is often used to indicate the fan size when using the fan laws (see section 7.1.1).

Flow \[ \alpha \quad d^3 \]
Pressure \[ \alpha \quad d^2 \]
Power \[ \alpha \quad d^5 \]

The fan laws are only accurate for a series of geometrically similar fans (Homologous series). Often the impeller diameter is changed whilst maintaining the impeller width. Generally, by increasing the impeller diameter alone a performance curve approximately parallel to the original is produced.

ie. Large diameter & Narrow width = Steep performance curve
    Small diameter & Wide width = Flat performance curve
6.1.3 Impeller Width [36]

Most centrifugal fan impellers have shrouds, the angle of which should be designed to maintain the area between the blades at a constant value. The mass flow rate along the blade passages remains constant, however, impeller restrictions and sudden expansions should be minimised to maintain the volume flow rate.

The volume of flow through the impeller will be the product of velocity and the area normal to the direction of flow, see fig.51.

\[ Q = \pi d_1 b_1 v_{m1} = \pi d_2 b_2 v_{m2} \]

Where:

- \( Q \) = Volume flow rate \((m^3/s)\)
- \( d_1 \) = Inner Diameter \((m)\)
- \( d_2 \) = Outer Diameter \((m)\)
- \( b_1 \) = Width at inner diameter \((m)\)
- \( b_2 \) = Width at outer diameter \((m)\)
- \( v_{m1} \) = Radial velocity at inner diameter \((m/s)\)
- \( v_{m2} \) = Radial velocity at outer diameter \((m/s)\)

In practice \( v_{m1} = v_{m2} = 0.2u_2 \) [36, page 125]

Therefore

\[ b_1 = \frac{Q}{\pi d_1 (0.2u_2)} \]
\[ = \frac{5Q}{\pi d_1 u_2} \]

\[ b_2 = \frac{Q}{\pi d_2 (0.2u_2)} \]
\[ = \frac{5Q}{\pi d_2 u_2} \]

also

\[ d_1 b_1 = d_2 b_2 \]
6.1.4 Eye Diameter [36]

The volume flow through the eye of the impeller will be the product of velocity and the area normal to the direction of flow.

\[
\begin{align*}
Q &= A \cdot v_e \\
Q &= (\pi d_1^2 / 4) \cdot v_e \\
d_1 &= \sqrt{4Q / \pi v_e}
\end{align*}
\]

Where:
- \(Q\) = Volume flow rate \((\text{m}^3/\text{s})\)
- \(A\) = Area normal to flow \((\text{m}^2)\)
- \(d_1\) = Impeller eye diameter \((\text{m})\)
- \(v_e\) = Impeller eye velocity \((\text{m/s})\)
Ideal Design
Air enters the impeller eye usually through a reducing section from the fan casing inlet, and then turns through a right angle prior to entering the blade passage. The loss here is proportional to $\frac{1}{2} \delta v_e^2$ [36, page 134], therefore the eye velocity $v_e$ should be minimised. Ideally the velocity through the impeller eye ($v_e$) should not be more than the radial velocity at the impeller inlet ($v_{m1}$).

$$ v_e \leq v_{m1} \quad [36, \text{page 125}] $$

$$ d_1 \geq \sqrt{4.4Q / \pi v_{m1}} $$

Since $v_{m1} = 0.2 \ u_2$

Where: $u_2 = \text{Impeller peripheral velocity (m/s)}$

$$ d_1 \geq \sqrt{4.4Q / \pi (0.2u_2)} $$

$$ d_1 \geq \sqrt{20.4Q / \pi u_2} $$

Practical Design
Often the ideal design cannot be achieved since the inlet diameter becomes too large. Generally the velocity through the impeller eye ($v_e$) is of the order of twice $v_{m1}$.

$$ v_e = 2v_{m1} \quad [36, \text{page 125}] $$

$$ d_1 = \sqrt{4.4Q / \pi (2v_{m1})} $$

Since $v_{m1} = 0.2 \ u_2$

Where: $u_2 = \text{Impeller peripheral velocity (m/s)}$

$$ d_1 \geq \sqrt{4.4Q / \pi (0.4u_2)} $$

$$ d_1 \geq \sqrt{10.4Q / \pi u_2} $$

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6.1.5 Angled front plate (Shroud)

Fig. 52, Angled front plate (Shroud)

Most centrifugal fan impellers have shrouds which give mechanical strength and reduce leakage between the blades and casing. The angle of the shroud should be designed to maintain the area between the blades at a constant value. The volume flow rate along the blade passages therefore remains constant and is not restricted or allowed to expand uncontrollably. This feature encourages laminar flow and reduces turbulence. The shroud angle is also very important when calculating the mechanical stress in an impeller, (see section 11.3).

6.1.6 Welded Construction

A rivetted construction generates noise as the air passes over the rivet heads and a welded construction helps to reduce turbulence in these areas.
6.1.7 Blade Number

Most fan theory assumes that the air follows the blade profile exactly, which is only justified if the number of blades is infinite. Since in practice an impeller has a finite number of blades the air flow through the impeller is not ideal. As well as flow separation losses there will also be losses due to inter-blade circulation.

![Inter-blade Circulation](image)

Fig. 53, Inter-blade Circulation

A large number of blades will reduce the flow separation and inter-blade circulation losses. However, if an impeller has a large number of blades the blades themselves constrict the flow especially at the blade inlet. Fluid friction is also increased by increasing the blade number which reduces performance. It is therefore difficult to predict the optimum number of blades using theoretical methods.

The number of blades should also be chosen so as to generate the blade passing component in a frequency regime of minimum acoustic radiation efficiency. This is the reason for the Fast Fourier Transform (FFT) analysis in section 11.1.2.
6.1.8 Aero-foil blades [36]

The object of blade camber design should be to provide the minimum of flow separation, which is generally best achieved in backward curved fans by having blades of aerofoil section.

**Aerofoil Theory**

If a flat plate is inclined at an angle to a moving stream of air, there will be a net force exerted on the plate by the fluid. This force may be split into the following components;

(a) Drag force  - In the direction of flow.
(b) Lift force    - At right angles to the flow.

If the angle of inclination of the plate to the flow (angle of attack) is altered, the drag and lift forces will change. The lift force may be looked upon as the useful component of force whilst the drag is the energy loss component of the force. The aerofoil shape has been developed to give high lift to drag ratios.

The lift force is created by the shape of the upper surface which causes an increase in the air velocity locally, and by Bernoullis equation, a reduction in static pressure. The local velocity at the under surface is little changed and a net upward force, or lift, results (see fig.54). Most of the lift force comes from the first 20% or so of the upper surface. The remainder of this surface being shaped to give as little drag as possible, by giving a gradual return to normal flow conditions. Even so, if the angle of attack exceeds 12 to 16° (depending on the type of section) severe flow separation occurs at the leading edge and then rapidly extends over almost the whole of the upper surface. The result of this is that the drag increases rapidly, whilst the lift reduces and may even fluctuate wildly. This phenomenon can occur quite suddenly and is known as "stalling".

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This aerofoil theory can be applied to centrifugal fan impeller blades, except in reverse. ie. the blades are adding energy to the fluid rather than extracting it.

6.1.9 Angled blades at eye of impeller

This feature helps to turn the air through 90° and therefore reduces turbulence.
6.1.10 Angled blades across impeller

In conventional centrifugal fan design, the cut-off edge and the trailing edge of the blades are parallel to one another. The pressures along the cut-off which are excited by the flow are in phase. When either the cut-off or the blades are inclined, a phase shift of the pressure is introduced which results in local cancellation and hence lower sound radiation into the acoustic far-field.

For this method to be effective the blades should be inclined so that they overlap each other.

Embleton [25] studied the effect of angled blades on impellers with backward curved blades. For a slope of 12 deg. the sound pressure level was reduction by 2 to 3 dB at the blade passing frequency. There was also a slight reduction in the pressure head developed by the fan.

Note.
An inclined fan case cut-off is easier to manufacture and is more effective in reducing noise levels, see 6.2.3.
6.1.11 Staggered blades

By spacing the blades in an irregular pattern around the impeller axis, the blade passing sound energy is spread over a wider frequency band. This is desirable from a subjective point of view, but the total sound intensity generated remains unchanged. Irregular blade spacing was suggested by Hübner [26] for the centrifugal impellers used for cooling electric motors. Krishnappa [30] showed that at certain operating conditions the irregularly spaced blades generate side band tones which can expose higher levels than the original blade passing tone.

Irregular blade spacing makes the manufacture of the impeller more difficult especially from the balancing point of view, i.e. the blade spacing needs to be designed so that the completed impeller can be balanced relatively easily.
6.1.12 Transition meshes

This noise control method was proposed by Petrov, et al [38 & 39]. The figure below shows a centrifugal impeller with mesh screens mounted along both the inner and outer circumference of the blades.

The mesh at the leading edge of the blades is meant to form a small scale turbulent flow behind the mesh and a turbulent blade boundary layer in order to shift the point of flow separation further downstream. The outer transition mesh is to smooth the outlet velocity field and to further reduce the turbulence.

The fan used in the experiment had blades with a forward curved heel and radial tip. The fine mesh (1.3 x 1.3 x 0.25mm) reduced the sound pressure level by 4 to 9 dB in the frequency range 100Hz to 10KHz. The penalty to pay was a 6% loss in the pressure head and a drop of approximately 8% in fan efficiency. The use of a coarse mesh (5.4 x 5.4 x 1.4mm) reduced the sound pressure level by up to 10 dB in the frequency range 100Hz to 2KHz. However, in the frequency range 2KHz to 10KHz the sound pressure levels were increased by up to 8 dB. The losses in pressure head and fan efficiency were also slightly greater than before.
It is therefore apparent that the mesh dimensions are important parameters with respect to noise reduction. However, data given would allow selection of the optimum mesh type for a given fan. The meshes were found to be most effective when used on fans with poor mean flow conditions in the rotating impeller ducts. One has therefore to anticipate that there would be larger losses in pressure head and efficiency for impellers of higher aerodynamic quality, i.e., backward curved blades. The use of meshes is not suitable for applications where contaminated air or gases are to be handled because the meshes are likely to become blocked after a short period of time. This noise reduction method was not investigated because its effect are likely to be limited for backward curved impellers and this technique would significantly complicate the manufacture of DCE impellers.

6.1.13 Aerodynamic Hub

![Aerodynamic nose](image)

Fig.59, Aerodynamic nose

The hub of a centrifugal impeller is directly in the air flow of the fan. It is therefore a possibility that if the aerodynamic properties of the hub are improved there may be an improvement in fan efficiency, and an associated reduction in noise levels.
6.2 Fan Case Design Features

A well designed fan case scroll should provide a uniform flow field for the impeller and have a diffusing section from the fan case throat to the fan case outlet.

A free vortex permits flow without loss and ideally a casing should be designed to permit such flow. A free vortex is where the total energy as expressed by Bernoulli's equation remains constant. It can be proven [36, page 25] that for a free vortex the tangential velocity varies inversely as the radius. The ideal casing profile is a streamline of a free vortex and therefore at any radius $r$ in the casing the following equation applies:

$$\text{Tangential Velocity } \times \text{ Radius of Case} = \text{ Constant}$$

$$v_{u_1}r_1 = v_{u_2}r_2$$

However, a free vortex casing is often uneconomically large and a further moderate sacrifice in efficiency is often accepted to reduce the casing size. Frequently, an arithmetic spiral of the following form is used:

$$r_c = r_1 (1 + k\phi)$$

Where:
- $\phi$ is spiral angle in degrees.
- $r_c$ is the fan case radius.
- $r_1$ is the impeller radius.
- $k$ is of the order of 0.0023 for backward curved fans.
Another simple construction which is often used, consisting of four circular arcs is shown below [36];

![Diagram of fan case construction using 4 circular arcs](image-url)

**Fig. 60, Fan case construction using 4 circular arcs**

Keeping the overall casing dimensions to a minimum results in a large impeller exposure of up to half the impeller diameter. Such large exposures tend to permit reverse flow in the fan outlet at the cut-off which reduces efficiency, see fig 61.

Many fan manufacturers fit a throat plate to reduce vortex formation at this point whilst having little effect on the air velocity at the outer edge of the casing.

The absolute velocity of air leaving the scroll section is often very high and the diffusing section should be designed to reduce the velocity of this air to the casing outlet velocity with as little loss as possible.
6.2.1 Fan case outlet dimensions

**Width**
The ratio of the impeller width to the fan case width is typically 2.5 for most backward curved fans, [36, page 132]. However, if the fan case width is significantly larger than the impeller width large impeller outlet losses will result.

**Length**
Increasing the length of the outlet will help to reduce losses due to sudden expansion at the fan outlet. However, if the outlet is too long a large impeller exposure will result which will tend to permit reverse flow in the fan outlet at the cut-off, resulting in reduced efficiency, see below [32, page 132];

![Diagram](image.png)

**Fig.61, Reverse flow due to large impeller exposure.**
6.2.2 Cut-off dimensions

Cut-off clearance

Very close to the impeller, the circumferential velocity profile exhibits sharp minima and maxima due to the blade wakes. A cut-off placed in this region would experience strong pressure fluctuations which in turn result in an effective sound radiation at the blade passing frequency and its harmonics. When the distance of the cut-off from the impeller is enlarged, the circumferential velocity profile becomes smoother, and the amplitude of the pressure fluctuations at the cut-off become diminished.

A second effect is that a cut-off close to the impeller blocks the flow between two blades periodically. The unsteady flow sets up varying blade forces which are a cause of discrete frequency rotor noise. When the cut-off is placed further away this flow blockage is less severe.
Leidel [32] and Smith, et al [41] carried out studies using centrifugal fans with backward curved impeller blades. They both used conventional volutes, Leidel increased the size of the complete fan casing together with the cut-off clearance. However, Smith, et al only changed the cut off clearance of a single fan case.

The main characteristics of the fans used by references [32] and [41] were as follows;

<table>
<thead>
<tr>
<th></th>
<th>IMPELLER DIAMETER</th>
<th>BLADE NUMBER</th>
<th>IMPELLER SPEED</th>
<th>FLOW COEFFICIENT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smith, et al</td>
<td>464mm</td>
<td>12</td>
<td>1500 rpm</td>
<td>Optimum</td>
</tr>
<tr>
<td>Leidel</td>
<td>199mm</td>
<td>6</td>
<td>3150 rpm</td>
<td>Optimum</td>
</tr>
</tbody>
</table>

The results given by Leidel [32] and Smith, et al [41] agree very closely with each other. In the graphs below the difference in blade passing frequency is plotted versus the cut-off clearance radius ratio (δ/R). The reference point being taken as δ/R =0.06

![Fig.62, Effect of cut-off clearance [32] and [41].](image-url)
The change in blade passing frequency level is very similar in both cases. However, Leidel also found a reduction in the broadband noise of 3 to 5 dB (presumably owing to the lower flow velocities in the enlarged casing), whereas Smith, et al [41] found no change.

In both references [32] and [41] the fan efficiency was not affected by the increase in cut-off clearance.

Leidel found that the minima in fan noise was associated with a maxima in fan efficiency. He found the optimum operation to be at the following cut-off clearance;

\[
\frac{\text{cut-off clearance}}{\text{impeller radius}} = 0.25
\]

Note: This was with a cut-off radius \((r/R)\) of 0.2  
(This is defined on page 119)

An enlarged cut-off clearance will affect the diffusing section of the fan case and a larger fan case will also normally result. For DCE the size of the fan case is often very critical as the fan must fit within the existing dust collector dimensions.

An alternative maybe to reduce the impeller diameter but this will require the impeller design to be modified, such as changing the blade angle and width etc to counteract the change in diameter.
Cut-off radius

Leidel [32] found that the influence of the radius at the lip of the cut-off upon the blade passing frequency level to be much less significant than that of the cut-off clearance. When the cut-off radius \((r/R)\) was enlarged from 0.01 to 0.2 the noise reduction was 6 dB. Fan performance and efficiency remain unaffected by the cut-off radius.

He found the optimum operation to be at the following cut-off radius;

\[
\frac{\text{cut-off radius}}{\text{impeller radius}} = 0.2
\]

Note.
This was with the following cut-off clearances, \((\delta/R)=0.06, 0.10, \text{ and } 0.25\)
In conventional centrifugal fan design, the cut-off edge and the trailing edge of the blades are parallel to one another. The pressures along the cut-off which are excited by the flow are in phase. When either the cut-off or the blades are inclined, a phase shift of the pressure is introduced which results in local cancellation and hence lower sound radiation into the acoustic far-field.

An inclined fan case cut-off is easier to manufacture and is more effective in reducing noise than sloping the blades.

The cut-off should be inclined so as to span at least two blades simultaneously. The slope angle of the cut-off required to encompass one blade spacing is relatively small in the case of multi-vane impellers, but it becomes large in the case of impellers with less than 12 blades. This results in a rather complicated geometry at the fan outlet which causes additional manufacturing costs.
Khoroshev and Petrov [27] obtained a considerable reduction in the sound pressure level at the blade passing frequency with a cut-off slope of 70 deg. However, the pressure head developed by the fan and the efficiency were marginally effected, see below;

![Fig.64, Effect of angled cut-off, Khoroshev and Petrov [27]](image)

**Notes.**

(a) The cut-off clearance, other fan dimensions, and impeller speed were not reported.

(b) Lyons and Platter [33] measured a 10 dB reduction in the sound pressure level at the blade passing frequency for a sharp cut-off placed close to a multi-vane impeller. This reduction was measured for the fan operating at optimum efficiency, at other operating points inclining the cut-off was less effective. Inclining the cut-off did not effect the fan efficiency.

(c) Ploner and Hertz [40] showed that the noise reduction obtained by this method strongly depends on the distance between rotor and cut-off. Both increasing the cut-off clearance and inclining the cut-off edge effect the blade passing noise. Once a considerable noise reduction has been obtained with either of the two methods, the other will be much less effective. ie. their effects are not additive.
6.2.4 Material gauge

Increasing the fan case material gauge should slightly reduce the noise level by reducing break out noise from the fan, (especially if the inlet and outlet are both ducted). However, under certain circumstances, increasing the material gauge can increase noise levels if resonance occurs. This will occur if the natural frequency of the fan case matches the blade passing frequency of the fan.

6.2.5 Inlet Cone

A well designed inlet cone encourages a smooth, streamline flow into the fan and reduces turbulence.

![Inlet Cone Diagram](image-url)

Fig.65, Inlet cone
6.2.6 Impeller and inlet cone clearance

The inlet cone of a centrifugal fan is normally somewhat smaller than the inlet diameter of the impeller. This is so that it can reach into the impeller mouth with an axial overlap of a few millimetres, depending on the size of the fan. Leakage occurs between the impeller eye and the inlet cone and by having a small clearance this leakage can be minimised (see Page 149 for more detail). Also, since the static pressure in the casing is higher than that in the fan inlet, there is a relatively fast flow through the ring gap. This helps the main flow to turn from the axial into the radial direction and tends to reduce unstable flow in the boundary layer by supplying kinetic energy, see below;

![Fig.66, Inlet cone clearance](image-url)
Suzuki and Ugai [44] have found that the size of the annular clearance plays an important role not only for the aerodynamic performance, but also for the noise. They used a fan with a 604mm diameter impeller having backward curved blades. The radial distance between the impeller mouth and inlet nozzle was varied from $T = 1$ to $10\text{mm}$, while the axial overlap was kept at a constant $S = 4\text{mm}$. The smallest ring clearance, $T = 1\text{mm}$ yielded the best results for the experimental fan used, see below;

![Figure 67](image)

*Fig. 67, Effect of inlet cone clearance, Suzuki and Ugai [44]*

It is probable that this value will not give optimum results for all fan sizes, but Suzuki and Ugai [44] did not make any suggestions how to scale ring clearance with fan dimensions. Tests at DCE found that a 2mm ring clearance is an acceptable compromise between fan performance and manufacturing tolerances. Smaller clearances than this may result in the impeller and inlet cone catching when the fan is in service.
6.2.7 Perforated Scroll

The figure below shows the fan used by Bartenwerfer, et al [2], where the volute is made of perforated sheet metal covered by a layer of non woven fabric. The volute itself is mounted in a closed box filled with rockwool.

![Perforated Scroll Diagram](image)

**Fig.68**, Perforated scroll used by Bartenwerfer, et al [2]

This acoustic lining to the fan casing effectively reduces both the harmonic and broad band noise.

The figure below shows the effect on the 'A' weighted sound pressure level in the fan outlet.

![Effect of Acoustic Lining on Outlet dB(A)](image)

**Fig.69**, Effect of acoustic lining on outlet dB(A), reference [2]
6.2.8 Resonators at cut-off

Neise and Koopmann [35] investigated experimentally a method by which an acoustic resonator can be used to reduce at source the aerodynamic noise generated by fans. The casing of a small centrifugal blower was modified by replacing the cut-off of the scroll with the mouth of a quarter wavelength resonator, see below:

Schematic of the fan with λ/4 resonator mounted. \( D = 140 \text{ mm} \).

Enlarged view of cut-off with λ/4 resonator mounted. Dimensions in mm.

Fig. 70, Diagram of resonator used by reference [35]
The resonator was constructed from a series of perforated plates with the same curvature as the original geometry of the casing. Tuning of the resonator was achieved by changing the length via a movable end plug. The noise measurements were taken in an anechoically terminated outlet duct at nearly a free delivery operating condition of the blower. The figure below shows the level of the blade passing frequency as a function of fan speed for the fan without and with resonator, for different tuned frequencies.

![Figure 7.1: Effect of resonator on blade passing frequency [35]](image)

With appropriate tuning, reductions of the blade passing frequency tones of up to 29 dB can be accomplished with corresponding overall sound pressure level reductions of up to 7 dB(A). Parameters which influence the bandwidth of the resonator response are the porosity and hole size of the resonator mouth and the flow velocity near the cut-off region. Throughout the tests, the aerodynamic performance of the blower was unaffected by the addition of the resonator to the casing.
In a second experiment, Koopmann and Neise [29] extended their work to a larger 235mm diameter commercially produced centrifugal blower having 12 backward curved blades. Noise measurements were taken in the anechoically terminated inlet and outlet ducts, under three different loading conditions. In this study the influence of the exposed resonator mouth area was investigated. The permutations involved eight discrete areas as indicated in the schematic of the resonator mouth below:

Fig. 72, Diagram of resonator used by reference [29]
Generally, it appeared that the inlet duct tone reductions were sensitive to the 1-3 regions of the perforations, while those of the outlet duct were most sensitive to regions 4-8. The results obtained with this configuration are shown below;

![Graph showing tone reductions for inlet and outlet ducts with different resonators](image)

Fig. 73, Effect of resonator used in reference [29]
The optimum resonator configuration found involved a splitter-type arrangement which split the resonator tip into two halves. Reductions of 12 to 15 dB are achieved in both ducts simultaneously and for a wide range of loading conditions ($\phi_{\text{min}} - \phi_{\text{max}}$). Although the upper and lower sections of the resonator mouth which influence the noise reduction in the inlet and outlet duct, respectively, share the same volume, the maximum reduction in each duct occurs at a different frequency, depending on the loading conditions. This is due to the different flow regimes over the upper and lower section of the mouth.

The advantage of the noise reduction using resonators seems to be that it does not add substantially to the bulk of the machine.

The suitability of this method is limited for DCE fans because the fans operating point is not fixed. When dust builds up on the filter bags the air volume reduces and changes the impeller velocity triangles which means that the resonator will need to be re-tuned. This is possible but would require additional mechanical or electrical equipment which would be expensive.
6.3 Fan Design Calculations

Velocity triangles for various centrifugal blade forms are shown below [36, page 126];

![Velocity triangles for centrifugal fans](image_url)

Fig. 74, Velocity triangles for centrifugal fans

Using these velocity triangles and various formula [36, section 5.2] it is possible to produce theoretical fan performance curves. However, these predictions assume that the air flow follows the blade profile exactly. This can only be justified if the number of impeller blades is infinite. Equations are also available [36, section 5.3] to predict the effect of these interblade circulation losses on the ideal equations.

The following example (based on [36, appendix I]) uses these equations to calculate the impeller dimensions required to obtain a particular duty;
Example.

**Duty Required**

Fan Static Pressure, \( P_{SF} \) = 230 mmwg  
Volume Flow Rate, \( Q \) = 1500 m³/hr

**Assumptions**

12 bladed backward curved impeller  
Outlet Blade Angle, \( \beta \) = 40°  
Total Efficiency = 75%  
\( \phi \) = 0.18  
\( v_{m1} = v_{m2} = 0.2 U_2 \)  
\( v_o = 2 v_{m1} \)  
\( v_3 = 0.3 U_2 \)  
Casing Width = 2.5 \( b_2 \)  
Air Density, \( \delta_a \) = 1.2 kg/m³  
Water Density, \( \delta_w \) = 1000 kg/m³

**Calculations**

Fan Static Pressure, \( P_{SF} \) = \( \delta_w \cdot g \cdot h \)  
= (1000) \cdot (9.81) \cdot (0.23)  
= 2256 Pa

Volume Flow Rate, \( Q \) = 1500 m³/hr  
= (1500)/(60x60)  
= 0.42 m³/s

\[
\frac{P_{SF} + (1/2 \cdot \delta_a \cdot (0.3 U_2)^2)}{E_{sf}} = \delta_a U_z \left[ \frac{U_2 - \pi \cdot U_2 \cdot \sin \beta_z}{z} - v_m \cdot \cot \beta_z \right]
\]

= \delta_a U_z^2 \left[ \frac{1 - \pi \cdot \sin \beta_z}{z} - \frac{v_m \cdot \cot \beta_z}{U_2} \right]

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Since \( \frac{v_m}{U_2} = 0.2 \) & \( \cot \beta_2 = 1/(\tan \beta_2) \)

\[
\frac{P_{ef} + (1/2 \cdot \delta_a \cdot (0.3 \cdot U_2)^2)}{E_{ef}} = \delta_a U_2^2 \left[ 1 - \frac{\pi \cdot \sin \beta_2}{z} - \frac{0.2}{\tan \beta_2} \right]
\]

\[
\frac{P_{ef} + (1/2 \cdot \delta_a \cdot (0.3 \cdot U_2)^2)}{E_{ef}} = 1.2 U_2^2 \left[ 1 - \frac{\pi \cdot \sin 40^\circ}{z} - \frac{0.2}{\tan 40^\circ} \right]
\]

\[
\frac{2256 + ((1/2) \cdot (1.2) \cdot (0.09 U_2^2))}{0.75} = 1.2 U_2^2 \left[ 1 - \frac{\pi \cdot (0.643)}{12} - \frac{0.2}{0.84} \right]
\]

\[
\frac{2256 + 0.054 U_2^2}{0.75} = 1.2 U_2^2 \left( 1 - 0.168 - 0.238 \right)
\]

\[
\frac{2256 + 0.054 U_2^2}{0.75} = 1.2 U_2^2 \left( 0.594 \right)
\]

\[
2256 + 0.054 U_2^2 = 0.534 U_2^2
\]

\[
0.48 U_2^2 = 2256
\]

\[
U_2^2 = 68.6 \text{ m/s}
\]

\[
\phi = \frac{4 \cdot Q}{\pi \cdot d_2^2 \cdot U_2}
\]

\[
d_2^2 = \frac{4 \cdot Q}{\pi \cdot \phi \cdot U_2}
\]

\[
d_2^2 = \frac{4 \times 0.42}{\pi \times 0.18 \times 68.6}
\]

\[
d_2^2 = 0.043
\]

\[
d_2 = 0.208 \text{ m}
\]

\[
d_2 = 208 \text{ mm}
\]
\[ U_2 = r_2 \cdot w \]
\[ U_2 = \frac{(d_2/2)}{d_2} \cdot w \]
\[ w = 2 \cdot \frac{(U_2/d_2)}{d_2} \]
\[ w = \frac{2 \times 68.6}{0.208} \]
\[ w = 660 \text{ rad/sec} \]
\[ w = 660 \times (60/2\pi) \text{ rev/min} \]
\[ w = 6300 \text{ rev/min} \]

\[ v_{m1} = v_{m2} = \frac{0.2 \cdot U_2}{0.2 \times 68.6} \text{ m/s} \]
\[ = 13.7 \text{ m/s} \]

\[ v_o = 2 \cdot v_{m1} \]
\[ = 2 \times 13.7 \text{ m/s} \]
\[ = 27.4 \text{ m/s} \]

\[ d_1^2 = \frac{4 \cdot Q}{\pi \cdot v_o} \]
\[ d_1^2 = \frac{4 \times 0.42}{\pi \cdot 27.4} \]
\[ d_1^2 = 0.0195 \]
\[ d_1 = 0.14 \text{ m} \]
\[ d_1 = 140 \text{ mm} \]

\[ b_1 = \frac{Q}{\pi \cdot d_1 \cdot v_{m1}} \]
\[ = \frac{0.42}{\pi \times 0.14 \times 13.7} \]
\[ = 0.07 \text{ m} \]
\[ = 70 \text{ mm} \]

\[ b_2 = d_1 \cdot \frac{b_1}{d_2} \]
\[ = 0.14 \times \frac{0.07}{0.208} \]
\[ = 0.047 \text{ m} \]
\[ = 47 \text{ mm} \]
\[ \beta_1 = \tan^{-1} \left( \frac{v_{m1}}{U_1} \right) \]
\[ = \tan^{-1} \left( \frac{v_{m1} \cdot d_2}{U_2 \cdot d_1} \right) \]
\[ = \tan^{-1} \left( \frac{13.7 \times 0.208}{68.6 \times 0.14} \right) \]
\[ = \tan^{-1} (0.297) \]
\[ = 16.5 \text{ deg} \]

Casing outlet velocity, \( v_3 \)
\[ = 0.3 \ U_2 \]
\[ = 0.3 \times 68.6 \]
\[ = 20.58 \text{ m/s} \]

Casing width
\[ = 2.5 \ b_2 \]
\[ = 2.5 \times 47 \]
\[ = 118 \text{ mm} \]

Casing outlet area
\[ = \frac{Q}{v_3} \]
\[ = \frac{0.42}{20.58} \]
\[ = 0.02 \text{ m}^2 \]

Casing outlet height
\[ = \frac{\text{Area}}{\text{Width}} \]
\[ = \frac{0.02}{0.118} \]
\[ = 0.169 \text{ m} \]
\[ = 169 \text{ mm} \]
6.4 Summary & Conclusions

6.4.1 Summary
The following impeller design concepts have been identified which have a direct effect on fan efficiency, performance and noise;

(1) **6.1.1 Blade design** - Curved blades will produce less flow separation than straight blades and will therefore increase fan efficiency and reduce broad band noise.

(2) **6.1.2 Impeller diameter** - The impeller diameter is often used to indicate the fan size when using the fan laws. Generally by increasing the diameter a corresponding increase in fan performance is achieved (along complete performance curve).

(3) **6.1.3 Impeller width** - Generally by increasing the width a corresponding increase in fan performance is achieved (particularly at maximum volume).

(4) **6.1.4 Impeller eye diameter** - The eye diameter should be sized to ensure that the air flow is uniform (ie. is not restricted).

(5) **6.1.5 Angled front plate (Shroud)** - The angle of the shroud should be designed to maintain the area between the blades at a constant value.

(6) **6.1.6 Welded construction** - A welded construction generally reduces noise by preventing turbulence when compared with rivetted construction.
(7) **6.1.7 Blade number** - A large number of blades will tend to reduce flow separation and inter-blade circulation losses. However, large numbers of blades restrict the air flow (particularly at the inlet) and also increase the fluid friction. It is therefore difficult to predict the optimum blade number using theoretical methods.

(8) **6.1.8 Aero-foil blades** - Aerofoil blades reduce flow separation and therefore increase efficiency and reduce noise levels.

(9) **6.1.9 Angled blades at impeller eye** - This feature helps to turn the air through 90° and therefore reduces turbulence.

(10) **6.1.10 Angled blades across impeller** - This feature reduced the blade passing frequency noise by introducing a phase shift of the pressure.

(11) **6.1.11 Staggered blades** - This feature spreads the blade passing frequency over a wider band which is desirable from a subjective point of view.

(12) **6.1.12 Transition mesh** - This feature is intended to reduce turbulence by creating a blade boundary layer.

(13) **6.1.13 Aerodynamic hub** - This feature is intended to reduces turbulence as air enters the eye of the impeller and therefore increase fan efficiency.
The following fan case design features have been identified which effect fan performance, efficiency, and noise;

(1) 6.2.1 Fan case outlet dimensions

**Width** - To allow sufficient outlet area to prevent restriction of the air flow whilst maintaining a compact design the outlet width should be approximately 2.5 times the width of the impeller.

**Length** - The outlet should be long enough to reduce loses due to sudden expansion at the fan outlet, but should not be too long as this will tend to permit reverse flow at the fan cut-off.

(2) 6.2.2 Fan case cut-off dimensions

**Cut-off clearance** - To reduce the blade passing frequency a large cut-off clearance should be used. However, as a large cut-off clearance results in large fan case dimensions the following ratio is often used;

\[
\frac{\text{cut-off clearance}}{\text{impeller radius}} = 0.25
\]

**Cut-off radius** - A large cut-off radius tends to reduce the blade passing frequency, however the following ratio is often used to maintain a reasonably compact fan case;

\[
\frac{\text{cut-off radius}}{\text{impeller radius}} = 0.2
\]

(3) 6.2.3 Angled cut-off - This feature helps to reduce the blade passing frequency by introducing a phase shift of the outlet pressure. Ideally the cut-off should be inclined so as to span at least two blades simultaneously.

(4) 6.2.4 Material gauge - Increasing the fan case material gauge should help to reduce breakout noise from the fan, provided that resonance does not occur.
6.2.5 Inlet cone - A profiled inlet cone will increase fan efficiency and reduces noise by reducing turbulence as the air enters the fan.

6.2.6 Impeller and inlet cone clearance - The inlet cone should protrude into the impeller eye and the smaller the ring clearance the higher the efficiency. However, a 2mm clearance was found to be a satisfactory compromise between fan efficiency and manufacturing tolerances.

6.2.7 Perforated Scroll - Acoustic lining the perforated fan casing effectively reduces the harmonic and broad band noise.

6.2.8 Resonators at cut-off - This feature can be used to reduce noise levels by tuning the resonator to lower the blade passing frequency component.

6.4.2 Conclusions

By incorporating the above features a more efficient and quieter fan should result. The fan calculations detailed in section 6.3 should also be used to arrive at the required fan and impeller dimensions to achieve the required fan performance. Chapter 9.0 details the prototype test results which were carried out to confirm the theory detailed in this chapter.
7.0 PERFORMANCE & NOISE LEVEL PREDICTION METHODS.

7.1 Fan Performance Prediction Methods.

It is not practicable to test the performance of every size of fan in a manufacturers range at all speeds at which it may be designed to run, and with every gas density it may be required to handle. Accurate methods are therefore required to predict the performance of fans.

7.1.1 Fan Laws

By use of the fan laws, the performance of geometrically similar fans of different sizes can be predicted accurately enough for practical purposes. Exact accuracy would require that the effects of, for example surface roughness of the fan, the viscosity of the gas and scale effect be taken into account, but for the vast majority of fan calculations this extreme accuracy is not necessary.

It is important to note, however, that these laws apply to the same point of operation on the fan characteristic. They cannot be used to predict other points on the fans performance curve.

These laws are most often used to calculate changes in flow rate, pressure, and power of a fan when the size, rotational speed or gas density is changed. Therefore, in the following laws the suffix "1" has been used for initial known values and suffix "2" for the changed values and the resulting calculated value.

\[
\begin{align*}
Q_2 &= Q_1 \times \left(\frac{n_2}{n_1}\right) \times \left(\frac{d_2}{d_1}\right)^3 \\
P_2 &= P_1 \times \left(\frac{n_2}{n_1}\right)^2 \times \left(\frac{d_2}{d_1}\right)^2 \times \left(\frac{\delta_2}{\delta_1}\right) \\
P_{w2} &= P_{w1} \times \left(\frac{n_2}{n_1}\right)^3 \times \left(\frac{d_2}{d_1}\right)^5 \times \left(\frac{\delta_2}{\delta_1}\right)
\end{align*}
\]

Where:

\[
\begin{align*}
Q &= \text{Volume flow rate} \\
P &= \text{Pressure} \\
\delta &= \text{Gas density} \\
n &= \text{Impeller rotational speed} \\
d &= \text{Impeller diameter} \\
P_w &= \text{Power}
\end{align*}
\]

(Section 9.3 shows a comparison of test results compared with fan law predictions).
When a significant change of density occurs between the fan inlet and discharge the laws apply to the arithmetic mean of the density and volume. However, for fans operating at pressures below 2kPa (8"wg) the above fan laws may be taken to apply to inlet volume and inlet density.

### 7.1.2 Slip and loss analysis

The theoretical work done by a rotating impeller can be calculated by considering the inlet and outlet velocity triangles produced at the inner and outer diameters of a given impeller. This theoretical work done by the impeller is known as the Euler equation and is derived as follows:

\[
\text{Work done by impeller} = m(v_{u2}u_2 - v_{u1}u_1) \quad \text{[36, page 123]}
\]

The head developed by the impeller may be defined as the height to which the same weight of air may be raised by the same amount of work;

\[
mgH = m(v_{u2}u_2 - v_{u1}u_1) \]

\[
H = \frac{v_{u2}u_2 - v_{u1}u_1}{g}
\]

Using the relationship between head and pressure \((P=\delta gH)\), then;

\[
\text{Euler pressure} = \delta v_{u2}u_2 - \delta v_{u1}u_1
\]

By considering the outlet velocity triangles (fig.75 & fig.76) the following equation can be derived;

\[
\text{Euler pressure} = \left[\delta . (u_2)^2\right] - \left[\delta . \frac{(u_2)}{\pi . b_2 . d_2} . \cot \beta_2 . Q\right]
\]

Where;

- \(\delta\) = Air density \((\text{kg/m}^3)\)
- \(u_2\) = Impeller peripheral velocity \((\text{m/s})\)
- \(Q\) = Volume flow rate \((\text{m}^3/\text{s})\)
- \(b_2\) = Blade width at impeller outer edge \((\text{m})\)
- \(d_2\) = Impeller outer diameter \((\text{m})\)
- \(\beta_2\) = Blade angle at outer edge \((\text{degrees})\)

Reference [36], page 127, equation 5.6 & reference [43], page 1 equation (1) (The full derivation can be found in reference [36], chapter 5.)
Fig. 75, Outlet velocity triangles

Fig. 76, Inlet velocity triangles (showing preswirl)
The actual performance of a centrifugal fan at the design point of operation differs from that predicted by Euler's equation due to a number of losses. The factors that contribute to this reduction in output are as follows:

(a) Inter-blade circulation.
(b) Impeller loss.
(c) Outlet loss.
(d) Inlet loss.
(e) Internal volumetric leakage.

![Slip & loss curves](image)

**Fig. 77, Slip & loss curves**
(a) **Inter-blade circulation**
The Euler equation assumes the air follows the blade profiles exactly. This can only be justified if the number of blades is infinite. The cause of the deviation of flow from the ideal is known as inter-blade circulation sometimes called Slip, see fig. 75;

The effective radius of the area between blades is difficult to assess, but is generally taken as half of the perpendicular distance between the blade tangents at the impeller periphery [36, page 129], therefore;

\[
\text{Effective radius, } a = \frac{\pi d_2 \sin \beta_2}{2z}
\]

Angular velocity, \( w = \frac{u_2}{r_2} = \frac{2u_2}{d_2} \)

\[
\text{Inter-blade Circulation Loss} = \delta . (u_2). a . w \quad [36, \text{ page 129}]
\]

\[
\text{Inter-blade Circulation Loss} = \delta . (u_2) . \frac{\pi d_2 \sin \beta_2}{2z} . \frac{2u_2}{d_2}
\]

\[
\text{Inter-blade Circulation Loss} = \delta . (u_2)^2 . \frac{\pi}{z} \sin \beta_2
\]

Where;

\[
\delta = \text{Air density} \quad (\text{kg/m}^3)
\]

\[
u_2 = \text{Impeller peripheral velocity} \quad (\text{m/s})
\]

\[
z = \text{Number of blades}
\]

\[
\beta_2 = \text{Blade angle at outer edge} \quad (\text{degrees})
\]

Reference [43] page 1, equation (2a).
(b) Impeller loss

Pressure loss will occur within the blade passage due to flow separation resulting from excessive decreases in relative velocity. This loss may be written as follows, see fig.75 & 76:

Impeller pressure loss = \( k_{\text{imp}} \cdot \frac{1}{2} \frac{Q^2}{\pi \cdot d_1 \cdot b_1 \cdot \sin \beta_1} \) \[36, \text{page 134}\]

Inner circumference = \( \pi \cdot d_1 \)

Impeller inner area = \( \pi \cdot d_1 \cdot b_1 \)

\( Q = v_{m1} \cdot \text{inner area} \)

\( v_{m1} = \frac{Q}{\pi \cdot d_1 \cdot b_1} \)

Outer circumference = \( \pi \cdot d_2 \)

Impeller outer area = \( \pi \cdot d_2 \cdot b_2 \)

\( Q = v_{m1} \cdot \text{outer area} \)

\( v_{m2} = \frac{Q}{\pi \cdot d_2 \cdot b_2} \)

From velocity triangles:

\( \sin \beta_1 = \frac{v_{m1}}{v_{r1}} \)

\( v_{r1} = \frac{v_{m1}}{\sin \beta_1} \)

\( v_{r2} = \frac{Q}{\sin \beta_2 \cdot \pi \cdot d_2 \cdot b_2} \)

\( \sin \beta_2 = \frac{v_{m2}}{v_{r2}} \)

\( v_{r2} = \frac{v_{m2}}{\sin \beta_2} \)

Impeller pressure loss = \( k_{\text{imp}} \cdot \frac{1}{2} \frac{Q^2}{\pi \cdot d_1 \cdot b_1 \cdot \sin \beta_1} \) \[36, \text{page 134}\]

Where:

- \( \delta \) = Air density (kg/m\(^3\))
- \( Q \) = Volume flow rate (m\(^3\)/s)
- \( b_1 \) = Blade width at impeller inner edge (m)
- \( d_1 \) = Inlet diameter (m)
- \( b_2 \) = Blade width at impeller outer edge (m)
- \( d_2 \) = Impeller outer diameter (m)
- \( \beta_1 \) = Blade angle at inner edge (degrees)
- \( \beta_2 \) = Blade angle at outer edge (degrees)

\( k_{\text{imp}} \) = Loss Factor (of the order of 0.2 to 0.3 for sheet metal blades, but rather less for blades of aerofoil section).

\( k_{\text{imp}} \) will be greater at points away from the design point since the fans efficiency will be lower at these points on the fan curve.

Reference [43] page 2, equation (3a)
(c) Outlet loss.

Although the casing may be designed with the intention of permitting free vortex conditions, such perfect flow is unlikely in practice. There is almost certainly an increase of area from the blade passage to the volute casing of up to about 2.5 times. There is thus a tendency for eddy formation due to the retardation of flow velocity. It is not easy to compare flow under these conditions with that at a sudden enlargement in normal pipe flow. It does however seem reasonable to assume that the pressure loss in the casing may be written as follows, see fig.75;

\[
\text{Outlet Pressure Loss} = k_{(out)} \delta/2 \cdot (v_2 - v_3)^2
\]

Where; Average velocity at fan outlet, \( v_3 = \frac{Q}{A_3} \)

Using the outlet velocity triangles fig.75 the following can be equated;

\[
v_2 = \sqrt{(u_2)^2 + (v_{m_2}/\sin\beta_2)^2 - (2u_2v_{m_2}/\tan\beta_2)}
\]

Therefore;

\[
\text{Outlet Pressure Loss} = k_{(out)} \delta/2 \cdot (\sqrt{(u_2)^2 + (v_{m_2}/\sin\beta_2)^2 - (2u_2v_{m_2}/\tan\beta_2)} - (Q/A_3))^2
\]

Where;

\[
\begin{align*}
\delta &= \text{Air density} \quad (\text{kg/m}^3) \\
u_2 &= \text{Impeller peripheral velocity} \quad (\text{m/s}) \\
v_{m_2} &= \text{Radial velocity at outer edge} \quad (\text{m/s}) \\
\beta_2 &= \text{Blade angle at outer edge} \quad (\text{degrees}) \\
Q &= \text{Volume flow rate} \quad (\text{m}^3/\text{s}) \\
A_3 &= \text{Fan case outlet area} \quad (\text{m}^2) \\
k_{(out)} &= \text{Loss Factor (of the order of 0.4)}
\end{align*}
\]

Note.

\( k_{(out)} \) will be greater at points away from the design point since the fans efficiency will be lower at these points on the fan curve.

Reference [43] page 4, equation (5b)
(d) Inlet loss

Inlet losses can be divided into two main components, these being;

(a) Loss due to the expansion of the air after passing through the eye of the inlet cone into the impeller and a turning loss due to the change from axial to radial flow, these inlet losses can be calculated as follows, see fig.78;

\[
\text{Inlet eye area, } A_e = \pi d_e^2/4
\]
\[
\text{Inlet eye velocity, } v_e = Q/A_e = 4Q/\pi d_e^2
\]
\[
\text{Inlet pressure loss (1)} = k_{(in)} \cdot \delta/2 \cdot v_e^2 \quad \text{[36, Page 134]}
\]

Inlet pressure loss (1) = \( k_{(in)} \cdot \delta/2 \cdot (4Q/\pi d_e^2)^2 \)

Where;
\[
\delta = \text{Air density (kg/m}^3) \\
Q = \text{Volume flow rate (m}^3/\text{s}) \\
d_e = \text{Inlet cone eye diameter (m)} \\
k_{(in)} = 0.7 \text{ to } 1.1 \text{ (for sheet metal blades)}
\]

Reference [43] page 3, equation (4a) - It should be noted that there is an error in this paper as the \( \pi \) symbol has been omitted.

(b) Expansion or contraction loss as the air enters the blade passages and a turning loss due to pre-swirl, these losses can be calculated as follows;

\[
\text{Inlet pressure loss (2)} = k_{(in(2))} \cdot \delta/2 \cdot (v_e - v_{r1})^2 \quad \text{[36, Page 134]}
\]

From velocity triangles, fig.76;
\[
v_{r1} = \sqrt{v_{m1}^2 + (u_1 - v_{m1} \tan \beta_s)^2}
\]
\[
\text{Inlet pressure loss (2)} = k_{(in(2))} \cdot \delta/2 \cdot \left( \frac{(4Q)(\pi d_e^2)}{\pi} - \sqrt{v_{m1}^2 + (u_1 - v_{m1} \tan \beta_s)^2} \right)^2
\]

Where;
\[
Q = \text{Volume flow rate (kg/m}^3) \\
d_e = \text{Impeller inlet diameter (m)} \\
v_{m1} = \text{Radial velocity at inlet (m/s)} \\
u_1 = \text{Impeller velocity at inner dia. (m/s)} \\
\beta_s = \text{Pre-swirl angle (degrees)} \\
\delta = \text{Air Density (kg/m}^3) \\
n = \text{Speed of rotation (rev/min)} \\
k_{(in)} = 0.7 \text{ to } 1.1 \text{ (for sheet metal blades)}
\]

Reference [43] page 4, equation (4g)
Fig. 78, Diagrammatic arrangement of a centrifugal fan
(e) Internal volumetric leakage.

Internal volumetric leakage occurs where the drive shaft enters the casing and also between the inlet cone and impeller eye. The inlet cone and impeller eye leakage is normally the most significant and can be considered in a similar fashion to flow through an orifice, see fig.78;

\[
\text{Leakage volume, } Q_L = C_d A \pi d_1 \alpha \sqrt{2 \left( \frac{P_s}{\delta} \right)}
\]

Leakage volume, \( Q_L \) = Leakage Volume (m³/s)
\( C_d \) = Discharge Coefficient
\( A \) = Leakage area (m²)
\( \delta \) = Air density (kg/m³)
\( P_s \) = Static pressure difference across clearance (Pa)
\( d_1 \) = Inlet diameter (m)
\( \alpha \) = Inlet clearance (m)

If the edge of the gap is sharp, it is reasonable to assume that the value of \( C_d \) will be of the order of 0.6.

Reference [43] page 4, equation (6).
Comparison with actual test data

The predicted fan performance is very much dependant on the loss factors that are assumed. Reference [36] & [43] recommend the following factors;

\[
\begin{align*}
    k_{(\text{imp})} &= 0.2 \text{ to } 0.3 \\
    k_{(\text{out})} &= 0.4 \\
    k_{(\text{in})} &= 0.7 \text{ to } 1.1
\end{align*}
\]

Note.

These loss factors will be greater away from the design point since the fan efficiency will be lower.

The predicted performance band using the above loss factors is shown below compared with actual test data;

Fig.79, Slip & loss (range) : K15-50Hz
Rather than having a predicted performance band a single predicted performance curve can be obtained by assuming average loss factors.

i.e.

\[ k_{\text{imp}} = 0.25 \]
\[ k_{\text{out}} = 0.4 \]
\[ k_{\text{in}} = 0.9 \]

The predicted performance curve for the K15-50Hz when compared with actual test data is as follows;

![Graph showing predicted and test data for K15-50Hz](image)

**Fig.80, Slip & loss : K15-50Hz**
Using the same loss factors \( K_{\text{imp}} = 0.25, K_{\text{out}} = 0.4, K_{\text{in}} = 0.9 \) the performance of the K5-50Hz and K21-50Hz can be predicted as follows:

**Fig.81, Slip & loss : K5-50Hz**

**Fig.82, Slip & loss : K21-50Hz**
The predictions using the recommended loss factors appear to be reasonably accurate. However, by modifying the K factors for different fan ranges/families more accurate predictions can be obtained.

For the K15-50Hz fan the following prediction is obtained by changing the value of the $K_{(in)}$ from 0.9 to 1.0 (other loss factors remain unchanged);

Fig.83, Slip & loss (Modified loss factor) : K15-50Hz
7.1.3 N.E.L. Turbo-machinery Software

For a number of years the National Engineering Laboratory (N.E.L.) has been involved in research and development work on pumps, fans, and turbines. During this period several computer models have been produced, using both empirical and theoretical methods, for predicting the flow in these machines. In order to make the results of these computer programs more widely available some have been released to industry as design tools. The programs that are presently available are listed below:

1. Mean line analysis
2. Stokes solution
3. Through flow analysis
4. Section Analysis
5. Blade section design
6. Axial flow design suite
7. Super-cavitating design suite
8. HPMF system

The effectiveness of these programs has not been assessed in detail and is an area that may warrant further investigation, see 13.0 (b).
7.2 Fan Noise Level Prediction Methods

7.2.1 Overall Sound Power level

Empirical formulae are available to estimate the likely sound power level that will be produced by a fan. The following seven formulae have been published by the authors shown below;

Erskine & Brunt [22]

\[
SWL = 91 + 10 \log P_w + 30 \log M/M_{20} \quad \ldots \ldots \ldots \ldots \ldots \ldots (1)
\]

Beranek, Kamperman and Allen [3]

\[
\begin{align*}
SWL &= 77 + 10 \log P_w + 10 \log P_{SF} \quad \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots 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\ldOTS
\]

Where:

\[
\begin{align*}
SWL &= \text{Overall Sound Power Level} \quad (\text{dB}) \\
P_w &= \text{Motor Shaft Power} \quad (\text{kW}) \\
P_{SF} &= \text{Fan Static Pressure} \quad (\text{mmWg}) \\
Q &= \text{Volume flow rate} \quad (\text{m}^3/\text{hr}) \\
u &= \text{Impeller peripheral velocity} \quad (\text{m/s}) \\
P_A &= \text{Air power} \quad (\text{kW}) \\
M &= \text{Mach no. at gas temperature} \\
M_{20} &= \text{Mach no. at 20°C}
\end{align*}
\]
The following graph shows a comparison of these equations, compared with actual test data;

It can be seen that the accuracy of the predictions and most suitable prediction method is dependent on the fan range.

ie.  
- **G range**: Equations (2) & (5) are most accurate  
- **K range**: Equations (5), (1) & (7) are most accurate  
- **Solyvent**: Equations (1) and (7)

**Note.**  
These comparisons have been compared with test results taken under full flow conditions.
The following two graphs show a comparison of test data with equation (1),

\[ \text{SWL} = 91 + 10 \log P_w \]

When compared with measurements taken at full flow the predicted levels are generally too low.

When compared with measurements taken at typical operating point the prediction levels are generally too high.
Noise predictions at operating point are more useful than at full flow as these will represent noise levels obtained under typical conditions.

The graph below shows that for DCE fans in general the following equation is applicable:

\[
\text{Sound Pressure level} = 89 + 10 \log P_w
\]

(this equation is accurate to ± 5 dB, see below)

---

More accurate equations can be obtained by dividing the DCE fan range into the following distinct ranges:

1. **F & G range**
2. **Solyvent & K range**

---

**Fig.87, Noise level prediction for DCE fan range**

---

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(1) F & G Range

Sound Pressure level = 91 + 10 log $P_w$

(this equation is accurate to ± 2 dB)

---

Fig. 88, Sound Pressure level prediction for F & G range

(2) Solyvent & K Range

Sound Pressure level = 87.5 + 10 log $P_w$

(this equation is accurate to ± 2 dB)

---

Fig. 89, Sound pressure level prediction for Solyvent & K range

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7.2.2 Octave band sound power spectrum

To calculate the octave band sound spectrum the decibel corrections shown below can be applied to the overall sound power level, [42, page 83];

<table>
<thead>
<tr>
<th>Octave band frequency (Hz)</th>
<th>63</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1k</th>
<th>2k</th>
<th>4k</th>
<th>8k</th>
</tr>
</thead>
<tbody>
<tr>
<td>CENTRIFUGAL</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Forward</td>
<td>-2</td>
<td>-6</td>
<td>-13</td>
<td>-18</td>
<td>-19</td>
<td>-22</td>
<td>-25</td>
<td>-30</td>
</tr>
<tr>
<td>Backward</td>
<td>-3</td>
<td>-5</td>
<td>-11</td>
<td>-12</td>
<td>-15</td>
<td>-20</td>
<td>-23</td>
<td>-26</td>
</tr>
<tr>
<td>Radial</td>
<td>-7</td>
<td>-9</td>
<td>-7</td>
<td>-7</td>
<td>-8</td>
<td>-11</td>
<td>-16</td>
<td>-18</td>
</tr>
<tr>
<td>AXIAL</td>
<td>0</td>
<td>-3</td>
<td>-6</td>
<td>-6</td>
<td>-10</td>
<td>-15</td>
<td>-21</td>
<td>-27</td>
</tr>
<tr>
<td>MIXED FLOW</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Table 9**, Correction to obtain octave band spectra

**Example.** [42, page 83]

A backward curved centrifugal fan is required to extract 13000 m³/hr against a system resistance of 17 mmWg. Estimate the likely sound power level and octave band sound power spectrum for the fan.

Using equation (3);

\[
\text{SWL} = 25 + 10 \log Q + 20 \log P_{SF} \quad \text{dB}
\]

\[
\begin{align*}
\text{SWL} &= 25 + 10 \log (13000) + 20 \log(17) \quad \text{dB} \\
\text{SWL} &= 91 \quad \text{dB}
\end{align*}
\]

<table>
<thead>
<tr>
<th>Octave Band Frequency (Hz)</th>
<th>63</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1k</th>
<th>2k</th>
<th>4k</th>
<th>8k</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overall SWL (dB)</td>
<td>91</td>
<td>91</td>
<td>91</td>
<td>91</td>
<td>91</td>
<td>91</td>
<td>91</td>
<td>91</td>
</tr>
<tr>
<td>Spectrum Correction (dB)</td>
<td>-4</td>
<td>-6</td>
<td>-9</td>
<td>-11</td>
<td>-13</td>
<td>-16</td>
<td>-19</td>
<td>-22</td>
</tr>
<tr>
<td>Octave Band SWL (dB)</td>
<td>87</td>
<td>85</td>
<td>82</td>
<td>80</td>
<td>78</td>
<td>75</td>
<td>72</td>
<td>69</td>
</tr>
</tbody>
</table>

**Table 10**, Example of octave band spectra correction
7.3 Summary & Conclusions

7.3.1 Performance Predictions

To reduce the amount of prototype testing to a minimum the following fan performance prediction methods can be used:

**Fan laws**

The fan laws can be used to predict the performance of geometrically similar fans of different sizes. This prediction method gives accurate results and is normally used to calculate changes in flow rate, pressure, and power when the size, rotational speed, or gas density is changed.

**Slip and loss analysis**

The theoretical work done by a rotating impeller can be calculated using the Euler equation by considering the inlet and outlet velocity triangles. The actual performance is then predicted by applying the following losses to the performance predicted by the Euler equation:

(a) Inter-blade circulation.
(b) Impeller loss.
(c) Outlet loss.
(d) Inlet loss.
(e) Internal volumetric leakage.

Each of the above losses have associated loss factors that are used to adjust each of the losses depending on the fan's design parameters. By adjusting these loss factors the accuracy of the predictions can be increased. i.e. the impeller loss factor used for curved bladed impellers will be less than that used for straight bladed impellers which have greater losses due to flow separation.

This prediction method is a useful method to determine the initial dimensions when manufacturing a prototype fan.
N.E.L. Turbo-machinery software

The National Engineering Laboratory have developed several computer models which can be used to predict the flow in fans. The accuracy of these computer programs has not been assessed during this thesis.

7.3.2 Noise level predictions

Overall sound power level

The following authors have published equations that can be used to prediction the overall sound power level of a fan:

(a) Erskine & Brunt [22]
(b) Beranek, Kamperman and Allen [3]
(c) Lawrence [31]
(d) Deeprose [21]

The accuracy of the equations is very much dependant on the design of the fan impeller.

By modifying the equation published by Erskine & Brunt [22] sound pressure levels within 5 dB can be predicted for all DCE fan ranges. By modifying the equation still further for distinct fan ranges sound pressure levels within 2 dB can be predicted.

Octave band sound power spectrum

By applying a decibel correction to the overall sound power level the octave band spectrum can be predicted for various impeller types.
8.0 EXPERIMENTAL METHODS

8.1 Performance Tests

The following three distinct types of pressure are commonly measured when determining the performance of a fan or a fan in a system:

**Total Pressure** - The pressure that would be obtained if a fluid were brought to rest isentropically (all dynamic head recovered without loss). The algebraic sum of static and velocity pressure.

**Velocity Pressure** - A proportion of the energy contained in a flow of gas exists because of its velocity. The pressure equivalent of this part of the energy is termed velocity pressure. It is always positive.

**Static Pressure** - The pressure of a fluid with the velocity pressure discounted. The static gauge pressure is positive when the pressure at the point is above the ambient pressure and negative when below. It acts equally in all directions and is independent of velocity.

8.1.1 Static Pressure Measurement

Static pressure in a stream of flowing air can only be determined accurately by measuring it in a manner such that the velocity of the air has no influence on the measurement. This is done by measuring it through a small hole or series of holes arranged at right angles to the flow in a surface lying parallel with the lines of flow. The surface must not cause any disturbance to the flow apart from friction. Holes arranged in this manner are referred to as "static holes", or "static tappings" when provided with short connecting pipes at the opposite side of the boundary wall.
8.1.2 Total Pressure

Total pressure is measured using a tube with its open end facing directly into the flow.

8.1.3 Velocity Pressure Measurement

Velocity pressure cannot conveniently be measured directly but can very easily be measured as the difference between total pressure and static pressure by joining the total pressure connection to the positive side of a pressure measuring instrument and the static pressure connection to the negative side.

Various methods are used to measure velocity pressure, these being:

(a) Pitot-tube
(b) Cross tubes
(c) Orifice plate
(d) Venturi
(e) Conical inlet
(f) Anemometers

(a) Pitot-tube

Fig.90, Pitot-tube
The pitot tube is a probe for inserting into a duct to provide in a single instrument, both static pressure holes (90° to flow) and a total pressure hole (forward facing). This allows static and total pressure to be measured simultaneously and if correctly connected also velocity pressure.

The pitot tube traverse pattern recommended for circular and rectangular section ducts are shown below [46, page14];

![Recommended pitot-tube traverse patterns](image)

Fig.91, Recommended pitot-tube traverse patterns

When averaging the readings taken on a duct traverse it is strictly correct to average the air velocities at the points (which is equivalent to averaging the square roots of the velocity pressures) but where no single velocity pressure reading is greater than twice any other the result will not be affected by more than about 1 or 2% if a straight average of velocity pressure is taken for calculating the mean velocity.
Notes.
(a) Measurements should not be taken less than six duct diameters or duct widths downstream of any bend, obstruction, or abrupt change of section to avoid errors due to turbulence.
(b) The quantity of air handled by a fan may be measured either at the entrance to or exit from the system or somewhere in the system itself, provided that all the air handled by the fan passes through the chosen section.

(b) Cross tubes

This method is based on the same principle as the pitot tube traverse. The cross tubes have forward facing holes drilled in the same positions as would be measured for a pitot tube traverse. The cross tubes measure the total pressure and the static pressure is measured on static tappings. The velocity is then measured as the difference between the total and static pressures.
At low air velocities, the velocity pressure is very small, and although sensitive measuring instruments are available it is advantageous to adopt a device which will magnify the velocity pressure, even at the expense of some loss of total pressure.

Bernoulli's equation may be used to show that a restriction in a duct will result in a magnified reading.

By continuity:

\[ A_1 \times v_1 = A_2 \times v_2 \]
\[ v_1 = \frac{(A_2 \times v_2)}{A_1} \]

Using Bernoulli's equation, and assuming no loss of total pressure:

\[ P_1 + \frac{1}{2} \delta \cdot v_1^2 = P_2 + \frac{1}{2} \delta \cdot v_2^2 \]
\[ P_1 - P_2 = \frac{1}{2} \delta \cdot (v_2^2 - v_1^2) \]
\[ P_1 - P_2 = \frac{1}{2} \delta \cdot v_2^2 \cdot (1 - \frac{v_1^2}{v_2^2}) \]
\[ P_1 - P_2 = \frac{1}{2} \delta \cdot v_2^2 \cdot (1 - \frac{A_2^2}{A_1^2}) \]

\[ v_2 = \sqrt{\frac{2(P_1 - P_2)}{\delta \cdot (1 - \frac{A_2^2}{A_1^2})}} \]
The volume flow rate is therefore given as:

\[ Q = A_2 \cdot v_2 \]
\[ Q = A_2 \sqrt{\frac{2(P_1 - P_2)}{\delta (1 - A_2^2/A_1^2)}} \]

Where:
- \( A_1 \) = Duct area (m\(^2\))
- \( A_2 \) = Orifice area (m\(^2\))
- \( v_1 \) = Duct velocity (m/s)
- \( v_2 \) = Orifice velocity (m/s)
- \( \delta \) = Air density (kg/m\(^3\))
- \( P_1 \) = Pressure before orifice (Pa)
- \( P_2 \) = Pressure after orifice (Pa)
- \( Q \) = Volume flow rate (m\(^3\)/hr)

A discharge coefficient \( C_d \), to allow for the total pressure loss, is determined experimentally. The formula to calculate the volume flow rate then becomes:

\[ Q = C_d \cdot A_2 \sqrt{\frac{2(P_1 - P_2)}{\delta (1 - A_2^2/A_1^2)}} \]

Typically for an orifice plate the discharge coefficient \( C_d = 0.60 \) to 0.61
The venturi method uses the same principle as the orifice plate. However, the loss of total pressure incurred is less for the venturi than for the orifice plate (at same mass flow rate).
Typically for a venturi the discharge coefficient $C_d = 0.93$ to 0.98
The entry nozzle of a conical inlet may be regarded in a similar manner to a restriction if the area $A_1$ is infinitely large, when the area ratio $A_2/A_1$ becomes zero. The resulting pressure differential is greater than that given by a pitot tube but will give results to a similar degree of accuracy with a single reading instead of a rather tedious traverse.

Typically for a conical inlet the discharge coefficient $C_d = 0.94$ to 0.96
Anemometer

Anemometers are mainly used for measurement of relatively low air velocities in free field, or point velocities at grilles. However, they require special calibration and are often provided with rather short probes. There are two main types that are available, these being;

**Vane Anemometers** - A rotating vane is used to measure the air velocity.

**Hot-wire Anemometer** - These use the cooling effect on a hot wire or element, to measure the air velocity.

8.1.4 Air flow calculations

(a) Duct velocity

The air velocity at the point of measurement can easily be calculated from the velocity pressure reading using the following formulae;

For standard air (1.2 kg/m³)

\[
\text{Air velocity, } v = 1.291 \times \sqrt{P_v}
\]

Where:

- \( v \) = Air velocity (m/s)
- \( P_v \) = Velocity Pressure (Pa)

OR

\[
\text{Air velocity, } v = 4005 \times \sqrt{P_v}
\]

Where:

- \( v \) = Air velocity (ft/min)
- \( P_v \) = Velocity Pressure (inch wg)
Alternatively the following chart can be used, which shows the relationship between velocity pressure and air velocity for standard air:

![Chart showing relationship between velocity pressure and air velocity](chart)

Fig. 96, Relationship between velocity pressure and air velocity

For non-standard air conditions

Air velocity, \( v \) = \( 1.291 \times \sqrt{\frac{1000}{P_o}} \cdot \frac{T}{289} \cdot \frac{100000}{100000+P_s} \cdot P_v \)

Where:
- \( v \) = Air velocity (m/s)
- \( P_v \) = Velocity Pressure (Pa)
- \( P_o \) = Atmospheric pressure (millibars)
- \( P_s \) = Static pressure in duct (Pa)
- \( T \) = Absolute temperature (°K)
(b) Air volume flow rate

Air Volume flow rate, \( Q = v_D \times A \)

Where:
- \( v_D \) = Duct velocity (m/s)
- \( A \) = Duct area (m²)

(c) Example of a typical air flow calculation

Calculate the air volume flow rate through a 14 inch duct, if the duct velocity pressure is 0.2 inch wg.

Duct velocity = \( 4005 \cdot \sqrt{p_v} \)
= \( 4005 \cdot \sqrt{0.2} \)
= 1791 ft/min
= 9.1 m/s

Duct area = \( \frac{\pi \cdot (\text{duct diameter})^2}{4} \)
= \( \frac{\pi \cdot (14 \times (25.4/1000))^2}{4} \)
= 0.1 m²

Volume flow rate = Duct velocity \( \times \) Duct area
= (9.1 \times 60 \times 60) \times 0.1
= 3276 m³/hr

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8.1.5 Performance test method normally used by DCE

The conical inlet test method is used as a basis to performance test fans at DCE, clause 21 of BS848, Part 1 [4].

**Set-up Procedure**

![Diagram of performance test set-up](image)

**Fig.97, Typical performance test set-up**

(a) Attach suitable conical inlet (test duct) to fan inlet. The conical inlet is generally sized by matching its diameter, as closely as possible, to the inlet diameter of the fan. Standard conical inlets have been made in a variety of diameters to dimensions specified in BS848, Part 1 [4] see below;
In addition, the conical inlet should not be used if the Reynolds Number (Re) is less than 2000.

\[
Re = \frac{\delta v_p d}{\mu}
\]

or

\[
Re = \frac{d}{\mu} \cdot \sqrt{20 \cdot \delta \cdot P_{VI}}
\]

Where:

- \(\delta\) = Air density \((\text{kg/m}^3)\)
- \(\mu\) = Air viscosity \((\text{Ns/m})\)
- \(v_p\) = Test duct air velocity \((\text{m/s})\)
- \(d\) = Test duct diameter \((\text{m})\)
- \(P_{VI}\) = Velocity pressure at fan inlet \((\text{mmWg})\)
Note.
Under standard air conditions (20°C)
\[
\begin{align*}
\delta &= 1.2 \text{ kg/m}^3 \\
\mu &= 1.85 \times 10^{-5} \text{ Ns/m}
\end{align*}
\]
Therefore:
\[
Re = 2.65 \times 10^5 \cdot d \cdot \sqrt{P_{VI}}
\]

(b) Connect measuring devices to conical inlet tapping points.
(Solomat MPM 500e or manometer usually used by DCE).

(c) Connect electrical power measuring device to motor terminals.
(Voltech PM3000 power analyzer and current clamps usually used by DCE).

Test Procedure

The following measurements are taken at various volume flow rates:

(i) Velocity pressure at fan inlet, \((P_{VI})\) .......... (mmWg)
(ii) Static pressure at fan inlet, \((P_{SI})\) .......... (mmWg)
(iii) Motor absorbed power, \((P_{WA})\) .......... (kW)
(iv) Motor input current.................................. (amps)
(v) Motor power factor
(vi) Motor input voltage................................. (volts)

The volume flow rate into the fan is adjusted by inserting grilles of differing flow resistances into the conical inlet (see fig.97). Typically approximately ten flow adjustments are made.

Presentation of results

Fan performance is usually presented in the form of a characteristic curve of air volume flow against the fan static pressure developed by the fan. Motor shaft power is also normally plotted against the air volume flow rate, (see 2.1).
The values are calculated from the test measurements as follows;

Volume flow rate, \( Q \) = \( 11520 \cdot d^2 \cdot \sqrt{P_{VI}} \)

Fan Static Pressure, \( P_{SF} \) = \( P_{SI} - P_{VI} \)

Shaft power, \( P_{WS} \) = \( P_{WA} \times \) Motor efficiency

Where;

- \( Q \) = Volume flow rate \( (m^3/hr) \)
- \( P_{SF} \) = Fan static pressure \( (mmWg) \)
- \( P_{WS} \) = Shaft power \( (kW) \)
- \( P_{VI} \) = Velocity pressure at fan inlet \( (mmWg) \)
- \( P_{SI} \) = Static pressure at fan inlet \( (mmWg) \)
- \( d \) = Test duct diameter \( (m) \)
- \( P_{WA} \) = Absorbed power \( (kW) \)

Motor efficiency is obtained from the motor supplier for the particular motor size on test.

**Correction for temperature**

Fan performance curves should normally be corrected to a standard air density of 1.2 kg/m\(^3\) at 20°C. However, there is no need to consider changes for normal ambient air conditions, (see 2.1.3).
8.2 Noise Tests

8.2.1 In duct sound power level, [5]

The fan is simply connected to a long duct along which all the energy it produces must travel. At some distance along the duct we choose a measuring plane, and the average sound pressure level in the frequency band required is determined by measuring at a number of points on the plane. The total power in that band travelling along the duct (and hence the sound power level of the fan at that frequency) is given by:

\[ SWL = SPL + 10 \log S \ \text{dB} \]

Where;

- SPL - The average sound pressure level across the plane of measurements (dB)
- S - The cross-sectional area of the duct at the plane of measurement (m²)

The following precautions must be observed when performing in duct tests;

(a) The microphone must be fitted with a windshield. Otherwise, aerodynamic pressure fluctuations on the microphone diaphragm, due to the passage of the air over it, will add to and be indistinguishable from acoustic pressure fluctuations.
(b) Sound energy reflections due to impedance changes must be minimised. Once the sound energy has passed the measuring plane, we have to ensure that none is reflected back up the duct from the end to give an artificially high reading. Sound energy would be reflected from the end of the tube if it were left as a plain end. This is because the acoustic impedance inside the tube is different from the impedance in the large volume of air outside. When an acoustic wave in any medium meets a change of impedance some of the energy is always reflected. To minimise this reflection an expansion duct should be fitted at the end of the duct to give a gradual change from the duct impedance to the open air impedance.

(c) Sound energy reflections due to hard surfaces must be minimised. The sound emitted from the end of the duct must not be reflected back into the duct by nearby hard surfaces. To do this an anechoic termination (which is in effect a miniature anechoic chamber) should be fitted over the end of the expansion duct. The following diagram show a typical layout for in-duct sound power measurements of a fan.

Fig.99, In-duct noise test method
8.2.2 Semi-reverberant field tests

While we can measure sound power very accurately with an anechoic chamber, or with a reverberant room, they both require a special construction which makes them rather expensive. Fortunately, it is still possible to obtain a value of SWL sufficiently accurate for most engineering purposes with a semi-reverberant room. As its name implies, it lies in terms of acoustic properties somewhere between the fully absorptive anechoic room and fully reflective reverberant room. In fact, most rooms are semi-reverberant. The fact that we can obtain sound power from a semi-reverberant room, makes this method an attractive one for machine manufacturers.

Most semi-reverberant test procedures call for a substitution test. For this we have to use a standard noise source, which has been previously calibrated so that its sound power level is known. This standard source is placed in the room and the sound level is sampled at a number of points to ensure a fair average is obtained, (BS4196 details recommended measuring positions). The standard source is then replaced by the test machine and the measurements repeated. The test machine sound power level will then differ from the standard source sound power level by the same amount in decibels as the difference between the average sound pressure level in the room with the test machine, and that with the standard source. If the standard source sound power level is SWLs, and it produces a sound pressure level SPLs in the room, and if the test machine produces a sound pressure level of SPLT, then the sound power level of the machine SWLT is given by:

\[
SWLT = SWLS + (SPLT - SPLS) \quad \text{dB}
\]

Note.

BS4196:1981 (ISO3744-1981) "Sound power levels of noise sources" details this test procedure, [10 to 16].
8.2.3 Sound Pressure Levels

This is the method presently used by DCE, and the procedure is as follows;

Sound pressure level measurements are taken 1 metre from the inlet face of the fan case, at the same height as the motor shaft. These measurements are carried out under semi-reverberant conditions, ie. in typical industrial areas with hard floor and walls, and with local equipment turned off. Measurements are normally carried out at full flow and also at a typical volume flow rate using an outlet damper.

Why DCE use this method;
(a) Historical reasons - can easily compare measurements with existing test data.
(b) Direct reading - no calculations required.
(c) Cost - Calibrated noise source or windshield for microphone are not required.
(d) Comparison only - Fan noise levels are only used internally for comparison of fans during fan development research. DCE publicity information quotes dust collector noise levels, and these are carried out in accordance with the machinery directive 89/392/EEC, [34].
(e) Fan tests are normally carried out in the same position and in the same test area.
8.3 Summary & Conclusions

The following methods can be used to measure air flow:

(a) Pitot-tube
(b) Cross tubes
(c) Orifice plate
(d) Venturi
(e) Conical inlet
(f) Anemometer

Depending on the application each of the above have advantages, however, the conical inlet cone is normally used for fan testing.

The following noise test methods are generally used:

(i) In duct sound power measurement
(ii) Semi-reverberant sound power measurement
(iii) Semi-reverberant sound pressure measurement

Methods (i) and (ii) are the methods recommended by BS848 and BS4196 respectively. However, since the fan noise test data is normally only used for comparison purposes during fan development DCE use method (iii). DCE publicity information quote dust collector sound pressure levels in accordance with the Machinery Directive (89/392/EEC) [34].
9.0 EXPERIMENTAL RESULTS

9.1 Impeller Design Features
A large number of prototype fans have been manufactured and tested during the development of the new K-range fans. From these tests various experimental results have been selected that can be used to support or contradict the fan design concepts outlined in chapter 6.

When comparing the fan performance curves it is important to assess the suitability of the various fan designs based on the primary and secondary ranges detailed in fig.50. However, sound pressure level measurements have been taken at a single typical operating point based on the primary range.

Note.
Appendices (15.3) contains a more detailed description of the impellers, fan cases and inlet cones used for these experiments.
9.1.1 Blade Design

(a) Blade radius

From the above graph it can be seen that the impeller with curved blades is more efficient than the impeller with straight blades, i.e. performance curves same but power curve of curved bladed impeller is lower. This higher efficiency is also reflected in the lower noise levels. These results confirm the blade design theory detailed in section 6.1.1.

See appendices (15.3) for details of fans used for the above tests.

- **Straight Blades**: SPL at 5000 m³/hr = 101 dB(A)
- **Curved Blades**: SPL at 5000 m³/hr = 99 dB(A)
Fig. 101, Effect of blade radius on performance

Blade radius 70mm : SPL at 5000 m$^3$/hr = 98 dB(A)
Blade radius 94mm : SPL at 5000 m$^3$/hr = 97 dB(A)
Blade radius 135mm : SPL at 5000 m$^3$/hr = 98 dB(A)

From the above graph it can be seen that the performance curves are very similar for all three blade radii. However, the power curve and noise levels for the impeller with blade radius 94mm are slightly lower, indicating higher efficiency. This shows that there is an optimum blade radius and supports the theory in section 6.1.1 that the blades should be designed to provide the minimum of flow separation.

See appendices (15.3) for details of fans used for the above tests.

Blade radius 70mm : impeller R, fan case 2, inlet cone 2.
Blade radius 94mm : impeller P, fan case 2, inlet cone 2.
Blade radius 135mm : impeller Q, fan case 2, inlet cone 2.
(a) Blade length

![Graph showing Fan Static Pressure vs. Volume Flow Rate](image)

**Fig.102, Effect of blade length on performance**

- **Blade length 104mm**: SPL at 3000 m³/hr = 97 dB(A)
- **Blade length 110mm**: SPL at 3000 m³/hr = 95 dB(A)
- **Blade length 123mm**: SPL at 3000 m³/hr = 94 dB(A)

From the above graph it can be seen that the performance & power curves of a fan are dependant on the blade length. For an impeller of fixed inner and outer diameter the blade angle will vary as the blade length is changed. The above three impellers all have approximately the same efficiency however the impeller with the longest blades and hence the smallest blade angle has the lowest noise level. This supports the theory in section 6.1.1 that the object of good blade design should be to produce the minimum of flow separation.

See appendices (15.3) for details of fans used for the above tests.

- **Blade length 104mm**: impeller P, fan case 2, inlet cone 2.
- **Blade length 110mm**: impeller O, fan case 2, inlet cone 2.
- **Blade length 123mm**: impeller N, fan case 2, inlet cone 2.
9.1.2 Impeller Diameter

Fig. 103, Effect of impeller diameter on performance (K11)

Fig. 104, Effect of diameter on performance (K15)
K11 (470mm Diameter) : SPL at 5000 m$^3$/hr = 102 dB(A)
K11 (490mm Diameter) : SPL at 5000 m$^3$/hr = 103 dB(A)
K15 (510mm Diameter) : SPL at 9000 m$^3$/hr = 108 dB(A)
K15 (580mm Diameter) : SPL at 9000 m$^3$/hr = 111 dB(A)

It can be seen from the two previous graphs that by increasing the impeller diameter the performance of the fan is increased (performance curve is approximately parallel to the original curve). The power requirements of the fan are increased, however at shut-off conditions the shaft power values are very similar. This supports the theory outlined in section 6.1.2.

See appendices (15.3) for details of fans used for the above tests.

K11 (470mm Diameter) : impeller V, fan case 5, inlet cone 4.
K11 (490mm Diameter) : impeller W, fan case 5, inlet cone 4.
K15 (510mm Diameter) : impeller Z, fan case 6, inlet cone 6.
K15 (580mm Diameter) : impeller Y, fan case 6, inlet cone 6.
9.1.3 Impeller Width

Fig.105, Effect of impeller width on performance (G3 & G5)

Fig.106, Effect of impeller width on performance (K3 & K5)
G3 (64mm Wide) : SPL at 1500 m³/hr = 92 dB(A)
G5 (102mm Wide) : SPL at 2000 m³/hr = 95 dB(A)
K3 (64mm Wide) : SPL at 1500 m³/hr = 86 dB(A)
K5 (102mm Wide) : SPL at 2000 m³/hr = 88 dB(A)

By increasing the impeller width it can be seen that the performance of the fan can be increased, this is most apparent at high volume flow rates. The power requirements of the wider impeller are also higher. This supports the theory outlined in section 6.1.3.

See appendices (15.3) for details of fans used for the above tests.

G3 (64mm wide) : impeller A, fan case 1, inlet cone 1.
G5 (102mm wide) : impeller B, fan case 2, inlet cone 1.
K3 (64mm wide) : impeller C, fan case 1, inlet cone 2.
K5 (102mm wide) : impeller D, fan case 2, inlet cone 2.
9.1.4 Eye Diameter

From the theory, since a low eye velocity is preferable, the eye diameter should be as large as possible. However, this was not tested as it is very difficult to set-up a realistic test condition. Generally the blades start at the edge of the impeller eye. For a given size impeller if the eye diameter is changed the length of the blades or their blade angle will also change.

ie. the following would not be normal practise;

![Diagram of impeller with small inlet eye diameter](image)

**Fig.107, Impeller with small inlet eye diameter**
9.1.5 Angled front plate (Shroud)

![Graph showing the effect of angled front plate on performance](image)

**Fig. 108, Effect of angled front plate (shroud) on performance**

- **Straight front plate**: SPL at 2000 m³/hr = 95 dB(A)
- **Angled front plate**: SPL at 2000 m³/hr = 92 dB(A)

It can be seen from the above graph that having an angled front plate reduces the effect of stall points (i.e., from 2000 to 3500 m³/hr the stall condition has been reduced). The maximum power requirements and noise levels of the fan with angled front plate are also slightly lower, this indicates that the efficiency of the fan has been slightly improved. This supports the theory in section 6.1.5 that this feature helps to maintain a constant volume flow rate along the blade passage and prevents losses due to sudden expansions.

See appendices (15.3) for details of fans used for the above tests.

- **Straight front plate**: impeller B, fan case 2, inlet cone 1.
- **Angled front plate**: impeller L, fan case 2, inlet cone 1.
9.1.6 Welded Construction

![Graph showing Fan Static Pressure vs Shaft Power](image)

**Fig.109, Effect of welded construction on performance**

Huckbolted construction : SPL at 2000 m³/hr = 95 dB(A)
Welded construction : SPL at 2000 m³/hr = 92 dB(A)

The welded impeller noise levels are lower than for the huckbolted impeller (huckbolts are a type of fixing used by DCE that is similar to a large rivet) which supports the theory in section 6.1.6 that a rivetted or huckbolted construction generates noise as air passes over the rivet heads. The fan performance curves for the huckbolted and welded impellers are approximately the same.

See appendices (15.3) details of fans used for the above tests.

Huckbolted Construction : impeller B, fan case 2, inlet cone 1.
Welded Construction : impeller E, fan case 2, inlet cone 1.
9.1.7 Blade Number

![Graph showing the effect of blade number on fan performance](image)

**Fig.110, Effect of blade number on performance**

10 Blades : SPL at 2000 m³/hr = 95 dB(A)
9 Blades : SPL at 2000 m³/hr = 97 dB(A)
11 Blades : SPL at 2000 m³/hr = 96 dB(A)

From the above graph it appears that the blade number has little effect on the fan characteristics. It appears that by reducing the number of blades the performance curves and power curves are slightly higher. However, the noise levels of the impeller with 10 blades is slightly lower than the others, which may be the optimum number for this size of fan. This supports the theory in section 6.1.7 that a balance between reducing flow separation, inter-blade circulation & fluid friction is required when finding the optimum blade number.

See appendices (15.3) for details of fans used for the above tests.

- 10 Blades : impeller B, fan case 2, inlet cone 1.
- 9 Blades : impeller F, fan case 2, inlet cone 1.
9.1.8 Aero-foil blades

From 4000 to 10000 m$^3$/hr the power curve is significantly lower whilst the performance curve is unaffected which indicates that the efficiency over this range has been increased. Aero-foils are only effective under certain conditions and may stall outside these limitations which accounts for the improved efficiency being limited to part of the fans characteristic. This confirms the theory in section 6.1.8 that aero-foil blades improve efficiency at the design point. The aerofoil blades tested only had a small camber and greater difference in performance compared with flat blades would be expected with greater cambers.

See appendices (15.3) for details of fans used for the above tests.

Flat Blades : impeller BB, fan case 6, inlet cone 5.
Aerofoil Blades : impeller DD, fan case 6, inlet cone 5.
9.1.9 Angled blades at eye of impeller

![Graph showing FAN STATIC PRESSURE vs. SHAFT POWER for Straight and Angled Blades](image)

**Fig.112, Effect of angled blades at eye of impeller**

- **Straight Blades**: SPL at 2000 m³/hr = 95 dB(A)
- **Angled Blades**: SPL at 2000 m³/hr = 92 dB(A)

From the above graph it can be seen that performance of the fan is slightly higher and the power requirements slightly lower. This indicates that the efficiency of the fan is slightly higher and supports the theory in section 6.1.9 that this design feature helps to turn the air through 90° and therefore reduces turbulence.

See appendices (15.3) for details of fans used for the above tests.

- **Straight Blades**: impeller B, fan case 2, inlet cone 1.
- **Angled Blades**: impeller H, fan case 2, inlet cone 1.
9.1.10 Angled blades across impeller

![Graph showing fan static pressure and shaft power vs volume flow rate]

Fig.113, Effect of angled blades across impeller

- **Straight Blades**: SPL at 2000 m³/hr = 95 dB(A)
- **Angled Blades**: SPL at 2000 m³/hr = 90 dB(A)

The performance curve is significantly lower and the power curve approximately the same indicating that the efficiency of the fan is lower. However, the noise is significantly lower which supports the theory in section 6.1.10.

**Note.**

Embleton [25] also noticed a slight reduction in pressure head developed by the fan when the SPL was reduced by 2-3 dB. For a 5 dB noise reduction a more significant reduction in pressure is likely to occur, which was confirmed by the above test results.

See appendices (15.3) for details of fans used for the above tests.

- **Straight Blades**: impeller B, fan case 2, inlet cone 1.
- **Angled Blades**: impeller I, fan case 2, inlet cone 1.
9.1.11 Staggered blades

![Graph showing FAN STATIC PRESSURE (mmWg) vs SHAFT POWER (kW) for Even Blades and Staggered Blades.]

**Fig.114, Effect of staggered blades on performance**

Even Blades : SPL at 2000 m$^3$/hr = 95 dB(A)
Staggered Blades : SPL at 2000 m$^3$/hr = 94 dB(A)

The noise levels for the impeller with staggered blades are slightly lower than for the impeller with even blade spacing which supports the theory in section 6.1.11 that this feature spreads the blade passage frequency over a wider frequency band.

The performance curve is very slightly lower and the power curve is very slightly higher. This would suggest that the fan efficiency is marginally lower for this design concept.

See appendices (15.3) for details of fans used for the above tests.

Even Blades : impeller B, fan case 2, inlet cone 1.
Staggered Blades : impeller K, fan case 2, inlet cone 1.
9.1.12 Transition meshes

This concept was not investigated because the theory suggests that the effect of this concept would be limited for backward curved impellers. This technique would significantly complicate the manufacture of DCE impellers and there is also the possibility the mesh would become blocked by any dust leakage through the dust filters. Serious impeller balance problems could result from this mesh blockage.
9.1.13 Aerodynamic Hub

![Graph showing fan performance metrics](image)

Fig.115, Effect of aerodynamic hub on performance

Standard Hub : SPL at 2000 m³/hr = 95 dB(A)
Aerodynamic Hub : SPL at 2000 m³/hr = 95 dB(A)

The aerodynamic hub had little effect on the fan efficiency or noise levels. This test neither confirmed or contradicted the theory outlined in section 6.1.13. This feature may be more effective for very large impellers were small efficiency improvements are very advantageous because of the potential power savings.

See appendices (15.3) for details of fans used for the above tests.

9.2 Fan Case Design Features

9.2.1 Fan case outlet dimensions

Width

<table>
<thead>
<tr>
<th>Width</th>
<th>Fan Case Static Pressure (mmWg)</th>
<th>Shaft Power (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>350mm</td>
<td>550</td>
<td>20</td>
</tr>
<tr>
<td>300mm</td>
<td>500</td>
<td>15</td>
</tr>
<tr>
<td>250mm</td>
<td>450</td>
<td>10</td>
</tr>
<tr>
<td>200mm</td>
<td>400</td>
<td>5</td>
</tr>
</tbody>
</table>

Fig.116, Effect of fan case outlet width on performance

350mm wide case : SPL at 9000 m³/hr = 103 dB(A)
215mm wide case : SPL at 9000 m³/hr = 103 dB(A)

It can be seen from the above graph that by increasing the width of the fan case the performance of the fan is increased and the power requirements of the fan are reduced. This shows that an increase in width and associated outlet area helps to reduces the loss due to sudden expansion at the fan outlet. It also confirms the theory in section 6.2.1 that the ratio of impeller width to fan case width should be of the order of 2.5 for backward curved fans.

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350mm wide case, ratio = \frac{\text{fan case width}}{\text{Impeller width}} = \frac{350}{122} = 2.9 \\

215mm wide case, ratio = \frac{\text{fan case width}}{\text{Impeller width}} = \frac{215}{122} = 1.8 \\

See appendices (15.3) for details of fans used for the above tests.

350mm wide case : impeller Z, fan case 6, inlet cone 6.

215mm wide case : impeller Z, fan case 8, inlet cone 5.
The above graph shows the effect of a large impeller exposure. By fitting a throat plate to the fan case cut-off a significant increase in performance was achieved whilst the power curve remained approximately the same. This confirms the theory in section [6.2.1] that large impeller exposures tend to permit reverse flow in the fan outlet at the cut-off which reduces efficiency.

See appendices (15.3) for details of fans used for the above tests.

**Large exposure**
- Impeller AA, fan case 7, inlet cone 5.

**Small exposure**
- Impeller AA, fan case 6, inlet cone 5.
9.2.2 Cut-off dimensions

Cut-off clearance

Fig.118, Effect of fan case cut-off clearance on performance

Large clearance : SPL at 9000 m³/hr = 103 dB(A).
Small clearance : SPL at 9000 m³/hr = 105 dB(A)

The noise levels of the fan with a large cut-off clearance (clearance=105mm, δ/R=0.2) are slightly lower than for the fan with a small cut-off clearance (clearance=20mm, δ/R=0.04). However, a large cut-off clearance is not always possible when fan case dimensions are restricted due to space limitations. This is often the case with DCE fans which are normally required to fit within the existing dust collector dimensions.

Note.
The noise reductions do not appear as significant as those measured by Leidel [32] and Smith, et al [41]. However, they were mainly concerned with the blade passage frequency reduction which would be more noticeable. See appendices (15.3) for details of fans used for the above tests.

Large clearance : impeller AA, fan case 6, inlet cone 5.
Small clearance : impeller AA, fan case 10, inlet cone 5.
The noise levels of the fan with a rounded cut-off are lower than for the fan with a sharp cut-off which supports the theory outlined in section 6.2.2. The above graph shows that by having a rounded cut-off rather than a sharp cut-off the performance of the fan can be increased whilst maintaining the same power requirements. This was not the case for the tests carried out by Leidel [32] who found that the fan performance and efficiency were unaffected.

See appendices (15.3) for details of fans used for the above tests.

Rounded cut-off : impeller AA, fan case 6, inlet cone 5.
Sharp cut-off : impeller AA, fan case 9, inlet cone 5.
9.2.3 Angled cut-off

Fig. 120, Effect of angled fan case cut-off on performance

Straight cut-off : SPL at 2000 m³/hr = 95 dB(A)
Angled cut-off : SPL at 2000 m³/hr = 94 dB(A)

A slight reduction in noise levels was found to result by angling the cut-off of the fan. However, the reduction was not as significant as predicted by the theory in section 6.2.3.

The performance curves shows that having an angled cut-off has little effect on the fan performance or power requirements.

See appendices (15.3) for details of fans used for the above tests.

Straight cut-off : impeller B, fan case 2, inlet cone 1.
9.2.4 Material gauge

Two fan cases where manufactured to the same dimensions, however the material gauges of these cases were as follows:

<table>
<thead>
<tr>
<th>Scroll</th>
<th>Thin Gauge Case</th>
<th>Thick Gauge Case</th>
</tr>
</thead>
<tbody>
<tr>
<td>Back &amp; Front Plate</td>
<td>2mm</td>
<td>3mm</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>VOLUME FLOW RATE (m³/hr)</th>
<th>FAN STATIC PRESSURE (mmWg)</th>
<th>SHAFT POWER (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Thin gauge</td>
<td>Thick gauge</td>
</tr>
<tr>
<td></td>
<td>Pressure</td>
<td>Power</td>
</tr>
<tr>
<td></td>
<td>SPL at 2000 m³/hr = 95 dB(A)</td>
<td></td>
</tr>
</tbody>
</table>

It was found that both the fan performance, power and noise levels were unaffected by the change in fan case material gauge (slight differences are due to repeatability of testing). The effect of material gauge on noise levels may have been more noticeable if the difference in material gauge had been greater. However, it would not be practical for DCE to use a material gauge greater than 3mm as our factory is mainly equipped with machines that are designed for light gauge material. It is also important to ensure that material gauge changes do not cause resonance problems.

See appendices (15.3) for details of fans used for the above tests.

Thin Gauge Case : impeller B, fan case 2, inlet cone 1.
Thick Gauge Case : impeller B, fan case 4, inlet cone 1.
9.2.5 Inlet Cone

The above graph shows that the aerodynamic properties of a profiled inlet cone increase the performance of a fan and also slightly reduce the maximum power requirements of the fan. This confirms the theory in section 6.2.5 that a well designed inlet cone encourages a smooth, streamline flow into the fan, reduces turbulence and leads to increased fan efficiency.

See appendices (15.3) for details of fans used for the above tests.

Simple inlet cone : impeller J, fan case 2, inlet cone 3.
Profiled inlet cone : impeller J, fan case 2, inlet cone 2.
9.2.6 Impeller and inlet cone clearance

The above graph shows that by having a small inlet cone clearance the performance of the fan is increased whilst the power requirements remain unchanged. It should be noted that the difference between the two performance curves becomes more apparent at high volume flow rates when the effect of the inlet cone clearance is more critical. This supports the theory in section 6.2.6 that by having a small inlet cone clearance the losses due to leakage are reduced.

See appendices (15.3) for details of fans used for the above tests.

1mm clearance : impeller EE, fan case 12, inlet cone 7.

6mm clearance : impeller EE, fan case 12, inlet cone 7.
9.2.7 Perforated Scroll

Tests were carried out using a variety of perforated scroll designs which were based on the K5 fan design.

![Perforated Scroll Design](image)

**Fig.124, Perforated scroll design**

<table>
<thead>
<tr>
<th>Standard Scroll</th>
<th>impeller D, fan case 2, inlet cone 2.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Perforated Scroll</td>
<td>impeller D, fan case 13, inlet cone 2.</td>
</tr>
</tbody>
</table>

The following designs were tested with and without an acoustic diffuser which was connected to the fan outlet as shown below;

![Unit with and without A/D](image)

**Fig.125, Unit with and without A/D**
(a) A standard fan scroll was used (no perforations) and the fan was not enclosed in a box.

(b) A standard fan scroll was used (no perforations), the fan was enclosed in a box and the void between the fan case and box was filled with sheets of acoustic foam.

(c) A perforated fan scroll was used, the fan was enclosed in a box but the void between the fan case and box was empty.

(d) A perforated fan scroll was used, the fan was enclosed in a box and the void between the fan case and box was filled with sheets of acoustic foam.

(e) A perforated fan scroll was used, the fan was not enclosed in a box but the outer surface of the scroll was covered by a 1/2" sheet of acoustic foam.

(f) A perforated fan scroll was used, the fan was enclosed in a box and the outer surface of the scroll was covered by a 1/2" sheet of acoustic foam.

(g) A perforated fan scroll was used, the fan was enclosed in a box and the void between the box and fan case was injected with expanding foam.

Sound pressure level measurements were taken at the position shown below to assess the effectiveness of each design;
The following table shows the results of these tests:

<table>
<thead>
<tr>
<th>Description</th>
<th>Without A/D</th>
<th>With A/D</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Standard Scroll</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No Foam (Without Box)</td>
<td>95 dB(A)</td>
<td>85 dB(A)</td>
</tr>
<tr>
<td>Solid Foam (With Box)</td>
<td>92 dB(A)</td>
<td>84 dB(A)</td>
</tr>
<tr>
<td><strong>Perforated Scroll</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>No Foam (With Box)</td>
<td>92 dB(A)</td>
<td>84 dB(A)</td>
</tr>
<tr>
<td>Solid Foam (With Box)</td>
<td>85 dB(A)</td>
<td>76 dB(A)</td>
</tr>
<tr>
<td>Sheet Foam (Without Box)</td>
<td>86 dB(A)</td>
<td>78 dB(A)</td>
</tr>
<tr>
<td>Sheet Foam (With Box)</td>
<td>85 dB(A)</td>
<td>76 dB(A)</td>
</tr>
<tr>
<td>Injected Foam (With Box)</td>
<td>86 dB(A)</td>
<td>78 dB(A)</td>
</tr>
</tbody>
</table>

Table 11, Noise levels for dust collector with perforated scroll

The most practical method for DCE to adopt was to use a perforated scroll with a 1/2" thick sheet of acoustic foam covering the outer surface and enclosed in a box.

This design was developed into a new fan range which forms a major part of a new DCE product known as the Siloair dust collector. This method could be developed to other fan ranges, however, most other DCE fans are fitted to dust collectors which would require major modifications to incorporate this development.

9.2.8 Resonators at cut-off

This was not tested because the suitability of this method is limited for DCE fans because the fans operating point is not fixed. When dust builds up on the filter bags the air volume flow is reduced which would require the resonator to be re-tuned.
9.3 Speed Variation

The following graph shows the curves for a K3-50Hz production fan (normal speed approx. 3000 rev/min) when tested at 60Hz speed (approx. 3600 rev/min).

![Graph showing pressure and power variation with volume flow rate](image)

Fig.127, Effect of impeller speed on performance

The graph below shows a comparison of the actual test results and the curves predicted by the fan laws. It can be seen that the test results are very similar to the curves predicted by the fan laws (section 7.1.1.)

![Graph comparing test results and fan laws](image)

Fig.128, Performance prediction due to speed variation

See appendices (15.3) for details of fan used for the above tests.

Impeller C, fan case 1, inlet cone 2.
9.4 Summary & Conclusions

The results of the prototype testing detailed in this chapter supports the theory detailed in chapter 6.0 for the following design features:

(1) Blade design (6.1.1 & 9.1.1)
(2) Impeller diameter (6.1.2 & 9.1.2)
(3) Impeller width (6.1.3 & 9.1.3)
(4) Angled front plate - Shroud (6.1.5 & 9.1.5)
(5) Welded construction (6.1.6 & 9.1.6)
(6) Blade number (6.1.7 & 9.1.7)
(7) Aero-foil blades (6.1.8 & 9.1.8)
(8) Angled blades at eye of impeller (6.1.9 & 9.1.9)
(9) Angled blades across impeller (6.1.10 & 9.1.10)
(10) Staggered blades (6.1.11 & 9.1.11)
(11) Fan case outlet dimensions (6.2.1 & 9.2.1)
(12) Cut-off dimensions (6.2.2 & 9.2.2)
(13) Angled cut-off (6.2.3 & 9.2.3)
(14) Inlet cone (6.2.5 & 9.2.5)
(15) Impeller and inlet cone clearance (6.2.6 & 9.2.6)
(16) Perforated scroll (6.2.7 & 9.2.7)

The design theories in chapter 6.0 were neither proven or disproved by the prototype testing as its effect could not be observed:

(1) Aerodynamic hub (6.1.13 & 9.1.13)
(2) Material gauge (6.2.4 & 9.2.4)

The following design theories were not tested:

(1) Eye diameter (6.1.4 & 9.1.4) - It was not possible to set-up a realistic test condition.
(2) Transition Mesh (6.1.12 & 9.1.12) - Not suitable for DCE application.
(3) Resonators at cut-off (6.2.8 & 9.2.8) - Not suitable for DCE applications.

Note.

Prototype testing also confirmed the fan laws were very accurate when predicting the effect of speed variation on fan performance (section 9.3).
10.0 ACTUAL DESIGN

10.1 Rationalisation / Standardisation

DCE like any other successful company is continually trying to improve its efficiency. The manufacturing efficiency can be increased by reducing stock levels and reducing work in progress. By rationalising the design of fans to standardise on parts significant manufacturing benefits can be achieved.

The existing F and G ranges have the same impellers and fan cases but are designed to fit particular dust collector ranges. The K range fans have been designed to be interchangeable and are therefore not unit specific as detailed below:

**G Range**

(a) Tall pedestal - For fitment to Unimaster units.
(b) Stiffener bars - Only one fan discharge required (ie. fan case does not need to rotate)
(c) Simple inlet cone - Open inlet

Fig.129, G type fan
**F Range**

(a) Short pedestal  - For fitment to Dalamatic units.
(b) Stiffener plate - Various fan discharges required.
                    (ie. need to allow for rotation of fan case)
(c) Profiled inlet cone - To accommodate fixings to attach to unit.

**K Range**

(a) Short pedestal  - For fitment to Dalamatic units.
                    (Make-up piece used for Unimaster fitment)
(b) Stiffener plate - Various fan discharges required for Dalamatics.
                    (ie. need to allow for rotation of fan case)
(c) Profiled inlet cone - To accommodate fixings to attach to unit.
Advantages of K range construction

(1) Reduced stock levels
(2) Fans become unit specific only at assembly, therefore fans can be assembled to meet urgent requirements. (ie. Parts for a Unimaster fan can be used to manufacture a Dalamatic fan if urgently required).
(3) Fans can be easily adapted to suite different unit if required.

Disadvantages of K range construction

(1) Slightly more expensive than F & G range (approximately 50p more per fan assembly). However, this cost increase is minimal and can be easily offset by the cost reductions due to the use of smaller motors detailed in section 10.3 of this chapter.
10.2 Spark Resistant Construction

Applications may involve the handling of potentially explosive or flammable particles, fumes, or vapours. Such applications require careful consideration of all system components to ensure the safe handling of such gas streams.

10.2.1 AMCA Standard

The AMCA (Air Movement and Control Association) standards deals with fan units installed in systems. The standard contains widely used guidelines which should be followed by both manufacturers and users as a means of establishing general methods of construction. The standard that deals with spark resistant construction is 99-0401-86. This standard which was established in December 1986, lists the following three degrees of protection that comply with this standard:

<table>
<thead>
<tr>
<th>TYPE</th>
<th>CONSTRUCTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>All parts of the fan in contact with the air or gas being handled shall be made of non ferrous materials. Steps must also be taken to ensure that the impeller, bearings, and shaft are adequately attached and/or restricted to prevent a lateral or axial shift in these components.</td>
</tr>
<tr>
<td>B</td>
<td>The fan shall have a non ferrous impeller and non ferrous ring about the opening through with the shaft passes. Ferrous hubs, shafts, and hardware are allowed provided construction is such that a shift of impeller or shaft will not permit two ferrous parts of the fan to rub or strike. Steps must also be taken to ensure that the impeller, bearings, and shaft are adequately attached and/or restrained to prevent a lateral or axial movement in these components.</td>
</tr>
<tr>
<td>C</td>
<td>The fan shall be so constructed that a shift of the impeller or shaft will not permit two ferrous parts of the fan to rub or strike.</td>
</tr>
</tbody>
</table>

Table 12, Types of spark resistant construction
10.2.2 F and G fan ranges

The existing F and G fan ranges when fitted with brass impellers may appear to comply with AMCA'B'. However, in fact they do not comply with any AMCA standard, because if the motor moves its ferrous shaft may rub on the ferrous fan case. A non ferrous ring about the opening through which the shaft passes is required to comply with this standard.

10.2.3 K fan range

The K fan range spark resistant construction method complies with AMCA'C', see below;

![Diagram](image)

Fig.132, K range Spark Resistant Construction

Non ferrous materials other than brass can be used, which may be more cost effective.

If the impeller moves backwards or forwards the ferrous impeller will rub on the ferrous brass rubbing disc or brass inlet cone respectively. If the motor moves the ferrous shaft will rub on the brass rubbing disc and not the ferrous fan case, see fig.133.
Since a shift of the impeller or shaft will not permit two ferrous parts of the fan to rub or strike, the K range fans comply with AMCA'C'.

10.2.4 Brass Impeller vs Brass inlet cone

The method used for the K range fans was chosen because it has the following advantages and disadvantages when compared to having a brass impeller;

Advantages
(a) An inlet cone made from brass is easier to manufacture than a brass impeller.
(b) Brass has a lower yield strength than steel. If a material with a relatively low strength to weight ratio is used for construction the impeller design is more difficult. ie. Thicker gauge material is required, which adds to the impeller weight and increases the stress levels, (see section 11.3 for stress calculations).
(c) Brass inlet cones can be spun using the same tool as standard steel inlet cones. ie. no additional tooling costs.

Disadvantages
(a) A brass impeller construction would comply with AMCA'B' provided a disc is fitted to prevent the motor shaft from rubbing on the fan case.
10.3 Cost Justification

The table below shows the following;
(a) 1988 F & G range costs - ie. costs when market survey was carried out.
(b) 1994 costs for both F & G and K range fans.
(c) Percentage increase in cost of F & G range from 1988 to 1994.

<table>
<thead>
<tr>
<th>FAN DESIGNATION</th>
<th>1988 COSTS</th>
<th>1994 COSTS</th>
<th>PERCENTAGE COST INCREASE</th>
</tr>
</thead>
<tbody>
<tr>
<td>F1</td>
<td>£ 55</td>
<td>£ 67.47</td>
<td>22.7%</td>
</tr>
<tr>
<td>F3</td>
<td>£ 77</td>
<td>£ 93.93</td>
<td>22.0%</td>
</tr>
<tr>
<td>F5</td>
<td>£ 91</td>
<td>£ 109.29</td>
<td>20.1%</td>
</tr>
<tr>
<td>F6</td>
<td>£ 108</td>
<td>£ 133.60</td>
<td>23.7%</td>
</tr>
<tr>
<td>F10</td>
<td>£ 143</td>
<td>£ 183.06</td>
<td>28.0%</td>
</tr>
<tr>
<td>F11</td>
<td>£ 162</td>
<td>£ 203.68</td>
<td>25.7%</td>
</tr>
<tr>
<td>F12</td>
<td>£ 251</td>
<td>£ 314.85</td>
<td>25.4%</td>
</tr>
<tr>
<td>G8</td>
<td>-</td>
<td>£ 201.31</td>
<td>-</td>
</tr>
<tr>
<td>K3</td>
<td>-</td>
<td>£ 95.71</td>
<td>-</td>
</tr>
<tr>
<td>K5</td>
<td>-</td>
<td>£ 102.74</td>
<td>-</td>
</tr>
<tr>
<td>K11</td>
<td>-</td>
<td>£ 221.47</td>
<td>-</td>
</tr>
<tr>
<td>K15</td>
<td>-</td>
<td>£ 330.48</td>
<td>-</td>
</tr>
<tr>
<td>K18</td>
<td>-</td>
<td>£ 503.60</td>
<td>-</td>
</tr>
<tr>
<td>K21</td>
<td>-</td>
<td>£ 593.22</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 13, F, G & K range costs

Note. Average percentage increase in costs = 24%
The following table shows the significant cost saving the K range fans offer over bought out fans:

<table>
<thead>
<tr>
<th></th>
<th>F&amp;G3</th>
<th>F&amp;G5</th>
<th>F&amp;G11</th>
<th>F&amp;G12</th>
</tr>
</thead>
<tbody>
<tr>
<td>ORIGINAL FANS</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1988 costs</td>
<td>£ 77</td>
<td>£ 91</td>
<td>£ 162</td>
<td>£ 251</td>
</tr>
<tr>
<td>1994 costs</td>
<td>£ 93.93</td>
<td>£ 109.29</td>
<td>£ 203.68</td>
<td>£ 314.85</td>
</tr>
<tr>
<td>NEW FANS</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1994 cost</td>
<td>£ 95.71</td>
<td>£ 102.74</td>
<td>£ 221.47</td>
<td>£ 221.47</td>
</tr>
<tr>
<td>cost increase</td>
<td>£ 1.78</td>
<td>(£ 6.55)</td>
<td>£ 17.79</td>
<td>(£ 93.38)</td>
</tr>
<tr>
<td>% increase</td>
<td>1.9%</td>
<td>(6.0%)</td>
<td>8.7%</td>
<td>(29.6%)</td>
</tr>
<tr>
<td>BOUGHT OUT FANS</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1988) Engart</td>
<td>£ 571</td>
<td>£ 614</td>
<td>£ 751</td>
<td>£ 939</td>
</tr>
<tr>
<td>Hampson Alamein</td>
<td>£ 465</td>
<td>£ 500</td>
<td>£ 575</td>
<td>£ 680</td>
</tr>
<tr>
<td>Punker</td>
<td>£ 134</td>
<td>£ 128</td>
<td>£ 195</td>
<td>£ 245</td>
</tr>
<tr>
<td>Standard &amp; Pochin</td>
<td>£ 289</td>
<td>£ 308</td>
<td>£ 403</td>
<td>£ 456</td>
</tr>
<tr>
<td>Utile</td>
<td>£ 374</td>
<td>£ 374</td>
<td>£ 496</td>
<td>£ 581</td>
</tr>
<tr>
<td>BOUGHT OUT FANS</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1994 Predicted) Engart</td>
<td>£ 708</td>
<td>£ 761</td>
<td>£ 931</td>
<td>£ 1164</td>
</tr>
<tr>
<td>Hampson Alamein</td>
<td>£ 577</td>
<td>£ 620</td>
<td>£ 713</td>
<td>£ 843</td>
</tr>
<tr>
<td>Punker</td>
<td>£ 166</td>
<td>£ 159</td>
<td>£ 242</td>
<td>£ 304</td>
</tr>
<tr>
<td>Standard &amp; Pochin</td>
<td>£ 358</td>
<td>£ 382</td>
<td>£ 500</td>
<td>£ 565</td>
</tr>
<tr>
<td>Utile</td>
<td>£ 464</td>
<td>£ 464</td>
<td>£ 615</td>
<td>£ 720</td>
</tr>
<tr>
<td>COST SAVINGS</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Based on 1994 costs) Engart</td>
<td>£ 612.29</td>
<td>£ 658.26</td>
<td>£ 709.53</td>
<td>£ 942.53</td>
</tr>
<tr>
<td>Hampson Alamein</td>
<td>£ 481.29</td>
<td>£ 517.26</td>
<td>£ 491.53</td>
<td>£ 621.53</td>
</tr>
<tr>
<td>Punker</td>
<td>£ 70.29</td>
<td>£ 56.26</td>
<td>£ 20.53</td>
<td>£ 82.53</td>
</tr>
<tr>
<td>Standard &amp; Pochin</td>
<td>£ 262.29</td>
<td>£ 279.26</td>
<td>£ 278.53</td>
<td>£ 343.53</td>
</tr>
<tr>
<td>Utile</td>
<td>£ 368.29</td>
<td>£ 361.26</td>
<td>£ 393.53</td>
<td>£ 498.53</td>
</tr>
<tr>
<td>% COST SAVINGS</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(Based on 1994 costs) Engart</td>
<td>86%</td>
<td>86%</td>
<td>76%</td>
<td>81%</td>
</tr>
<tr>
<td>Hampson Alamein</td>
<td>83%</td>
<td>83%</td>
<td>69%</td>
<td>74%</td>
</tr>
<tr>
<td>Punker</td>
<td>42%</td>
<td>35%</td>
<td>8%</td>
<td>27%</td>
</tr>
<tr>
<td>Standard &amp; Pochin</td>
<td>73%</td>
<td>73%</td>
<td>56%</td>
<td>61%</td>
</tr>
<tr>
<td>Utile</td>
<td>79%</td>
<td>78%</td>
<td>64%</td>
<td>69%</td>
</tr>
</tbody>
</table>

**Table 14**, Cost comparison with bought out fans
The following tables use the cost savings from the previous table and calculate annual cost savings:

<table>
<thead>
<tr>
<th>ORIGINAL FANS</th>
<th>F3 &amp; G3</th>
<th>F5 &amp; G5</th>
<th>F11 &amp; G11</th>
<th>F12 &amp; G12</th>
</tr>
</thead>
<tbody>
<tr>
<td>NEW FANS</td>
<td>K3</td>
<td>K5</td>
<td>K11</td>
<td>K11</td>
</tr>
<tr>
<td>ORIGINAL FANS (1994)</td>
<td>£ 93.93</td>
<td>£ 109.29</td>
<td>£ 203.68</td>
<td>£ 314.85</td>
</tr>
<tr>
<td>NEW FANS (1994)</td>
<td>£ 95.71</td>
<td>£ 102.74</td>
<td>£ 221.47</td>
<td>£ 221.47</td>
</tr>
<tr>
<td>1989 ANNUAL FAN SALES</td>
<td>1565</td>
<td>1575</td>
<td>360</td>
<td>175</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>ANNUAL COST SAVING</th>
<th>Engart</th>
<th>£ 958,234</th>
<th>£ 1,036,760</th>
<th>£ 255,431</th>
<th>£ 164,943</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Hampson Alamein</td>
<td>£ 753,219</td>
<td>£ 814,685</td>
<td>£ 176,951</td>
<td>£ 108,768</td>
</tr>
<tr>
<td></td>
<td>Punker</td>
<td>£ 110,004</td>
<td>£ 88,610</td>
<td>£ 7,391</td>
<td>£ 14,443</td>
</tr>
<tr>
<td></td>
<td>Standard &amp; Pochin</td>
<td>£ 410,483</td>
<td>£ 439,835</td>
<td>£ 100,271</td>
<td>£ 60,118</td>
</tr>
<tr>
<td></td>
<td>Utile</td>
<td>£ 576,374</td>
<td>£ 568,984</td>
<td>£ 141,671</td>
<td>£ 87,243</td>
</tr>
</tbody>
</table>

Table 15, Annual cost saving due to K3, K5 & K11 compared with bought out fans

Total annual cost saving due to development of K3, K5 & K11 when compared with cost of buying fans from fan manufacturers (assuming that new range of fans is required):

<table>
<thead>
<tr>
<th></th>
<th>£ 2,425,368</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engart</td>
<td></td>
</tr>
<tr>
<td>Hampson Alamein</td>
<td>£ 1,853,623</td>
</tr>
<tr>
<td>Punker (not inc. case)</td>
<td>£ 220,448</td>
</tr>
<tr>
<td>Standard &amp; Pochin</td>
<td>£ 1,010,707</td>
</tr>
<tr>
<td>Utile</td>
<td>£ 1,374,272</td>
</tr>
</tbody>
</table>

Table 16, Summary of cost savings due to K3, K5 & K11

The above table shows that if DCE had bought the K3, K5, and K11 fans from a fan manufacturer in order to obtain quieter more efficient fans than the F and G range it would have cost the company over 1 million pounds per year, (Punker prices do not include fan cases).
There is also another significant benefit to DCE following the implementation of the K11 fan. The K11 replaces both the F/G11 and the F/G12. The F/G12 uses an 11 kW motor which would normally require a star/delta starter to run the fan whereas the K11 uses a 7.5 kW motor which normally only requires a direct on line starter.

7.5 kW direct on line starter = £38
11 kW star/delta starter = £265
Fan starter cost saving per fan = £227 (86% saving)
F12 and G12 annual sales (1989) = 175

Annual cost saving = £227 x 175 = £39,725

<table>
<thead>
<tr>
<th>BOUGHT OUT FAN DESIGNATION</th>
<th>-</th>
<th>CD15</th>
<th>CD18</th>
</tr>
</thead>
<tbody>
<tr>
<td>DCE FANS</td>
<td></td>
<td>K15</td>
<td>K18</td>
</tr>
<tr>
<td>BOUGHT OUT COST</td>
<td>£700.00</td>
<td>£905.00</td>
<td>£988.00</td>
</tr>
<tr>
<td>DCE COSTS</td>
<td>£330.48</td>
<td>£503.60</td>
<td>£593.22</td>
</tr>
<tr>
<td>SAVINGS PER FAN</td>
<td>£369.52</td>
<td>£401.40</td>
<td>£394.78</td>
</tr>
<tr>
<td>% SAVINGS PER FAN</td>
<td>53%</td>
<td>44%</td>
<td>40%</td>
</tr>
<tr>
<td>ANNUAL SALES (1994)</td>
<td>Concept units</td>
<td>-</td>
<td>6</td>
</tr>
<tr>
<td>UMA750 units</td>
<td>27</td>
<td>46</td>
<td>7</td>
</tr>
<tr>
<td>Total</td>
<td>27</td>
<td>52</td>
<td>17</td>
</tr>
<tr>
<td>ANNUAL COST SAVING (1994)</td>
<td>Concept units</td>
<td>-</td>
<td>£2,408</td>
</tr>
<tr>
<td>UMA750</td>
<td>£9,977</td>
<td>£18,464</td>
<td>£2,763</td>
</tr>
<tr>
<td>Total</td>
<td>£9,977</td>
<td>£20,872</td>
<td>£6,711</td>
</tr>
</tbody>
</table>

Table 17, Annual cost saving due to K15, K18 & K21 compared with bought out fans

Total annual cost saving due to development of K15, K18 & K21 = £37,560
Summary of cost savings

Cost comparisons can be made using the following two criteria:

(a) New K fan range compared with old F & G fan range.
(b) New K fan range compared with bought out fans.

(a) New K range compared with old F & G fan range

<table>
<thead>
<tr>
<th>Fan Type</th>
<th>Saving per Fan (£)</th>
<th>Annual Usage (Per Fan)</th>
<th>Annual Savings (£)</th>
</tr>
</thead>
<tbody>
<tr>
<td>K3 (compared with F3)</td>
<td>1.78</td>
<td>1565</td>
<td>2,786</td>
</tr>
<tr>
<td>K5 (compared with F5)</td>
<td>6.55</td>
<td>1575</td>
<td>10,316</td>
</tr>
<tr>
<td>K11 (compared with F11)</td>
<td>17.79</td>
<td>360</td>
<td>6,404</td>
</tr>
<tr>
<td>K11 (compared with F12)</td>
<td>93.38</td>
<td>175</td>
<td>16,342</td>
</tr>
</tbody>
</table>

Table 18, Summary of K range cost savings compared with F & G range

Fan cost saving per year due to development of the above fans = £ 17,468

Annual controller cost saving due to the K11 implementation (Direct on line starting rather then Star-delta) = £ 39,725

Total cost saving per year due to development of the above fans = £ 57,193

Total tooling cost to implement the above fans into production = £ 12,000

Pay back period for investment = \[
\frac{\text{Implementation cost}}{\text{Annual savings}} = \frac{12,000}{57,193} = 0.21 \text{ years} = \text{Approx. 11 weeks}
\]

Total annual cost saving = Approximately £ 57,000
(b) New K range compared with bought out fans

<table>
<thead>
<tr>
<th></th>
<th>SAVING PER FAN</th>
<th>ANNUAL USAGE</th>
<th>ANNUAL SAVINGS</th>
</tr>
</thead>
<tbody>
<tr>
<td>K3 (compared with Utile F3)</td>
<td>£ 368.29</td>
<td>1565</td>
<td>£ 576,374</td>
</tr>
<tr>
<td>K5 (compared with Utile F5)</td>
<td>£ 361.26</td>
<td>1575</td>
<td>£ 568,984</td>
</tr>
<tr>
<td>K11 (compared with Utile F11)</td>
<td>£ 393.53</td>
<td>360</td>
<td>£ 141,671</td>
</tr>
<tr>
<td>K11 (compared with Utile F12)</td>
<td>£ 498.53</td>
<td>175</td>
<td>£ 87,243</td>
</tr>
<tr>
<td>K15 (compared with Stockbridge Airco)</td>
<td>£ 369.52</td>
<td>27</td>
<td>£ 9,977</td>
</tr>
<tr>
<td>K18 (compared with Stockbridge Airco)</td>
<td>£ 401.40</td>
<td>52</td>
<td>£ 20,872</td>
</tr>
<tr>
<td>K21 (compared with Stockbridge Airco)</td>
<td>£ 394.78</td>
<td>17</td>
<td>£ 6,711</td>
</tr>
</tbody>
</table>

Table 19, Summary of K range cost savings compared with bought out fans

Fan cost saving per year due to development of the above fans = £ 1,411,832

Annual controller cost saving due to the K11 implementation
(Direct on line starting rather then Star-delta) = £ 39,725

Total cost saving per year due to development of the above fans = £ 1,451,557
Total tooling cost to implement the above fans into production = £ 17,000

Pay back period for investment = \(\frac{\text{Implementation cost}}{\text{Annual savings}}\)

= \(\frac{17,000}{1,451,557}\)

= 0.012 years

= Approx. 5 days

Total annual cost saving = Approximately £ 1.45 Million
10.4 Summary & Conclusions

The F and G range of fans have the same impellers but are designed to fit particular dust collector ranges. The K range have been designed to be interchangeable and are therefore not unit specific. This allows stock levels and work in progress to be reduced and therefore increases manufacturing efficiency.

Spark resistant F and G fans are manufactured with brass impellers to prevent explosions when handling flammable particles, fumes or vapours. However, the K range fans us a standard steel impeller with a brass rubbing disc attached to the fan case backplate and a brass inlet cone. This method is advantageous as the steel impeller is easier to manufacture and is a stronger/lighter design because of the increase in material yield strength.

Cost comparisons can be made using the following two criteria;

(a) New K range fan range compared with old F & G fan range.

(b) New K range fan range compared with bought out fans.

The table below shows the annual cost savings for each of these criteria;

<table>
<thead>
<tr>
<th></th>
<th>Compared with F&amp;G fans</th>
<th>Compared with typical Bought out fans</th>
</tr>
</thead>
<tbody>
<tr>
<td>K3</td>
<td>£ 2,786</td>
<td>£ 576,374</td>
</tr>
<tr>
<td>K5</td>
<td>£ 10,316</td>
<td>£ 568,984</td>
</tr>
<tr>
<td>K11</td>
<td>£ 9,938</td>
<td>£ 228,914</td>
</tr>
<tr>
<td>K15</td>
<td>-</td>
<td>£ 9,977</td>
</tr>
<tr>
<td>K18</td>
<td>-</td>
<td>£ 20,872</td>
</tr>
<tr>
<td>K21</td>
<td>-</td>
<td>£ 6,711</td>
</tr>
<tr>
<td>Controller saving</td>
<td>£ 39,725</td>
<td>£ 39,725</td>
</tr>
<tr>
<td>Total saving</td>
<td>£ 57,193</td>
<td>£ 1,451,557</td>
</tr>
</tbody>
</table>

Table 20, Summary of K range cost savings (from table 18 & 19)

It can be seen that the K range fans offer significant cost benefits especially when compared with bought out fans.
11.0 ASSOCIATED INVESTIGATION

During the design of a fan the following tests, calculations and analysis also need to be carried out to ensure that a safe design is achieved:

(a) Vibrations tests
- Fan assembly vibration tests
- Impeller FFT analysis
- Fan assembly tracking tests
- Unit resonance tests

(b) Balancing

(c) Stress analysis
- Stress calculations
- Finite stress analysis
- Brittle lacquer tests
- Strain gauge tests
- Overspeed tests

(d) Moment of inertia calcs.

Included in this chapter is a case study which outlines how impeller failures can occur if all the above are not carried out during the design of a fan.

11.1 Vibration Tests

11.1.1 Fan Assembly Vibration levels

For machines the most common specification deals with the vibratory effect of unbalance. This is a limit placed on the displacement, velocity, or acceleration that can be measured on a critical vibrating part. The amount of unbalance that can be tolerated varies with the speed and mass of the rotating parts, the sturdiness of the bearings and the supporting structure. BS848 : Part6 : 1989, [7] details a method of measuring fan vibration, however, BS4675 : Part1 : 1976 (ISO2372-1974), [17], is the main standard that recommends acceptable fan assembly vibration levels. The full title of this standard is "Mechanical vibration in rotating machinery". Part 1 of this standard is "Basis for specifying evaluation standards for rotating machines with speeds from 10 to 200 revolutions per second".
For class I machines (typically electric motors of up to 15kW) the vibration levels must be below 4.5 mm/s rms to achieve a grade C classification, see table;

<table>
<thead>
<tr>
<th>Ranges of vibration severity</th>
<th>Examples of quality judgement for separate classes of machines</th>
</tr>
</thead>
<tbody>
<tr>
<td>Range</td>
<td>mms</td>
</tr>
<tr>
<td>0.28</td>
<td>0.28</td>
</tr>
<tr>
<td>0.45</td>
<td>0.45</td>
</tr>
<tr>
<td>0.71</td>
<td>0.71</td>
</tr>
<tr>
<td>1.12</td>
<td>1.12</td>
</tr>
<tr>
<td>1.8</td>
<td>1.8</td>
</tr>
<tr>
<td>2.8</td>
<td>2.8</td>
</tr>
<tr>
<td>4.5</td>
<td>4.5</td>
</tr>
<tr>
<td>7.1</td>
<td>7.1</td>
</tr>
<tr>
<td>11.2</td>
<td>11.2</td>
</tr>
<tr>
<td>18</td>
<td>18</td>
</tr>
<tr>
<td>28</td>
<td>28</td>
</tr>
<tr>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>71</td>
<td></td>
</tr>
</tbody>
</table>

Table 21, Recommended machinery vibration severity ranges
11.1.2 FFT Analysis

Introduction
The various components of a fan impeller have natural resonant frequencies and it is possible to excite the components at these frequencies during operation.

The result of operating a fan at a rotating speed that coincides with the shafts natural frequency are:
(a) Very high amplitude vibration.
(b) Extreme sensitivity to unbalance.
(c) Damage to the bearings and shaft (Possibility of catastrophic failure).

In the same way any single component of a fan impeller is subject to unexpected failure if excited at its natural frequency by some outside force.

The purpose of this section is to explain how one may avoid such failures using the following approach;
(i) Measure the vibration response of the various components.
(ii) Anticipate the frequencies that will be excited in the field installation.
(iii) Change the natural frequency of components (if necessary) to avoid excitation in the field.

This procedure was successfully used to determine a resonance mode near to running speed for a standard Fl1-60Hz impeller, see 11.5.
Test Set-up

The figure below shows a typical impact test set-up:

![Diagram of impact test set-up](image)

**Fig.134, Typical FFT impact test set-up**

**Spectrum Analyzer** - This is a dual channel fast fourier transform real time analyzer which accepts the incoming signals and processes them into the frequency domain almost instantaneously.

**Accelerometer** - The output signal is proportional to the acceleration of the test specimen.

**Impact Hammer** - A single strike of the hammer excites the test specimen over a wide frequency range and the output of the force transducer is proportional to the impact force.
To prepare the impeller for testing it should be hung with its shaft centre in a horizontal plane so that there is no contact between the impeller and the floor or any other obstruction. It should be hung to simulate the on-site mounting arrangement as below:

![Diagram of Centre-hung and Over-hung rotor suspensions]

**Fig.135, Support of Impeller**

The following locations are normally tested on a centrifugal impeller:

(a) Shroud O/D - midway between two blades.
(b) Shroud I/D - midway between two blades.
(c) Blade leading edge - midway between shroud and backplate.
(d) Blade trailing edge - midway between shroud and backplate.
(e) Backplate O/D - midway between two blades.
Test Procedure

For each location the accelerometer is attached to one face of the metal and the hammer strikes the other face directly opposite the accelerometer. The most convenient way to effectively mount the accelerometer is with beeswax. Under extreme temperatures or high frequency measurements (above 1 kHz) other methods such as screw stud or adhesive are required. The hammer should strike each location several times (typically 4 to 8), allowing time between each hit for the vibration to dissipate. By averaging the readings the effect of ambient vibrations and non-linearities are greatly reduced.

The following graphs can now be displayed on the screen of the spectrum analyzer;

(a) Transfer function vs Frequency
The transfer function is essentially the input of channel B (Acceleration) divided by the input channel A (Force).

(b) Phase angle vs Frequency
This is used in the following two ways;
(i) For each test point a phase shift indicates a resonant frequency. Theoretically, this is a shift of $180^\circ$ to $0^\circ$ (or vice versa) with a value of $90^\circ$ at the actual frequency.
(ii) Mode shapes can be determined from the phase relationship. This is done by moving the accelerometer to different points on the impeller and having the excitation point constant. Alternatively, leave the accelerometer at one location and vary the excitation point. By comparing the phase values of all the points at each resonance, a mode shape can be defined.
Analysis of test results

Once the measurements are complete the results are ready to be analyzed. The first thing to look for is resonance at or near one of the fan operationally induced frequencies described below:

**Operating speed** - This is always of concern, however, most component parts have natural frequencies well above the operating frequency of industrial fans.

**Blade passing frequency** - This is the primary concern as a source of excitation, and is equal to the number of blades multiplied by the operating speed.

**Rotating stall** - This sometimes occurs in a system with a partially closed inlet damper. Typically this results in a vibration frequency of approximately two thirds of the operating speed.

**Duct induced vibration** - This can be due to stack length, duct work turns, unusual fan inlet configurations. These may be a source of low frequency excitation.

**Mechanical drive vibration** - This can be caused by bearing problems, coupling misalignment, gearbox fault, bent fan or motor shaft, etc.

**Shaft torsional frequency** - This can be an important exciting force especially in variable speed drive applications.

**Corrective action**

Normally the stiffness of components with resonance problems is changed. Increasing the stiffness raises the natural frequency whereas reducing the stiffness lowers the natural frequency. The stiffness can be increased by increasing the material gauge or by adding stiffeners. The mode shape determination outlined earlier is very useful in deciding the type and location of stiffener required.
Impulse excitation vs Swept-sine excitation
The impulse excitation method offers several advantages over swept-sine excitation, these being:

(1) It is much more convenient because the amount of time and equipment is greatly reduced. The impulse hammer excites a wide frequency range in one strike, whereas the swept-sine wave technique involves a slow process of scanning through frequency ranges. The shaker and corresponding hardware used in the swept-sine wave procedure is all replaced with a simple hammer with integral force transducer and a small power supply. Therefore, set-up time and actual test time are much less.

(2) The excitation point can be easily moved anywhere on the fan impeller. This provides a much better response for each individual component and allows the mechanical compliance (displacement per unit input force) to be determined anywhere on the impeller. The shaker can usually only be attached to one point with its output oriented in one direction. If that particular point happens to be a node for a frequency of interest, then the full response frequency will be distorted.

The impulse excitation method does however have the following disadvantage:

(1) Its accuracy is very sensitive to structural non-linearities, however this is of limited concern with integrally welded fan impellers.

(The impulse excitation method was successfully used to determine an impeller resonance mode for the F11-60Hz impeller).
11.1.3 Fan assembly tracking tests.

Introduction

This test is used to produce a plot of vibration amplitude against fan speed. It is very useful for quickly determining vibration peaks and resonance points of fan assemblies. This test was successfully used to determine a vibration peak near to running speed for a standard F11-60Hz fan assembly, see 11.5.

Test Procedure

(1) Fix transducer to a critical Vibrating Part (ie. fan pedestal).
(2) Bring fan up to a speed of at least normal operating speed, preferably faster.
(3) Actuate vibration measuring instrument, (ie. Shenck Vibroport30).
(4) Switch of fan.
(5) Allow fan to run down until impeller has stopped.
(6) Store test data.
(7) Plot graph.

The graph below shows a typical fan assembly tracking plot;

![Graph](image)

**Fig.136, Typical fan assembly tracking plot.**
11.1.4 Unit Resonance Tests

Introduction
This test is used to produce a plot of vibration amplitude against fan speed for critical areas of machinery such as dust collector panels. It is very useful for quickly determining vibration problems of critical components. This test was successfully used in confirming the existence of a standing wave in the header box of a DU dust collector, see 11.5.

Test Procedure
(1) Fix transducer to a critical component (eg. dust collector header box ).
(2) Bring fan up to a speed of at least normal operating speed, preferably faster.
(3) Actuate vibration measuring instrument, (ie. Shenck Vibroport 30).
(4) Switch of fan.
(5) Allow fan to run down until impeller has stopped.
(6) Store test data.
(7) Plot graph.

The graph below shows a typical unit resonance plot;

![Graph of Vibration Amplitude against Fan Speed](image.png)

Fig.137, Typical unit resonance plot
11.2 Balancing

Vibrations occur in all rotating machines. As long as the vibrations remain within permissible limits then no special attention is required. However, if they exceed the permissible limits the safety of the machine and operating personnel are threatened.

Fan impellers must be balanced to reduce vibration levels to permissible levels. Balancing is accomplished by redistributing the mass so that the principal inertia axis more closely coincides with the axis of rotation. Because perfect balance is never achieved specifications usually list the permissible unbalance, and are normally as the following:

**Permissible Residual Unbalance,** \( (U_{\text{per}}) \) - this is the balancing mass required at a particular radius to perfectly balance the impeller and is usually expressed in g.mm.

\[
U_{\text{per}} = m \times r \quad \text{(g.mm)}
\]

**Eccentricity,** \( (e) \) - this is the distance between the principle inertia axis and the axis of rotation and is usually expressed in \( \mu \text{mm} \).

**Permissible Residual Specific Unbalance,** \( (e_{\text{per}}) \) - this is residual unbalance divided by impeller mass and is usually expressed in g.mm/kg.

\[
e_{\text{per}} = \frac{U_{\text{per}}}{M}
\]

\( e_{\text{per}} \) can also be considered as equivalent to a permissible displacement of the mass centre of the rotor from the shaft axis, expressed in \( \mu \text{m} \).

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Balance quality grade (G-grade) - Balance quality grades have been established which permit a classification of quality requirements.

\[
\text{G-grade} = \frac{e_{\text{per}} \times w}{1000} \quad \text{(mm/s)}
\]

\[
w \quad \text{(rad/sec)} = \frac{\text{Motor speed (rpm)} \times 2.\pi}{60}
\]

\[
\text{G-grade} = \frac{U_{\text{par}} \times w}{1000 \cdot M} \quad \text{(mm/s)}
\]

\[
\text{G-grade} = \frac{m \cdot r}{1000 \cdot M} \times \frac{2 \cdot \pi \cdot n}{60}
\]

\[
\text{G-grade} = \frac{m \cdot r \cdot n \cdot \pi}{3000 \cdot M}
\]
The graph below can also be used to calculate the G-grade as an alternative to the above equation:

Fig.138, G-grade calculation graph
11.2.1 Types of unbalance

It is important to distinguish between static and dynamic unbalance. In the figure on the following page a rotating impeller is idealized as two axially supported discs on a shaft between bearings, and centrifugal forces are shown as vectors.

Diagram A: This diagram depicts a statically unbalanced condition. This condition causes a parallel displacement of the central principal inertia axis relative to the shaft axis, the separation corresponding to the displacement of the centre of gravity of the impeller. If the impeller was not rotating it would tend to assume a position with the unbalanced mass at the bottom. This can be statically balanced by adding a single balancing mass at the proper position at the opposite side of the impeller as in (B) and (C). However method (B) is the better solution because it is dynamically balanced as well as statically balanced.

Diagram B: This diagram depicts a statically balanced and dynamically balanced condition.

Diagram C: This diagram depicts a statically balanced but dynamically unbalanced condition. This is a special case of dynamic unbalance called a couple unbalance which only occurs when the impeller is statically balanced. In this condition the main axis of the inertia is inclined by an angle to the shaft and intersects this at the centre of gravity of the impeller. If the impeller is both statically and dynamically unbalanced the main axis of inertia does not intersect the shaft axis at the centre of gravity of the impeller, i.e. is skewed relative to the shaft axis.

Diagram D: This diagram depicts a balanced couple which is a statically balanced and dynamically balanced condition.
Fig. 139, Static & Dynamic unbalance
11.2.2 Balancing Techniques

In order to balance an impeller it is necessary to restore its mass symmetry by removing or adding mass. Since perfect balance is never achieved acceptable balance quality grades are listed in the following Standard. BS6861 : Part 1 : 1987 (ISO1940/1-1986) Part 1 Balance quality requirements of rigid rotors [18]. This standard states that the recommended balance quality grade for a fan is G6.3, see below;

<table>
<thead>
<tr>
<th>Balance quality grade</th>
<th>Product of the relationship ( r \times \omega u )</th>
<th>Rotor types — General examples</th>
</tr>
</thead>
<tbody>
<tr>
<td>G4 000</td>
<td>4 000</td>
<td>Crankshaft/drives of rigidly mounted slow marine diesel engines with uneven number of cylinders</td>
</tr>
<tr>
<td>G3 000</td>
<td>1 600</td>
<td>Crankshaft/drives of rigidly mounted large two-cycle engines</td>
</tr>
<tr>
<td>G2 000</td>
<td>830</td>
<td>Crankshaft/drives of rigidly mounted large four-cycle engines</td>
</tr>
<tr>
<td>G250</td>
<td>250</td>
<td>Crankshaft/drives of rigidly mounted large four-cycle diesel engines</td>
</tr>
<tr>
<td>G100</td>
<td>100</td>
<td>Crankshaft/drives of fast diesel engines with six or more cylinders</td>
</tr>
<tr>
<td>G40</td>
<td>40</td>
<td>Compressors (gasoline or diesel) for cars, trucks and locomotives</td>
</tr>
<tr>
<td>G16</td>
<td>16</td>
<td>Crankshaft/drives of engines, trucks, and locomotives</td>
</tr>
<tr>
<td>G8.3</td>
<td>6.3</td>
<td>Drive shafts (propeller shafts, cardan shafts) with special requirements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Parts of crushing machinery</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Parts of agricultural machinery</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Individual components of engines (gasoline or diesel) for cars, trucks and locomotives</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Crankshaft/drives of engines with six or more cylinders under special requirements</td>
</tr>
<tr>
<td>G2.5</td>
<td>2.5</td>
<td>Crankshaft/drives of engines under special requirements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Marine main turbine gears (merchant service)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Centrifuge drums</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Paper machinery rolls; print rolls</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Fans</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Assembled aircraft gas turbine rotors</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Flywheels</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Pump impellers</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Machine-tool and general machinery parts</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Medium and large electric armatures (of electric motors having at least 60 mm shaft heights) without special requirements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration-insulating mountings</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Individual components of engines under special requirements</td>
</tr>
<tr>
<td>G1</td>
<td>1</td>
<td>Gas and steam turbines, including marine main turbines (merchant service)</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Rigid turbo-generator rotors</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Computer memory drums and discs</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Turbo-compressors</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Machine-tool drives</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Medium and large electric armatures with special requirements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Small electric armatures not qualifying for one or both of the conditions specified for small electric armatures of balance quality grade G6.3</td>
</tr>
<tr>
<td>G0.4</td>
<td>0.4</td>
<td>Turbine-driven turbines</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Tape recorder and phonograph (gramophone) drives</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Grinding-machine drives</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Small electric armatures with special requirements</td>
</tr>
</tbody>
</table>

Table 22; Recommended G-grades

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Static Balancing
Static balancing serves to compensate for the static component of unbalance only. The unbalance can be determined by rolling on bezels but this is only sufficient for narrow rotors rotating without axial run-out. However, a higher degree of accuracy can be obtained if the unbalance measurement is performed during rotation, namely on a balancing machine. For unbalance correction a mass correction has to be performed in a radial plane of the rotor, at best in the centre of gravity of the plane.

Dynamic Balancing
In the case of elongated rotors the couple unbalance must not be neglected. For these rotors dynamic balancing is often indispensable. In contrast to the static unbalance the effect of the couple unbalance are only obvious during rotation. Therefore, it can only be measured as a rotating body on a balancing machine, i.e. not by rolling on bezels. The unbalance correction for a rotor supported on two bearings requires mass correction in at least two radial planes.
Two distinct dynamic balancing techniques are used, these being:

(i) Balancing impellers
(ii) Balancing fan assemblies

Balancing impellers
Impellers are normally balanced prior to assembly of the fan. The impeller is mounted on a mandrel and then balanced on the balancing machine at a relatively low speed (typically 500 rpm). This method is relatively quick and is ideal as a production method for most fan manufacturing companies.

The figure below shows a typical impeller balancing machine.

Fig. 140, Typical impeller balancing machine

Balancing fan assemblies
A portable balancing machine can be used to balance fan assemblies. Since the impeller is mounted on its own shaft and is balanced at its operating speed this method should give a better balance quality than balancing the impeller alone.
11.3 Stress Analysis

11.3.1 Stress Calculations

The various parts must be designed not only for their aerodynamic function but also for mechanical integrity.

**Torque**

The shaft torque $T_s$ delivered to a fan by its driving motor can be calculated from the shaft power $P_{ws}$ and the fan speed using the following equation [20, Section 7.1].

$$Shaft\ Torque,\ T_s = \frac{1000 \cdot P_{ws}}{2 \cdot \pi \cdot n} \quad N.m$$

Where:

- $P_{ws}$ = Shaft Power (kW)
- $n$ = Speed of rotation (rev/sec)

**Centrifugal Force**

Consider a body rotating at constant angular velocity about a fixed axis. Each element of the body has an acceleration towards the centre of rotation, the value of which can be calculated using the following equation:

$$\text{Acceleration towards centre} = \frac{(\text{linear velocity})^2}{\text{radius}}$$

The restraining force exerted by adjacent elements (called centripetal force), which equals the product of this acceleration and the mass, is therefore directed toward the centre of rotation. The force exerted by the mass on its restraints must equal the restraining force and act in the opposite direction. This force, is known as the centrifugal force $F_c$, and can be calculated using the following equation:

$$\text{Centrifugal Force, } F_c = \frac{m \cdot v^2}{r}$$
Since \( v = r \cdot w \)

then

Centrifugal Force, \( F_c \) = \( \frac{m \cdot (r \cdot w)^2}{r} \)

= \( m \cdot r \cdot w^2 \)

and since \( w = 2 \cdot \pi \cdot n \)

Centrifugal Force, \( F_c \) = \( 4 \cdot \pi^2 \cdot n^2 \cdot m \cdot r \)

Where

- \( w \) = Angular velocity (rad/sec)
- \( v \) = Linear velocity (m/s)
- \( n \) = Speed of rotation (rev/sec)
- \( m \) = Mass of element (Kg)
- \( r \) = Radius of centre of gravity (m)

The above equation can be used to calculate the centrifugal force related to each of the elements of a rotating structure by using the mass \( m \) of the element and the radius \( r \) of its centre of gravity. Similarly the centrifugal force of the entire rotating structure will be the same as if the total mass were concentrated at its centre of gravity.
Example
This example is based on some of the calculations carried out during the development of the K5-60Hz impeller. Appendices 15.4 contains a print out from a spreadsheet that was written to carry out these calculations.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fan Speed</td>
<td>3440 rev/min</td>
</tr>
<tr>
<td>Impeller outside diameter</td>
<td>320 mm</td>
</tr>
<tr>
<td>Front plate inner diameter</td>
<td>182 mm</td>
</tr>
<tr>
<td>Back plate inner diameter</td>
<td>40 mm</td>
</tr>
<tr>
<td>Blade width</td>
<td>102 mm</td>
</tr>
<tr>
<td>Blade length</td>
<td>115 mm</td>
</tr>
<tr>
<td>Blade number</td>
<td>10</td>
</tr>
<tr>
<td>Blade thickness</td>
<td>3.0 mm</td>
</tr>
<tr>
<td>Backplate thickness</td>
<td>3.0 mm</td>
</tr>
<tr>
<td>Frontplate thickness</td>
<td>2.5 mm</td>
</tr>
<tr>
<td>Material Density, δ</td>
<td>7790 Kg/m³</td>
</tr>
<tr>
<td>Poissons ratio, σ</td>
<td>0.29</td>
</tr>
<tr>
<td>Blade Angle, β</td>
<td>27.5 deg</td>
</tr>
</tbody>
</table>

**Front Plate Stresses**

Maximum Radial Stress = \( \frac{\delta \cdot w^2}{8} \left( \sigma + 3 \right) \left( r_2 - r_1 \right)^2 \)

= \( \frac{(7790) \cdot (360.236)^2 \cdot (0.29 + 3) \cdot (0.16 - 0.091)^2}{8} \)

= \( 1.98 \times 10^6 \text{ N/m}^2 \)

= \( 1.98 \text{ N/mm}^2 \)

Max. Tangential Stress = \( \frac{\delta \cdot w^2}{4} \left[ (\sigma + 3)r_2^2 + (1 - \sigma)r_1^2 \right] \)

= \( \frac{(7790) \cdot (360.236)^2 \cdot [(0.29+3)0.16^2+(1-0.29)0.091^2]}{4} \)

= \( 22.77 \times 10^6 \text{ N/m}^2 \)

= \( 22.77 \text{ N/mm}^2 \)
Back Plate Stresses

Since the backplate has to support the blades and shroud the density is corrected to compensate for this increased loading.

Back Plate Mass = \[ \frac{\pi \cdot (d_2 - d_1) \cdot t \cdot \delta}{4} \]
= \[ \frac{\pi \cdot (0.32^2 - 0.04^2) \cdot (0.003) \cdot (7790)}{4} \]
= 1.85 Kg

Mass of blades = (No. of blades) \cdot (Width) \cdot (Length) \cdot (Thickness) \cdot (Density)
= (10) \cdot (0.102) \cdot (0.115) \cdot (0.003) \cdot (7790)
= 2.743 Kg

Corrected Density = \[ \frac{\delta \cdot (\text{Backplate Mass} + \text{Mass of Blades})}{\text{Backplate Mass}} \]
= \[ \frac{7790 \cdot (1.85 + 2.743)}{1.85} \]
= 19340 Kg/m³

Maximum Radial Stress = \[ \frac{\delta \cdot w^2 \cdot (\sigma + 3) \cdot (r_2 - r_1)^2}{8} \]
= \[ \frac{(19340) \cdot (360.236)^2 \cdot (0.29 + 3) \cdot (0.16 - 0.02)^2}{8} \]
= 20.22 x 10⁶ N/m²
= 20.22 N/mm²

Max. Tangential Stress = \[ \frac{\delta \cdot w^2 \cdot [ (\sigma + 3)r_2^2 + (1 - \sigma)r_1^2 ]}{4} \]
= \[ \frac{(19340) \cdot (360.236)^2 \cdot [(0.29+3)0.16^2+(1-0.29)0.02^2]}{4} \]
= 53.0 x 10⁶ N/m²
= 53.0 N/mm²

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Blade Stresses

Bending Stress (at inside edge)

\[ \frac{b^2 \cdot r \cdot w^2 \cdot \cos \beta}{2t} \]

\[ = \frac{(0.102)^2 \cdot (7790) \cdot (0.091) \cdot (360.236)^2 \cdot (\cos 27.5)}{2(0.003)} \]

\[ = 141.49 \times 10^6 \text{ N/m}^2 \]

\[ = 141.49 \text{ N/mm}^2 \]

Bending Stress (at outside edge)

\[ \frac{b^2 \cdot r \cdot w^2 \cdot \cos \beta}{2t} \]

\[ = \frac{(0.102)^2 \cdot (7790) \cdot (0.16) \cdot (360.236)^2 \cdot (\cos 27.5)}{2(0.003)} \]

\[ = 248.78 \times 10^6 \text{ N/m}^2 \]

\[ = 248.78 \text{ N/mm}^2 \]
Rivet Stresses

If the blades are attached by rivets, these will have to withstand the total centrifugal force due to the blade. Thus shear stress on the rivets can therefore be calculated using the following equation, assuming equal loading on rivets;

\[
\text{Blade mean radius} = \frac{r_2 - r_1}{2}
\]

\[
= \frac{0.16 - 0.091}{2} = 0.1255 \text{ mm}
\]

Stress in rivets \[
= \frac{m \cdot r \cdot w^2}{z \cdot a}
\]

\[
= \frac{(0.274) \cdot (0.1255) \cdot (360.236)^2}{(3) \cdot (17.814 \times 10^{-6})} = 83.5 \times 10^6 \text{ N/m}^2 = 83.5 \text{ N/mm}^2
\]

Where

- \(m\) = mass of blade (kg)
- \(r\) = blade mean radius (m)
- \(w\) = angular velocity (rad/sec)
- \(a\) = cross-sectional area of each rivet (m)
- \(z\) = number of rivets (attaching blade to backplate)
Weld Stresses

If the blades are welded to the backplate the stress in the weld can be calculated from the following equation:

\[ \text{Stress in weld} = \frac{m \cdot r \cdot w^2}{A} \]

Where;

Cross-sectional area of weld, \( A = \) 2. (weld length). (weld width). cos 45 degrees
(for a double fillet weld)

\[ = 2 \cdot (0.115) \cdot (0.005) \cdot (0.7071) \]
\[ = 8.13 \times 10^{-4} \text{ m}^2 \]

Stress in weld

\[ = \frac{(0.274) \cdot (0.1255) \cdot (360.236)^2}{8.13 \times 10^{-4}} \]
\[ = 5.5 \times 10^6 \text{ N/m}^2 \]
\[ = 5.5 \text{ N/mm}^2 \]
11.3.2 Finite stress analysis

This technique is widely recognised as a cost effective method of identifying stress and dynamic problems in a structure while it is still on the drawing board, reducing both lead time and expenditure on prototype evaluation. It involves the computerised construction of a mathematical model of the subject structure whose response can then be analyzed against a variety of simulated conditions. Performance of the structure can then be viewed graphically, (see page 269 & 270 when this technique was used to analyse stresses in an impeller).

11.3.3 Brittle Lacquer tests

This tests has not been used by DCE however I have included a summary of its uses, based on the work of R. G. Patton [37]. Patton used a lacquer called "Stresscoat", manufactured by the Magnaflux corporation. It is good practise to use two lacquers of different sensitivities, the lacquer is sprayed onto the impeller and allowed to dry before testing the impeller. Firstly the less sensitive lacquer is used to find the areas of high tensile strain. Secondly the more sensitive lacquer is used to show the general strain pattern and hence find the directions of principal stresses.

However the brittle lacquer technique has the following drawbacks;

(a) Because the lacquer must be sprayed on when the component is stationary it is not possible to use the technique of spraying the component under stressed conditions and unloading, in order to measure compressive strains.

(b) When an impeller is run, especially in an enclosed space, the metal temperature quickly rises and reduces drastically the strain sensitivity of the brittle lacquer.
11.3.4 Strain gauge tests

This tests has not been used by DCE however I have included a summary of its uses, based on the work of R. G. Patton [37]. Strain gauges are often carried out after a brittle lacquer test to more accurately quantify the impeller stress levels. Patton used 45° rectangular rosette gauges on the backplate and shroud in order to confirm the directions of principle stresses indicated by the brittle lacquer tests. On both the backplate and shroud a series of gauge positions was chosen at various radii midway between blades. As far as possible each position on the outer surface had a corresponding position on the inside surface to enable bending stresses to be separated from direct stresses. A further series of positions were chosen round the blade profiles at points of high stress indicated by the brittle lacquer tests.

However, strain gauging of rotating parts such as impellers has several problems;

(a) The gauge leads must be attached to the impeller surface securely so that they do not fly off at speed and do not apply strain to the gauges themselves. Patton used plastic adhesive, but care had to be taken to keep this to minimum so as not to effect the stresses in the relatively thin parts of the impeller.

(b) Slip-ring units are required to overcome the problem of bringing leads from a rotating assembly to the measuring instrument.

(c) Temperature compensated gauges are sometimes required to overcome the problems associated with the impeller heating up during testing.
11.3.5 Overspeed tests

Overspeed tests are carried out on all the newly developed K-range fans. They are tested at normal running speed and at the following speed for a minimum of 8 hours;

<table>
<thead>
<tr>
<th>Impeller Frequency</th>
<th>Normal Running Speed</th>
<th>Minimum Overspeed</th>
</tr>
</thead>
<tbody>
<tr>
<td>50 Hz impeller</td>
<td>3000 rev/min</td>
<td>4200 rev/min</td>
</tr>
<tr>
<td>60 Hz impeller</td>
<td>3600 rev/min</td>
<td>4800 rev/min</td>
</tr>
</tbody>
</table>

Table 23, Overspeed test speeds

After testing the impeller is thoroughly inspected for any signs of fatigue or distortion, before the design can be approved for production. The figure below shows a comparison of a standard impeller (L.H.) and an impeller which distorted during overspeed testing (R.H.);

Fig.141, Impeller distortion caused by overspeed testing
11.4 Moment of inertia calculations

A suitable motor can often be chosen based on full load power requirements. However, it is also wise to examine the starting characteristics of the fan, in particular the flywheel effect. The flywheel effect of a fan rotor is its mass moment of inertia about the axis of rotation.

The moment of inertia of a particle is the product of its mass and the square of its distance from the axis of rotation. The moment of inertia of a group of particles is the algebraic sum of the individual moments of inertia.

\[ I = m \cdot k^2 \]

Where:
- \( I \) = Moment of inertia (kg \( m^2 \))
- \( m \) = Impeller mass (kg)
- \( k \) = Radius of gyration (m)

Lists of moments of inertia and radii of gyration for common bodies are available. Most fan elements can be considered as an assembly of one or more of these common bodies.

As an approximation the radius of gyration of an impeller is usually between 65% and 75% of the impeller tip radius [20, 18-2].

Example.

The following two pages are printouts from a spreadsheet that can be used to calculate the weight and moment of inertia of fan impellers (more detailed calculation contained in appendices 15.1).

Using the K11-50Hz impeller as an example it can be seen that the calculated moment of inertia is 0.3765 kg \( m^2 \).

Using the approximation for the radius of gyration outlined above [20, 18-2] similar values for the moment of inertia are obtained, see below;

\[ I = m \cdot k^2 \]

\[ I_{\text{Lower}} = 15.1 \times (0.65 \times d/2)^2 \]
\[ = 15.1 \times (0.65 \times 0.485/2)^2 \]
\[ = 0.375 \text{ kg m}^2 \]

\[ I_{\text{Upper}} = 15.1 \times (0.75 \times d/2)^2 \]
\[ = 15.1 \times (0.75 \times 0.485/2)^2 \]
\[ = 0.540 \text{ kg m}^2 \]
Example.

CALCULATION OF MOMENT OF INERTIA
K11-50Hz

INPUTS

<table>
<thead>
<tr>
<th>BACKPLATE OUTER DIAMETER</th>
<th>485 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>BACKPLATE INNER DIAMETER</td>
<td>60 mm</td>
</tr>
<tr>
<td>FRONT PLATE (SHROUD) INNER DIAMETER</td>
<td>290 mm</td>
</tr>
<tr>
<td>BLADE INNER WIDTH</td>
<td>140 mm</td>
</tr>
<tr>
<td>BLADE OUTER WIDTH</td>
<td>95 mm</td>
</tr>
<tr>
<td>BLADE LENGTH</td>
<td>130 mm</td>
</tr>
<tr>
<td>STIFFENING FOLD LENGTH</td>
<td>15 mm</td>
</tr>
<tr>
<td>NUMBER OF BLADES</td>
<td>13</td>
</tr>
<tr>
<td>WEIGHT PER UNIT AREA OF BACKPLATE</td>
<td>25.435 Kg/m²</td>
</tr>
<tr>
<td>WEIGHT PER UNIT AREA OF FRONT PLATE</td>
<td>20.699 Kg/m²</td>
</tr>
<tr>
<td>WEIGHT PER UNIT AREA OF BLADES</td>
<td>20.699 Kg/m²</td>
</tr>
<tr>
<td>WEIGHT OF HUB</td>
<td>2.85 Kg</td>
</tr>
<tr>
<td>K Hub</td>
<td>38.7 mm</td>
</tr>
</tbody>
</table>

NOTES:

<table>
<thead>
<tr>
<th>GAUGE</th>
<th>Moment of Inertia</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.2 mm</td>
<td>9.338 Kg/m²</td>
</tr>
<tr>
<td>1.6 mm</td>
<td>12.450 Kg/m²</td>
</tr>
<tr>
<td>2.0 mm</td>
<td>15.915 Kg/m²</td>
</tr>
<tr>
<td>2.5 mm</td>
<td>20.699 Kg/m²</td>
</tr>
<tr>
<td>3.0 mm</td>
<td>25.435 Kg/m²</td>
</tr>
</tbody>
</table>

CALCULATIONS

CALCULATION OF COMPONENTS WEIGHTS

BACKPLATE

<table>
<thead>
<tr>
<th>AREA</th>
<th>0.182 m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>WEIGHT</td>
<td>4.627 Kg</td>
</tr>
</tbody>
</table>

FRONT PLATE

<table>
<thead>
<tr>
<th>AREA</th>
<th>0.131 m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>WEIGHT</td>
<td>2.710 Kg</td>
</tr>
</tbody>
</table>
### BLADES

Area = 0.017 m²
Weight = 0.360 Kg

### WELDS

Assuming:
3mm fillet each side of blade, top & bottom.
Density of steel = 7750 Kg/m³

<table>
<thead>
<tr>
<th>Volume of Bottom Welds</th>
<th>Weight of Bottom Welds</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.17E-06 m³</td>
<td>0.0091 Kg</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Volume of Top Welds</th>
<th>Weight of Top Welds</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.2E-06 m³</td>
<td>0.0096 Kg</td>
</tr>
</tbody>
</table>

Total weight of welds = 0.0187 Kg

### IMPELLER

Total weight of impeller = 15.105 Kg

### RADIUS OF GYRATION CALCULATION FOR COMPONENT PART

\[
K_{\text{Backplate}} = 136.3 \text{ mm} \\
K_{\text{Frontplate}} = 193.8 \text{ mm}
\]

Assuming \( K_{\text{Blades}} \) is approximately equal to \( K_{\text{Frontplate}} \)

\[
K_{\text{Blades}} = 193.8 \text{ mm}
\]

### MOMENT OF INERTIA CALCULATIONS

<table>
<thead>
<tr>
<th>( I ) of Backplate</th>
<th>0.0859 Kg.m²</th>
</tr>
</thead>
<tbody>
<tr>
<td>( I ) of Frontplate</td>
<td>0.1017 Kg.m²</td>
</tr>
<tr>
<td>( I ) of Blades</td>
<td>0.1846 Kg.m²</td>
</tr>
<tr>
<td>( I ) of Hub</td>
<td>0.0043 Kg.m²</td>
</tr>
<tr>
<td>( I ) Total</td>
<td>0.3765 Kg.m²</td>
</tr>
</tbody>
</table>

Weight of Impeller = 15.1 Kg

Moment of Inertia = 0.3765 Kg.m²
11.5 Case study of Impeller failures

11.5.1 Summary
This report outlines the investigation undertaken to assess the reasons for two F11-60Hz fan failures in America and to recommend ways of preventing them re-occurring.

11.5.2 Introduction
The first fan had been in operation for about two months before failure occurred. After the first failure a replacement impeller was supplied which failed after only one week. The complete fan assembly was then replaced as it was thought the motor may have been damaged during the first failure which had then caused the second impeller to fail so quickly. However, the motor was checked by the manufacturers and was found to be within factory tolerance. An urgent investigation was therefore started to assess the cause of failure and to prevent further fan failures. The figure below shows photographs of one of the failed impellers;

Fig.142, Photographs of failed impeller
11.5.3 Investigation

The following organisations and companies were asked to inspect the failed impellers which had been returned to DCE:

(a) British steel.
(b) Loughborough University of Technology.
(c) PERA (Production Engineering Research Association).

They all agreed that the failures were due to a fatigue crack which started at the inside edge of the impeller shroud. It was also noted that the crack started at the edge of a blade as shown below;

![Diagram of impeller fatigue crack](image)

*Fig.143, Impeller fatigue crack*
Impeller Stress Calculations

Assuming that the impeller shroud is a disc the following calculation can be applied.

The maximum stress in a rotating disc is the tangential stress and is maximum at the inner radius, and can be calculated from the following equation [23];

$$\sigma_t = \frac{n^2 \cdot (3.3 \, r_2^2 + 0.7 \, r_1^2)}{497500}$$

Where:

- $\sigma_t$ = Tangential stress (psi)
- $n$ = Impeller speed (rev/min)
- $r_2$ = Outer radius (Inch)
- $r_1$ = Inner radius (Inch)

$$\sigma_t = \frac{3470^2 \cdot (3.3 \times (7.97)^2 + 0.7 \times (5.08)^2)}{497500}$$

$$\sigma_t = 5510 \text{ psi}$$

$$\sigma_t = 5510 \times (6.895 \times 10^{-3}) \text{ MN/m}^2$$

$$\sigma_t = 38 \text{ MN/m}^2$$

Strength of CR1 steel = 140 MN/m²
(BS1449 : Part 1) [8]

Safety Factor = 140/38
= 4

Since the safety factor of 4 is reasonably high, the above calculation supports the theory that the failure is due to a fatigue problem.
Impeller Resonance Tests

Impeller resonance tests were carried out at Loughborough University of Technology using the FFT (Fast Fourier Transform) equipment. As the failure was associated with the impeller shroud a mesh was drawn on the face using 12 angular coordinate points and 3 radial coordinates as shown below:

![Fig.144, Typical mesh for FFT analysis of impeller shroud](image)

An accelerometer was attached to the surface at location 10/2 and an instrumented hammer was used to apply impulse loading at all angular locations and half angular locations, on the radial coordinate 2 (25 sample points). The frequency spectrum for both the excitation force and the accelerometer response was obtained over a 2 KHz range. By manipulating the spectra a transfer function giving a measure of output response per unit force input over the whole frequency range was obtained.
Further manipulation then yielded the real and imaginary components of the transfer response so that modal parameters (natural frequencies and mode shapes) could be deduced. This technique identified clearly that the face plate was able to distort at specific frequencies in the fundamental disc modes. i.e. about one diameter as a 1D mode and about 2 and 3 diameters in the 2D and 3D mode, as shown below;

![Diagram of disc modes](image)

**Fig.145, Fundamental disc modes of vibration**

It was concluded that the 3D resonance mode was very near to the normal running speed of the impeller and was likely to cause severe vibration problems.

(Section 11.1.2 explains these tests in more detail)
Fan Assembly Tracking Tests

These tests were carried out using the Vibroport30 universal vibration measuring instrument. The fan was freely suspended and vibration measurements taken at the locations shown in the following diagram, whilst the fan was slowed from just above the normal running speed.

Fig.146, Fan tracking test measuring locations
Plots of the vibration amplitude against speed of rotation can be plotted for the two measurement locations. It can be seen from the following plots that severe vibrations occur at normal running speed;

Fig. 147, Tracking plots for F11 fan assembly

(Section 11.1.3 explains these tests in more detail)
Unit Resonance Tests

Failures have only occurred on a particular type and size of dust collector, this being the DU30. It was decided to investigate how the fan was fitted within this particular unit.

DU unit have a header box arrangement to which the eye of the fan is attached, see below;

![DU header box arrangement](image)

**Fig.148, DU header box arrangement**

It was found that the length of the DU30 header box was very nearly the same as the blade passage wavelength of the F11-60Hz fan, see calculation below;

\[
\text{Blade passage frequency} = 58 \times \text{No. Blades} \\
= 58 \times 12 \\
= 696 \text{ Hz}
\]

\[
\text{Blade passage wavelength} = \frac{\text{Speed of Sound}}{\text{Frequency}} \\
= \frac{330}{696} \\
= 0.474 \text{ m} \\
= 474 \text{ mm}
\]
Since the DU30 header box is 496mm long it is therefore very likely that a standing wave is being developed in the DU30 header box when an F11-60Hz fan is fitted. If a standing wave is developed in the header box the fan will be subjected to very large pressure surges which will compound the impeller resonance problems.

Unit resonance tests were therefore carried out on the header box to check this standing wave theory. The following vibration plot for the side of the header box shows that there is a large vibration peak very near to the running speed.

![Vibration plot for side of standard DU header box](image)

This vibration plot confirms the existence of a standing wave in the DU header box.

*(Section 11.1.4 explains these tests in more detail)*
11.5.4 Recommendations

Impeller Modification

The impeller shroud material gauge was increased from 2.0mm to 2.5mm. This modification has moved the vibration peaks to higher frequencies, which are above the running speed, and has therefore reduced the impeller vibration problem, see below;
Finite element stress analysis of impeller

A finite element stress analysis was carried out at Loughborough University of Technology to determine the effect of this change in shroud thickness. A quadrant of the impeller was modelled using 8-noded quadrilateral and 6-noded triangular thin facet shell elements, giving a problem with 4000 degrees of freedom. The attachment of the impeller blades to the shroud was modelled by making common the degrees of freedom on adjacent nodes of the blade and shroud at the position of the rivets attaching the blades. The nodal loading due to the centrifugal forces corresponding to the 60Hz rotational speed was produced automatically by the computer package. The following photograph shows the displaced shape of the impeller after loading;

Fig.151, Finite stress analysis displaced shape of impeller
The following photograph shows the shaded contours of the impeller maximum principle stresses:

Fig.152, Finite stress analysis impeller principle stress contours

Results were obtained for an impeller with a 2.0mm shroud and a 2.5mm shroud. It was found that the shroud stresses were slightly lower for the 2.5mm shroud. The blade stresses were however slightly higher for the 2.5mm shroud due to the extra blade bending stress associated with an increase in shroud mass. Although the blade stresses were slightly higher they were below the yield strength of mild steel and were unlikely to cause failure. Since the failures were due to the shroud failing it was recommended to precede with the recommended increase in shroud thickness from 2.0mm to 2.5mm gauge.

(Section 11.3.2 explains this technique in more detail)
DU header box modification

A metal dividing plate was fastened to the inside of the box to split the header box in half. It can be seen from the following vibration plot that this modification prevents the formation of a standing wave in the header box.

**Fig.153**, Vibration plot for side of modified header box

11.5.5 Addendum

These recommendations were implemented in November 1989 and there have been no further impeller failures reported.
11.6 Summary & Conclusions

As well as fan performance, noise, and power measurements the following must also be considered to ensure that a safe reliable design is achieved:

(a) Vibrations tests
- Fan assembly vibration tests
- Impeller FFT analysis
- Fan assembly tracking tests
- Unit resonance tests

(b) Balancing

(c) Stress analysis
- Stress calculations
- Finite stress analysis
- Brittle lacquer tests
- Strain gauge tests
- Overspeed tests

(d) Moment of inertia calcs.

If the above are not carried fan failure is a possibility as outlined during the case study of the failed F11-60Hz impeller.
12.0 CONCLUSIONS & ACHIEVEMENTS

By following the recommended design philosophy outlined below a suitable fan design to meet typical industrial requirements can be quickly and efficiently obtained;

(1) Determine fan requirements.
   (a) Applicable noise legislation (chapter 1 & 2).
   (b) Fan performance required (chapter 2).
   (c) Power requirements, ie. motor size (chapter 2).
   (d) Fan type required, ie. specific speed (chapter 3).

(2) Predict fan dimensions required to satisfy fan requirements using;
   (a) Performance coefficients (chapter 3).
   (b) Fan Performance prediction
      (i) Fan laws (chapter 7).
      (ii) Slip & loss analysis (chapter 7).
   (c) Fan noise level prediction (chapter 7).

(3) Check mechanical integrity of proposed prototype fan
   (a) Stress calculations (chapter 11).
   (b) Moment of inertia calcs. (chapter 11).

(4) Manufacture & test prototype fan to confirm mechanical integrity
   (a) Balance impeller (chapter 11).
   (b) Vibration test (chapter 11).
   (c) Overspeed test impeller (chapter 11).

(5) Test fan to check satisfies requirements
   (a) Performance test (chapter 8).
   (b) Noise test (chapter 8).

(6) Based on test results fine tune fan design to meet requirements
   (a) Fan design concepts (chapter 6).
   (b) Experimental results (chapter 9).

Note.

Stages (3) to (6) may need to be repeated to obtain a suitable design.
By using the recommended design philosophy it has been possible to develop a new range of fans (K range) to improve the competitiveness of DCE products. The table below shows a comparison with other DCE fan ranges to summarize the benefits achieved.

<table>
<thead>
<tr>
<th></th>
<th>PRESSURE AT TYPICAL AIR VOLUME</th>
<th>MOTOR SIZE</th>
<th>NOISE LEVEL</th>
</tr>
</thead>
<tbody>
<tr>
<td>F3/G3 K3</td>
<td>195 mmWg 175 mmWg</td>
<td>2.2 kW 1.5 kW</td>
<td>92 dB(A) 86 dB(A)</td>
</tr>
<tr>
<td>F5/G5 K5</td>
<td>200 mmWg 200 mmWg</td>
<td>3.0 kW 2.2 kW</td>
<td>95 dB(A) 88 dB(A)</td>
</tr>
<tr>
<td>F11/G11 F12/G12 K11</td>
<td>230 mmWg 300 mmWg 280 mmWg</td>
<td>7.5 kW 11 kW 7.5 kW</td>
<td>99 dB(A) 101 dB(A) 97 dB(A)</td>
</tr>
<tr>
<td>K15</td>
<td>200 mmWg</td>
<td>11 kW</td>
<td>99 dB(A)</td>
</tr>
<tr>
<td>CD15 K18</td>
<td>325 mmWg 315 mmWg</td>
<td>15 kW 15 kW</td>
<td>101 dB(A) 101 dB(A)</td>
</tr>
<tr>
<td>CD18 K21</td>
<td>345 mmWg 375 mmWg</td>
<td>18.5 kW 18.5 kW</td>
<td>102 dB(A) 102 dB(A)</td>
</tr>
</tbody>
</table>

Table 24, Benefits of K range

The achievements of this project can best be determined by assessing the extent to which the project objectives (5.1.1) have been satisfied.

(1) Maintain or reduce assembly external dimensions

The external dimensions of the fans have been maintained and for some of them the dimensions have actually been reduced since it has been possible to use smaller motors.
(2) Reduce fan noise levels

It can be seen from the previous table that noise levels have been significantly reduced when compared with the F and G ranges.

\[
\begin{align*}
\text{F3/G3} & \rightarrow \text{K3} & \rightarrow 6 \text{ dB(A) reduction} \\
\text{F5/G5} & \rightarrow \text{K5} & \rightarrow 7 \text{ dB(A) reduction} \\
\text{F11/G11} & \rightarrow \text{K11} & \rightarrow 2 \text{ dB(A) reduction} \\
\text{F12/G12} & \rightarrow \text{K11} & \rightarrow 4 \text{ dB(A) reduction}
\end{align*}
\]

The large K18 and K21 have the same noise levels as the bought out CD15 and CD18 fans.

Note.
Further noise reductions would be achieved if the perforated scroll design was used as detailed in section 9.2.7.

(3) Maintain performance (over typical operating range)

It can be seen from the previous table that the performance of the K range fans is approximately the same as the fans they replace.

\[
\begin{align*}
\text{F3/G3} & \rightarrow \text{K3} & \rightarrow 20 \text{ mmWg less} & \text{ (at 1500 m}^3/\text{hr)} \\
\text{F5/G5} & \rightarrow \text{K5} & \rightarrow \text{Same} & \text{ (at 2000 m}^3/\text{hr)} \\
\text{F11/G11} & \rightarrow \text{K11} & \rightarrow 50 \text{ mmWg More} & \text{ (at 5000 m}^3/\text{hr)} \\
\text{F12/G12} & \rightarrow \text{K11} & \rightarrow 20 \text{ mmWg less} & \text{ (at 5000 m}^3/\text{hr)} \\
\text{CD15} & \rightarrow \text{K18} & \rightarrow 10 \text{ mmWg More} & \text{ (at 10000 m}^3/\text{hr)} \\
\text{CD18} & \rightarrow \text{K21} & \rightarrow 30 \text{ mmWg less} & \text{ (at 10000 m}^3/\text{hr)}
\end{align*}
\]
The CD15 and CD18 are unstable at low volume flow rates, which would lead to noise and vibration problems if the fans were operated on these areas of the performance curves. However, the K18 and K21 are stable at any part of their performance curves which is a significant benefit over the CD15 and CD18 fans see below;

Fig.154, Comparison of unstable bought out fans with stable K fans
(4) Minimise costs

As detailed in section 10.3 cost comparisons can be made using the following two criteria:

(a) New K fan range compared with old F & G fan range.
(b) New K fan range compared with bought out fans.

It can be seen below that either criteria offer significant cost benefits, especially when compared with bought out fans.

(a) New K range compared with old F & G fan range

Fan cost saving per year due to development of K3, K5 & K11 fans = £17,468
Annual controller cost saving due to the K11 implementation (Direct on line starting rather then Star-delta) = £39,725

Total cost saving per year due to development of the above fans = £57,193

| Total annual cost saving | Approximately £57,000 |

(b) New K range compared with bought out fans

Fan cost saving per year due to development of K3, K5, K11, K15, K18, K21 = £1,411,832
Annual controller cost saving due to the K11 implementation (Direct on line starting rather then Star-delta) = £39,725

Total cost saving per year due to development of the above fans = £1,451,557

| Total annual cost saving | Approximately £1.45 Million |
(5) Reduce power consumption

Due to the increased efficiency of the K range fans it has been possible to reduce the maximum power requirements of the fans, and therefore reduce running costs, see below;

<table>
<thead>
<tr>
<th></th>
<th>Power Requirement</th>
</tr>
</thead>
<tbody>
<tr>
<td>F3/G3</td>
<td>2.2 kW</td>
</tr>
<tr>
<td>K3</td>
<td>1.5 kW</td>
</tr>
<tr>
<td>F5/G5</td>
<td>3.0 kW</td>
</tr>
<tr>
<td>K5</td>
<td>2.2 kW</td>
</tr>
<tr>
<td>F11/G11</td>
<td>7.5 kW</td>
</tr>
<tr>
<td>F12/G12</td>
<td>11 kW</td>
</tr>
<tr>
<td>K11</td>
<td>7.5 kW</td>
</tr>
</tbody>
</table>

Table 25, Benefits of K range (power requirements)

In most cases it has also been possible to reduce the motor sizes with associated cost savings.

(6) Standardise on parts to reduce stock levels

K range fan (shown fitted into a Unimaster unit);

Fig.155, K range fan (shown fitted in a Unimaster unit)
As detailed in section 10.1 the K range fans offer the following advantages over the F and G range:

(1) Reduced stock levels
(2) Fans become unit specific only at assembly, therefore fans can be assembled to meet urgent requirements. (ie. Parts for a Unimaster fan can be used to manufacture a Dalamatic fan if urgently required).
(3) Fans can be easily adapted to suite different unit if required.

(7) Completely interchangeable with existing fans.

The dimensions of the K fan range are comparable with the F and G fan range (same fan case in most instances). This allows complete interchangeability of fans and would allow retro fitting to existing units if required.

(8) Utilize current production practises.

DCE's factory is mainly equipped to cut, form and join sheet metal. The F & G impeller range are made by fastening the sheet metal blades to the front and back-plate with huckbolts. Most K range fans are assembled in the same way as the F & G fan range thereby utilising current production practises, (the larger K fans are welded which is the industrial norm for impellers of this size).
13.0 FUTURE WORK

The following areas have been identified that may warrant further investigation;

(a) The slip and loss analysis detailed in section 7.1.2 could be investigated further, in particular selection of suitable loss factors for various fan ranges. By increasing the accuracy of this analysis the number of prototypes required to obtain acceptable designs would be significantly reduced. This would reduce development time and would be a more cost effective method.

(b) Computer models for predicting fan performance such as that produced by N.E.L. could be investigated further, (section 7.1.3). This would also reduce development time and costs as (a) above.

(c) Some works has been carried out using smoke to determine the air flow pattern through fans and identify areas of turbulence, see below;

Fig. 156, Determination of air flow through fans using smoke
A similar technique has also been used to assess the air flow in a typical dust collector fan chamber. This helps to show the air flow into the fan inlet and can be used to determine flow restrictions and areas of turbulence in the fan chamber, see below;

Fig.157, Determination of air flow in fan chamber using smoke

Further work using this technique should allow further improvements of the K-range fans and the fan chamber design of the DCE dust collector range.
(d) The inlet and outlet grilles of a fan restrict the air flow through the fan. It has been found that the design of these grilles can significantly effect the fans efficiency and noise. It is apparent that the open area of the grilles is important, however, the effect of other parameters such as the shape of the holes and the grille material have not been determined. Further investigation in this area would be useful as any improvements made could be easily incorporated into most fans.

(e) DCE could achieve further noise reductions by using the perforated scroll technique to products other than the Siloair range (section 9.2.7).

(f) The noise level prediction methods detailed in section 7.2 could be investigated further to try and find an existing method or develop new more accurate techniques.

(g) The use of computer packages to predict air flow in systems is becoming more wide spread. This technique could be used to determine air flow pattern in the dust collector as well as in the fan. Flow Simulation Ltd. Sheffield distribute a fluid dynamics package called "Fluent" which appears to be suitable for modelling such systems. This and other fluid dynamic computer packages could be investigated further.

(h) The acoustic diffuser designs used by DCE (see fig.125 for typical unit) are merely foam lined boxes with an integral baffle. The design of these could be improved so that they help to diffuse the air leaving the fan outlet before discharge to atmosphere. This design improvement will not only reduce noise levels but should also improve the fan efficiency by improving the fan outlet conditions.
14.0 REFERENCES

[1] Publisher: Air Movement and Control Association inc.
Title: AMCA Standards Handbook
Publication 99-86
AMCA Standard 99-0401-86 (Adopted 2/12/86)
Classification for spark resistant construction

Title: Noise reduction in centrifugal fans by means of an acoustically lined casing.

[3] Author: Beranek, Kamperman & Allen
Title: Noise of centrifugal fans.

Fans for general purposes

Fans for general purposes
Part 2. Methods of noise testing.
Fans for general purposes
Part 5. Guide for mechanical and electrical safety.

Fans for general purposes

Steel plate, sheet & strip.
Section 1.1 : General Specification

Method for Rating industrial noise effecting mixed residential and industrial areas

Sound power levels of noise sources.
Part 0. Guide for the use of basic standards and for the preparation of noise codes.

Sound power levels of noise sources.
Part 1. Precision methods for determination of sound power levels for broad-band sources in reverberation rooms.
Sound power levels of noise sources.  
Part 2. Precision methods for determination of sound power levels for discrete-frequency and narrow-band sources in reverberation rooms.

Sound power levels of noise sources.  
Part 3. Engineering methods for determination of sound power levels for sources in special reverberation test rooms.

Sound power levels of noise sources.  
Part 4. Engineering methods for determination of sound power levels for sources in free-field conditions over a reflecting plane.

Sound power levels of noise sources.  
Part 5. Precision methods for determination of sound power levels for sources in anechoic and semi-anechoic rooms.

Sound power levels of noise sources.  
Part 6. Survey method for determination of sound power levels of noise sources.
Mechanical vibration in rotating machinery
Part 1. Basis for specifying evaluation standards for rotating machines with operating speeds from 10 to 200 revolutions per second.

Balance quality requirements of rigid rotors.
Part 1. Methods of determination of permissible residual unbalance

Test code for the measurement of airborne noise emitted by rotating electrical machinery.
Part 1. Engineering method for free-field conditions over a reflecting plane.

Buffalo Forge Company, New York
Title: Fan Engineering (Eight Edition)
Editor: Robert Jorgensen

DeeProse, W M and Brooks J M
Title: Effect of scale on fan noise generation of backward curved centrifugal fans.
[22] Author: Erskine & Brunt
Title: Prediction and control of noise in fan installations

[23] Publisher: The James F. Lincoln arc welding foundation
Title: Design of weldments (5.1-5 equation 10)
Author: Omer W. Blodgett

[24] Author: Eck, Bruno (Dr-Ing.)
Title: Design and operation of centrifugal, axial-flow, and cross flow fans. (First edition)

[25] Author: Embleton, T.F.W.

[26] Author: Hübner, G
Title: Decreasing the noise level of centrifugal ventilators.
Translated into English: Joint publication research service 31116.
(TT-65-31614. 1965)

Title: Some new methods of fan noise reduction.
Proceedings of the 7th International congress on acoustics.
Budapest, 1971, paper 19N16

[29] Author: Koopmann, G.H. and Niese, W.
Title: An application of resonators in reducing centrifugal fan noise.

[30] Author: Krishnappa, G
Title: Effect of moduled blade spacing on centrifugal fan noise.

[31] Author: Lawrence, H R
Title: Near field sound pressure measurement around an air-cooled heat exchanger.
[32] Author: Leidel, W

[33] Author: Lyons, L.A. and Platter, S.
Title: Effect of cut-off configuration on pure tones generated by small centrifugal blowers.
Journal of the acoustical society of america.
(vol.35, 1963, pp 1455-1456.)

[34] Title: Machinery Directive
89/392/EEC
As amended by Directive 91/368/EEC & 93/44/EEC

[35] Author: Neise, W and Koopmann, G.H.
Title: Reduction of centrifugal fan noise by use of resonators.

[36] Author: William C. Osborne
(Polytechnic of the South Bank, London, England)
Title: FANS. 2nd Edition (in SI/Metric Units)
Publisher: Pergamon Press
[37] Author: Patton, R.G. BSc (Eng) PhD.
Title: Stress in centrifugal fan impellers.
Conference on "Fan technology and practice (18-19th April 1972)."

[38] Author: Petrov, Yu. and Khoroshev, G.A.
Title: Improving the noise level of centrifugal fans.

Title: Reduction of the noise level in centrifugal fans by means of transition meshes.
(Translated into English: NAVSTIC-Translation 3322, 1972.)

[40] Author: Ploner, B. and Hertz, F.
Title: New design measures to reduce siren tones caused by centrifugal fans in rotating machines.

[41] Author: Smith, W.A. O'Malley, J.K. and Phelps, A.H.
Title: Reducing blade passage noise in centrifugal fans.
[42] Author: Sharland, Ian
Title: Woods practical guide to noise control (Fifth Edition)
Publisher: Woods of Colchester Limited

[43] Author: Shipway, J C
Title: Estimation of headflow curves for centrifugal fans.
British Hydromechanics Research Association, Hovercraft report 5.

[44] Author: Suzuki, S and Ugai, Y
Title: Study on high speed airfoil fans.

[45] Author: Carlos C. Chardon and Ira J. Roy
Title: Packing the maximum fan in the minimum space

[46] Author: Standard & Pochin Ltd.
Title: Fan application guide.
15.0 APPENDICES.

15.1 Moment of inertia calculations for K11-50Hz

**BACKPLATE**

\[
\text{AREA} = \left( \frac{\pi \times \text{OUTER DIAMETER}^2}{4} \right) - \left( \frac{\pi \times \text{INNER DIAMETER}^2}{4} \right) \\
= \left( \frac{\pi \times 0.485^2}{4} \right) - \left( \frac{\pi \times 0.06^2}{4} \right) \\
= 0.182 \text{ m}^2
\]

\[
\text{WEIGHT} = \text{AREA} \times \text{WEIGHT PER UNIT AREA} \\
= 0.182 \times 25.435 \\
= 4.627 \text{ kg}
\]

**FRONTPLATE**

\[
\text{AREA} = 3.146 \times (R + r) \\
= 3.146 \times (D + d)
\]

Where: \[S = \sqrt{(R - r)^2 + h^2} = \sqrt{0.2425 - 0.145^2 + 0.045^2} = 0.1074\]

\[
\text{AREA} = 3.146 \times 0.1074 \times (0.485 + 0.29) \\
= 0.131 \text{ m}^2
\]

\[
\text{WEIGHT} = \text{AREA} \times \text{WEIGHT PER UNIT AREA} \\
= 0.131 \times 20.679 \\
= 2.710 \text{ kg}
\]
**Blade**

**Total Area** = Area 1 + Area 2 + Area 3

\[
= (130 \times 95) + \left( \frac{1}{2} \times 130 \times 45 \right) + (15 \times 140)
\]

\[
= 17375 \text{ mm}^2
\]

\[
= 0.017 \text{ m}^2
\]

**Weight** = Area \times Weight per unit area

\[
= 0.017 \times 20.699
\]

\[
= 0.360 \text{ kg}
\]
WELD

Assuming 3mm fillet weld each side of blade

Cross-sectional area of weld = \( \frac{1}{2} \times 0.003 \times 0.003 \)
\[ = 4.5 \times 10^{-6} \text{ m}^2 \]

Volume of bottom weld = \( 0.13 \times 4.5 \times 10^{-6} \times 2 \)
\[ = 1.17 \times 10^{-6} \text{ m}^3 \]

Weight of bottom weld = \( 1.17 \times 10^{-6} \times 7750 \)
\[ = 0.0091 \text{ kg} \]

Length of top weld = \( \sqrt{130^2 + 45^2} \)
\[ = 138 \text{ mm} \]

Volume of top weld = \( 0.138 \times 4.5 \times 10^{-6} \times 2 \)
\[ = 1.2 \times 10^{-6} \text{ m}^3 \]

Weight of top weld = \( 1.2 \times 10^{-6} \times 7750 \)
\[ = 0.0096 \text{ kg} \]

Total weld weight = \( 0.0187 \text{ kg} \)

Hub

Weight = 2.85 kg

IMPELLER

Total weight = Backplate + Frontplate + 13 x Blade + Weld + Hub
\[ = 4.627 + 2.710 + (13 \times 0.36) + (13 \times 0.0187) + 2.85 \]
\[ = 15.1 \text{ kg} \]
Radius of Gyration for Component Parts

**Backplate**

\[ K_{\text{backplate}} = \frac{R + \Gamma}{2} \]
\[ = \frac{242.5 + 30}{2} \]
\[ = 136 \text{ mm} \]

**Front Plate**

\[ K_{\text{frontplate}} = \frac{R + \Gamma}{2} \]
\[ = \frac{242.5 + 145}{2} \]
\[ = 194 \text{ mm} \]

**Blades**

\[ K_{\text{blades}} = K_{\text{frontplate}} \]
\[ = 194 \text{ mm} \]

**Hub**

\[ K_{\text{hub}} = 36.7 \text{ mm} \]
**Moment of Inertia**

\[ I = m \cdot h^2 \]

**Backplate**

\[ I_{\text{backplate}} = 4.627 \times 0.136^2 = 0.086 \text{ kg m}^2 \]

**Frontplate**

\[ I_{\text{frontplate}} = 2.710 \times 0.194^2 = 0.102 \text{ kg m}^2 \]

**Blades (including weld)**

\[ I_{\text{blades}} = 13 \times 0.379 \times 0.194^2 = 0.1846 \text{ kg m}^2 \]

**Hub**

\[ I_{\text{hub}} = 2.85 \times 0.0387^2 = 0.0043 \text{ kg m}^2 \]

\[ I_{\text{total}} = I_{\text{backplate}} + I_{\text{frontplate}} + I_{\text{blades}} + I_{\text{hub}} \]

\[ = 0.086 + 0.102 + 0.1846 + 0.0043 \]

\[ = 0.377 \text{ kg m}^2 \]

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15.2 Slip & loss calculations for K15-50Hz

Air Density, \( \rho \) = 1.2 kg/m\(^3\)
Impeller Speed, \( n \) = 3000 rev/min
Blade Width (outer), \( b_2 \) = 100mm = 0.1m
Blade Width (inner), \( b_1 \) = 160mm = 0.16m
Outer Diameter, \( d_2 \) = 510mm = 0.51m
Blade Angle (outer), \( \beta_2 \) = 64°
Blade Angle (inner), \( \beta_1 \) = 32°
Blade Number, \( Z \) = 13
Discharge Coefficient, \( C_d \) = 0.6
Impeller Inlet Diameter, \( d_1 \) = 260mm = 0.26m
Inlet Clearance, \( \alpha \) = 2mm = 0.002m
Inlet Cone Eye Diameter, \( d_e \) = 230mm = 0.23m
Fan Case Width = 350mm = 0.35m
Fan Case Length = 350mm = 0.35m

Inlet Loss Coefficient \( K(\text{in}) \) = 0.9
Impeller Loss Coefficient \( K(\text{imp}) \) = 0.25
Outlet Loss Coefficient \( K(\text{out}) \) = 0.4

Volume Flow Rate, \( Q \) = 9000 m\(^3\)/hr
= 2.5 m\(^3\)/s

Impeller Tip Speed, \( V_{tu} \) = \( \frac{\pi \cdot \omega}{5} \)
= \( \frac{\pi \cdot 2\pi \cdot n}{5} \)
= \( \pi \cdot d_2 \cdot n \)
= \( \pi \times 0.51 \times 3000/60 \)
= 80.11 m/s
Fan Case Outlet Area, \( A_3 = \text{Length} \times \text{Width} \)
\[ = 0.35 \times 0.35 \]
\[ = 0.1225 \text{ m}^2 \]

**Inlet**

\[ V_{u1} = \frac{\dot{V}_1}{\eta \cdot d_i \cdot b_i} \]
\[ = \frac{2.5}{\pi \times 0.26 \times 0.16} \]

\[ V_{u1} = 19.13 \text{ m/s} \]

\[ \tan \beta_i = \frac{V_{m1}}{\infty} \]
\[ = \frac{19.13}{\tan 32'} \]
\[ \infty = \frac{19.13}{\tan 32'} \]
\[ \infty = 30.61 \text{ m/s} \]

\[ V_{m1} = U_1 - \infty \]
\[ = 40.84 - 30.61 \]
\[ = 10.22 \text{ m/s} \]
\[ \sin \theta_i = \frac{V_{m_i}}{V_{r_i}} = \frac{19.13}{V_{r_i}} \]

\[ \therefore V_{r_i} = \frac{19.13}{\sin 22^\circ} \]

\[ V_{r_i} = 36.10 \text{ m/s} \]

\[ \tan \beta_s = \frac{V_{w_i}}{V_{m_i}} \]

\[ = \frac{10.22}{19.13} \]

\[ \therefore \beta_s = 28^\circ \]
\[
V_{m_2} = \frac{Q}{\pi \cdot d_1 \cdot b_1}
\]
\[
= \frac{2.5}{\pi \times 0.51 \times 0.1}
\]

\[V_{m_2} = 15.6 \text{ m/s}\]

\[\sin \theta_2 = \frac{V_{m_2}}{V_{r_2}}\]

\[V_{r_2} = \frac{V_{m_2}}{\sin \theta_2}\]

\[V_{r_2} = \frac{15.6}{\sin 64^\circ}\]

\[V_{r_2} = 17.36 \text{ m/s}\]
Euler Pressure

\[
\text{Euler Pressure} = \left[ 8 \cdot (V_{u_3})^2 \right] - \left[ \frac{8 \cdot (V_{u_3})}{\pi \cdot b_2 \cdot d_2} \cdot \cot B_2 \cdot Q \right]
\]

\[
= \left[ 1.2 \times (80-11)^2 \right] - \left[ \frac{1.2 \times 80.1}{\pi \times 0.1 \times 0.51} \times \frac{1}{\tan 64^\circ} \times 2.5 \right]
\]

\[
= 7701 - 731
\]

\[
= \underline{6970 \ \text{Pa}}
\]

Interblade Circulation Loss (Blade Loss)

\[
\text{Interblade Circulation Loss} = 8 \cdot (V_{u_2})^2 \cdot \frac{V_2}{2} \cdot \sin B_2
\]

\[
= 1.2 \times (80-11)^2 \times \frac{V_2}{2} \times \sin 64^\circ
\]

\[
= \underline{1673 \ \text{Pa}}
\]

Impeller Loss

\[
\text{Impeller Loss} = (K_{imp}) \cdot (\lambda_2) \cdot (\delta) \cdot (Q^2) \cdot \left[ \frac{1}{\pi \cdot d_1 \cdot b_1 \cdot \sin B_1} - \frac{1}{\pi \cdot d_2 \cdot b_2 \cdot \sin B_2} \right]^2
\]

\[
= (0.25) \cdot (\lambda_2) \cdot (1.2) \cdot (2.5^2) \left[ \frac{1}{\pi \times 0.26 \times 0.16 \times \sin 32^\circ} - \frac{1}{\pi \times 0.51 \times 0.10 \times \sin 64^\circ} \right]
\]

\[
= 0.9375 \left[ \frac{1}{0.0873} - \frac{1}{0.1440} \right]^2
\]

\[
= 0.9375 \left[ 14.44 - 6.94 \right]^2
\]

\[
= 0.9375 \left[ 7.495 \right]^2
\]

\[
= 0.9375 \left[ 56.18 \right]
\]

\[
= \underline{53 \ \text{Pa}}
\]

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IMPELLER LOSS (CHECK)

IMPELLER LOSS = \((K_{\text{imp}}) \cdot (X) \cdot (R) \cdot (V_{\text{in}} - V_{\text{out}})^2\)

= \((0.25) \cdot (X) \cdot (1.2) \cdot (36.10 - 17.36)^2\)

= \((0.15) \cdot (351)\)

= 53 \text{ Pa}

OUTLET LOSS

OUTLET LOSS = \(K_{\text{out}} \cdot \frac{8}{\pi} \cdot \left[ \sqrt{\frac{(V_{\text{in}})^2 + (V_{\text{out}})^2}{\sin 64}} - \left(\frac{2 \cdot V_{\text{in}} \cdot V_{\text{out}}}{\tan 64}\right) - \left(\frac{Q}{A_3}\right)^2 \right] \)

= 0.4 \times \frac{1.2}{2} \cdot \left[ \sqrt{\left(80.11\right)^2 + \left(15.6\right)^2} - \left(\frac{2 \times 80.11 \times 15.6}{\tan 64}\right) - \left(\frac{2.5}{0.1225}\right)^2 \right]

= 0.24 \left[ \sqrt{6418 + 301 - 1219} - 20.4 \right]^2

= 0.24 \left[ 74.16 - 20.4 \right]^2

= 0.24 \left[ 53.8 \right]^2

= 673 \text{ Pa}

INLET LOSS

INLET LOSS (1) = \(K \cdot (m) \times \frac{8}{\pi} \times \left(\frac{4 \cdot Q}{\mu \cdot d^2}\right)^2\)

= 0.9 \times \frac{1.2}{2} \times \left(\frac{4 \times 2.5}{\pi \times (0.23^2)}\right)^2

= 0.54 \times (60.17)^2

= 0.54 \times 3621

= 1955 \text{ Pa}

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\[ \text{Inlet Loss (2)} = K(m) \times \frac{5}{4} \left( \frac{4 - Q}{A \cdot d_t} - V_m^2 + (V_u - V_m \tan B_5) \right)^2 \]

\[ = 0.9 \times 1.2 \times \left( \frac{4 \times (25)}{A \times (0.25)} - 19.13^2 + (40.84 - (15.13 \times \tan 45))^2 \right)^2 \]

\[ = 0.54 \times (47.09 - \sqrt{366 + 47})^2 \]

\[ = 0.54 \times (47.09 - 28.74)^2 \]

\[ = 178 \text{ Pa} \]

**Total Inlet Loss** = Inlet Loss (1) + Inlet Loss (2)

\[ = 1955 + 178 \]

\[ = 2133 \text{ Pa} \]

**Internal Volumetric Leakage**

\[ \text{Leakage Volume, } Q_L = C_d \times A \times \sqrt{2 \times \left( \frac{P_m}{\gamma} \right)} \]

\[ = 0.6 \times (1 \times 0.26 \times 0.002) \times \sqrt{2 \times \left( \frac{2418}{1.2} \right)} \]

\[ = (9.8 \times 10^{-4}) \times (13.5) \]

\[ = 0.0622 \text{ m}^3/\text{s} \]

**Summary of Losses**

<table>
<thead>
<tr>
<th>Volume</th>
<th>Euler Pressure</th>
<th>Blade Loss</th>
<th>Impeller Loss</th>
<th>Outlet Loss</th>
<th>Inlet Loss</th>
<th>Volume Leakage</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.5 m³/s</td>
<td>6970 Pa</td>
<td>1673 Pa</td>
<td>53 Pa</td>
<td>693 Pa</td>
<td>2133 Pa</td>
<td>0.0622 m³/s</td>
</tr>
</tbody>
</table>

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### INPUT DATA

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air Density (Kg/m(^3))</td>
<td>1.20</td>
</tr>
<tr>
<td>Impeller Speed (Rev/min)</td>
<td>3000</td>
</tr>
<tr>
<td>Impeller tip speed (m/s)</td>
<td>80.11</td>
</tr>
<tr>
<td>Blade Width (outer) (m)</td>
<td>0.100</td>
</tr>
<tr>
<td>Blade Width (Inner) (m)</td>
<td>0.160</td>
</tr>
<tr>
<td>Outside Diameter (m)</td>
<td>0.510</td>
</tr>
<tr>
<td>Blade Angle (Outer) (Degrees)</td>
<td>64</td>
</tr>
<tr>
<td>Blade Angle (Inner) (Degrees)</td>
<td>32</td>
</tr>
<tr>
<td>Blade Number</td>
<td>13</td>
</tr>
<tr>
<td>Discharge Coefficient</td>
<td>0.60</td>
</tr>
<tr>
<td>Impeller Inlet Diameter (m)</td>
<td>0.280</td>
</tr>
<tr>
<td>Inlet Clearance (m)</td>
<td>0.002</td>
</tr>
<tr>
<td>Inlet Cone Eye Diameter (m)</td>
<td>0.230</td>
</tr>
<tr>
<td>Tangential Velocity at Inlet (m/s)</td>
<td>40.84</td>
</tr>
<tr>
<td>Pre-Swirl Angle (Degrees)</td>
<td>45</td>
</tr>
<tr>
<td>Fan Case Length (m)</td>
<td>0.350</td>
</tr>
<tr>
<td>Fan Case Width (m)</td>
<td>0.350</td>
</tr>
<tr>
<td>Fan Case Outlet Area (m(^3))</td>
<td>0.1225</td>
</tr>
</tbody>
</table>

| K (in) | 0.90  |
| K (imp)| 0.25  |
| K (out)| 0.4   |

---

### SLIP & LOSS PREDICTION

**K15-50 Hz**

![Graph showing SLIP & LOSS PREDICTION for K15-50 Hz Fan](image)
## Pressure & Volume Losses

<table>
<thead>
<tr>
<th>VOLUME (m³/s)</th>
<th>VOLUME (m³/hr)</th>
<th>Euler</th>
<th>Blade</th>
<th>Impeller (Upper) (Pa)</th>
<th>Impeller (Lower) (Pa)</th>
<th>Outlet (Upper) (Pa)</th>
<th>Outlet (Lower) (Pa)</th>
<th>Inlet (Upper) (Pa)</th>
<th>Inlet (Lower) (Pa)</th>
<th>VOLUME (Upper) (m³/s)</th>
<th>VOLUME (Lower) (m³/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>0</td>
<td>7701</td>
<td>1673</td>
<td>0</td>
<td>0</td>
<td>1540</td>
<td>1540</td>
<td>901</td>
<td>901</td>
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<td>2</td>
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<td>1335</td>
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<td>496</td>
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<td>424</td>
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<td>0.0814</td>
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<td>3600</td>
<td>7409</td>
<td>1673</td>
<td>4</td>
<td>4</td>
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<td>438</td>
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<td>0.0814</td>
</tr>
<tr>
<td>1.25</td>
<td>4500</td>
<td>7335</td>
<td>1673</td>
<td>5</td>
<td>5</td>
<td>1061</td>
<td>1061</td>
<td>534</td>
<td>534</td>
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<td>5400</td>
<td>7262</td>
<td>1673</td>
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<td>6</td>
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<td>710</td>
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### Graph

**Predicted Loss Curves (Lower)**

**K15-50Hz**

- **Euler Line**
- **Blade Loss**
- **Impeller Loss**
- **Outlet Loss**
- **Inlet Loss**

**Legend**:
- ● Euler Line
- ▲ Blade Loss
- ○ Impeller Loss
- ◊ Outlet Loss
- △ Inlet Loss

**Axes**:
- **Y-axis**: Fan Static Pressure (Pa)
- **X-axis**: Volume Flow Rate (m³/hr)

**Grid**: Thousands

---

**Note**: The graph and table data are related to fluid dynamics and pressure loss in a fan system.
### 50 Hz Impellers

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*Notes:*
- HH: Impellers with straight blades.
- LL: Impellers with profiled shrouds.
- NN: Impellers with welded construction.
- OO: Impellers with welded construction.
## FAN CASES

<table>
<thead>
<tr>
<th>DCE Code</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>Special Features</th>
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<tr>
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<td>210</td>
<td>273</td>
<td>213</td>
<td>156</td>
</tr>
</tbody>
</table>

**Diagram:**

- **A**: Width of the case
- **B**: Height of the case
- **C**: Depth of the case
- **D**: Diameter of the inlet
- **E**: Diameter of the outlet
- **F**: Height of the scroll
- **G**: Length of the scroll

**Key:**

- **G5**: Standard case
- **K15**: Narrow case
- **K15-Exp.**: Large exposure
- **K15-Narrow**: Narrow case
- **K15-Sharp**: Sharp cut-off
- **K18/K21**: K18/K21
- **KS5**: Perforated Scroll
### INLET CONES

<table>
<thead>
<tr>
<th>DCE CODE</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>F</th>
<th>G</th>
<th>H</th>
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<td>G3 &amp; G5</td>
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<td>285</td>
<td>360</td>
<td>390</td>
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<td>170</td>
<td>190</td>
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![Diagram of an inlet cone with dimensions labeled A, B, C, D, E, F, G, and H.](image-url)
## Inputs

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<td>Fan Speed (rev/min)</td>
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<tr>
<td>Poisson's Ratio</td>
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<td>Front plate inner diameter (mm)</td>
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<td>Back plate inner diameter (mm)</td>
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<td>Blade width-inner (mm)</td>
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<td>Blade width-outer (mm)</td>
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<td>Blade thickness-outer (mm)</td>
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<td>Blade angle-outer (deg)</td>
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<td>Blade thickness (average) (mm)</td>
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## Calculations

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<td>Blade thickness-inner (m)</td>
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<td>Blade thickness-outer (m)</td>
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<td>Blade angle-inner (deg)</td>
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<tr>
<td>Blade angle-outer (deg)</td>
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<td>Blade angle-straight (deg)</td>
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<td>Blade inner radius (m)</td>
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<td>Blade mid radius-inner (m)</td>
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## Front & Back Plate Stresses

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## Blade Stresses

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<th>Outer</th>
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<td>258.89</td>
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<td>Z (mm³)</td>
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<td>Stress (N/mm²)</td>
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