COMPUTERSIED DESIGN AND SELECTION

PROCEDURES FOR CENTRIFUGAL FANS

by

JARNAIL SINGH

A THESIS SUBMITTED FOR THE

DEGREE OF

DOCTOR OF PHILOSOPHY

DEPARTMENT OF MECHANICAL ENGINEERING

THE UNIVERSITY OF ASTON IN BIRMINGHAM

FEBRUARY 1983

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SUMMARY

There is a great deal of literature about the initial stages of innovative design, a process whereby a completely new product is conceived and developed. In industry, however, the continuing success of a company is more often achieved by improving or developing existing designs to maintain their marketability, a process known as evolutionary design. This thesis reports the way in which this process was applied to the design of centrifugal fan impellers.Improvements were achieved by:-

1). Developing a computer aided selection system to select suitable fans to meet a customer's specific requirement. The present manual system was time consuming, resulting in lost business, due to a large proportion of the enquiries not being answered. The developed system was implemented successfully and showed a substantial improvement in the enquiry response time.

2). Using the finite element technique to gain a better understanding of the complex stress distribution in the centrifugal fan impeller. The need for more accurate methods of stress analysis has become necessary due to increased demands in performance, economy of material, reduction of weight and improved standards of safety. The numerical results obtained, using the Semi-Loof shell element agreed very well with those obtained experimentally, using strain gauges on a commerically produced impeller.

The improvements in the design and selection procedures will significantly help the company to retain/enhance its market share.

EVOLUTIONARY DESIGN

COMPUTER AIDED FAN SELECTION

STRESS ANALYSIS OF THE CENTRIFUGAL FAN IMPELLER

ACKNOWLEDGEMENTS

This project has been a co-operative venture and I would like to express my appreciation to all those who have so freely given assistance.

I would like to thank the management of Alldays, Peacock & Co. Ltd., for proposing the project and also for their continued support during a time of economic difficulties. Thanks are due to Mr. N. Moore who initiated and supervised the project for the first year, to Mr. R. Felton who took over the role of industrial supervisor subsequently. I am also grateful to all staff in the Technical Sales department for their co-operation in the development of the Computer Aided Selection system.

Thanks are also due to my supervisors within the university: Mr. P. Cooley for supervising the project for the first two years and Mr. T.H. Richards who switched roles from being an advisor on the project to a supervisor when Mr. Cooley took sabbatical leave. Dr. J. Edwards from the Management Centre and Dr. L. Hazlewood from the Computing department for being associate supervisors and the contribution they made to the research project. I would also like to thank the other members of the academic and technical staff of university, in particular Mr. K. Apperley for carrying out the strain gauging work on the slip ring adapter.

I am grateful to all staff in the IHD department, particularly Dr. A. Cochran for his role as the IHD supervisor. Thanks are also due to my fellow students on the IHD scheme and in the Mechanical Engineering department whose company has been extremely stimulating.

I would also like to thank everyone involved in the production of this thesis, particularly Mrs. D. Bird for typing the thesis, and Mrs. G. Harrison for tracing the illustrations. Finally I would like to thank all members of my family and friends for their continued support and encouragement during this period.

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NOMENCLATURE

A IN THE FAN SELECTION PROCESS.

t_{sec} .... Run-up time for the fan-motor combination

C_2 .... Motor constant supplied by the motor manufacturer to calculate the run-up time for the fan-motor combination.

Fvp .... Fan velocity pressure.

F.S.P. .... Fan static pressure.

J_{tot} .... Moment of inertia of the fan reduced to the motor speed plus moment of inertia of the motor.

N .... Fan speed.

P .... Static water gauge pressure.

$P_{adjusted}$ .... Adjusted static water gauge pressure due to operating temperature and or altitude.

Q, Q_1 .... Volume flow rate.

S .... Fan diameter.

T .... Gas temperature.

$T_{test\ data}$ ...... Temperature for which the test data is available.

TP_e .... Total pressure at the system entrance.

TP_{losses} .... Sum of the pressure losses in the system.

TP_x .... Total pressure at the system exit.

W .... Power

W_i .... Power applied to the shaft.

W_o .... Useful power output.

η .... Mechanical efficiency.

η_{max} .... Maximum mechanical efficiency.

ρ .... Density of the gas.

ν .... Number of intervals taken along the torque-speed characteristic curve.

B IN THE STRESS ANALYSIS OF THE FAN IMPELLER.

b .... Width of the blade at any radius R.

g .... Acceleration due to gravity.

h .... Blade thickness.

m .... Total mass of the blades in the impeller.

r .... Any radius on the impeller.

t .... Thickness of the backplate or conesheet.

x,y,z, ... Global Cartesian coordinates.

i,j,k, ... Node numbers.

E .... Youngs Modulus of Elasticity.

I .... Moment of Inertia of the blade at any width b.

M .... Bending moment.

M_x, M_y, M_z ..... Bending stress resultants.

N_x, N_y, N_y ..... Membrane stress resultants.

P .... Radially outward force on the blade due to rotation.

R .... Any radius on the impeller.

R_i .... Inner radius of the backplate or conesheet.

R_o .... Outer radius of the backplate or conesheet.

W .... Specific weight of the material.

β .... Local angle between the local radius R to the blade and the normal to the blade.

β_i .... Local angle between the local radius R_i to the blade and the local normal to the blade.

β_o .... Local angle between the local radius R_o to the blade and the local normal to the blade.

θ_1, θ_2 .... Normal rotations at the loof nodes.

ρ .... Density of the material.

σ .... Stress.

σ_r .... Radial stress.

σ_θ .... Hoop stress.

ν .... Poission's ratio.

ω .... Rotational speed.

VECTORS AND MATRICES

$[B]$ .... Strain-displacement matrix.

$[D]$ .... Elasticity matrix.

$\{F^e\}$ .... Element force vector.

$\{F\}$ .... Global force vector.

$[K^e]$ .... Element stiffness matrix.

$[K]$ .... Global stiffness matrix.

$\{u^e\}$ .... Element displacement vector.

$\{u\}$ .... Global displacement vector.

$\{\alpha\}$ .... Vector of unknown coefficients.

$\{\varepsilon\}$ .... Strain vector.

$\{\sigma\}$ .... Stress vector.

CHAPTER ONE

INTRODUCTION

There is a considerable amount of literature about the initial stages of innovative design. This is the process whereby a completely new product is conceived, and developed. In industry, however, the continuing success of a company is more often achieved by improving or developing existing designs to maintain their marketability. Unfortunately, this process of design by evolution is less well documented. This thesis is concerned with two aspects of the evolutionary design process as applied to centrifugal fan impellers:-

1). The selection of an impeller (type and size) to meet a specific requirement.

2). Stress analysis of the impeller.

The project was initiated by Alldays, Peacock & Company Limited, a member of Mitchell Cotts and Company Engineering Limited, which is itself part of the Mitchell Cotts Group. Alldays Peacock manufacture centrifugal fans ranging in size from 400 mm to 2400 mm diameter (with volume flows ranging from 300 M^3/Hr. to 260,000 M^3/Hr., and pressure rises from 125 N/M^2 to 6.25 KN/M^2 respectively), in varying blade widths and cross-sections. The fans are used in a variety of industrial applications including induced draught and dust collection equipment for industrial boilers in steelworks and power stations, large ventilation installations in mines and tunnels, induced and forced draught equipment in a large variety of ships, from passenger liners to oil tankers. In addition to the above large fan applications, smaller centrifugal fans are used in low cost, compact units for heating and ventilating applications.

In initiating this project, the sponsors wished to:-

a). Reduce the time spent on selecting a suitable fan from a given range to meet a customer's specific requirement.

b). Gain a better understanding of the stress distribution in the impeller, with a view to improving the design of the existing range and to instil more confidence in designing larger "one off" impellers.

The background to these two areas of research are discussed in detail, in Chapter 2 Section 2 of this thesis.

The project was accomplished through the Interdisciplinary Higher Degrees (I.H.D.) scheme at the University of Aston in Birmingham. Since the sponsoring company market only centrifugal fans, they were able to provide invaluable expertise, but all the development work was performed at the university. The I.H.D. scheme provides an ideal environment for joint industry/university projects of this type. The co-ordinating role of the I.H.D. office enabled assistance to be drawn from mechanical engineering, systems analysis and computing disciplines as well as from the company.

This arrangement laid the foundations of the project with the terms of reference being to:-

a). Develop a computer aided selection system, to enable the company to reduce the time spent on each enquiry, in selecting a suitable fan impeller to meet a customer's requirement from the standard manufactured range.

b). Develop a computer program to analyse the stress distribution in the centrifugal fan impeller (due to centrifugal forces only) using the finite element technique and to assess its effectiveness by comparing predictions with experimentally produced data.

CHAPTER TWO

BACKGROUND TO THE PROJECT

INTRODUCTION

Chapter 2 begins with a brief introduction to fans and fan terminology. Section 2.2 deals with the background to the project and identifies the areas of concern to the company. The two particular issues namely that of the enquiry response time and the stress analysis of the fan impeller towards which the project is directed are critically examined in Sections 2.3 and 2.4. In each case, the present procedure is reviewed, and the imperfections recognised, leading to the aims of the project to improve upon the current situation.

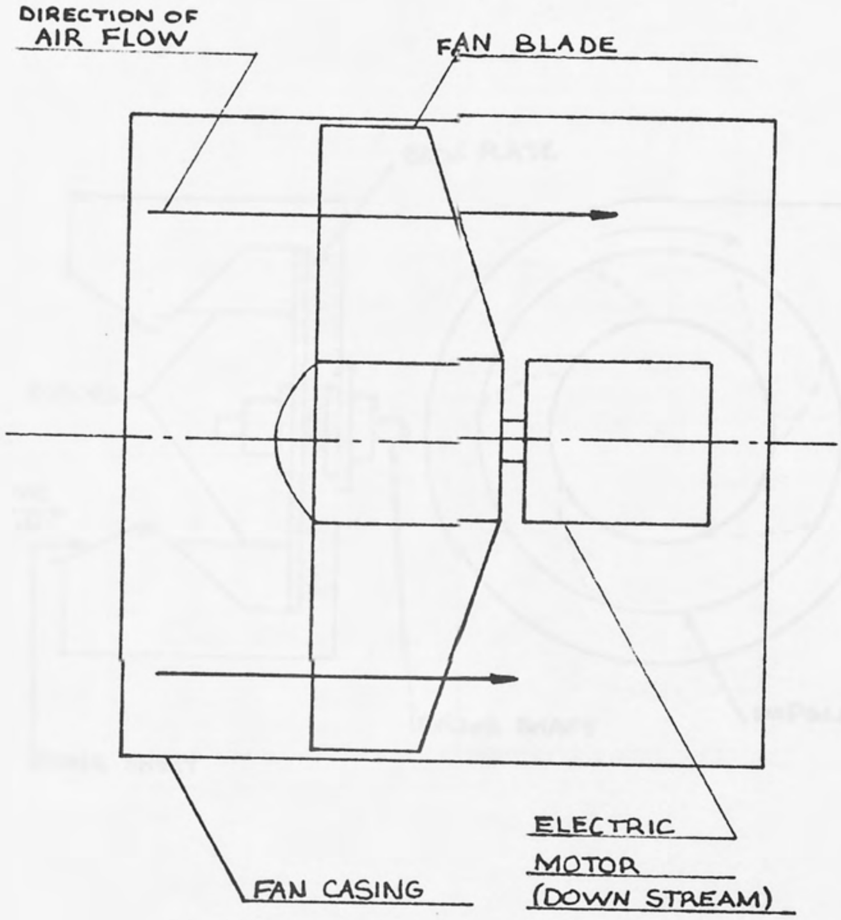
2.1. FANS AND FAN TERMINOLOGY

A fan is defined as a bladed rotary machine maintaining a continuous flow of air. Continuous because the air flows steadily into, through and out of the fan. This feature distinguishes it from positive displacement (piston, vane or lobe) machines, which generally produce a pulsating flow. A fan has a rotating impeller carrying blades of some kind. These blades exert force on the air, thereby maintaining the flow and raising the pressure. Fans do not normally raise the absolute pressure by more than 30% (30,000 N/M^2 fan total pressure with standard air at the inlet), distinguishing it from a compressor where the pressure rise is much greater.

Fans can be divided into two categories, termed "axial" and "centrifugal". In a pure axial fan (Fig. 2.1) the effective progress of the air is straight through the impeller at a constant distance from the axis. The primary component of the blade force on the air is directed axially from inlet to outlet, and thus provides the pressure rise by a process that maybe called direct blade action. The blade force necessarily has an additional component in the tangential direction, providing the reaction to the driving torque. This sets the air spinning about the axis independently of its forward motion.

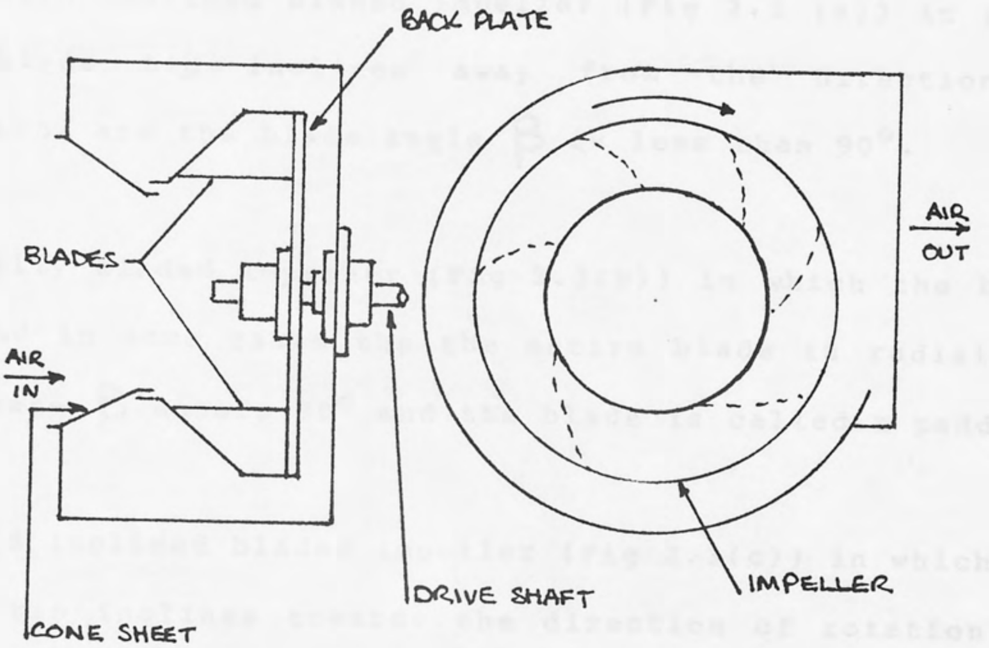
A centrifugal fan (Fig 2.2) consists of an impeller running in a casing which has a spirally sloped contour. The air enters the impeller at the centre in the axial direction and is discharged at the periphery in the tangential direction. The impeller is usually the most highly stressed part of the fan assembly, and therefore the component most likely to fail first. The stresses in the impeller are caused by rotation, temperature, and aerodynamic forces, but by far the most important (and the ones the project is concerned with) are those due to rotation.

The centrifugal fan impeller consists of a backplate (backsheet) and conesheet (inlet cone), with radially



AXIAL FAN NOTATION

Fig 2.1



CENTRIFUGAL FAN NOTATION

Fig 2.2

spaced blades continuously welded between them. The cross section of the blade can be plane or aerofoil. There are three main classes of impellers (Fig 2.3) depending on the type of blade. These are:-

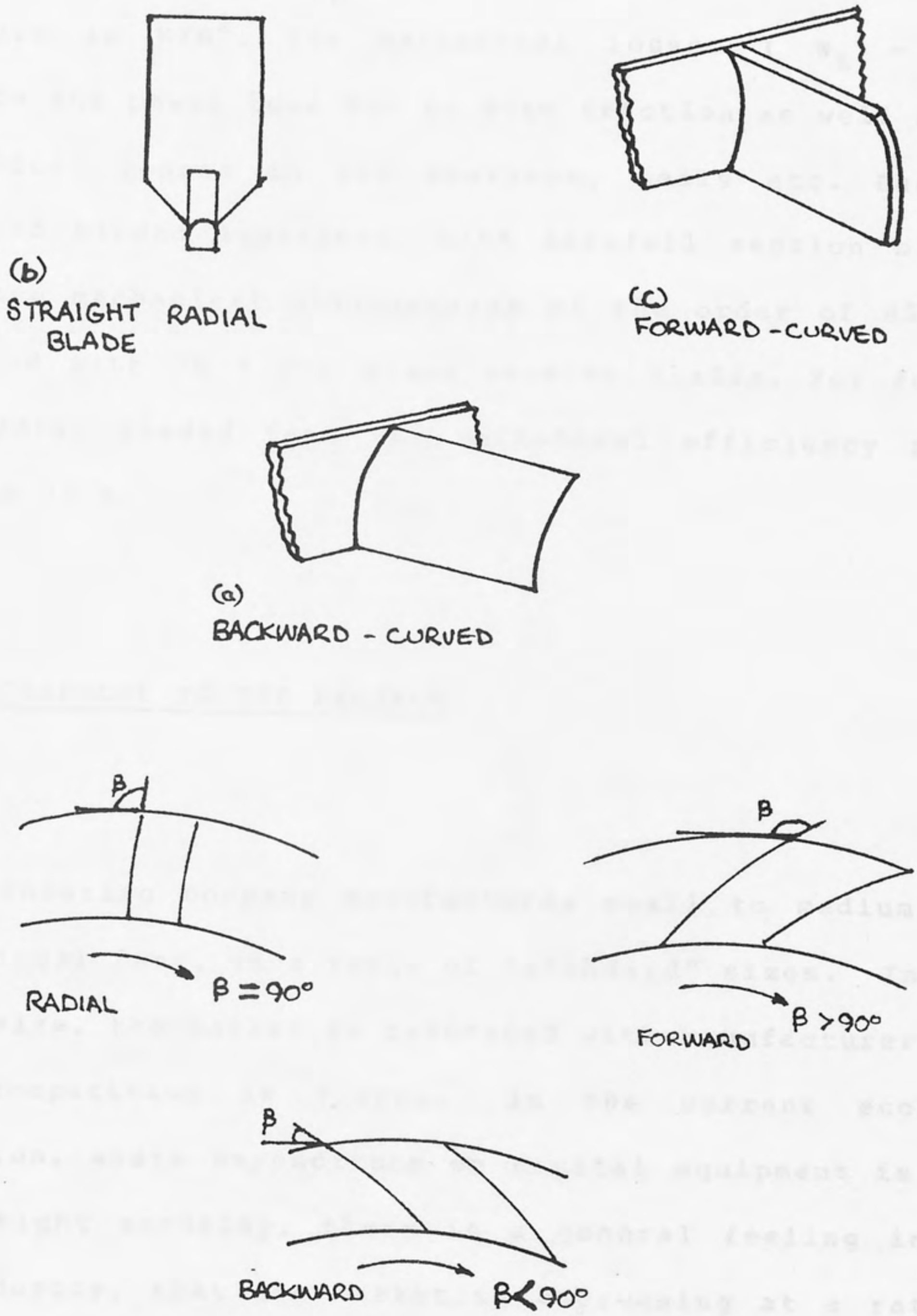
a). Backward inclined bladed impeller (Fig 2.3 (a)) in which the blade tip inclines away from the direction of rotation, and the blade angle β is less than 90^{o}.

b). Radially bladed impeller (Fig 2.3(b)) in which the blade tip and in some cases the the entire blade is radial. In this case β equals 90^{o} and the blade is called a paddle.

c) Forward inclined bladed impeller (Fig 2.3(c)) in which the blade tip inclines towards the direction of rotation and in this case β is greater than 90^{o}.

The mechanical efficiency (η_m) is defined as the ratio of the power transmitted to the fluid and converted into useful output **(W_o)** to the power applied to the shaft **(W_i)** ie,

$$\frac{W_o \text{ (Watts)}}{W_i \text{ (Watts)}} = \frac{Q_1 * P}{W_i} \qquad \text{..........(2.1)}$$



CENTRIFUGAL FAN BLADE FORMS

fig 2.3

where Q_1 is the flow rate in M^3/second and P is the pressure in N/M^2. The mechanical losses (W_i - W_o) include the power loss due to disc friction as well as the mechanical losses in the bearings, seals etc. Backward inclined bladed impellers, with aerofoil section blades, can have mechanical efficiencies of the order of 85 % as compared with 75 % for plane section blades. For forward and radial bladed fans the mechanical efficiency rarely exceeds 70 %.

2.2. BACKGROUND TO THE PROJECT

The sponsoring company manufactures small to medium size centrifugal fans, in a range of "standard" sizes. In this range size, the market is saturated with manufacturers, so that competition is fierce. In the current economic situation, where expenditure on capital equipment is kept under tight scrutiny, there is a general feeling in the fan industry, that the market is decreasing at a rate of some 3 - 5% per annum. Alldays, Peacock are holding their position well in the decreasing market. The fan industry is in general, highly labour intensive. This is because the method of manufacture employed by the industry is fabrication and welding. These two processes are labour intensive, and with the rate of inflation experienced in the late 70's and early 80's means the industry has to support very high overheads.

In the current economic climate, where resources for capital expenditure are scarce and because of the amount of capital already invested in this method of production, the manufacturers are reluctant to adopt other processes such as casting the impeller assembly (e.g. paddle blade type). Therefore, like most other industries in the engineering sector, they are looking to reduce their overheads. This has been achieved to some extent by most manufacturers by standardising their product ranges in terms of the sizes and blade types they produce. Some, who have enough resources behind them, have gone into batch production of units, with some degree of automation, at the smaller size end of the market.

The sponsoring company do not have the resources necessary for this type of production. The company also felt that as the smaller size market was already saturated with manufacturers, they would try and cater for the larger size market. This policy has the advantage that the company already has production facilities for this, as there is some 30-35% spare production capacity not being utilised at the present, and therefore no extra capital expenditure on plant or machinery would be necessary. Investigation by the company also showed that administration overheads could not be reduced further as these were at a bare minimum. To generate more business and thus improve the financial position the investigation showed that the following had to be accomplished:-

1). Reduce the time taken to respond to an enquiry, thereby generating more business with the present levels of staff.

2). Improve knowledge of the stress distribution in the impeller (as this is the most highly stressed component), so that high performance (i.e. faster running), low cost, large size fans can be designed with more confidence for the newer larger size market being served for.

3). Improve on production difficulties i.e. production scheduling and quality control during production.

This project is concerned with the first two areas of concern identified by the company.

2.3.1. ENQUIRY RESPONSE TIME

The company achieves sales through tendering in response to enquiries which it receives. The selection of a suitable size and type of impeller to meet the customer's requirement involves a certain amount of tedious, time consuming calculations. An analysis for the year 1978 showed that the average cost of adequately responding to the enquiry was seventeen pounds and fifty pence (at 1978 levels). For full details of this analysis please refer to Appendix No. (1) of this thesis. The analysis also revealed, that in this period, 889 enquiries received (i.e. 18% of the total) were not answered due to insufficient time. Assuming an order to tender ratio of 1 in 10 (a conservative estimate), this means approximately 89 potential orders lost. The situation is becoming more acute since there is also an upward trend in the number of enquiries received per annum, probably due to the fact that the prospective customers are sending enquiries to more manufacturers to obtain the most competitive price. The analysis also shows that the prospective customer is also demanding more information at the enquiry stage than previously. This information may comprise of performance curves, noise level data, engineering drawings, stress calculations etc. This will result in more time being spent on each enquiry, (say up to 30% more time) depending on how much of the above information the customer requests in his particular case.

Clearly, this means more and more enquiries not being answered, unless more staff are employed, or a means found of reducing the time spent on each enquiry. Employing more staff, on average costs 6500 pounds (at 1978 levels) including overheads, per man, per annum, with a 2-3 month training period, as the selection procedure involves a certain amount of "art". The selection process, described in Chapter 4, involves a certain amount of routine but time consuming calculations so that the process can very appropriately be carried out on a computer. The advent of low cost desk top (micro) machines, provides a highly attractive means of computer implementation, where ease of use and direct control, creates an atmosphere of ready acceptance by the user. Computer aided selection, therefore seems an ideal way of reducing the time spent on each enquiry.

2.3.2. CHARACTERISTICS OF THE TWO TYPES OF ENQUIRIES RECEIVED

The enquiries received by the company fall into one of the two following categories written and verbal (telephone).

To establish if there is any significance in the time taken to answer an enquiry and whether or not it subsequently resulted in a firm order, a random sample of 528 written enquiries (of which 121 were subsequently found to be orders) and 1030 verbal (of which 151 resulted in orders) were taken, and the statistical chi square (χ^2) test was applied. In both cases the null hypothesis (H_o) assumed was that there is no significant association between the time taken to answer the enquiry and whether it subsequently resulted in an order or not. For full details of the analysis, please refer to Appendix 2 of the thesis.

The analysis revealed the two categories to be very different in character from each other. In the case of written enquiries , the response time (which is related to the amount of information requested by the customer) has a significant influence on whether or not it results in a firm order. In this case, the customer has thought out his requirements in detail. In return, he wants to know if the fan is safe to operate (i.e. do calculations show that stresses and noise level spectra are acceptable), will it fit into the available space (engineering drawings) etc. The results suggest (Fig. 2.4 and 2.5) the ideal response time to be in the region of 2-3 days, although up to 5 days maybe acceptable for large "one off" units, or where the customer requests a substantial amount of information (e.g. material specification, motor performance characteristics etc.) which has to be obtained from outside suppliers and hence takes time to accumulate.

In the second case (i.e. verbal enquiries), the

BAR CHART OF THE TOTAL NUMBER OF ENQUIRIES (WRITTEN) ANSWERED AGAINST TIME

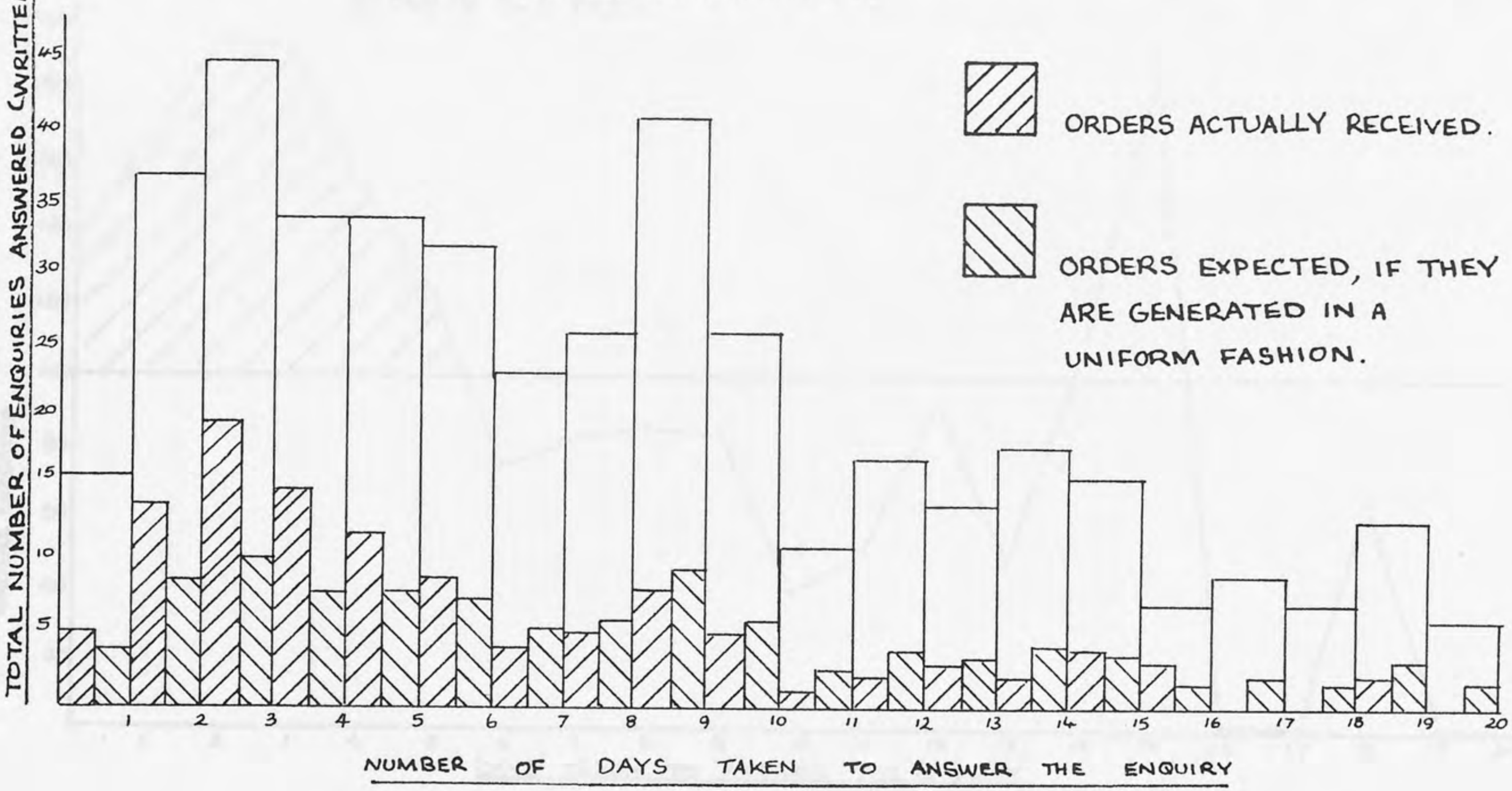


Fig 2.4

GRAPH OF $\frac{\text{ORDERS RECEIVED}}{\text{ORDERS EXPECTED}} \times 100\%$ AGAINST TIME

SHADED AREA SHOWS THE TIME SPAN WHEN MORE ORDERS ARE RECEIVED THAN EXPECTED, IF ORDERS ARE GENERATED IN A UNIFORM FASHION. ONE CAN THEREFORE CONCLUDE THAT THERE IS A GREATER CHANCE THAT THE ENQUIRY COULD RESULT IN A FIRM ORDER, IF THE TIME TAKEN TO ANSWER IS IN THE RANGE OF 1-5 DAYS, WITH THE HIGHEST CHANCE IF THE RESPONSE TIME IS BETWEEN 2-3 DAYS.

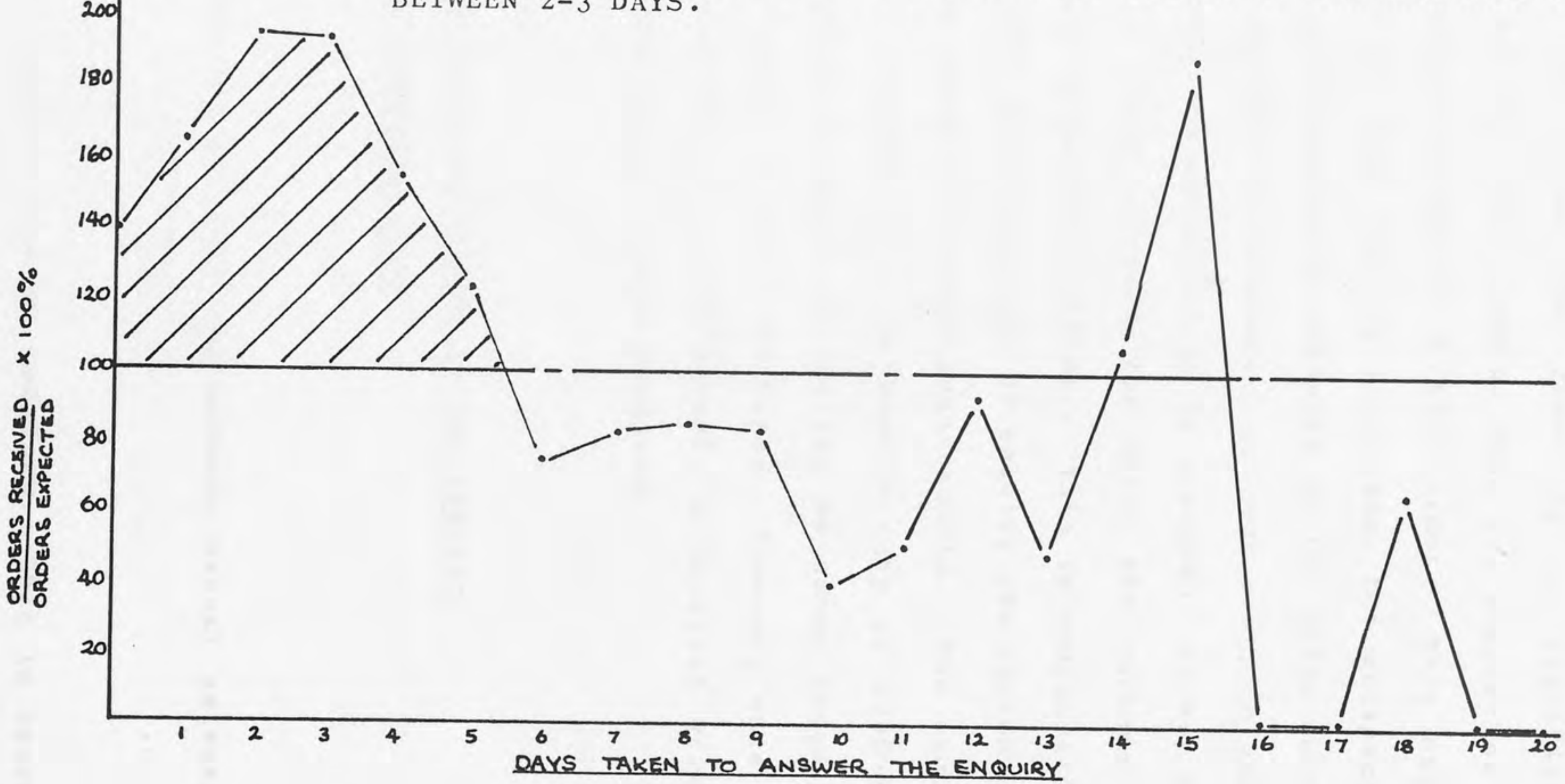


Fig 2.5

statistical test shows that there is no significance association between the response time and whether or not it subsequently resulted in a firm order. This can be explained by the fact that in this case, the prospective customer is only seeking an estimate of the price so that it could be included in an overall project cost, to enable capital expenditure sanctions to be granted. It may also be part of a large contract for which the customer is himself trying to prepare a tender. This is characterised by the fact that with this type of enquiry the customer is usually vague about his exact requirements. The enquiry might be for example a fan to meet a duty of 2000-3000 M^3/second with a pressure of 200-300 mm water gauge, or the duty may change a number of times. However, once the project or contract has been agreed, a detailed written "follow up" enquiry is usually received.

2.3.3. IMPERFECTIONS RECOGNISED IN THE PRESENT ENQUIRY RESPONSE TIME

The main disadvantages with the present manual selection system are:-

1). The enquiry response time is too long. This is because of the routine time consuming calculations necessary and the gathering of information from outside suppliers for components (e.g. electric motors, bearings, acoustic materials etc) bought in.

2). A large proportion of the enquiries received are not answered due to insufficient time, hence potential orders lost.

3). Poor overall service : In certain cases not all the information requested by the customer is supplied because of the time taken manually to: calculate fan-motor combination run-up time, produce performance curves and engineering drawings etc.

4). There is currently insufficient time for the technical salesman to "follow up" the enquiry once it has been sent out, unless the prospective customer contacts the company himself.

All the above factors results in lost business for the company. For example, assuming an order to quote ratio of 1 in 10 (a conservative estimate), the 889 enquiries received in 1978 but not answered, represents approximately 89 potential orders lost.

2.3.4. OBJECTIVES OF THE COMPUTER AIDED SELECTION SYSTEM

1). Provide an instant service for telephone enquiries i.e., the customer is given details of suitable selections, while he waits, with a written quotation to follow. This represents an improvement in the service provided; the fact that the customer is paying also results in reduced telephone and consequent overhead costs. At present a return telephone call is made and a written quotation follows. The telephone charges for the sales department totalled approximately 6000 pounds for the year 1977 and the aim is to reduce these costs substantially.

2). For written enquiries, achieve a response time of between 3-5 days. The statistical analysis suggested that this was the order of time scale tolerated by a customer if the enquiry is to result in a firm order

3). Respond to more enquiries with the existing levels of staff

4) "Following up" the enquiry with the customer. At present there is no such "follow up" procedure, as all the time is taken up preparing the initial tenders

The above aims are all to provide a better service and hence generate more business.

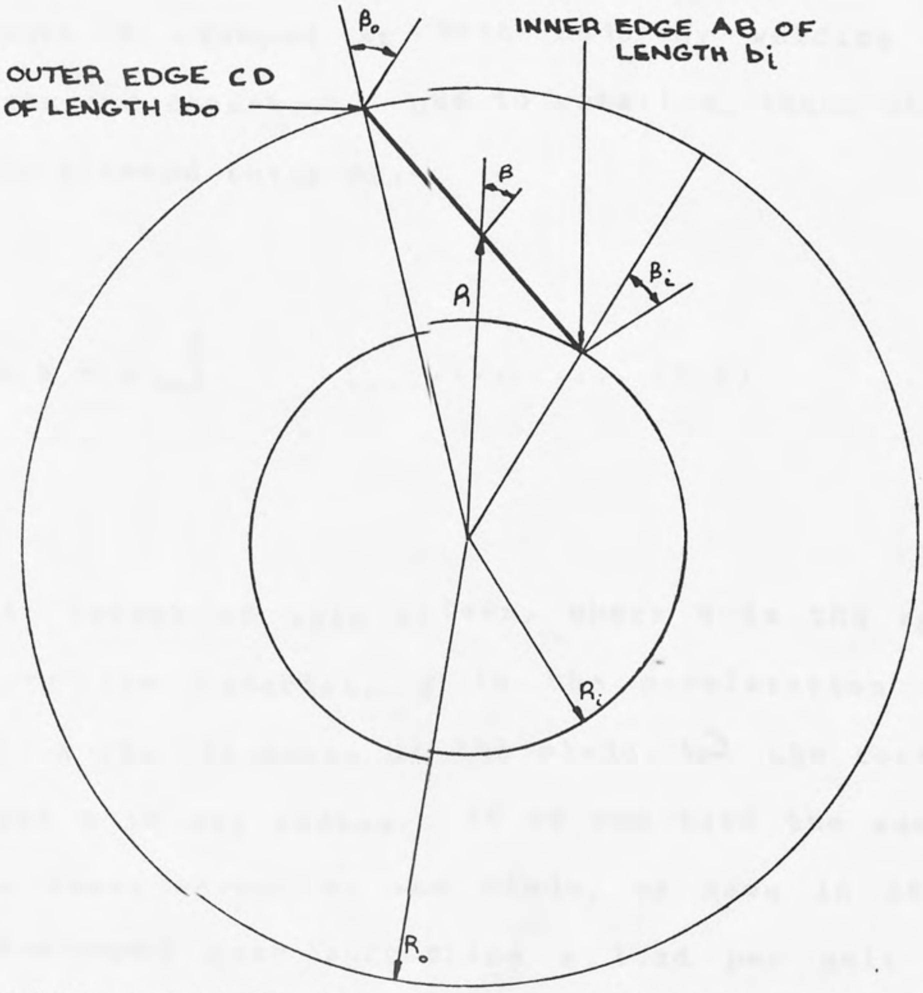
2.4. STRESS ANALYSIS OF THE IMPELLER

2.4.1. PRESENT PROCEDURES

The centrifugal fan impeller is usually the most highly stressed part of the fan assembly and is therefore the one requiring the most care in the analysis. Of the various actions on the impeller, the most important and the one the project is concerned with, are those due to the centrifugal forces. The impeller consists of a backplate and conesheet, with radially spaced blades continuously welded between them; a front view of such an impeller is shown in Fig. 2.6. The present stress analysis procedure is extremely simple and consists of treating the components of the impeller individually, and then superimposing the effect of the blades on the backplate and conesheet.

For the blades the procedure is as follows:-

Somewhere on the blade at the radius R, (Fig. 2.6) imagine an axial sliver of blade of unit width isolated as a beam



BLADE THICKNESS h

ROTATIONAL SPEED

ω RAD/SEC

EDGES AB AND CD ARE PARALLEL TO THE AXIS (PERPENDICULAR TO THE PLANE OF THE PAPER) AND ARE FREE OF SUPPORT.

NOTATION USED IN THE PRESENT STRESS ANALYSIS PROCEDURES FOR CENTRIFUGAL FANS

Fig 2.6

of length **b** clamped at both ends by welding to the backplate and conesheet. Due to rotation, there will be a radially outward force of:-

$$P = \frac{W\ h\ R\ \omega^2}{g} \qquad \ldots\ldots\ldots\ldots\ldots\ldots(2.2)$$

per unit length of this sliver, where **W** is the specific weight of the material, **g** is the acceleration due to gravity, **h** the thickness of the blade, ω the rotational speed and R is any radius. If we now take the component of this force normal to the blade, we have in effect a clamped-clamped beam supporting a load per unit length of:-

$$P = \frac{W\ h\ R\ \cos\beta\ \omega^2}{g} \qquad \ldots\ldots\ldots\ldots\ldots\ldots(2.3)$$

where β is the local angle between the local radius R to the blade and the local normal to the blade. Now for a clamped-clamped beam of length **b**, the maximum bending moment is at the ends and is given by:-

$$M = \frac{P\ b^2}{12} \qquad \ldots\ldots\ldots\ldots\ldots\ldots(2.4)$$

The corresponding maximum bending stress is given by

$$\sigma = \frac{M\,b}{2\,I} \qquad \text{...............(2.5)}$$

where **I** is the moment of Inertia of the beam. Substituting into 2.5 for **M** (equ 2.4) and for **P** (equ 2.3) and for **I** gives:-

$$\sigma = \frac{W\,b^2\,R\cos\beta\,\omega^2}{2\,g\,h} \qquad \text{................(2.6)}$$

The stress at the tip and root of the blade can now be calculated by inserting the appropriate values of the variables. It is worth emphasising that the angle β is different for different regions of the blade even when the blade is flat (i.e. β_i and β_o are different in Fig. 2.6).

The next step is to treat the conesheet and backplate as two rotating hollow discs, with their respective inner and outer radii. For a rotating disc, with a central hole, the maximum radial stress occurs at a radius **r** (Roark **[1]**) given by:-

$$r = \sqrt{R_o \, R_i} \qquad \ldots\ldots\ldots\ldots\ldots(2.7)$$

where R_i is the inner radius and R_o the outer, and the magnitude of the maximum stresses is given by :-

$$\sigma_r = \frac{(3+\nu)\rho\omega^2 (R_o - R_i)^2}{8} \qquad \ldots\ldots(2.8)$$

$$\sigma_\theta = \frac{(3+\nu)\rho\omega^2 (2R_o^2+(1-\nu)R_i^2)}{8 \qquad (3+\nu)} \qquad \ldots(2.9)$$

where ν is Poissions ratio and ρ is the density, and σ_r and σ_θ are the radial and hoop stresses respectively. The maximum stresses σ_r and σ_θ in each component are calculated using equations 2.8 and 2.9.

The interaction of the blades with the backplate and conesheet is analysed next. The procedure is to calculate the centrifugal force due to the total mass of the blades, acting at the mean of the two radii of the conesheet, i.e.

$$F = \frac{m\,(R_o + R_i)}{2}\,\omega^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots(2.10)$$

where **m** is the total mass of the blades on the impeller. The assumption is then made, that this force is distributed equally between the backplate and the conesheet. The hoop stress due to the centrifugal force of the blades in each component is calculated as:-

$$\sigma_\theta = \frac{F}{2\ \text{Area}} \quad \ldots\ldots\ldots\ldots\ldots\ldots(2.11)$$

where

$$\text{Area} = 2\,r\,t\,\pi \quad \ldots\ldots\ldots\ldots\ldots\ldots(2.12)$$

where **r** is that obtained from equation (2.7) and t is the thickness of the backplate or conesheet. The hoop stress (σ_θ) due to the blades given by equation 2.11 is then added to the hoop stress given by equation 2.9 to give the total hoop stress in each component. The values obtained for the hoop and radial stresses in the backplate and conesheet, and the stresses at the blade root and tip, are then compared with the maximum permitted values (with large factors of safety) for the material used for each component.

It is worth commenting, that whilst this model of the impeller is very crude, many years of trial and error have yielded permissible values for the stresses calculated in this way even though finer detail cannot be obtained.

2.4.2. IMPERFECTIONS IN THE PRESENT STRESS ANALYSIS PROCEDURES

The above method has certain disadvantages in that it assumes the impeller stresses are axisymmetric and takes no account of the complex geometry of the fan, or the discontinuous nature of the blade loading which can in some cases cause large bending stresses in the backplate and conesheet. It is for these reasons that this method of analysis applied to the centrifugal fan impeller must be considered to be very approximate.

The need for more accurate methods of stress analysis of engineering components has become necessary due to increased demands in performance, economy of materials, reduction of weight, appearance and improved standards of safety. In the case of centrifugal fans the performance demands are increasing to such an extent that the present methods of estimating the stress distribution are no longer considered adequate.

2.4.3. AIMS OF THE PROJECT TO IMPROVE KNOWLEDGE OF THE STRESS DISTRIBUTION IN THE FAN IMPELLER

The aim is to provide improved knowledge of the stress distribution (due to centrifugal force only) in the impeller utilising the Finite Element process. The advantages of this numerical method includes the ability to:-

a). Deal with highly irregular shapes and surfaces.

b). Analyse areas of high stress gradients, or of particular interest in greater detail.

c). Apply a greater variety of loading and boundry conditions than would be possible by a purely mathematical method.

CHAPTER THREE

REVIEW OF THE LITERATURE

INTRODUCTION

This chapter provides a review of the literature relevant to the design of fans. The chapter commences with a consideration of the design process in general and continues with a discussion of fan selection as part of the design process.

Section 3 discusses the hallmarks of industrial marketing as part of the design process and its relevence to the proposed computer aided selection system.

Section 4 deals with the process of selecting impellers for specific duties, while section 5 is devoted to the stress analysis of the centrifugal fan.

A centrifugal fan impeller can be regarded as an assemblage of plates and shells and as this is the basis for the finite element formulation of the elements, section 6 of the chapter summarises relevant aspects of plate and shell theory. The section continues with a brief review of the finite element process as applied to plate and shell type structures.

3.1 . THE DESIGN PROCESS.

3.1.1. DESIGN CRITERIA AND RESOURCES

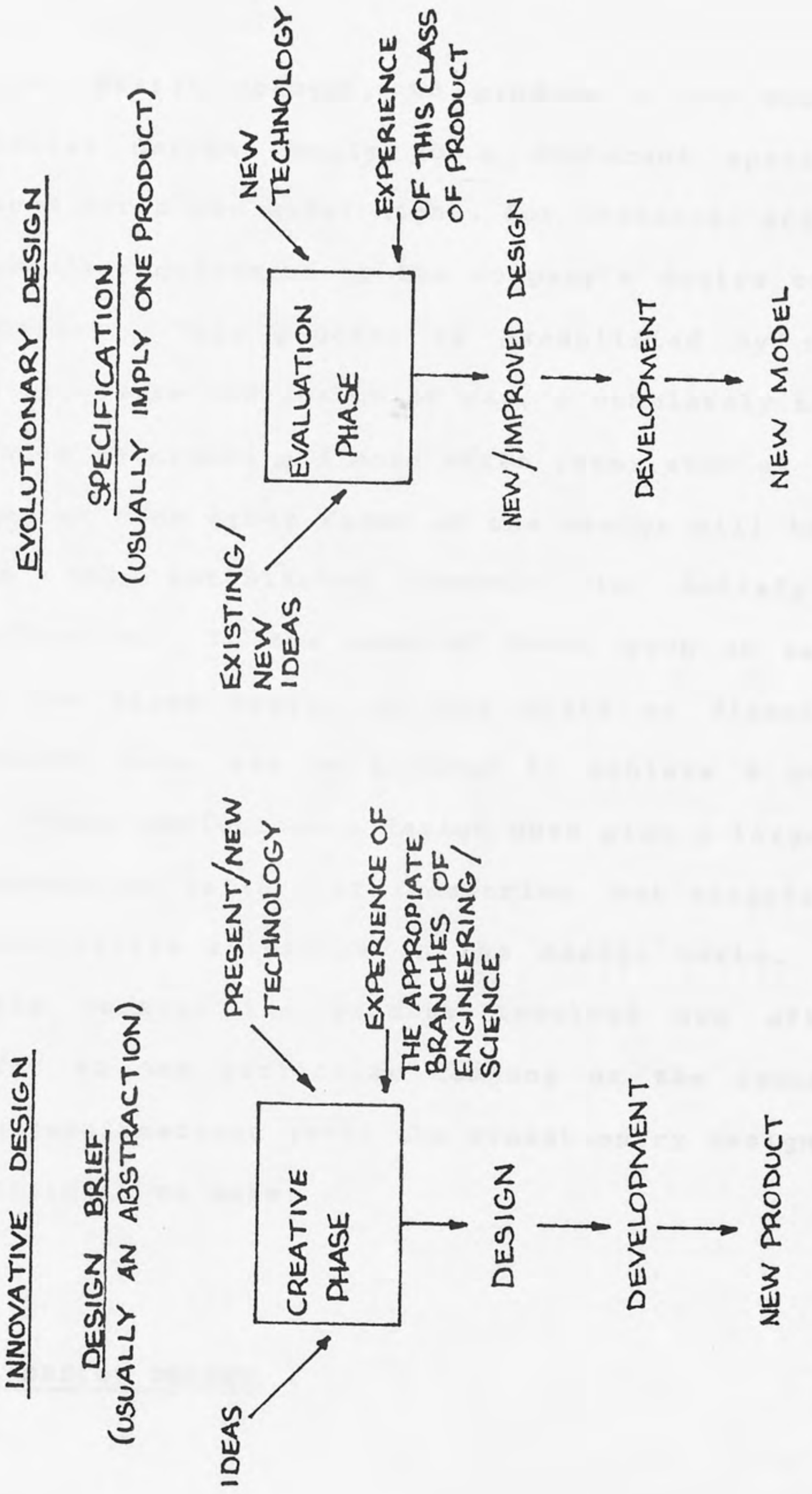
Design consists of satisfying objectives subject to certain constraints or criteria. The basic criteria are defined in a specification or brief which should contain sufficient information to attempt a solution. The criteria fall into a number of categories, but the technical specification is probably foremost in the designer's mind during initial design, although cost and ease of maintenance of a product, for example, are just as important as the technical adequacy. Beakley and Chilton [2] demonstrate this with several case studies. In many cases, the complexity of the technical considerations monopolises the designer's concentration.

The fundamental resources necessary for carrying out any task is a substantial allocation of capital and time. Both these factors are what the economists call scarce resources. Some companies spend most of their capital on engineering and enter bankruptcy after designing the finest product of its kind. Other companies choose to emphasise production to the exclusion of both engineering and marketing. Other firms are extremely sales minded and tend to sacrifice engineering and production on the altar of increased volume. Lack of proper balance in any of these aspects can be disastrous not only for the product but for the company. Capital is also necessary for acquiring labour, machinery, space and materials. These four quantities have a considerable bearing on a company's design capability, especially when it is desired to manufacture a product in house with the available machinery, materials and expertise.

3.1.2. EVOLUTIONARY AND INNOVATIVE DESIGN PROCEDURES

Many texts agree that different types of design work can be distinguished according to the nature of mental activity involved. Furman [3] distinguishes between repetitive, evolutionary and innovative design. Aismov [4] simply distinguishes between innovative and evolutionary design, which are depicted in Fig. 3.1.

There is significant emphasis being placed on innovative design. This is the process whereby a completely new product is conceived, invented and developed. In industry, however, the continuing success of a company can more often be achieved by evolutionary design. Design by evolution, is concerned with modifying or improving an



COMPARISON OF INNOVATIVE AND EVOLUTIONARY DESIGN PROCESSES

Fig 3.1

existing design concept, to produce a new member of a particular design family to a different specification. The need for a new model might, for instance, arise from a customer's requirement or the company's desire to enter a new market. This process is exemplified by the motor industry, where the design of say, a completely new engine is a rare occurence and more often items such as the bore, stroke, or some other facet of the design will be altered within the established concept to satisfy a new specification. In the case of fans, such an example is where the blade angle, or the width or diameter of a "standard" size, may be altered to achieve a particular duty. Thus, evolutionary design does play a large part in the success or failure of industries, but surprisingly it attracts little attention in the design texts. This is probably because the process involved are often very specific to one particular company or the industry, so that generalisations about the evolutionary design process are difficult to make.

3.1.3. DESIGN PHASES

Engineering design, whether innovative or evolutionary in nature passes through three phases. These are:-

3.1.3.1. THE FEASIBILITY STUDY PHASE

A design project begins with a feasibility study, the purpose is to achieve a set of useful solutions to the design problem. Sometimes a design group is assigned to a project for which a design concept has already been fixed. This implies one of three possibilities:-

1). A feasibility study had been previously made.

2). The technical management has had so much experience with the particular design problem that further study would be superfluous.

3). The management, by omitting the feasibility study, is proceeding on unsupported intuition.

The first step in the study is to establish whether the original need, which was presumed to be valid, does indeed have current existence, or strong evidence of latent existence.

The next step is the identification and formulation of the design problem. Before attempting a solution of the means of satisfying the need, the design problem needs to be identified and formulated. At this stage in the design process, there is a strong temptation to seize mentally on some concept of hardware that seems to provide a feasible solution before the real problem is understood, and there after to patch up the concept in perilous ways as deficiencies in the solution begin to appear. This temptation should be resisted, for it tends to lead to a mental rut which blocks the truly creative effort that should follow once the problem has been fully grasped.

In the following step, that of the synthesis of possible solutions an effort is made to conceive a number of plausible solutions to the problem. It is the synthesis step which most characterises the project as a design undertaking. This, more than any other step requires inventive and creative effort. Creativity is therefore a very essential ingredient of engineering design. Creativity according to Asimov is the talent for discovering combinations of principles, materials or components, which are especially suitable as solutions to the problem in hand. None of the individual elements which comprise the synthesis need be new or novel. Developing new and novel elements is more the object of research than of design. After a new principle is discovered by the researcher, the designer is often assigned the task of evolving some new components that are advantaged, for the first time made possible by the new principle. In doing so the designer will generally combine the new principle with several old ones to achieve the final result.

Finally, at this stage of the design process, the potentially useful solutions are sorted out of the plausible set in three steps on the basis of physical realisability, economic worth-whileness and financial feasibility. In conclusion, the completed study indicates whether a current or a potential need exists, what the design problem is, and whether useful solutions can be found, i.e., it investigates the feasibility of the proposed project.

3.1.3.2. THE PRELIMINARY DESIGN PHASE

The preliminary design phase is intended to establish an overall concept for the project which will serve as a guide for the detailed design. An evaluation of the design concept is carried forward far enough so that a decision can be made about committing funds for the next phase.

The preliminary design phase starts with the set of useful solutions, developed in the previous phase, to establish which of the possible alternatives is the best design concept. Each of the possible solutions is subjected to order of magnitude analyses until the evidence suggests either that the particular solution is inferior to some of the others, or that it is superior to all the others. In principle, the approach is straight forward, in practice, the decision is difficult. In principle, the advantages and disadvantages of the attributes of each solution are listed and the one with the most favourable set is selected. Asimov [4] lists some of the difficulties which can occur in practice. The surviving solution is tentatively accepted for closer examination. Synthesis studies are initiated for establishing to a first approximation, the fineness of the range within which the major design parameters of the system must be controlled. The results of the sensitivity analysis step are a greater insight to the inner workings of the system or device, an identification of the critical design parameters as distinct from the less critical, an indication whether some of the constraints should be loosened or tightened, a more quantitative idea about the expected overall performance of the system.

Futher studies investigate the tolerances in the characteristics of major components and critical materials which will be required to insure mutual compatibility and proper fit into the system, while other studies examine the extent to which changes in environmental or internal forces will affect the stability of the system.

Next, projective-type studies are undertaken, addressed to the question of how the solution will fare in time. The socio-economic conditions, such as consumers' tastes, competitors' offerings, or availability of critical raw materials, may change, the state of technical art may advance, and eventually corrosion, fatigue and deterioration of performance may set in. Time will almost certainly erode the quality of the product. The question is, how fast? The rate of obsolescence or wear must be accounted as one of the important design considerations, and its economic impact must be put in the balance.

Before the preliminary design phase can be considered to have been completed, the solution should be subjected to a rigorous study to reveal any unnecessary complicating factors and to discover every possible simplification. Good design has an aesthetic quality: the quality of simplicity.

3.1.3.3. THE DETAILED DESIGN PHASE

The detailed design phase begins with the concept evolved in the preliminary design. Its purpose is to furnish the engineering description of a tested and producible design.

Up to this point the design project was characterised by great fluidity. Major changes in concept could be accomodated without great financial loss. Indeed, for the first two phases such fluidity is essential, for they are primarily exploratory in nature, seeking to reveal an adequate range of possible solutions. At this point, however, either exploration on a large scale must come to a close, and final decision for a particular design concept be made, or the project must be abandoned as infeasible. In the latter event, final abandonment if warranted by favourable circumstances, may be deferred pending the results of an additional search for possible new solutions.

With the design concept in mind and the preliminary synthesis information at hand, an overall, but provisional, synthesis is accomplished. It is developed as a master layout. With this as a basis the detailed design or specification of components is carried forward. From time to time, requirements in the detailed work at the component level may dictate changes in the master layout, therefore it has provisional status. As the paper design progresses, experimental design is appropriately initiated.

Experimental models are constructed to check out untried ideas which are not suitable to the final disposition by analysis. Components, partial prototypes, and finally complete prototypes are tested as the need for information arises. This information, accruing from the testing programs, provides a basis for redesign and refinement until an engineering description of a proven design is accomplished.

3.1.3.4. PLANNING THE PRODUCTION PROCESS

Whereas the preceding three phases were peculiarly in the province of the engineering designer, much of the responsibility for the production stage will be shared with other segments of management. A new combination of skills, those of tool design and production engineering come into play. The decision to produce often involves an enormous economic commitment. The level of confidence in the success of the product must be very high to support a positive decision. The decision itself must be made at that level of management at which the final responsibility for the success of the enterprise rests. The evidence on which the designer, responsible for the design project, basis his confidence must be communicated in a concise, but nevertheless in a fully revealing form to the appropriate decision-maker. Ideally, the designer's confidence will be shared by his superior, who will re-evaluate this confidence using additional information concerning financial capability, business conditions, and like considerations, before rendering a final decision.

The production planning phase involves many steps which will vary in form and detail with the industry concern. The more common ones are:-

1). Detailed planning of the manufacturing processes as required for every part, every sub-assembly, and the final assembly. The information is usually contained on process-sheets, one for each part or sub-assembly. The process-sheets contain a sequential list of operations which must be performed to produce the part, it specifies the raw material, clarifies special instructions, and indicates tooling and machines required. This step is particularly important, because design features that lead to difficulties in production are revealed. Such difficulties should have been minimised by previous timely consultations between product designers and tool designers. Similarly, questions about materials should have been resolved by consultation with process-metallurgists.

2). Design of tools and fixtures. The design work proceed generally from the information developed in the operations analysis on the process sheet.

3). Planning, specifying or designing new production and plant facilities.

4). Planning the quality control system.

5). Planning for production personnel i.e. job specifications are developed, standard times determined and labour costs are estimated.

6). Planning for production control i.e. work schedules and inventory controls are evolved. Standard costs for labour, material, and services, are established and integrated with the accounting system.

7). Planning the information-flow system necessary for transmission of instructions and provision of feed-back for control is determined. Appropriate forms and records are designed, flow pattern and routings are also established.

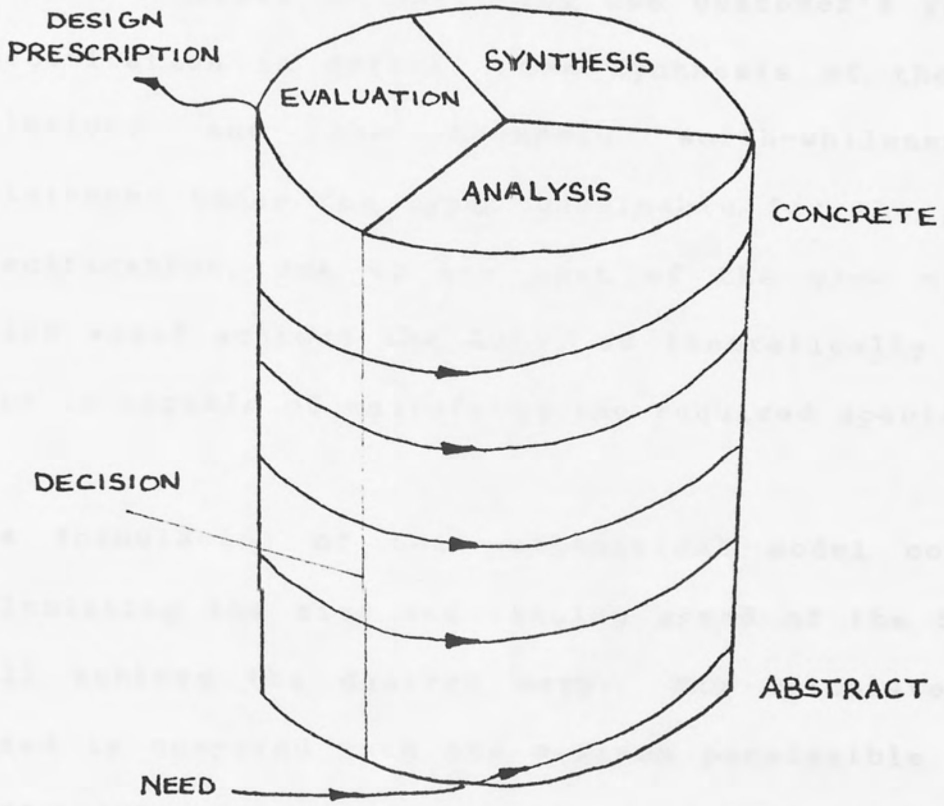
8). Financial planning, because large sums of capital are required to initiate production of a new product. The source of financing must be carefully established, and the means and rate of recovering the capital determined.

3.1.3.5. THE DESIGN PROCESS IN PRACTICE

From the foregoing it is seen that the design phases are clearly defined. Ideally, a design would pass through each of these phases once, but in practice it is often necessary to feed back to an earlier phase to crystallise or improve the ideas. As these phases are being pursued, the main activities of the designer are analysis, synthesis and evaluation. Mesarovic [5] suggest the model of the design process as shown in Fig. 3.2. In this model, the designer repeatedly circulates the cylinder gaining greater and greater confidence in his concept, until they are sufficiently concrete to form a design prescription. Again in practice, not all designs follow a steady upward trend, indeed, poor designs should never reach the top of the cylinder. The model suggests that a systematic design method might be achieved by controlling the progress of the line on the cylinder, perhaps to avoid sudden drops.

3.2. THE DESIGN PROCESS APPLIED TO THE FAN SELECTION PROCESS

Selecting a fan for a specific duty passes through most of



MESAROVIC'S MODEL OF THE DESIGN PROCESS

Fig 3.2

the design phases discussed in the previous section. The need is established by the client when he sends out a request to tender for a suitable unit to meet his requirements. The identification and formulation of the problem consists of analysing the customer's requirement specification in detail. The synthesis of the possible solutions and the economic worth-whileness phase eliminates those fan types unsuitable for the particular specification, due to the cost of the size of the fan which would achieve the duty, as theoretically any blade type is capable of satisfying the required specification.

The formulation of the mathematical model consists of calculating the size and running speed of the fans which will achieve the desired duty. The calculated running speed is compared with the maximum permissible speed for that particular type and size. The maximum speed is established as a result of the stress analysis procedure (which is also a part of the formulation of the mathematical model phase) carried out as part of the initial design process, where the various dimensions of the fan impeller are fixed for that particular size. Having determined a possible suitable size, the compatibility analysis consists of establishing if the size will fit into the available space and with the rest of the system in which the fan will operate.

The examination of the location of the duty point on the fan characteristic (volume-pressure) consists of the stability analysis, since a certain portion of the characteristic (the unsteady flow region shown in Fig. 4.1) is considered unsuitable for operation because the flow there is is unstable. The testing of the design has usually been done during the development of the fan range when certain of the design parameters are fixed (e.g. blade type, blade angle, impeller diameter, impeller width etc.). For the fan range to be competitive in the market, the initial design goes through simplification stages so that final design is as competitive as possible.

When the enquiry results in an order, the detailed design phase of the design process begins. Drawings of the overall system (i.e. layout drawings of the fan-motor and ancillary equipment) and the sub-system (i.e. impeller, casing, shaft etc.) are prepared. Drawings of the items to be manufactured in house are prepared for the production department. Specifications of the manufactured items to be brought in (e.g. motor, bearings, pulleys, belts etc) are prepared for the purchasing department. Production schedules are drawn up for the assembly to be manufactured, and the materials required purchased. Finally the assembly is manufactured and despatched to the customer.

3.3. INDUSTRIAL MARKETING

The fan selection aspect of the project is concerned with the marketing of the product produced by the company. Since the sponsors cater solely for the industrial market a brief review of industrial marketing was carried out, as marketing of the product is part of the design process, which the designer has to be aware of when designing the product.

Industrial marketing is the marketing of goods and services to industrial and institutional customers. These include manufacturing companies, government departments, public utilities, educational institutions, wholesalers, retailers and other formal organisations. Consumer marketing, in contrast, is concerned with marketing to individual families, and households purchasing goods and services for their own consumption. The distinguishing feature of industrial customers is that they use purchased goods and services in their own production of goods and services. Purchased products such as raw materials, components, and sub-assemblies, may become part of the customer's final product or may be added to physical facilities, in the form of construction and equipment. In other cases, the purchases may be for supplies used in operations, repair and maintenance activities.

Industrial marketing is distinguished from consumer marketing more by the nature of the customer than by the nature of the product. That is, industrial customers tend to be relatively few in number for any given supplier compared with the size and scope of consumer markets. Individual transactions, in contrast, typically have a much higher value. Compared with consumer decision making, industrial buying behaviour is characterised by the participation of many people interacting with each other in the context of a formal organisation. The buying decision process therefore takes a long time and maybe more highly structured (but not necessarily more "rational") than the consumer buying decision process. Complexity in the buying decision process is also reflected by the following factors:-

1). The complex technical and economic factors that must be considered.

2). The environment in which the company operates.

3). The large sums of money involved in the transactions.

Industrial buying is a complex organisational decision making process, rather than a single purchase event which takes time to make. In view of the time delay between the initial enquiry and the subsequent order, evaluation of the computer aided selection system in assessing its success or failure would have to be on a long term basis. That is, if the written enquiries are answered within the period indicated by the enquiry response analysis, do they subsequently result in firm orders, would have to be monitored over a considerable period.

Another aspect of industrial marketing uniqueness is that of technical product complexity. The major barrier to a true marketing orientation in the company remains excessive product, engineering, manufacturing and technical orientation. One class of business strategies available to industrial companies calls for a high degree of technical innovation and risk taking with related high expenditures for research and development. In such companies, top management is likely to have grown in the engineering and research areas, and technical values maybe predominant in management decision making. The real risk in these cases is becoming so attached to the technical accomplishment that the necessary flexibility for responding to customer needs in a competitive market place disappears. As a result, one of the most common marketing sins committed is that of trying to change the customer or his requirements to fit the product. Corey [6] dealt with this problem by observing that in industrial marketing strategy, the product must always be regarded as a variable, not as a constant. He offers the following as the key concepts for understanding the nature of industrial market selection and product planning:-

1). The basic and most important decisions in planning marketing strategy are those related to the choice of a market or markets to serve (i.e. market segmentation). In the case of fans, the size range to cater for i.e. small, medium, large, or "one off".

2). The form of the product is a variable, not a constant, in developing marketing strategy. Products are planned and designed to serve customers, i.e. various options must be evaluated and the best one selected to serve the needs of the particular customer. In the case of the sponsors, this means evaluating all the possible types and sizes of fans which will achieve the customer's requirement, and recommending the best type and size, and not of deciding one particular size and recommending it as is the case at present. This will have to be the main requirement of the computer aided selection system, i.e., of listing all the possible types and sizes manufactured by the sponsoring company, that achieve the particular requirement, so that the best type and size(s) can be offered to the customer.

3). Not only is the functional utility of the product important, but the total package of benefits the customer receives when he enquires or buys. Corey points out that these include, the supplier's reputation for the technical assistance the customer receives, the assurance of dependable supply, the product and service.

From what has already been said about the nature of industrial customers and products, it is clear that the buyer-seller interdependence is indeed a hallmark of industrial marketing, especially for products used in the customer's operations as is the case with fans, which is usually a small part of say a new plant being built by the customer. In such cases the buyer becomes crucially dependent on the supplier for:-

1). An assured supply of assemblies.

2). Continued supply of maintenance and repair parts and skilled repair service for capital equipment.

3). Efficient order handling and delivery.

4). Extension of credit.

5). Technical support.

The technical support aspect is the part of the service which is lacking with the present manual enquiry response procedure. As a result of the pressures on the technical sales staff in answering the initial enquiry, the full amount of the information requested by the customer may not be supplied to him, due to the time taken to accumulate the information. There is no form of "follow up" procedure to contact the customer once the enquiry has been answered. It was this lack of technical assistance to the prospective customer, that prompted the company to review the manual selection process with a view to reducing the time spent on the initial enquiry, but still provide the customer with all the information he requests, so that additional technical assistance can be provided to the prospective customer. This improvement in service is hoped will generate more business for the company.

In conclusion, the hallmark of industrial marketing is providing good "back up" service to the product being marketed. The aim of the sponsoring company in the development of the computer aided selection system is to improve on the service being provided at present with the existing level of staff. This will become even more important in the larger size market, where unit costs are much higher. Knowing exactly what the customer wants and selecting the best alternative will be very important, as any mistakes maybe very costly to rectify.

3.4.THE CURRENT PRACTICE OF FAN SELECTION

As the author had no previous experience with the fan selection process, before any work could commence on the development of the computer aided selection system, the current practice was reviewed. The process varies from company to company to a certain extent and as other companies were not willing to discuss their practice, the method discussed is that used by the sponsors. All the competitors however did confirm the selection process was carried out manually without any form of computerisation.

In the majority of fan applications it is neither necessary nor desirable to design a completely new fan for a specific requirement. Standard designs in numerous sizes are available in each of the various blade types, produced by the manufacturer, suitable for a wide range of applications. Fan selection is, therefore usually a matter of choosing the best type and size for the particular application. Fan selection is a procedure which begins with the specification of requirements and ends with the evaluation of alternative possibilities. Of the many fans which maybe capable of satisfying a particular capacity and pressure requirement, the best selection is the one that does the job most economically. Capital, operation and maintenance costs are all costs that need to be considered.

3.4.1. SPECIFICATION NECESSARY FOR FAN SELECTION

A fan specification should give the fan manufacturer all the relevant information regarding performance, operating environment, drive arrangement etc., so that the best selection can be offered. A typical specification requirement sheet is shown in Fig. 3.3. The type of service for which the fan is intended should be specified to inform the manufacturer of any unusual conditions. In some instances a duct layout maybe included with the specification. The capacity of the fan must be specified by the client. Due to the nature of the application, the client may find it convenient to calculate the capacity as a weight rate of flow. Fan capacity is usually expressed as a volume rate of flow and is commonly specified in cubic feet per minute at inlet conditions. If an adjustable flow is contemplated the maximum, minimum, and any intermediate capacities at which power requirements are to be evaluated should also be specified.

CLIENT .... CLIENT'S REF. ....

CONTACT .... OUR REF . ...

DATE .... TELE. NO. ....

SPECIFICATION ....

....

....

ALTITUDE .... GAS HANDLED .... DENSTY ....

ITEM NO .... 1 .... 2 .... 3 ....

VOLUME INLET ....

PRESSURE/SUCTION ....

TEMPERATURE ....

EQUIVALENT PRESSURE ....

FAN SIZE ....

FAN TYPE ....

ASSEMBLY NO. ....

R.P.M. ....

EFFICENCY ....

POWER ABSORBED ....

MOTOR POWER ....

ELECTRICITY SUPPLY ....

COMMENTS:

....

....

....

FAN SPECIFICATION / ESTIMATION SHEET.

FIG 3.3

The fan must provide the air or gas with sufficient energy to overcome the losses encountered in passing through the system. The usual method of specifying this energy requirement is to stipulate the static pressure in inches water gauge which the fan must develop at each capacity. If the fan total pressure is specified, a clear statement of the fact should be given so as to avoid giving the impression that the gross static pressure is being specified. In any case the distinction between suction and discharge pressure should also be specified. When a fan is located at the downstream end of the system (i.e. exhausting), the static pressure required of the fan is equal to the sum of the total pressure losses in all of the system components, plus any difference in total pressure that may exist between the two ends of the system. When the fan is located at any position other than the downstream end, i.e. blowing or boosting, an amount equal to the fan outlet velocity pressure should be subtracted from the requirement as stated above. The fan static pressure **(FSP)** may, therefore, be expressed in terms of the sum of the total pressure losses **(TP_{losses})**, the difference between the total pressure at the system exit **(TP_x)** and the total pressure at the system entrance **(TP_e)** and the fan velocity pressure **(FVP)** ie:

$$FPS = TP_{losses} + (TP_x - TP_e) - FVP \quad ....3.1$$

The term **FVP** is zero when the fan is located at the downstream end of the system. However, the fan velocity pressure **(FVP)** depends on its size and since this is not usually known in advance the term is usually ignored regardless of the fan location. Improved accuracy when necessary, can be obtained by using the **FVP** for a trial size and corrected accordingly to the actual value when the final size has been selected. The sum of the total pressure losses through the system should include an allowance for any elements required to connect the fan to the system. This is usually negligibly small compared with the other quantities, unless the size of the fan opening differs greatly from the size of the connecting ductwork. Usually there is no difference in the total pressure between the entrance and exit of the system. The exception is when there is more than one device supplying energy to the air.

The performance of a fan is a function of the density of the air or gas at the fan inlet. The inlet density not only determines the volumetric capacity for a specified weight rate of flow, but the pressure which the fan is able to develop as well. The factors which affect the density and therefore need to be specified are the barometric pressure, temperature and relative humidity at the inlet, as well as the composition of the gas being handled. Information about any entrained material (e.g. dust loading etc) should also be specified, so that the manufacturer can offer the best blade type for that particular type of environment based on any previous experience.

If the gas composition and conditions are not specified standard conditions are assumed. Standard conditions for the fan industry are dry air at 20° C (68° F) and 30" Hg barometric pressure.

Certain other items in the specification maybe given by the customer because he is in a better position than the manufacturer, e.g. the material from which the fan is to be manufactured maybe specified, because the customer has experience of using such materials in the environment in which the fan is to operate. Items such as the rotation and the discharge angle have no influence on the cost of the unit. Others, such as the type of drive (direct or indirect) and ancillary equipment (e.g. flanged inlets, inspection door, drain plug etc) will have a significant influence on the capital cost but has no effect on the size and type of fan which should be selected. Still other items, like the number of inlets and the type of drive, determine the size and type of fan which should be selected.

3.4.2. SELECTING A FAN FOR A SPECIFIC DUTY

Theoretically any size and type of fan could conceivably be used to satisfy the requirements of a particular specification. Practical engineering and economic considerations reduce the possibilities to a relatively narrow range of sizes and a few types. The suitability of a particular type of fan depends more on the relationships between the various performance requirements than on their exact values. This is particularly true if the speed is specified (e.g. directly coupled fan). In such cases the size necessary to achieve the duty maybe calculated, and the types of fans which exhibit a reasonable (economic) efficiency at this condition is calculated.

Certain types or designs of fans are designated according to their usual field of application. There are ventilating fans, mechanical draft fans, industrial exhausters and pressure blowers with sub-classifications in each case. Ventilating fans are designed for clean air service at normal temperatures. Some heavy duty ventilating fans maybe used for more severe conditions. Centrifugal fan types may have backward or forwardly curved blades, backward or forwardly laminar blades, or radially straight blades. Maximum efficiencies (as defined by Equation 2.1) are obtained with backward type, particularly when the blade is aerofoil shaped in cross-section. Forwardly curved blade types are used when space and capital purchase price are more important than efficiency. Belt drives are normally used so that a particular duty can be obtained using a standard fan size.

Mechanical draft fans are those which are designed for forced draft, induced draft, gas recirculating, primary air and similar service. Mechanical draft fans have similar characteristics to heavy duty ventilating fans. Temperatures and dust loadings maybe comparatively high in such applications.

Industrial exhausters are designed for various kinds of industrial service. In most cases efficiency is sacrificed for the sake of simplicity and ruggedness. Belt drives are usually employed in these applications. In many highly corrosive or abrasive environments, the impeller and other parts are considered expendable. In other cases a considerable effort is made to extend the life of such parts by using special materials. Whenever "stringy" material must be passed through the fan, a centrifugal type with "cone" or "open" impeller should be used. The root (heel) of the blade should be shaped so that the material will slide off due to the centrifugal force. When sticky materials are to be handled, the internal surface should be minimised as is done in most axial types designed for paint spraying boots.

Pressure blowers are designed to withstand the high tip speeds required to produce high pressures. The impeller is frequently mounted directly on the motor shaft. If not, some other form of direct drive is usually used. Therefore, numerous designs are required to provide maximum efficiency for all ratings.

After determining the type or types of fans that are suitable for an application, it is necessary to determine the best size in each type. There is only one fan size in each type that will operate at the point of maximum efficiency for any given rating. This optimum size fan must be operated at a certain speed to produce the required rating. A smaller size fan could be selected that would have to operate at a higher speed, or a larger size fan selected that would operate at a lower speed. In either case the efficiency would be lower than for the optimum size. Fans which rate at the right of the peak efficiency (i.e. where the rated volume is greater than volume at peak efficiency) are called under-size fans, while those to the left of the peak efficiency (i.e. where the rated volume is less than the volume at peak efficiency) maybe called over-sized fans. Over-sized fans are hard to justify unless future increases in capacity are contemplated. Occasionally, the required operating speed of an over-sized fan will match a motor speed. Slightly under-sized fans are usually selected because optimum sizes are rarely standard ones. Sometimes a fan which is considerably under-sized will be the best choice (i.e. when customer is limited by the size of space available for the fan to fit into with the rest of system). In such cases, the additional operating costs occasioned by the lower efficiency must be offset by the saving in the capital cost, or some other engineering or economic factor. Such a size would only be recommended if the customer cannot relax his specification e.g. the size of the fan cannot be changed due to the rest of the system in which the fan will operate.

3.5. STRESS ANALYSIS OF CENTRIFUGAL FANS

Having reviewed the design and the fan selection processes, the next area reviewed is that of the stress analysis of the centrifugal fan impeller, the other main area of this research. The stress analysis procedure presently used by the sponsoring company has already been described in chapter 2. Again, as with the selection process the competitors were not willing to discuss the methods of stress analysis of the impellers currently used by them.

However, two references were found that dealt specifically with the type of impellers manufactured by the sponsoring company. The first by Patton [7] deals with an experimental analysis. Patton presents results of strain gauge and brittle lacquer tests on three impellers of different sizes, and running at different speeds. These results are compared with the stresses predicted by the Haerle [8] chart method. However, it should be emphasised that there is no attempt at an analytical analysis of the blades and that there is poor correlation between the experimental and analytical results. This is particularly noticeable at the critical points of high stress areas on the inlet cone and backplate. For both the backplate and conesheet on the outer surface this was just above the nose (tip) of the blades, and on the inner surface just below the tail (root).

Patton also states that attempts were made using thin plate theory [9] to predict the stresses in the blades but this met with very little success. This was due not only to the complexity of the geometry of the impeller, but also due to the complex interaction of the backsheet and conesheet with the blades. He concludes that this strengthens the case for a more sophisticated way of analysing the entire structure e.g. the finite element method.

The second by Bell [10], is the first real attempt at analysing the impeller as a complete unit, using the finite element method, and comparing the analytical results with those obtained experimentally using strain gauges and brittle lacquer. In this work, in an attempt to understand the complex stress distribution in the impeller, he follows a step by step approach. First he treats the backplate and conesheet separately, and then examines the interaction between these two components and the blades. The conesheet was first simulated as a truncated thin cone, fixed at its inner edge and analysed using a classical elasticity approach, the finite element method and strain gauges. In the next stage of his step by step approach, a simplified model of the impeller was studied, where the backplate and the conesheet were connected by six thin rods simulating the blades.

Finally, a complete commercially produced impeller was analysed using the finite element method, strain gauges and brittle lacquer. The main disadvantage of the brittle lacquer technique applied to rotating components is that no quantitative results can be obtained because of the difficulties in maintaining strict control of the test conditions. However, in this particular case, the technique did give valuable insight into the general stress pattern in the impeller. Using high sensitivity lacquer (ST 70) the directions of the principal stresses could be seen, whilst with the low sensitivity lacquer (ST 75) just the areas of high tensile strain were revealed. Using these results, strain gauges were placed at the appropriate locations, so that quantative results could be obtained.

The results obtained using the finite element method for the analysis of the blades were on the whole disappointing, because it was necessary to analyse the blade as part of the integral structure and, therefore the element mesh of this component was quite coarse. In these circumstances, according to Bell, a comparison of the finite element and strain gauge data is unrealistic. The element used in this work was the plane triangular shell element described by Zienkiewicz **[11]**, in which the bending and membrane behaviour are separated. The element has 18 degrees of freedom i.e. 3 displacements and 3 rotations per node. The stiffness matrix of this element is obtained by a combination of the plane stress triangular element **[11]** and the triangular bending element of Bazeley et al **[12]**. The element is attractive since the only geometric input required is the cartesian coordinates of the three vertices. However, the element is a relatively poor performer according to Knowles et al [13], at times notoriously so. In general, they conclude that, adequate performance can be obtained when the sought response is either predominately membrane or bending. However, when there is a strong coupling between membrane and bending (as in the case of the centrifugal fan) it is extremely poor. However, the performance can be enhanced slightly with the refinement suggested by Irons and Razzaque [14] which employs the concept of derivative smoothing of the rational functions.

As these were the only two references found, dealing specifically with the stress analysis of the types of centrifugal fan impellers manufactured by the sponsoring company, this then is the present state of the art in the stress analysis of the centrifugal fan impeller. The fan impeller can be regarded as an assembly of plates and shells. However, due to the complex geometry and the interaction of the three main components of the impeller, classical plate and shell theory cannot be applied in this case. In such instances, one has to resort to using an analytical solution technique, such as the finite element method. The technique has the advantage of being able to deal with structures with complex geometry.

Due to the rapid development in micro electronic technology and with the advent of low cost mini computers, the finite element technique has emerged as a powerful tool in the stress analysis of modern complex structures.

Before discussing the development of the finite element method of stress analysis as applied to plate and shell type structures, a brief summary of the relevant aspects of plate and shell theory is given.

3.6. PLATE AND SHELL THEORY

3.6.1. PLATE THEORY

The theory of plates can be categorised into two classes:-

1). Kirchhoff plate theory in which shear deformation is ignored.

2). Mindlin (or Reissner) plate theory which includes shear deformation, thus allowing analysis of thick plates.

The theory is based on several assumptions similar to those in the analysis of beams. These maybe stated as follows:-

a). The lateral deflections (Fig. 3.4) are small in comparison to the plate thickness.

b). The middle surface of the plate remains unstrained during bending and is consequently a neutral surface (this assumption is not valid if large displacements are allowed).

c). Normals to the mid-surface before deformation remain straight but not necessarily normal to the mid-surface after deformation. This is the "Mindlin" assumption which allows transverse shear deformation to be included in the theory. If the shear deformation is neglected (i.e. normals before deformation remains normal to the deflected mid-surface after deformation), the classical "Kirchhoff" thin plate theory is recovered.

d). Normal stresses in the transverse **(z)** direction of the plate are small in comparison to the in-plane stresses and hence neglected. (This is unreliable even for thin plates in the vicinity of a concentrated transverse load.

e). The material is homogeneous, isotropic and linearly elastic. This permits a simple description of the stress-strain relations in terms of Youngs Modulus **(E)** and Poisson's ratio **(ν)**. These conditions could be relaxed, but in this work, they are considered to be valid.

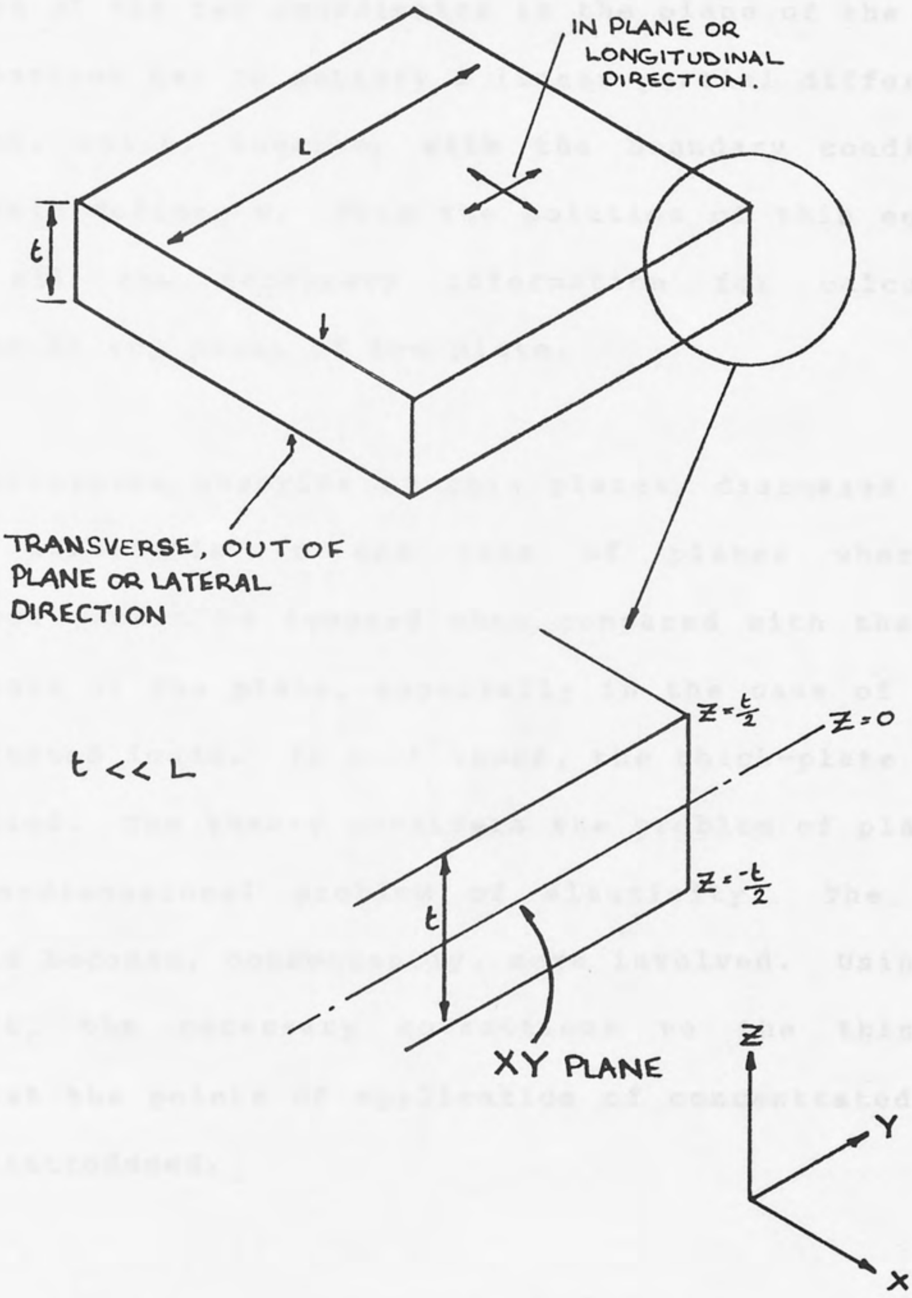


PLATE THEORY DIRECTION TERMINOLOGY

Fig 3.4

Using these assumptions, all the stress components can be expressed by the deflection **w** of the plate, which is a function of the two coordinates in the plane of the plate. This function has to satisfy a linear partial differential equation, which, together with the boundary conditions, completely defines **w**. Thus the solution of this equation gives all the necessary information for calculating stresses at any point of the plate.

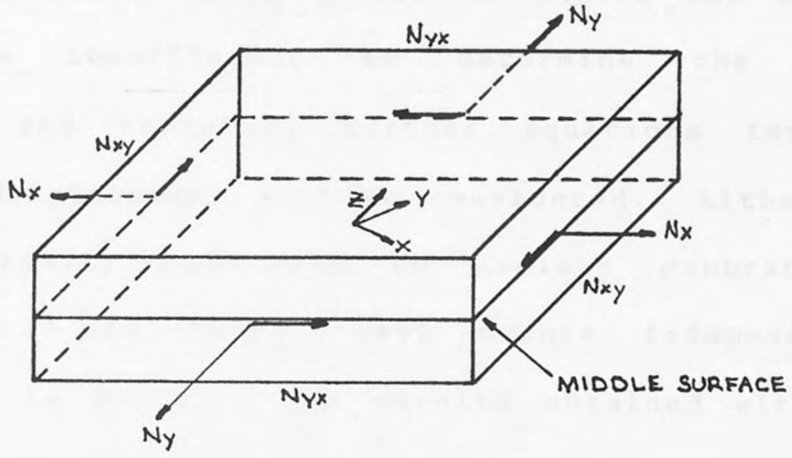
The approximate theories of thin plates, discussed above, become unreliable in the case of plates where the thickness cannot be ignored when compared with the other dimensions of the plate, especially in the case of highly concentrated loads. In such cases, the thick-plate theory is applied. The theory considers the problem of plates as a three-dimensional problem of elasticity. The stress analysis becomes, consequently, more involved. Using this analysis, the necessary corrections to the thin-plate theory at the points of application of concentrated loads can be introduced.

3.6.2. SHELLS

A shell is defined as the solid material between two doubly curved surfaces. The thickness (t) of the shell is the distance between these two surfaces and the mid-surface is defined as the loci of points which lie equidistant between the curved surfaces. Again, as with plates, it is necessary to differentiate between thin and thick shell. A thin shell is one in which the ratio of the thickness (t) to the radius of curvature (R) i.e. (t/R), can be neglected in comparison with unity. In the case of thick shells this ratio cannot be ignored. Again, as with plates, it follows from this that the theory of thin shells is much simpler than the theory of thick shells. In practice a shell which has a ratio of ($t/R < 1/20$) can be considered a thin shell.

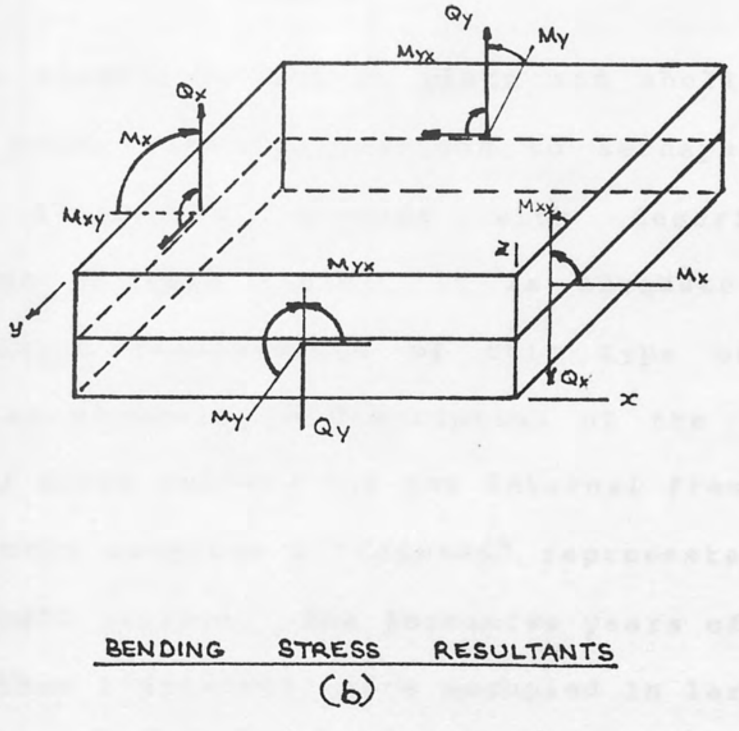
In some of the literature, shells are often described as curved plates, as they bear the same relationship to plates as curved beams do to beams. In fact it was from the theory of plates that the theory of shells was first derived. The Kirchhoff theory of plates was first applied to shells by Love [15] in 1888 and the resulting theory is known as Love's first approximation to the theory of thin elastic shells. The theory is based on the same assumptions as those in the theory of thin plates. There exists, however, a substantial difference in the behaviour of plates and shells under the action of external loading. The static equilbrum of a plate element under the action of a lateral load is only possible by action of bending and twisting moments accompanied by shearing forces, while a shell, in general, is able to transmit the surface load by membrane stresses, which act parallel to the tangential plane at a given point of the middle surface and are distributed uniformly over the thickness of the shell. This property of shells makes them, as a rule, a much more rigid and more economical structure than a plate would be under the same conditions. When considering membrane action it is assumed that the shell is incapable of withstanding any bending moments and that the external loads are carried by internal forces induced in the surface of the shell. Therefore, in deriving a membrane theory for shells, only the elementary laws of statics are involved. It is general practice to define the forces acting on the sides of a shell element as forces/unit length, and these "forces" are termed stress resultants. The membrane stress resultants N_x, N_y, N_{xy} for a shell as shown in Fig. 3.5 (a).

Membrane stress resultants are insufficient to define the true elastic behaviour of a shell and therefore bending action must also be considered. The application of external forces must give rise to bending of the shell and such bending can only be resisted by internal moments and forces induced in the shell. Again, it is general practice to define the moments acting on the edges of a shell element as moments/unit length and they are termed



MEMBRANE STRESS RESULTANTS

(a)



BENDING STRESS RESULTANTS

(b)

STRESS RESULTANTS

Fig 3.5

bending stress resultants M_x, M_y and M_z as shown in Fig. 3.5 (b). In the bending theory of shells the laws of statics are insufficient to determine the stress resultants, and therefore further equations involving strain and displacement must be considered. Although it is mathematically incorrect to isolate membrane and bending action as though each occurs independently, nevertheless in practice the results obtained with this procedure agree well with the exact results.

3.6.3. THE EVOLUTION OF FINITE ELEMENTS IN STRESS ANALYSIS

The finite element method in plate and shell structural analysis, began with applications to aerospace vehicles and the literature abounds with descriptions of applications in this field. It is adequate to form a finite element idealisation of this type of structure using planar elements in description of the sheet metal and beam or axial members for the internal framework. The plate elements comprise a "faceted" representation of the aircraft shell surface. The formative years of the finite element method (1950-1960) were occupied in large measure, with the continual refinement of such idealisations. The early 1960's saw the shift from aircraft to space vehicles in the aerospace industry and with it a change to highly-curved unstiffened thin shell structures. However, limitations and shortcomings of the "faceted" form of finite element representation of shells using plate elements were soon perceived. This focused attention on the need for curved thin shell finite elements, whose formulation required a new examination of modes of element geometric representation, strain-displacement equations and assumed displacement fields.

Initially displacement fields which had previously been used for flat plate elements were adopted, but such an approach was found to be inadequate for realisation of an acceptable level of solution accuracy. By 1969 a large number of curved thin shell elements had appeared in the literature. A comprehensive survey of plate and shell elements was carried out by Gallagher [16]. These encompassed a large proportion of the ideas which still hold promise for acceptable formulation. Irons [17] motivated by a deep dissatisfaction with almost all known shell elements developed the Semi-Loof general purpose shell elements. These elements are general curved triangles and curved quadrilaterals with degrees of freedoms at the corners, midsides and the Gauss points (the so-called "Loof" nodes) along the edges. They are essentially degenerate isoparametric solid elements specialised, by modifying the three dimensional elasticity matrix, to give zero pinching stresses and by imposing zero transverse shears at discrete points and boundaries. An important feature of the element is the fact that the normal rotations are interpolated (quite independently of the translations) only at the Loof nodes. After the rotations at the Loof nodes have been eliminated (by imposing the normality hypothesis in the form of shear constraints) a very elegant, yet practical, element is obtained, capable of representing shells with junctions and discontinuities. The degrees of freedom are the three cartesian translations, X,Y,Z, at each of the corner and midside nodes and the normal rotations θ_1 and θ_2 at the two Loof nodes on each side (for identification purposes the two rotations are treated as midside variables giving effectively five degrees of freedom at the midside node). The geometric input consists simply of the cartesian coordinates of the corner and midside nodes. The curved boundaries are described in isoparametric form. The element is thus simple to used for practical problems and can be used with membrane elements and a compatible curved beam element. The principle advantages of the Semi-Loof element compared with other general shell elements are:-

1). Sharp corners and multiple junctions are allowed.

2). Loof nodes avoid both basic inconveniences of plate elements, i.e. singularities at corners and high order nodal variables.

3). Discontinuities in element thickness are allowed from element to element.

4). The problem of slope conformity at corners in shells is avoided.

5). The problem of a third in-plane rotation is avoided.

6). The elements maybe joined to plane two dimensional quadratic elements.

7). The Semi-Loof element degenerates to a plate element when the curvature is zero. Hence the shell element can be used as a plate element.

In view of the above advantages of the Semi-Loof element in representing the behaviour of shells and since the centrifugal impeller can be regarded as an assembly of plates and shells, it was decided to use this element in the analytical stress analysis of the impeller. It is hoped that this work will provide a basis for computer aided stressing package, giving results of the stress distribution in the impeller which are in good agreement with those obtained experimentally on a commercially produced impeller, and thus giving more detail than is currently available. The detailed information will improve the understanding of the complex interaction of the main components of the impeller, resulting in the design of high performance fans, and also giving the sponsoring company more confidence in the designing of larger units than currently being produced. The development of this aspect of the research is discussed in Chapter 5.

CHAPTER FOUR

DEVELOPMENT, IMPLEMENTATION, AND EVALUATION OF THE COMPUTER AIDED SELECTION SYSTEM

INTRODUCTION

Chapter 4 deals with the development, implementation and evaluation of the computer aided selection system. The chapter commences with a brief introduction to the centrifugal fan characteristics and the "fan laws" which can be applied to geometrically similar fans.

Section 4.2. discusses the present manual selection system, leading to section 4.3 which deals with the requirements of the proposed computer aided selection system, as a result of discussions with senior management and the technical sales staff.

Section 4.4. deals with the development of the computer aided selection system. Section 4.4.1. discusses the procedure developed for setting up the data base and section 4.4.2. discusses the development of the fan selection program itself.

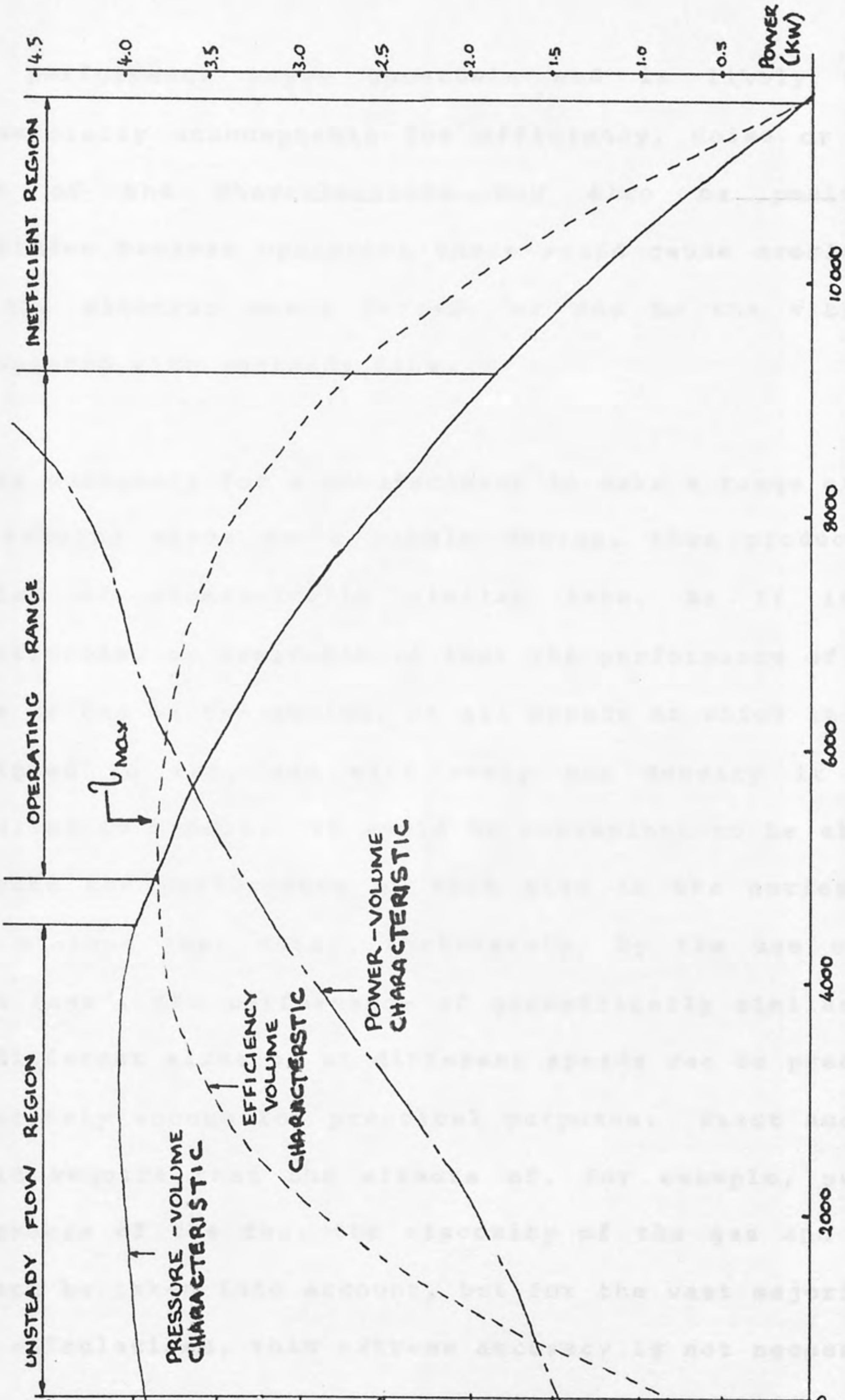
The decisions concerning the acquisition of the computer hardware are discussed in section 4.5, and section 4.6 is concerned with the implementation of the computer aided selection system.

The criteria to be used, in assessing the success or failure of the implemented system is discussed in section 4.7, with section 4.8 discussing the results of the evaluation.

4. 1. FAN CHARACTERISTICS AND THE FAN LAWS

The performance of a fan cannot be adequately described by single values of pressure rise or volume flow, but they have a fixed relationship with one another. This relationship is best defined graphically in the form of a fan characteristic. Fig. 4.1 is an example, the volume flow rate is always plotted along the base, with the fan pressure (and other performance quantities e.g. power, efficiency) as the ordinates. One point on the characteristic can always be found at which the efficiency of the fan is a maximum. This is called the best efficiency point -η_{max} on the diagram. As well as providing the lowest power consumption for a given duty, operation at this point, experience has shown, usually secures the lowest noise level for that particular fan.

However, the fan can be operated at other points on the characteristic. The region of satisfactory operation is defined by the manufacturer as the operating range and such a range is indicated on Fig. 4.1. Outside this range



FAN CHARACTERISTICS

fig 4.1

the performance maybe uncertain and is likely to be commercially unacceptable for efficiency, noise or cost. Part of the characteristic may also be positively forbidden because operation there would cause overloading of the electric motor fitted, or due to the vibration associated with unsteady flow.

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. As it is not practicable, or desirable to test the performance of every size of fan in the series, at all speeds at which it maybe designed to run, and with every gas density it maybe required to handle. It would be convenient to be able to compute the performance of each size in the series from the minimum test data. Fortunately, by the use of the "fan laws", the performance of geometrically similar fans of different sizes or at different speeds 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.

The variables involved, when using the "fan laws" are the fan size (S), rotational speed (N), gas density (ρ), volume (Q), pressure (P), power (W) and efficiency (η).

The fan laws are the mathematical expressions of the fact that when two fans are geometrically similar, their performance curves are similar. That is, at similarly situated points of operation, efficiencies are equal. The ratios of all the other variables can be shown by dimensional analysis [18] to be inter-related as follows :-

1). For a change in speed (N) at constant size (S).

$$\text{Volume} \propto N \qquad \text{..........(4.1)}$$

$$\text{Pressure} \propto N^2 \qquad \text{..........(4.2)}$$

$$\text{Power} \propto N^3 \qquad \text{..........(4.3)}$$

2). For a change of size (S) at constant speed (N).

$$\text{Volume} \propto S^3 \qquad \text{..........(4.4)}$$

$$\text{Pressure} \propto S^2 \qquad \text{..........(4.5)}$$

$$\text{Power} \propto S^5 \qquad \text{..........(4.6)}$$

3). For a change in density (ρ)

$$\text{Volume} = \text{constant} \qquad \text{..........(4.7)}$$

$$\text{Pressure} \propto \rho \qquad \text{..........(4.8)}$$

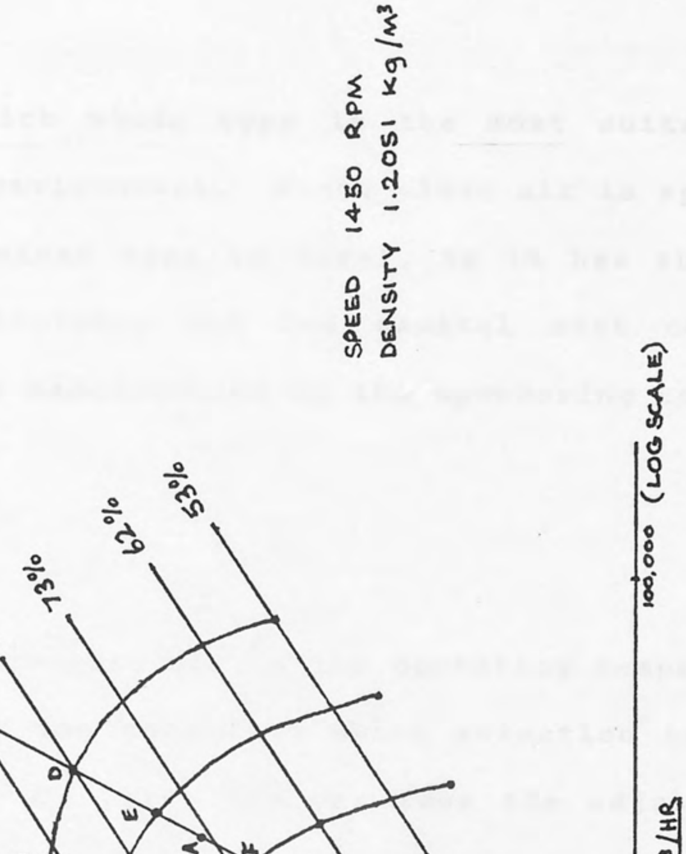
$$\text{Power} \propto \rho \qquad \text{..........(4.9)}$$

Before the "fan laws" can be used to compute the performance of geometrically similar fans in the series, it is necessary to have test data for a fan in the same series. In the "fan laws" any convenient dimension maybe used for the size (S) ratio, usually its the diameter of the fan impeller that is used.

4.2. THE PRESENT MANUAL SELECTION PROCEDURE

In the present manual selection procedure, the most common form of selection chart prepared, is that of the sizes in the series at a constant speed (e.g. 1450 r.p.m.). Part of such a chart for the backward laminar (BL) range is shown in Fig. 4.2, at a density of 1.2014 Kg/M^3 (i.e. the density of air at 20^oC, and at standard atmospheric pressure). The charts are usually prepared on log-log scales, showing only the operating ranges of each fan. The fan selection process involves a step by step approach, which is best explained by means of an example.

Consider a fan required to deliver 40000 M^3/Hr of clean air at a pressure of 200 mm water gauge, operating at 100^oC.



FAN SELECTION CHART

Fig 4.2

Step I

Deciding which blade type is the most suitable for the particular environment. Since clean air is specified, the backward laminar type is ideal, as it has the advantages of high efficiency and low capital cost compared with other series manufactured by the sponsoring company.

Step II

Adjust the pressure due to the operating temperature being 100^{o} C while the data from which selection is to be made being at 20^{o} C. Using the gas laws the adjusted pressure is given by:-

$$P_{adjusted} = \frac{P_{required}\ (273+T_{operating})}{(273+T_{test\ data})} \qquad \text{.........................(4.10)}$$

Equation 4.10 gives the adjusted pressure to be 255 mm water gauge. If the operating altitude of the fan had been above or below sea level, then a similar correction would also have been required.

Step III

Locate the duty point (40000 M^3/Hr., 255 mm water gauge) on the selection chart, point A, Fig. 4.2. As the duty point does not lie exactly on the performance curves we use the fan laws to interpolate the operating speed and efficiency of each fan at the duty point. This is done by drawing the system pressure loss-volume flow relationship defined by:-

$$P \propto Q^2 \qquad \text{..........(4.11)}$$

through the duty point, line BC in Fig. 4.2. The operating speed and the efficiency at the duty point for a particular size are given by:-

$$\text{Operating speed} = \frac{\text{Curve speed} \times \text{Duty Volume}}{\text{Volume at intersection of performance curve with the line BC.}} \qquad \text{.....(4.12)}$$

$$\text{Efficiency } (\eta) = \text{constant} \qquad \text{..........(4.13)}$$

In the example, the line BC intersects the performance curve for the 1100mm, 1000mm, 900mm, 800mm diameter fans at points D, E, F and G respectively. Using equation 4.12 this gives operating speeds of 1150 rpm, 1333 rpm, 1589 rpm and 1966 rpm for the 1100mm, 1000mm, 900mm and 800mm diameter fans respectively.

Step IV

The operating speed necessary for the fan to achieve the duty is then compared against the maximum speed permitted for that particular size. Where the operating speed is greater than the maximum permitted, the size is considered unsuitable for the particular case. In this instance, the computed operating speed for each of the above fans is well within the maximum permitted for that size.

Step V

Calculate the power absorbed by each fan according the following expression:-

$$W \text{ (Watts)} = \frac{\text{Duty volume } (M^3/S) \times \text{Pressure } (N/M^2)}{\eta} \quad \ldots\ldots\ldots\ldots\ldots(4.14)$$

The efficiency (η) is the efficiency of the fan at the operating point, that is point D for the 1100mm (η = 0.76), point E for the 1000mm (η= 0.755), point F for the 9000mm (η= 0.72) and point G for the 800mm (η= 0.63) fans. Using equation 4.14. this gives power consumptions of 35.8 kw, 36.1 kw, 37.8 kw, 43.2 kw for the 1100mm, 1000mm, 900mm and 800mm diameter fans respectively. This is the power required by the fans at 20° C (i.e. at a volume flow of 40000 M3/Hr. and pressure (corrected) 255mm water gauge), however, in this instance the required operating temperature is 100° C (where the flow rate is the same, but the pressure is 200mm water gauge). This gives power consumption figures of 28.7 kw, 28.8 kw, 30.3 kw and 34.6 kw for the four fans at the operating temperature.

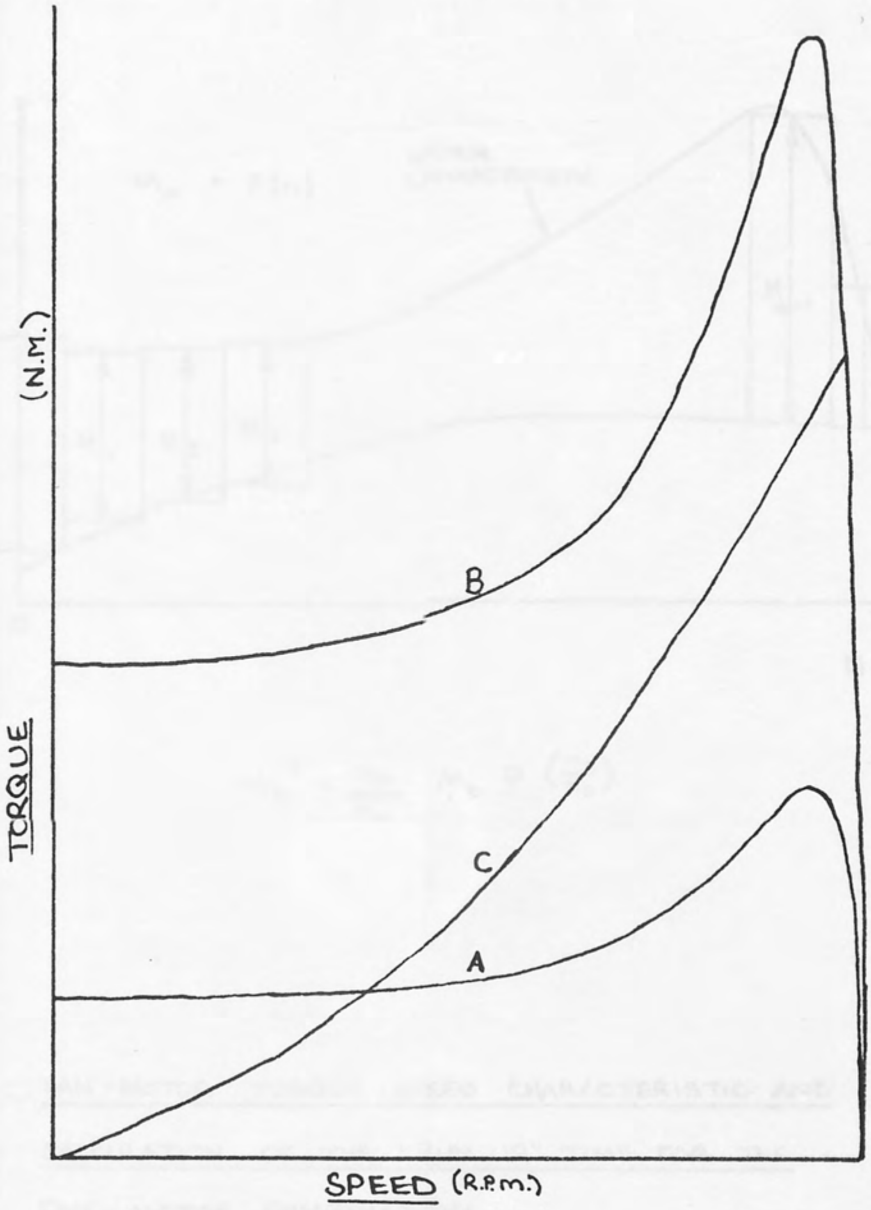
Step VI

Finding a suitable motor for each fan. Here, the torque-speed characteristic as well as the power output are important. Thus if two motors A and B and the desired fan have the characteristics of Fig. 4.3, it is clear that motor B is required. Additionally, it is necessary to estimate the time to reach the operating speed to ensure that overloading of the motor is avoided, which is given by:-

$$t_{start}(sec) = J_{tot}\, C_2\, \frac{n_a - n}{\nu} \left(\frac{1}{2}\frac{1}{m_0} + \frac{1}{m_1} + \frac{1}{m_2} + \cdots \frac{1}{m_{\nu-1}} + \frac{1}{2m_\nu} \right)$$

..........(4.15)

where J_{tot} is the moment of inertia of the fan reduced to the motor speed plus the moment of inertia of the motor, and ν represents the number of intervals see Fig. 4.4. C_2 is a constant depending on the motor characteristic given by the motor manufacturer. This gives motor sizes

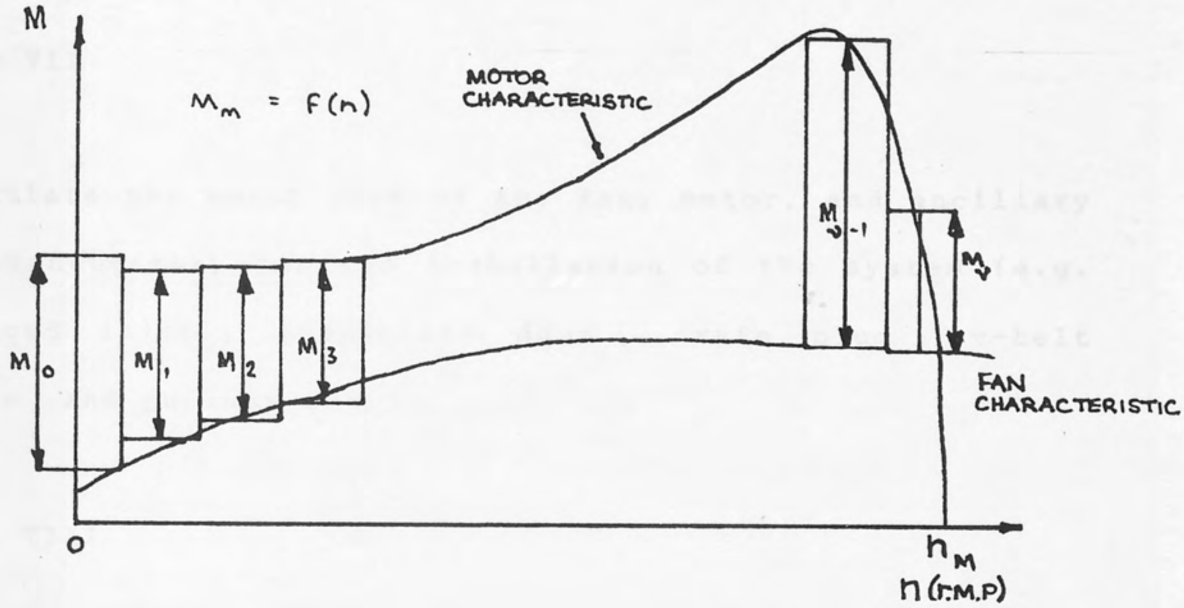


A AND B TYPICAL MOTOR TORQUE - SPEED CHARACTERISTICS.

C TYPICAL FAN TORQUE - SPEED CHARACTERISTICS.

FAN AND MOTOR TORQUE - SPEED CHARACTERISTICS

Fig 4.3



$$M_b' = \frac{n_b}{n_m} M_b \, P\left(\frac{n_i}{n_b}\right)$$

FAN - MOTOR TORQUE - SPEED CHARACTERISTIC AND CALCULATION OF THE "RUN UP" TIME FOR THE FAN - MOTOR COMBINATION.

Fig 4.4

of 30 kw, 30 kw, 30 kw and 37 kw for the 1100, 1000, 900 and 800 mm diameter fans respectively.

Step VII

Calculate the total cost of the fan, motor, and ancillary items necessary for the installation of the system (e.g. flanged inlet , inspection door , drain plug , v-belt drive and pulleys etc.).

Step VIII

Having obtained the cost details of the various units, a suitable size has to be recommended from the list of possibilities. This is straight forward, when the customer has specified the requirements or preferences in precise details. For example the lowest capital cost assembly or the most efficient operating assembly may have been requested, or the approximate size of the impeller may have been prescribed. In the case where the customer has not specified such details, the list of possible sizes are discussed with the customer before recommending a particular size or sizes.

Step IX

Plot the fan characteristics for the chosen size (or sizes) at the operating conditions, showing the duty point.

Step X

Prepare a formal tender to be sent to the customer in response to his enquiry.

From the above example, it will be seen that apart from Step VIII, that of recommending a particular size from a list of possibilities, the selection procedure is systematic, but time consuming. Such steps can very appropriately be carried out on a computer.

4.3. SPECIFICATION OF THE COMPUTER AIDED SELECTION SYSTEM

Before any work could start on the development of the system, a considerable amount of discussion took place with senior management of the sponsoring company and the technical sales staff, the people who would be using the system daily. The discussion with management was centred on the type of system they wanted to implement, i.e. fully computerised selection where no fan specialist engineers would be required, or some form of computer aided selection where specialist engineers would still be required. The sponsoring company had no computing facilities and did not know if the introduction of such technology, with a fully computerised selection would be opposed. In view of this and the fact that close co-operation of the staff in the technical sales department was needed in the development of the system, as the author had no previous experience of selecting fans, the author suggested that a computer aided selection be developed with the technical salesman as an essential part of the system, with a gradual development towards a fully computerised selection system later. The advantages of this course of action being:-

1). The development time for the software would be short compared with the time for a fully computerised system.

2). Computerisation of the other aspects of the company's operations (e.g. accounts, stock control, production scheduling, pay roll et.) maybe readily accepted, if employment security is seen not to be at risk by the employees concerned should the company consider this at a later date.

Senior management agreed with this approach, and discussion with the technical sales staff began. The discussion was centred on the following:-

1). What aspects of the selection process they would like computerised?

2). How much dialogue should there be between the user and the machine?

3). What information they wanted the computer to provide them with, to enable them to determine if the size was suitable or not and how it should be displayed?

During these discussions, it was indicated by the technical sales staff that with the expertise in the department, it would be a relatively easy task for the person to determine which blade type would be the most suitable having examined the duty, the environment in which the fan is required to operate and any previous experience of the customer's preferences (blade types). Due to the many factors which determine this, the dialogue between the man and the machine would be long and the user would have no control over the final selection. For these reasons it was agreed that the blade type would be specified and selection made only from that type. This also has the advantage that the technical salesman is retained as an essential part of the new system, and thus was likely to be readily accepted, as it would relieve them of the mundane time consuming calculations. The main requirements of the system were to be:-

1). Short dialogue between the man and machine.

2). Quick response time in giving details of the possible sizes.

During the discussions it was agreed that the input to the selection program should consist of the following information:-

1). Volume required and units.

2). Pressure required and units.

3). Operating temperature and units.

4). Type of fan-motor drive arrangement required (i.e. directly coupled drive or indirect drive via v-belt drives and pulleys).

5). Blade type from which selection is to be made.

The output should give the following details for each suitable size:-

1). Fan size, operating speed and efficiency.

2). Power required at the operating temperature and at $20^{o}C$.

3). Motor size required at the operating temperature and at $20^{o}C$.

4). Sound spectra levels.

5). Prices of fan and all ancillary options available.

6). Price of motor selected.

7). Performance characteristic curves showing the duty point.

The sales staff agreed that with the above information for each size, recommending a size from a list of the possibilities would present no problems and the decision could be made quickly. The time taken for this would depend on how much information the customer had given initially, and whether there was any need to go back to him and discuss the possible sizes.

Having reviewed the requirements of the proposed computer aided selection system, the solution that emerged required the setting up of a data base system, which would then be used by the fan selection program. The development work of the computer aided selection system therefore consisted of:-

1). Developing the procedures (software) in setting up a suitable data base sytem.

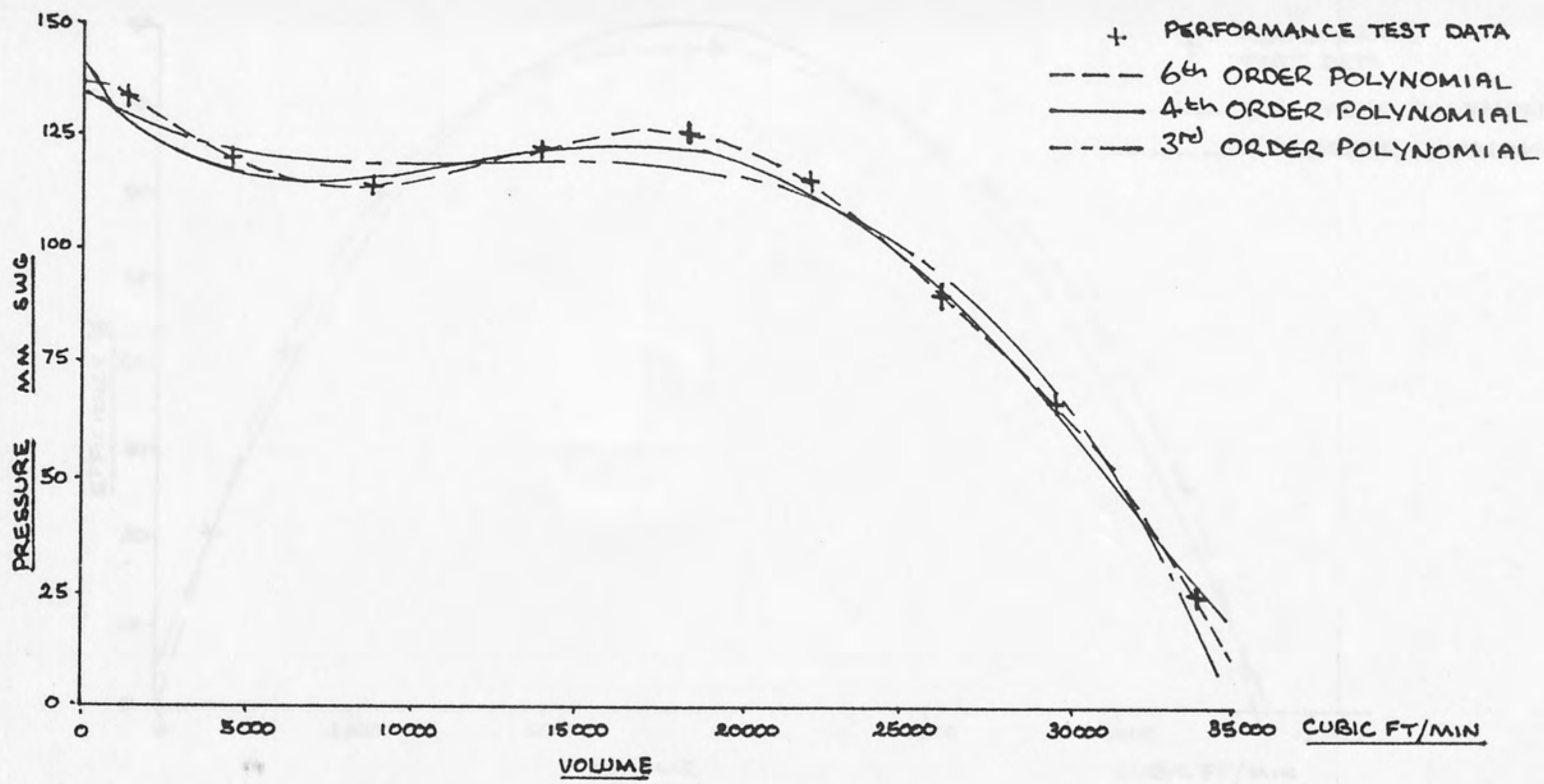
2). Developing the fan selection program which would use the data base and select fans for a specific requirement.

4.4. DEVELOPMENT OF THE COMPUTER AIDED SELECTION SYSTEM

4.4.1. PROCEDURE DEVELOPED FOR SETTING UP THE DATA BASE TO BE USED BY THE FAN SELECTION PROGRAM

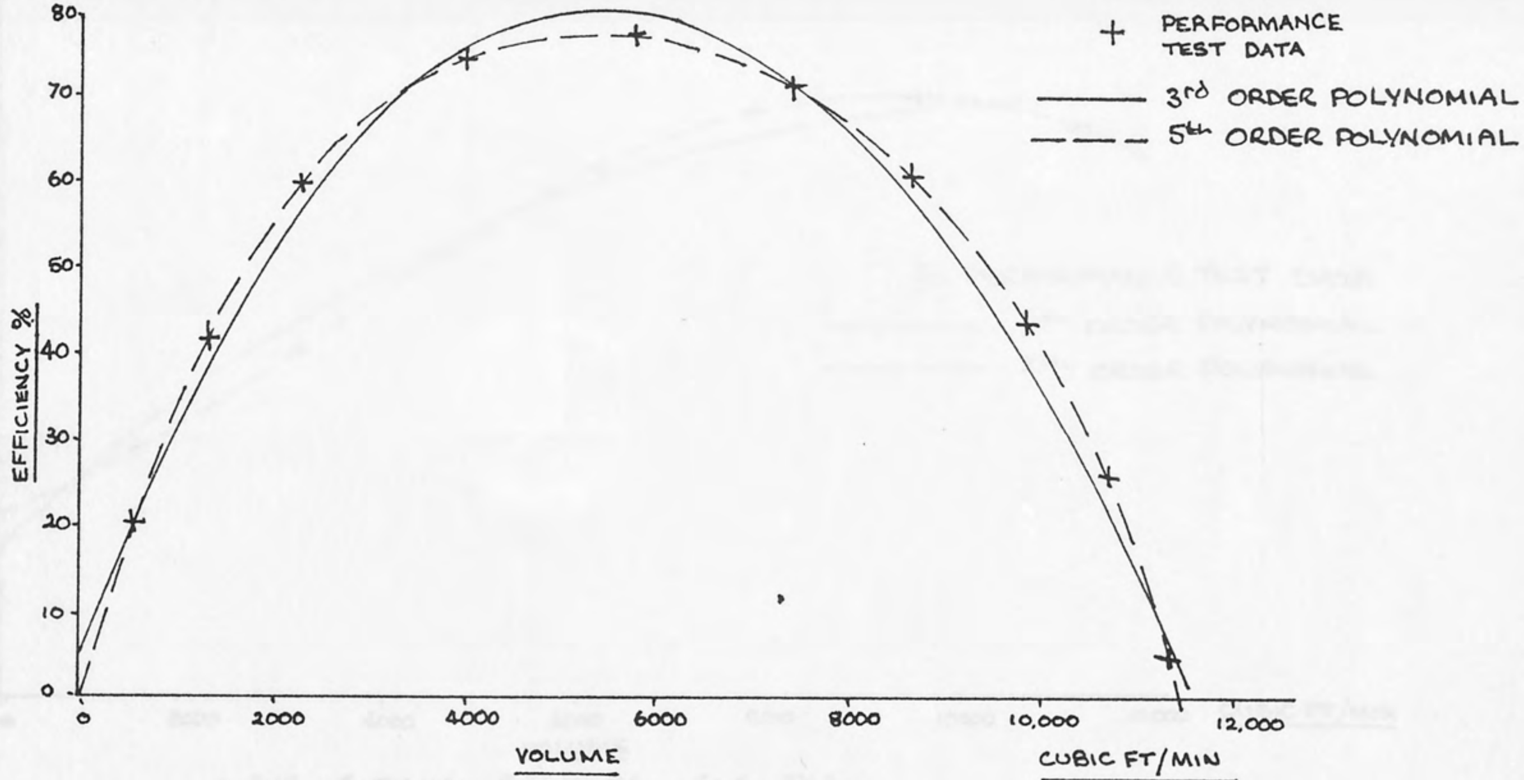
The development work on the Computer Aided Selection system was carried out on the Hewlett Packard 9830 and 9845 desktop computers at Aston University. The first problem in the development of the system, was that of representing the performance characteristics on the data base. A polynomial regression program, using the least squares fit, together with a X-Y plotter was available. Using this program, with the test data obtained from the sponsors, for their "Europa" series of fans, coefficients of the polynomial defining the volume-pressure, volume-power and volume-efficiency characteristics were calculated and the results were encouraging. Figures 4.5 to 4.7 show the results for the backward laminar (BL) and multivane (MV) types and it is seen that a good fit through the data points can be obtained with the order of polynomial ranging from 3-6. Similar results were obtained for the other range of fans manufactured by the company. An important point to note here, is that it is only necessary to obtain a good fit over the operating (selection) range. The advantage of polynomials is the ease with which mathematical operations can be performed. For example representing the performance characteristics in the form of polynomials, for a given volume, the other quantities (i.e. pressure, power, efficiency) can be calculated easily.

Having determined that this approach would be suitable in representing the fan performance charcteristics on the data base, the next step was to arrange this data and the other information necessary for selection purposes in a suitable manner, which could then be used by the selection program. The development of the setting up of a suitable data base involved a certain amount of trial and error.



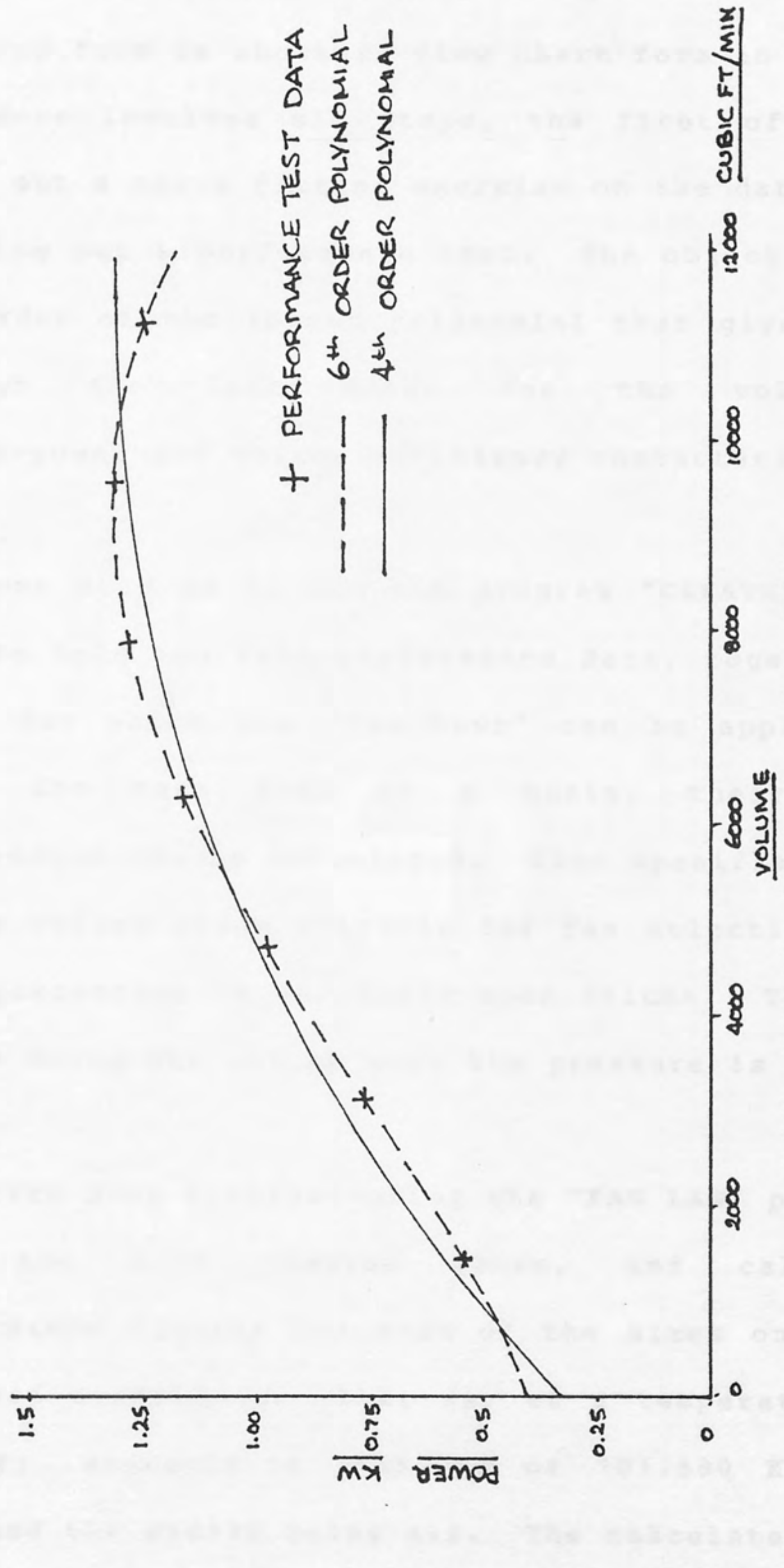
CURVE FITTING RESULTS FOR THE VOLUME - PRESSURE CHARACTERISTIC FOR THE "MY" BLADED IMPELLER

Fig 4.5



CURVE FITTING RESULTS FOR THE
VOLUME — EFFICIENCY CHARACTERISTIC FOR THE "BL" BLADED IMPELLER

Fig 4.6



CURVE FITTING RESULTS FOR THE VOLUME - POWER CHARACTERISTIC FOR THE "BL" BLADED IMPELLER

Fig 4.7

The final procedure adopted to get the data into the required form is shown in flow chart form in Fig. 4.8. The procedure involves six steps, the first of which is to carry out a curve fitting exercise on the data obtained by carrying out a performance test. The object is to obtain the order of the lowest polynomial that gives a good fit through the test data, for the volume-pressure, volume-power and volume efficiency characteristics each.

The next step is to use the program **"CREATE"** to create a file to hold the test performance data, together with the sizes for which the "fan laws" can be applied, so that using the test data as a basis, their theoretical performance can be calculated. Also specified for the fan is the volume range suitable for fan selection, expressed as a percentage of the fully open volume. The fully open volume being the volume when the pressure is zero.

The third step involves using the **"FAN LAW"** program, which uses the file created above, and calculates the performance figures for each of the sizes on the file at standard conditions. That is, at a temperature of 20^{o} C (68^{o} F), atmospheric pressure of 101.580 KN/M^{2} (762 mm Hg), and the medium being air. The calculated figures for each size are stored in individual files.

Step four involves using the polynomial regression program

Carry out a curve fitting exercise on the test data to establish suitable order of polynomial for the test performance curves (see Figs 4.5, 4.6, 4.7).

Create a file to hold test performance data using "CREATE" program.

Using the "FANLAW" program and the above data file, create files to store the calculated performance data for the range of sizes applicable to the test data using the fan laws.

Use "Curve" program to calculate the coefficients of the polynomials, (the order being established from the curve fitting exercise above) for the above calculated performance data, and stored on file, together with the fully closed pressure and fully open volume.

Use "DAFILE" program to combine the above files containing the coefficients of the polynomials. Also typed in is the maximum running speed and inertia for each size. This data is stored on file and used by the selection program.

Use "COST" to created data file to hold cost data for the above series. This file is also used by the selection program.

PROCEDURE USED TO CREATE DATA BASE FOR

THE FAN SELECTION PROGRAM.

FIG 4.8

"CURVE" to calculate the coefficients of the polynomials defining the volume-pressure, volume-power and volume-efficiency characteristics of the fan performance, using the calculated performance data contained in the files created by the **"FAN LAW"** program. The program requires the order of the polynomial for each of the three characteristics to be specified, which has already been determined in step one of the procedure. The calculated coefficients of the polynomials, together with the lower volume and upper volume limits (i.e. the selection range limits) for each size are again stored in individual files.

Step 5 involves combining all the individual files containing the polynomial coefficients into one master file for the series. This is achieved using the program **"DAFILE"**. Also typed in for each size, at this point is the maximum operating speed allowed for the fan and its moment of inertia. The master data file created is the data base for the selection program.

The final step is creating the file containing the price details of the fan unit and the extra options available with it, and this is acheived using the program **"COST"**. It was decided to have the price details on a separate file rather than combine them with the master file, because the price data will need to be updated periodically. One master data file and cost file needs to be created for each fan type manufactured by the company.

The "MOTOR" program is used to create a file to hold the performance and price details of the range of motors used by the company. The range included in this file is large enough to cater for all the sizes ever likely to be used by the sponsors, so that the one performance data motor file can be used irrespective of the type of fan being selected. As with the fan price details the motor prices are stored on a separate file.

4.4.2. DEVELOPMENT OF THE SELECTION PROGRAM

Having arranged the data base described in the previous section, work on the development of the fan selection program commenced. The logic of the developed program is shown in flow chart form in Appendix No.3 , and it will be seen that it follows the steps of the selection process described in Section 4.2. The developed program has the advantages of:-

1). Quick response time.

2). Short dialogue between the man and the machine.

As the computer aided selection system had been developed at Aston University, and since the technical sales staff had no previous experience of using such technology, short trials were arranged at the sponsors using the Hewlett Packard and Tektronix desktop computers to assess the response of the users to the proposed system. The trial periods showed that the technical sales staff experienced no difficulties in using the developed system, in spite of the fact that they had no experience of using such technology. The system was well received by the technical sales staff, as it relieved them of the laborious routine time consuming calculations, but left them with the more rewarding task of decision making.

4.5. BACKGROUND TO THE DECISION CONCERNING COMPUTER HARDWARE ACQUISTION

Two short trials at Alldays, Peacock & Co. Ltd., using the Hewlett Packard and Tektronix desk top computers showed that the developed system achieved its aims, and the technical sales staff indicated that there would be no opposition to the introduction of such a system.

A cost benefit analysis (see Appendix 1) was carried out to assess the benefits of the proposed system. The analysis showed senior management the advantages of such a system, and they accepted the recommendation that the system be implemented on a Hewlett Packard micro-computer. Senior management indicated that the company's accounts needed to be computerised soon, as the present mechanical accounting machines were continually breaking down, and that they were considering computerisation of other aspects of the company's operations (e.g. stock control, production scheduling and computer-aided draughting). They requested advice on suitable machines for such purposes. The author suggested that in view of the extent of computerisation the company was now contemplating, they might consider a multi-user mini computer. The advantages of such a system being:-

1). It is multi-user, i.e. more than one user can have access to the machine at the same time.

2). Processing times are faster than on a micro-computer.

3). Much larger disk storage capacity is available with mini-computers compared with micro-computers.

4). Mini-computers are much more flexible than micro-computers when up grades are required e.g. more memory, more storage, or more users require access to the machine.

The disadvantages of micro-computers pointed out were:-

1). The micro-computers are stand alone machines and cannot be linked to each other easily.

2). Storage devices (floppy disks) limited to between 1/2 Mbytes - 1 Mbytes.

3). Storage devices (hard disks) have larger capacities range from 5 Mbytes to 10 Mbytes but costs are high ranging from 10,000 to 15,000 pounds.

In view of such remarks, senior management requested details of mini-computer systems which would be capable of implementing the computer aided selection system, and the accounting activities of the company, but which could be up graded to include other activities indicated as resources permitted.

Tenders were requested from various mini-computer system manufacturers, for suitable machines within their ranges, suitable for the implementation of the computer aided selection system and the company's financial accounts and easy to up grade as needs change.

This resulted in tenders being received for the following systems:-

1). Data General Nova 4X mini computer system.

2). Digital Equipment Corporation PDP11/34 mini computer system.

3). Prime model 250 mini computer system.

4). Honeywell Level 6 mini computer system.

5). Texas Instruments 990 mini computer system.

The proposed systems were discussed with the supervisor from the computing department in the supervisory team. The outcome was the recommendation of the Prime (250) and the Digital Equipment Corporation (PDP11/34) as the two most suitable machines, with the Prime machine as the first choice. The advantages of these two machines being:-

1). Operating system easy to use.

2). Machines easy to up grade and more importantly software will not require modification when an up grade is made. This is very important because software development is still labour intensive.

3). Good back-up service available.

4). Hardware well established and has good reputation regarding reliability.

5). Machines suited for both commercial and scientific applications.

6). Large quantities of commercial and scientific software available.

The Prime also has the added advantage that it is a virtual memory machine. This facility gives each user a 512 Mbyte virtual address space of which 32 Mbytes is reserved for the user program and data space. As these facilities are transparent, users can create, test, modify and execute very large programs without defining overlays (i.e. complex partitioning schemes) and without concern for how system resources perform these functions. More over, programmers can write programs with no concern about the memory configuration of their particular system. Also as the operating system functions are embedded in each users virtual address space, all functions are immediately available as if they were an integral part of the program, thereby reducing overheads and streamlining the program development process.

The Prime system was recommended to senior management, who in turn passed the recommendation to the parent company for capital sanctions approval. An associate company within the engineering group of the parent company, in the meantime expressed the need for a computer for their operations. In view of the above recommendation, the question was posed, could the one machine serve the needs of both the companies? The requirements of the associate company were reviewed, and it was felt that the work load would be too demanding for the model 250 and that the model 450 would be more suitable. The arrangement with such a system, would be that the computer is installed at the sponsoring company and the associate company having access to the machine via G.P.O. lines. The proposed system was then passed for final approval from the parent company. The approval was granted and the final configuration of the system is:-

Prime 450 mini computer with 1/2 Mbyte main memory processor, 64 Mbytes of disk storage (of which 48 Mbytes are fixed and 16 Mbytes removable), with 16 multi-user lines.

The model 450 system can be up graded to 2 Mbytes of main memory and can support up to 63 simultaneous users.The ancillary hardware is as follows :-

For the technical sales department:-

1 Graphics terminal.

1 X-Y plotter.

1 Letter quality printer (daisy wheel type).

1 Visual display terminal.

For the Accounts department:-

1 Printer terminal.

2 Visual display terminals.

For the associate company:-

2 Visual display terminals.

1 Printer terminal.

Multiplexors and Modems for the G.P.O. lines.

The above system was installed in July 1981.

Finally it is worth pointing out that micro development is proceeding so rapidly now, that some of the remarks made earlier concerning micro computers are no longer relevant. The main ones being:-

1). Storage capacity. The cost of hard disks have fallen with the advent of high storage capacity, low cost Winchester disks compatible with micro computers.

2). Compilers available for micro computers in a variety of high level languages, giving quicker processing times.

3). Graphics available on most micro computers.

4). Multi access to one disk, thus giving the effect of a multi-user system using micro computers.

4.6. IMPLEMENTATION OF THE COMPUTER AIDED SELECTION SYSTEM

The original computer aided selection system developed on the Hewlett Packard 9845 desk top system was developed in the "BASIC" computing language. Since the desk top machine is a stand alone machine, and the data and programs were stored on cassettes, to increase the speed of response, all the files containing the data required by the selection program were read into the computer's memory from the cassette once the fan type had been specified before the start of any calculations. The advantage being that the transfer of information in the main memory is considerably faster than transferring it from the cassette as required. The main disadvantage with reading all the information into memory in one go, is that it uses more memory than reading the information serially, however, the memory size of the desk top machine was not a problem in this case. In fact it had excess capacity which could not be used by anyone else because the desk top machine is a one user machine.

Since the developed system was now to be implemented on a multi-user system, and because the "Fortran" computing language was available, it was decided to implement the computer aided selection system using Fortran. The advantage with this language is that it is compiled and is much faster in executing the instructions in this mode. During discussions with the software house who supplied the Prime computer, it was felt that the data required by the selection program should be set up as a data base system. Since the author had no experience of setting up a data base system on the Prime machine and as time was running short to complete the project, the software house agreed to write the software necessary to set up the data base on the Prime machine using the procedure described earlier in Section 4.3.1. They also developed the software to read the individual records of data in the selection program, with the rest of the selection program being developed by the author. The input required and the output produced by the selection program, apart from a few minor details, however remains the same as originally developed for the desk top machine.

The advantages of implementing the system on the Prime multi-user system compared with desk top micro are:-

1). Quicker response time up to 2-3 times quicker.

2). Can handle a much larger data base.

3). Multi-user access to the same program at the same time.

4). Suitable size details are stored on file and can be used to prepare a specification data sheet of the selected size once the necessary software has been developed, thereby enabling the tender to be prepared using the computer.

4.7. CRITERIA USED IN EVALUATING THE COMPUTER AIDED SELECTION SYSTEM

In evaluating the performance of the implemented computer aided selection system, it was decided after discussion with the senior management of the sponsoring company, to compare the enquiry response time before and after the implementation of the system. This form of analysis, strictly, will not give an accurate account of the performance of the computer aided selection system, since the selection process is only part of the overall enquiry response time. A procedure could have been set up where the selection process alone, was monitored before and after the introduction of the system, to give an accurate account, but due to practical difficulties in recording the information necessary for such an evaluation, also because of the work load in the technical sales department, and the extra work load this would entail it was not done. For social reasons also, it was felt that such an evaluation, which would require an accurate record of the time the technical salesman spent on each enquiry, would not be accepted easily, and may lead to the resentment of the introduction of the new technology.

For these reasons, it was decided to see if there was any improvement in the overall enquiry response time, as it would not impose any extra work load on anyone, but still give some meaningful evidence in the evaluation.

4.8. EVALUATION OF THE COMPUTER AIDED SELECTION SYSTEM

Since the original enquiry response time analysis was carried out, the sponsoring company, along with everyone else has been affected by the world wide recession. Like most companies in such situations, the company had reviewed its overheads, and taken steps to reduce them. This has meant a reduction in staff in the Technical Sales department from 6 to 3 and a reduction from 5 to 2 in the typing pool staff. In view of these changes and the time lapse between the original analysis and the implementation of the computer aided selection system, it was decided to carry out another enquiry response time analysis (with a smaller sample size) just prior to the introduction of the system to see if there had been any significant changes in the response time. The analysis was carried out for the 3 months (April to June 1981) period prior to the introduction of the system, which was introduced in the middle of July 1981.

The analysis showed (Fig. 4.9, Fig. 4.10 and Table 4.1) a slight improvement in the response time. In the case of written enquiries, the percentage of enquiries answered in the period 0 to 5 days increased from 45.2% in 1978 to 50.6% prior to the introduction of the system. For verbal enquiries for the same period the improvement was from

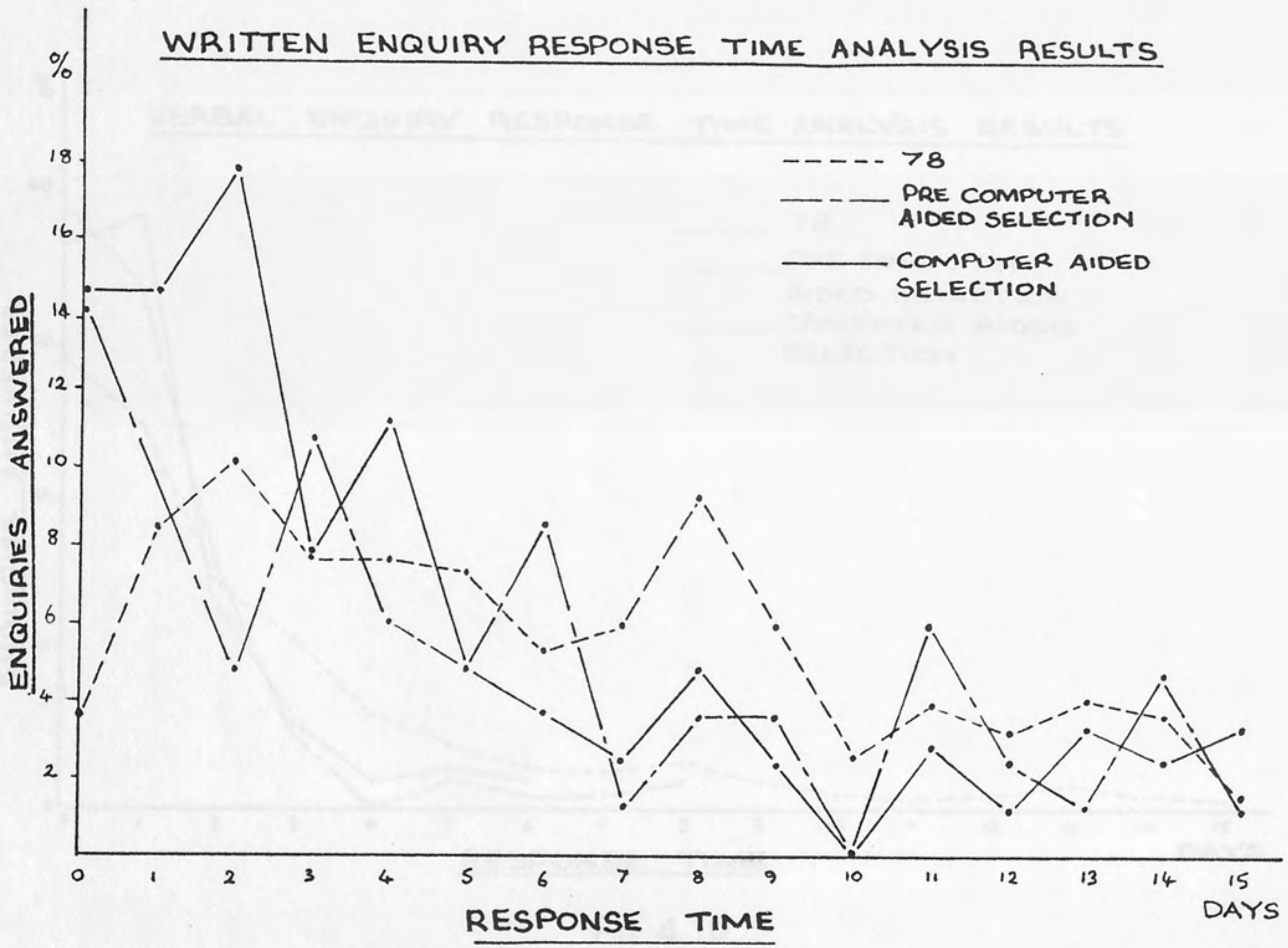

WRITTEN ENQUIRY RESPONSE TIME ANALYSIS RESULTS
%
ENQUIRIES ANSWERED
18
16
14
12
10
8
6
4
2
0
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
DAYS
78
PRE COMPUTER AIDED SELECTION
COMPUTER AIDED SELECTION
RESPONSE TIME

Fig 4.9

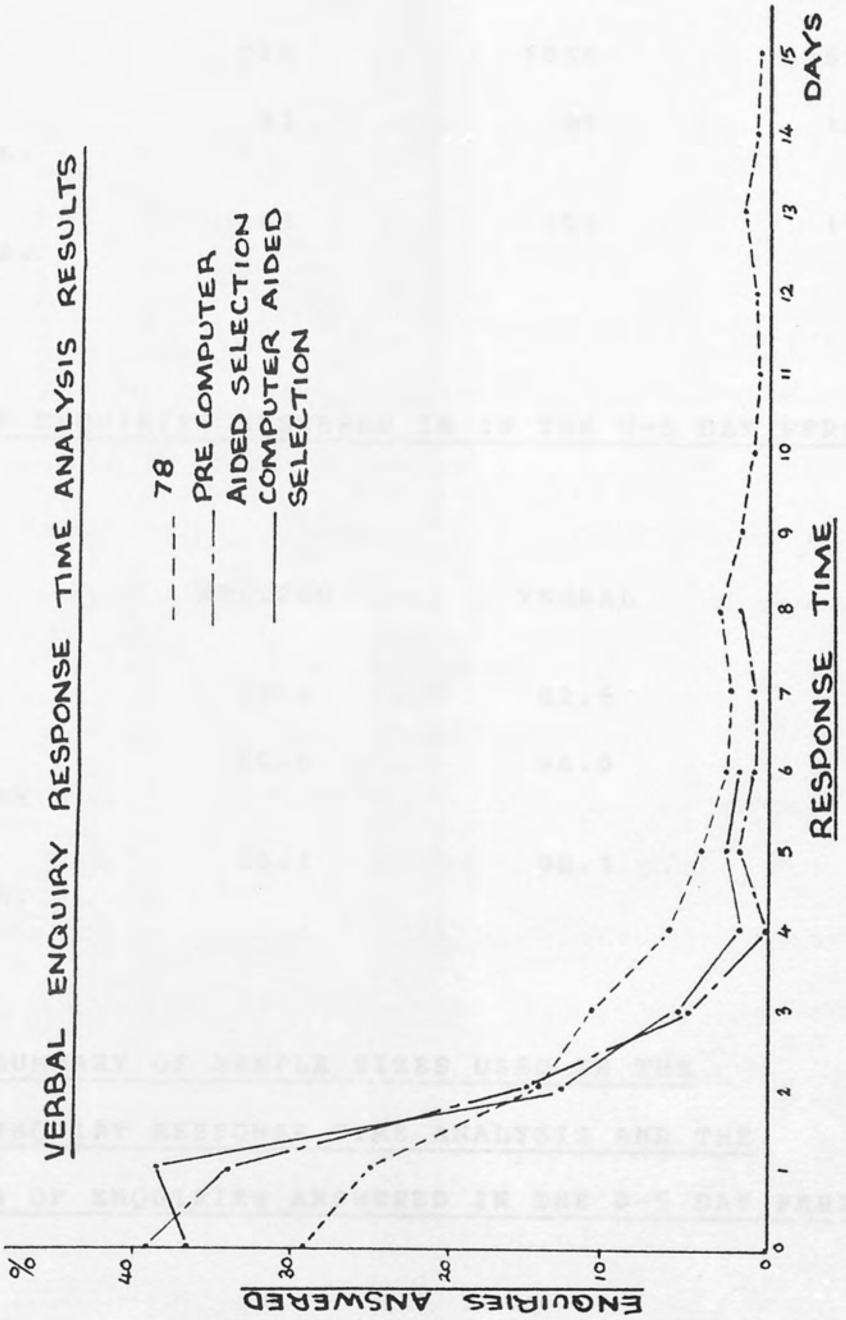

VERBAL ENQUIRY RESPONSE TIME ANALYSIS RESULTS
78
PRE COMPUTER AIDED SELECTION
COMPUTER AIDED SELECTION
%
ENQUIRIES ANSWERED
40
30
20
10
0
0 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
DAYS
RESPONSE TIME

fig 4.10

SAMPLE SIZES

	WRITTEN	VERBAL	TOTAL
1978	528	1030	1558
Pre-computer aided selection.	83	99	182
Post-computer aided selection.	89	109	198

% OF ENQUIRIES ANSWERED IN IN THE 0-5 DAY PERIOD.

	WRITTEN	VERBAL
1978	45.2	82.5
Pre-computer aided selection.	50.6	96.0
Post-computer aided selection.	70.1	98.1

SUMMARY OF SAMPLE SIZES USED IN THE ENQUIRY RESPONSE TIME ANALYSIS AND THE % OF ENQUIRIES ANSWERED IN THE 0-5 DAY PERIOD.

TABLE 4.1

82.5% to 95%. This improvement can be explained by the fact that the Technical Sales Manager whose main function was in the administration of the Technical Sales department and of the typing pool has himself had to take a more active role in answering the enquiries due to the reduction in staff and the reduction of the administration as a result of this.

In view of the fact that there has been no significant change in the enquiry response time, it was decided to carry out a similar analysis after the introduction of the system, in assessing the sucess or failure of the computer aided selection system. As time was running short for the completion of the project, the analysis was carried out during the 3 month period from August to October 1981. Table 4.1 shows the number of enquiries received during this period are approximately the same as for the period prior to the introduction of the system. Since there has been no changes in staff during the two periods, any change in the response time in the second period, one can confidently assume is as a direct result of the introduction of the system.

For written enquiries the results of the analysis (Fig. 4.9, Fig 4.10 and Table 4.1) for the enquiry response time, show that after the introduction of the computer aided selection system there is a substantial improvement in the percentage of enquiries answered in the period 0-5 days from 50.6% prior to the introduction of the system to 70.1% after its introduction. This is a significant increase bearing in mind that during the evaluation period:-

1). The data base to be used by the selection program was gradually being increased.

2). The selection program was being developed and refined to the user's requirements.

3). The technical sales staff were still on the learning curve with the multi-user system.

In the case of verbal enquiries, the analysis showed that there was no significant overall improvement in the response time. In this case the percentage of enquiries answered in the period 0-5 days increased from 95.9% prior to the system to 98.1% after its introduction. This is not totally unexpected as in this case, the time taken to select a suitable fan is short compared with the overall time taken to prepare the tender. One important point however, not high-lighted by the analysis is the fact that with most verbal enquiries, the customer is given details of suitable size(s) instantly at the time of the enquiry with a written tender to follow, which was the aim of the project.

In general there has been an overall improvement in the service provided to the prospective customer, in that the possible selections are being discussed with the customer before recommending a particular size or sizes. Prior to the system, the Technical Salesman would, using his experience, calculate the details for one particular size and the tender prepared for that size. The amount of information provided has also increased, in that performance curves, sound spectra levels, fan-motor combination run up time, are provided as standard information. This improvement in service is not high-lighted by the analysis, but should be borne in mind when assessing the system, such improvements in service are difficult to quantify in the short term analysis as any gains as a result of the improvement in the service provided are long term. That is, the customer is likely to send future enquiries to companies which provide good service. The importance of providing good service which is a hallmark of industrial marketing was discussed in Chapter 3.

In view of the delay in preparing the tender by the typing staff, which affects the overall performance of the computer aided selection system, when an analysis is carried out of the enquiry response time, the author proposed a short term solution. The solution was to utilise the Commodore micro-computer, not utilised since the acquisition of the Prime mini computer, by acquiring a suitable word processing software package and developing "standard" letters and fan specification data sheets to reduce the typing time. This it was pointed out would be a short term solution whilst a tender preparation system by using the output produced by the selection program is developed on the Prime mini computer. This phase of the development will not be part of this project due to lack of time. This was accepted by senior management and suitable software acquired in January 1982.

In conclusion, although the analysis carried out does not measure the change in the selection process after the introduction of the computer aid, it does, however, give measurable evidence in the overall improvement in the response time. Since the introduction of the computer aid is the only change in the overall process, the improvement observed, one can confidently claim, is as a result of the introduction of the new aid. In view of the short existence of the system and the measured improvement, one can claim the system as a success.

CHAPTER FIVE

NUMERICAL STRESS ANALYSIS

INTRODUCTION

Chapter 5 discusses the development of the finite element work; it begins with a brief review of the historical development of the method, leads onto a discussion of the advantages of this technique and discusses accuracy and the convergence of solutions.

The geometry and the shape function polynomials for the so called Semi-Loof element, which is used in the present work are discussed as are its benefits for the work in hand.

Implementation of the Finite Element process in a computer code and examples used for verification are also presented in this chapter.

5.1. HISTORICAL DEVELOPMENT OF THE FINITE ELEMENT METHOD IN STRESS ANALYSIS

5.1.1. BACKGROUND TO THE PROBLEM

Aircraft have always presented the structural engineer with some of his most difficult problems. The relentless drive for minimum weight, coupled with maximum safety, has left little room for guesswork. To meet these requirements, aircraft structural engineers pioneered the development of high-strength, light-weight alloys, and led the research that resulted in refined methods of structural analysis. By the early 1940's, as a result of this work, ingenious analytic methods had become available for treating both static and dynamic problems arising in connection with aircraft structures. In particular, matrix methods for calculating vibration frequencies and mode shapes. Hence a reasonably comfortable state of affairs existed at that time for the structural engineer. Suitable materials were available and reasonably adequate analytical procedures had been developed for checking the integrity of the structural design.

However, this stable period was short lived, when in the latter part of the 1940's jet power aircraft began to appear. The sequence of design changes introduced at that time are very much in evidence today. Due to increased flight speeds with jet engines, it became necessary to take compressibility of the air into account in designing the external shape of the aircraft. This led to the familar swept wings seen on present day passenger carrying aircraft. It also led to the short delta-shaped wings seen on high speed military aircraft. In either case the available structural analysis methods proved to be inadequate. Futhermore, for the higher speed ranges aluminium alloys were soon recognised to be unsuitable. This latter problem was eventually overcome by turning to titanium as a new structural material.

The structural analyst confronted with such basic design changes was faced with a dilemma. Those idealised problems that his theory could handle turned out to be too simple to represent the true problem. More realistic idealisations were beyond his available theory. As a result the problems that had suddenly arisen were pressing and they were difficult. The work carried out during this period to provide a solution, eventually led to the finite element method of today.

5.1.2. THE MATRIX STIFFNESS METHOD

The first steps at providing a solution followed traditional lines. Even these proved to be difficult, for it was not at all clear how the known methods could be generalised to include the new problems. The traditional structural analysis methods had grown out of the theory of elasticity and followed that general approach, namely, to calculate internal stresses by using fundamental conditions of equilibrum of forces and continuity of displacements. Deformations could then be obtained by a second, subsequent calculation. For the engineering structure, consisting of various connected structural members, this procedure is known as the redundant force method. In recent years it has often been called simply the force method.

In spite of the progress in development of the force method, it became clear in the first half of the 1950's that the ultimate solution might well have to be sought in other directions. The difficulty that led to these feelings arose in connection with the delta wing structure. Idealisation of the delta wing required elements incorporating characteristics for which flexibility matrices could not be derived. Since these matrices were essential to carrying out the force method, it was imperative that they either be found or that an alternative approach be found. One of the first to recognise this situation was Levy [19]. In this paper, he sought to define the problem in terms of the stiffness matrix. Although this work only partially succeeded in breaking away from the force method approach, it nevertheless did suggest that an alternative theoretical path was available.

During this same period a small group was formed by Turner at The Boeing Company to work on the problem. The goal was to provide a method of analysis that would yield structural stiffness data sufficiently accurate for use in subsequent structural dynamics calculations. Furthermore, it was desired to find an analysis procedure suitable for application to any structural configuration. The work by Turner et al [20] is regarded as one of the key contributions in the development of the finite element method. The finite element concept in its modern form was established through the derivation of the stiffness matrix for the triangular element, based on assumed displacements.

5.1.3. PLATE AND SHELL ELEMENTS IN THE FINITE ELEMENT METHOD

One of the first problems to receive attention was that of plate bending. Both the rectangle and the triangle shapes were examined as possible elements for bending analysis. The rectangular bending element was treated by Melosh [21]. Inter-element compatibility of displacements was not fully achieved in this early work. A quite different approach for developing a triangular plate element was also found and used by Greene et al [22]. It forms the basis for the first analysis by finite elements for thin shell problems. A totally different derivation for this element was given much later by Melosh [23]. This element is still in use and has characteristics that make it suitable for certain complex problems; for example, the buckling of plates and the elastic-plastic response of plates to dynamic loading.

Difficulty experienced with the plate elements served notice that even greater obstacles would be faced when shell elements were to be considered. Early attempts centred on using plate elements in analysing the behaviour of shells. This method proved to be unsatisfactory and attention had to be focused on the development of new curved shell elements. A comprehensive survey of plate and shell elements is given in reference 16. A very simple, but nevertheless useful, thin shell element was discussed by Grafton and Strome [24]. Known as the conical element, it made possible the finite element analysis of axisymmetric shells and pressure vessels. The element itself consisted of a frustum of a right circular cone. Nodes became circles at each end of the frustum. This was a new concept since all previous elements had nodes which were single points.

5.1.4. VARIATIONAL PRINCIPLES AND FINITE ELEMENTS

Most of the work carried out during the interim period of the development of the finite element method leaned strongly towards the physical based reasoning on which reference 20 was developed. Although the underlying role of the virtual work principle was at least recognised during this period, the appreciation and understanding was not sufficiently focused to enable the benefits of this association to be advanced to its full potential.

Numerically it had been observed for years that the finite element method often led to convergent results as the number of elements was increased. Gradually an interest developed in examining this question on purely theoretical grounds. Once this was seriously undertaken, it quickly became obvious that it would be necessary to place the entire finite element method completely on the firm foundation of classical infinitesimal theory. Only by doing this could a rigorous discussion of convergence be developed.

The earliest convergence studies of the finite element method were by McLay [25] and Melosh [26]. Melosh's work placed the finite element method on the principle of minimum potential energy. Additional papers concerned with the basic theory appeared at the same time. For example in 1963 Besseling [27] presented the analogy between the matrix equations of structural analysis and the continuous field equations of elasticity. The question of upper and lower bound was discussed by Fraeijs de Veubeke [28] in a paper that introduced the alternate possibility of defining stress or equilibrum elements based on the principle of minimum complementary energy. Other papers further demonstrated the rich theoretical base offered by the variational principles for defining finite elements. A considerable extension of previous work was presented by Jones [29] in 1964. Jones pointed out the advantages that could be secured using Reissner's general variational principle. This led to the so called mixed element, which depends on assumed displacements and stresses. However, the conditions to be satisfied by these assumed functions are considerably less stringent than those required when seeking a displacement (i.e., compatible) or stress (i.e., equilibrum) element. Hence this approach is very useful when certain complex elements, such as thin shell elements are to be derived.

A variant of this theory by Jones was also published in 1964 by Pian [30]. This also based the element specification in terms of both assumed displacements and stresses. However, the variational formulation was in terms of both the minimum potential and complementary energy principles. Again the added flexibility led to advantages of particular value for certain complex cases, or when certain degrees of freedom are desired for the element. Pian's approach has led to what is now known as the hybrid element.

Establishing the finite element on the variational principles led to advances that would have otherwise been impossible. First, it was recognised that the finite element method represented a new and numerically efficient procedure for applying the classical Rayleigh - Ritz method. Courant [31] pointed out the connection between the partial-differential equation and the variational problem, and discussed approximating methods then in use, treating in detail the Rayleigh - Ritz technique. In his final remarks Courant proposed a scheme based on the Rayleigh - Ritz method similar to the finite element method as it has unfolded to this date. In the Rayleigh - Ritz method the displacement field in a continuum is usually described by means of a sum of pre-selected functions, each multiplied by a constant. The constants are determined by means of a condition of minimum potential energy. While in the classical Ritz procedure one set of functions describes the displacement field in the entire continuum, thus leading to simultaneous algebraic equations in which no banding occurs and the coefficients matrix is full. In the finite element method this specification is piecewise, each nodal parameter influencing only adjacent elements, and thus a sparse and usually banded matrix of coefficients is obtained. By its nature the conventional Ritz process is limited to relatively simple geometrical shapes of the total region while this limitation only occurs in finite element analysis in the element itself. Thus complex, realistic, configurations can be assembled from relatively simple element shapes.

In addition, the variational formulation permitted a unified approach to be used for determining generalised nodal forces for surface tractions, body forces, thermal gradients, inertia forces, and so on. Hence it became possible to express fully the general elasticity problem in finite element terms. From a conceptual point of view the new theoretical foundations permitted the physical element to be replaced by its mathematical equivalent. In other words the element could now be visualised as a small region of space within which the unknown function (or functions) was to be prescribed in a simple manner. Furthermore, the conditions to be met in choosing the function could be stated with certainty. This immediately lifted the method outside the borders of solid mechanics. It was no longer necessary, for example, to define a physical fluid or temperature element in order to extend the method to problems in those fields. It was sufficient to have a variational formulation of the problem, and then to proceed with the element derivation by properly interpolating the unknown function. These ideas and the underlying theory became widely known with the publication of the text by Zienkiewicz [11]. This was the first comprehensive treatment of the subject, and the text had a far-reaching influence on subsequent developments.

Thus the overwhelming majority of finite element approaches are based on the principle of virtual displacements (or its equivalent, the principle of minimum potential energy). The elements derived on this basis are known as displacement or compatible elements.

5.2. THE ADVANTAGES OF THE FINITE ELEMENT TECHNIQUE

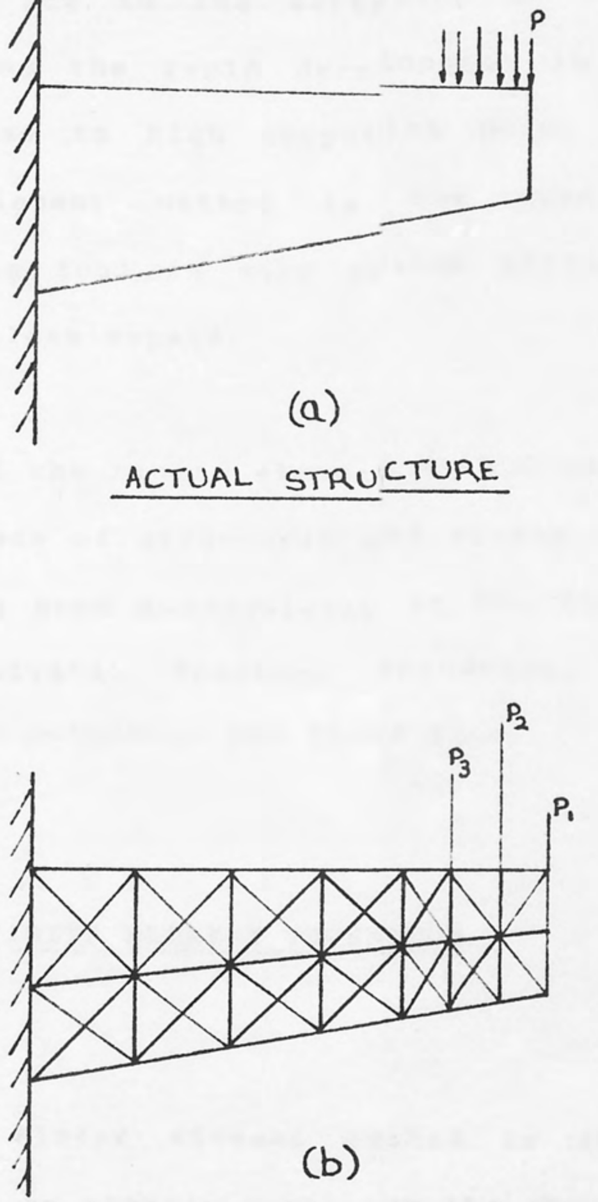
The finite element method allows the structural continuum to be replaced by a fictitious system consisting of discrete elements of finite dimensions. The system is analysed by means of the displacement method, which is already well known due to its extensive use in frame analysis. Hence, it may be said that the finite element technique extends the power of the available methods of analysis of statically indeterminate frames to include also structural continua such as plates, shells and solids.

Sizes and shapes of the elements may be selected in such a way that highly irregular geometric forms may be approximated to an almost arbitrary degree of accuracy. Triangular and rectangular shaped elements are commonly used. The element size may easily be varied such that areas of steep stress gradients may be examined in particular detail.

Any type of external loads may be considered. Distributed loads are replaced by equivalent concentrated nodal point loads as indicated in Fig. 5.1. The flexibility in choice of boundary conditions is also radically improved compared with purely analytical methods.

Anisotropic materials can also be handled, and each element of the structure may, if necessary be assumed to have its own set of specified properties. Hence inhomogeneity may also be considered. By means of an incremental load procedure it is, moreover, possible to study effects of non-linear properties of materials.

The finite element method also has the great advantage that it can be programmed for a digital computer thus relieving the analyst of long and involved calculations.



FINITE ELEMENT MODEL

Fig 5.1

As the finite element method is not purely a mathematical exercise a physical meaning of the problem is given to the analyst. Due to the versality of the finite element method, and the rapid development in micro electronics giving rise to high computing power at low cost, the Finite Element method is now used as an everyday engineering tool in many design offices and this trend will doubtless expand.

The use of the finite element method is not only confined to the areas of structural and stress analysis but it is also being used successfully in the fields of vibration, creep analysis, fracture mechanics, heat conduction, electrical potential and fluid flow.

5.3. THE FINITE ELEMENT TECHNIQUE

When the finite element method in stress analysis is applied to an elastic continuum the three basic principles of solid mechanics must be observed. These conditions are:-

1). Equilibrium of forces.

2). Compatibility of displacements.

3). Stress-strain relationships.

Any two or three dimensional body may be considered to be internally statically indeterminate with an infinite degree of indeterminancy. Therefore in order to solve for the internal forces and displacements both the conditions of equilibrium and compatibility of displacements must be satisfied simultaneously.

In the finite element method the complex body being analysed is modelled by a simplified idealised structure. The model, e.g. Fig. 5.1, consists of a number of "finite elements" obtained by a series of fictitious cuts through the original structure. The elements obtained by this method are assumed to be connected to each other only at the nodes. The external loads are also idealised by replacing them by a statically equivalent system of forces acting at the appropriate nodes.

The analysis of the assumed model may be approached by two distinct methods. The first and most widely used is known as the displacement or stiffness approach and the second is known as the flexibility or force method. In the stiffness approach the displacements at the nodes are considered as the unknown quantities, while in the flexibility method the internal forces acting at the nodes are the unknowns. The finite element method using the displacement approach can be best explained by considering the various steps necessary for complete solution of the problem. A simplified flow chart of the method is shown in Fig. 5.2.

The method can be divided into two main sections:-

1). The element analysis.

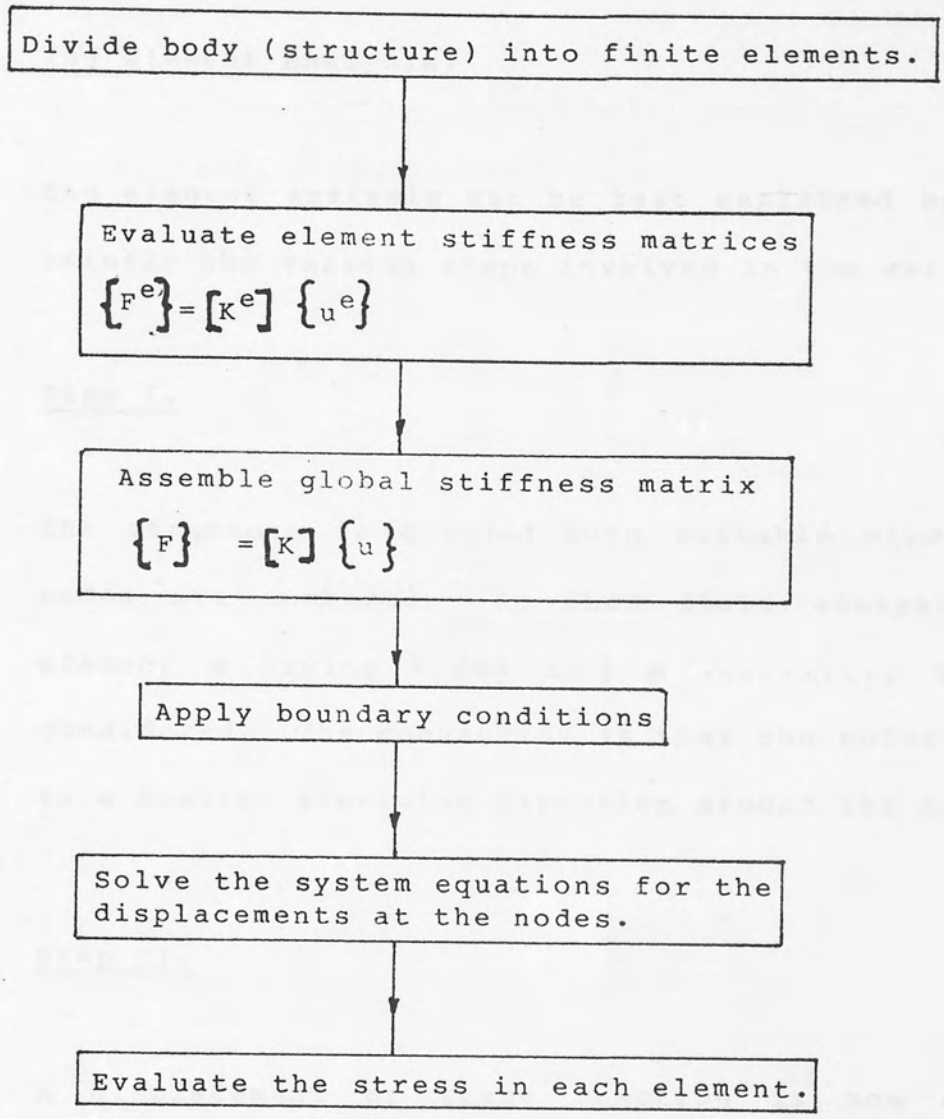
2). The system analysis.

The element analysis consists of the derivation of the relationship between the nodal point forces and the nodal point displacements for each element.

i.e. $\{F^e\} = [K^e] \{u^e\}$ 5.1

The system analysis involves the assembly and solution of the corresponding system of equations governing the behaviour of the complete body.

i.e. $\{F\} = [K] \{u\}$ 5.2



BASIC STEPS REQUIRED FOR THE FINITE ELEMENT METHOD USING THE DISPLACEMENT APPROACH.

FIG 5.2

(1) Element Analysis.

The element analysis can be best explained by considering briefly the various steps involved in the derivation.

Step 1.

The structure is divided into suitable elements and the nodes are numbered. In this short analysis a general element **e** having nodes **i j m** ...., etc. will be considered. The convention is that the nodes are numbered in a counter clockwise direction around the element.

Step II.

A displacement or shape function is now chosen which uniquely defines the state of displacement throughout the element in terms of the nodal displacements. This function is usually represented by a polynomial and the choice of this function is a critical step in the finite element method. The displacement at any point **(x, y, z)** maybe written in matrix notation as:

$$\{u(x,y,z)\} = [\ F\ (x,y,z)\ ]\{\alpha\} \qquad ....5.3$$

where $\{\alpha\}$ is a vector of the unknown coefficients of the displacement or shape function. Expressing 5.3. in terms of the nodal displacements $\{u^e\}$

$$\{u^e\} = [\ A\ ] \{\alpha\} \qquad \text{..........5.4}$$

Therefore from 5.3 and 5.4 the displacement of any point within the element in terms of the nodal displacements is:

$$\{u\ (x,y,z)\} = [\ F\ (x,y,z)\ ]\ [\ A\ ]^{-1}\ \{u^e\} \qquad \text{......5.5}$$

Step III.

The strains at any point **(x,y,z)** within the element may now be expressed in terms of the displacements, **u (x,y,z)** at any point and hence in terms of the nodal point displacements $\{u^e\}$. In plane elasticity problems the strains correspond to the first derivatives of displacement while in bending problems the strains are in effect curvatures and correspond to the second derivative of displacement. In general

$$\{\varepsilon(x,y,z)\} = [\partial] \{u(x,y,z)\} \qquad \ldots\ldots\ldots 5.6$$

Substituting for $\{u(x,y,z)\}$ and letting $[\partial][F(x,y,z)][A]^{-1} = [B(x,y,z)]$ the strain at any point within the element in terms of the nodal displacements maybe written as:

$$\{(x,y,z)\} = [B(x,y,z)] \{u^e\} \qquad \ldots\ldots\ldots 5.7$$

Step IV.

The stresses occuring in the element are related to the strains using the elastic properties of the element, i.e.

$$\{\sigma(x,y,z)\} = [D] \{\varepsilon(x,y,z)\} \qquad \ldots\ldots\ldots\ldots 5.8$$

Using this relationship and equation 5.7 the stresses in terms of the nodal displacements maybe written as

$$\{\sigma\} = [S] \{u^e\} \qquad \ldots\ldots\ldots\ldots 5.9$$

where $S = [D][B(x,y,z)]$.

Step V.

The generalised coordinate stiffness of the element $[K^e]$ can now be obtained by applying the Principle of Virtual Work, which may be stated in general terms as: "For a system of forces acting on a particle, the particle is in equilibrium if, when it is given a virtual displacement, the net work done by the forces is zero". This principle may be stated as follows for application to this problem. "When the element is subjected to a virtual displacement the internal work done by the stresses equals work done by the nodal forces". Using this principle reference [11]

$$\{F^e\} = \int^{Vol} [\, B(x,y,z)\,]^T \, [\, D\,] \, [\, B\,(x,y,z)\,] \, \partial(\mathrm{vol}) . \{u^e\} \qquad \ldots\ldots\ldots\ldots 5.10$$

$$\text{or} \quad \{F^e\} = [\, K^e\,] \, \{u^e\}$$

where the element stiffness matrix $[\, K^e\,]$ is given by :-

$$[\, K^e\,] = \int^{Vol} [\, B\,]^T \, [\, D\,] \, [\, B\,] \, \partial(\mathrm{vol}) \qquad \ldots\ldots\ldots 5.11$$

II) System Analysis.

The first step in the system analysis involves the assembly of all the element stiffness matrices into the correct position of the global stiffness matrix so that a system of equations of the form

$$\{F\} = [\ K\ ]\{u\} \qquad \ldots\ldots\ldots\ldots 5.12$$

may be obtained.

The global stiffness matrix is symmetric, positive definite and if care is taken in the order in which the nodes are numbered, the terms in this matrix will be banded about the principal diagonal. This property of the stiffness matrix results in saving in computer storage because banded and frontal solution subroutines have been written which require only the upper or lower portion of the banded section of the matrix to be stored in the computer.

The global stiffness matrix as formed from the element stiffness matrices alone is singular and therefore it cannot be solved for the displacements. It is rendered non-singular by the application of the geometric boundary conditions and thus can be solved for the displacements. In engineering terms the application of the boundary conditions prevents rigid body motion of the structure due to the action of the applied forces.

When the global stiffness matrix equations has been solved to obtain the displacements, the stresses on each element can then be calculated using equation 5.9.

5.4. ACCURACY AND CONVERGENCE.

If a continuum is assumed to consist of elements literally connected only at the nodes then, when external loads are applied the behaviour of the idealisation would only crudely approximate the actual structure. As the load is applied, the elements would distort quite independently of each other, except at the nodes, and gaps and overlapping would appear at their edges. Therefore the idealisation of the continuum will be much more flexible than the actual continuum.

In order that the finite element idealisation provides a reasonable approximation to the deformation of the actual continuum, each element must be made to deform in a pattern which is similar to the deformation pattern in adjacent elements. This is accomplished by applying displacement functions to the deformation of each element.

As these assumed displacement or shape functions limit the degrees of freedom of the continuum a true minimum of potential energy will never be reached. Therefore in order that a solution will converge to the correct solution, when the size of the elements are decreased, care must be taken in the choice of displacement functions.

If the displacement functions are chosen so that conformity between elements is obtained, then the solution will converge to the correct value as the element size is decreased. It is also possible to use non-conforming elements and obtain reasonably good results providing the non-conformity is of a minor nature. The convergence of such elements are checked by comparison of results for different mesh sizes with known analytical methods.

Therefore, to ensure convergence it is necessary that the displacement functions should follow the true displacement pattern as closely as possible. In order to achieve this result the displacement function should fulfil certain criteria which may be found in Irons and Draper[32],and Zienkiewicz [11].

5.5. THE SEMI-LOOF ELEMENT

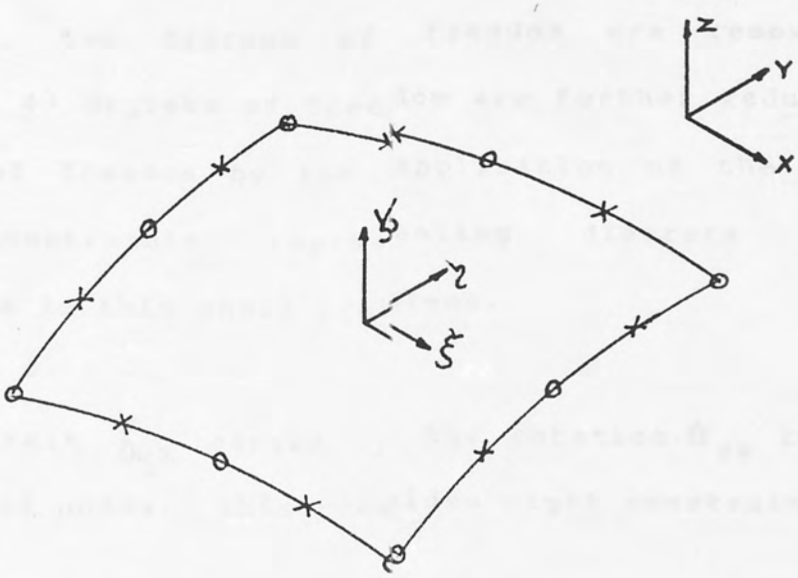
5.5.1. ELEMENT GEOMETRY AND NODAL VARIABLES

Figure 5.3. shows the quadrilaterial and triangular Semi-Loof shell element where the local coordinate system (u,v,w), isoparametric curvilinear coordinate system (ξ , η , ζ) and global reference system (X, Y, Z) are illustrated. There are three types of nodes for this element:-

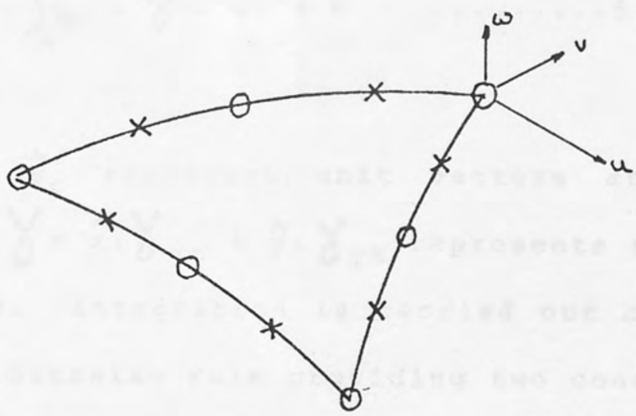
i) Corner and midside nodes, where there are three components of the global displacement vector taken as the nodal variables at each of these nodes.

ii) So called Loof nodes, these are located at the Gauss points along the element sides. The nodal variables at the Loof nodes are two rotations in the local coordinate system with unit vectors $(\hat{x},\hat{y},\hat{z})$ and denoted by the normal θ_{xz} and parallel θ_{xz} to the element edge.

iii) Central node, where there are three components of the local displacement vector and two rotations about the curvilinear coordinates ξ and η are taken as the nodal variables. These constitute 45 degrees of freedom for each element. By combining the displacements at the



QUADRILATERAL SEMI-LOOF ELEMENT
(32 D.O.F.)



TRIANGULAR SEMI-LOOF ELEMENT
(24 D.O.F.)

O CORNER / MIDSIDE NODE
X LOOF NODE

THE QUADRILATERAL AND TRIANGULAR SEMI-LOOF ELEMENTS

Fig 5.3

centre to produce only a normal deflection (bubble function), two degrees of freedom are removed. The remaining 43 degrees of freedom are further reduced to 32 degrees of freedom by the application of the following shear constraints representing discrete Kirchhoff hypothesis in thin shell problems.

a). Shear strain γ_{yz} caused by the rotation θ_{yz} to be zero at the Loof nodes. This provides eight constraints.

b). The area integrals

$$\int_A \hat{x}_c \cdot \gamma \cdot dA = \int_A \hat{y}_c \cdot \gamma \cdot dA = 0 \qquad ....5.13$$

where $\hat{x}_c$ and $\hat{y}_c$ represent unit vectors at the centre $\xi = \eta = 0$ and $\gamma = \hat{x} \cdot \gamma_{xz} + \hat{y} \cdot \gamma_{yz}$ represents the vector of lateral shears. Integration is carried out over the area using a 2 x 2 Gaussian rule providing two constraints.

c).

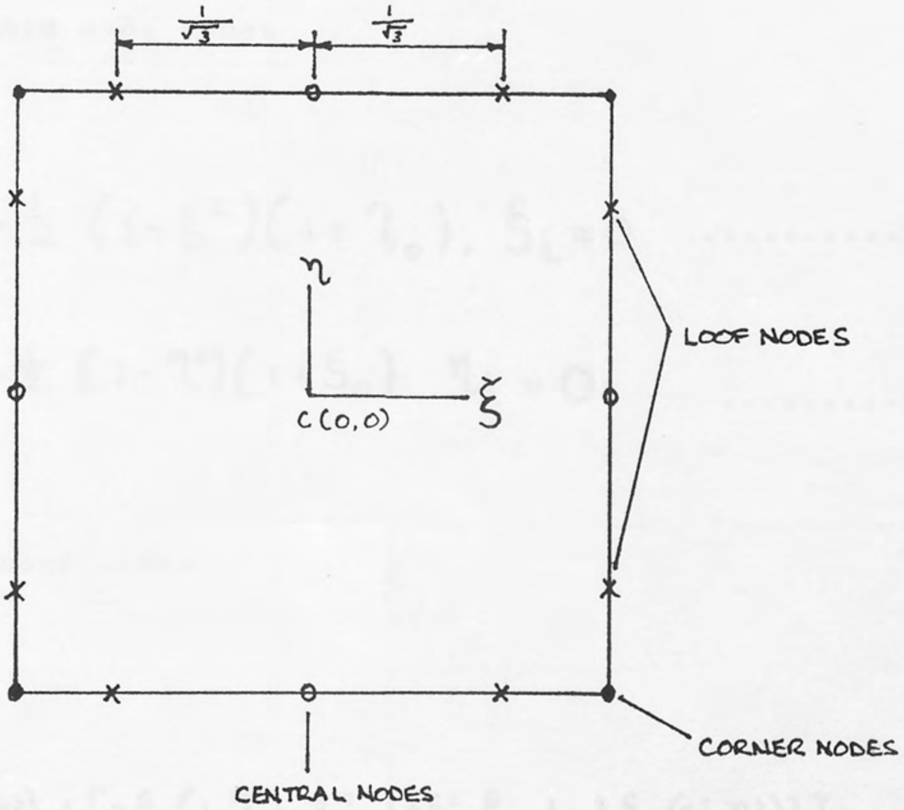
$$\int (\text{thickness})\, \gamma_{xz} \cdot d(\text{boundary}) = 0 \qquad ....5.14$$

This integral carried out over the element boundary provides another constraint.

With the application of the above eleven constraints, the final nodal variables consist of 3 components of displacement at every corner and midside nodes (24 degrees of freedom), together with normal rotations (8 degrees of freedom) at the Loof nodes resulting in an element with 32 degrees of freedom, which are sufficient to define linear stress fields in both membrane and bending action. Similar constraints (except constraint (c) above) are applied for the triangular Semi-Loof element with a final 24 degrees of freedom.

5.5.2. SHAPE FUNCTION POLYNOMIALS.

The Semi-Loof element is a non-conforming element which passes the patch test [33]. Basically the element adopts the well known iso-parametric 8 noded parabolic model. C^1 continunity is maintained by the introduction of the normal rotation variables at the Loof nodes on the element periphery. The following shape functions in terms of curvilinear coordinate systems (ξ, η) are used for quadrilateral elements ($\xi_0 = \xi_i \xi$, $\eta_0 = \eta_i \eta$) (see Fig. 5.4.):-



SEMI - LOOF NODES

Fig 5.4

a). For the corner nodes

$$N_i = \frac{1}{4}\left(1+\xi_0\right)\left(1+\eta_0\right)\left(\xi_0+\eta_0-1\right) \qquad \text{..........5.15}$$

b). For the mid side nodes

$$N_i = \frac{1}{2}\left(1-\xi^2\right)\left(1+\eta_0\right), \; \xi_i = 0 \qquad \text{..........5.16}$$

$$N_i = \frac{1}{2}\left(1-\eta^2\right)\left(1+\xi_0\right) \; \eta_i = 0 \qquad \text{..........5.17}$$

c). For the Loof nodes

$$N_j = \frac{3}{32}\left(3\xi^2-\eta^2\right)+\frac{1}{8}\left[3\xi_0\left(1-\eta^2\right)+3\eta_0\left\{3\xi^2+\xi_0-1+\frac{3}{2}\xi_0\left(\xi^2-\eta^2\right)\right\}\right] \qquad \text{..5.18}$$

$$\xi_j = \pm 1$$

$$N_j = \frac{3}{32}\left(3\eta^2-\xi^2\right)+\frac{1}{8}\left[3\eta_0\left(1-\xi^2\right)+3\xi_0\left\{3\eta^2+\eta_0-1+\frac{3}{2}\eta_0\left(\eta^2-\xi^2\right)\right\}\right] \qquad \text{...5.19}$$

$$\eta_j = \pm 1$$

d). The central node i.e. the bubble function

$$N_c = \left(1-\xi^2\right)\left(1-\eta^2\right) \qquad \text{..........5.20}$$

with the explicit forms for the shape function, it is straight forward to define the matrix of shape functions and its derivatives.

The centrifugal fan impeller is a complicated structure consisting of a backplate, conesheet with radially spaced blades continuously welded between them. The selection of a suitable element to model the actual structure when using the Finite Element technique is a very crucial step. The requirements for a suitable element to model the impeller is the ability to deal with the complex geometry and the discontinuous nature of the blade loading, which causes large bending stresses in the backplate and conesheeet without difficulty.

The Semi-Loof shell element has proved to be a very efficient element in many engineering applications having complicated geometry and has been included in many Finite Element packages (eg BERSAFE, ASAS, PAFEC). The degrees of freedom are all low-ordered, the only first derivatives being rotations, so that discontinunity incompatibilities inherent in high order shell elements are avoided. As the centrifugal impeller can be regarded as an assembly of plates and shells and since the Semi-Loof element has proved to be a efficient element, it appears as a suitable choice for the present work. The implementation of the Finite Element method incorporating the Semi-Loof element in a computer code is discussed in the following section.

5.6. THE FINITE ELEMENT STRESS ANALYSIS COMPUTER PROGRAM

5.6.1. DESCRIPTION OF THE SUBROUTINES IN THE FINITE ELEMENT PROGRAM

The subroutines discussed in this section, are those published in [34]. The modifications necessary to the published subroutines are discussed in section 5.6.4. The main segment **"FINITE"** defines the long vector **VEC** which is to be used by the flexible size arrays. The size of the vector needs to be changed depending on the size of the problem being solved i.e. the number of elements and nodes in the particular problem. The main segment then calls a series of subroutines.

The first subroutine to be called **"NURSE"** reads the first few sets of data and deduces from them the size of the job i.e the number of elements, number of nodes with fixed values, number of nodes with additional point loads, the number of nodes with additional stiffnesses and the number of right hand sides. If there are any "fatal" errors in the set of data (e.g. there is not enough storage due to the size of vector **VEC**, or the number of elements is zero), the subroutine **"DOCTOR"** is called, the error details are printed and the run is terminated. If there are no fatal errors the subroutine **"NURSE"** continues to read the next set of data which gives details of the element type (e.g. Semi-Loof beam or shell), element property number (e.g. where the type of element may have different properties e.g. thickness) and the element nodal connections. After reading this set of data, the subroutine **"DOCTOR"** may again be called if there are fatal errors and again the run will terminate. A full list of fatal errors (where the run terminates immediately) and non fatal errors (where the run is allowed to continue a little longer in the hope of discovering further errors) are given in Appendix No.4.

Assuming there are no fatal errors, the subroutine returns to the main segment where the next subroutine to be called is **"INPUT"**. This subroutine reads the remaining mass of input data, i.e. the nodal coordinates, the nodes with prescribed values and the prescribed values, nodes with extra point loads and the values, nodes with extra stiffness if any and the element properties. The element properties are : Young's Modulus of elasticity, Poisson's ratio, density, uniform pressure loading acting on it and thickness. Again the subroutine **"DOCTOR"** may be called if there are any fatal or non fatal errors. Assuming there are none, the subroutine returns to the main segment for the next subroutine **"MATRON"** to be called.

This subroutine does the entire pre-front search, and marks the last appearance of each node by making the entry in the array **LNODS** negative. This is in order to know when a nodal variable can be eliminated in the subroutine **"FRONT"**. Again the subroutine **"MATRON"** may call subroutine **"DOCTOR"** if any errors are detected. If these are again non fatal errors the subroutine returns to the main segment of the program, otherwise the error number is printed and the run terminated.

The next subroutine to be called by the main segment is **"ELFILE"**, which itself calls **"SHELL"**, the subroutine for the Semi-Loof shell element. The subroutine **"ELFILE"** selects the element subroutine (i.e. Semi-Loof shell or beam, in this work only the shell element subroutine has been coded) and prepares the files on back-up tape storage on which the element stress, element stiffness and their nodal load contributions matrices will be stored. The subroutine then calls the subroutine **"SHELL"** which is the subroutine for the Semi-Loof shell element with 2 x 2 point integration at the Gauss points. The resulting stresses are also calculated at the Gauss points. The subroutine extracts all the relevant material properties from the given list (i.e. the Young's Modulus of elasticity, Poission's ratio, the density of the material and the pressure loading on the element). The subroutine calculates the integration point and calls the subroutine **"HALOOF"**. This subroutine creates the shape function array for the Semi-Loof shell element at the curent integration point. In doing so it calls the subroutine **"SFR"**, which creates the polynominal terms and ξ, η derivatives. The subroutine **"HALOOF"** returns to the subroutine **"SHELL"** having created the shape function array. The subroutine **"SHELL"** then creates the Modulus matrix and generates the matrix [B] using the derivatives from the shape functions array. Having obtained the Modulus matrix and the [B] matrix the element stiffness is calculated. Also calculated at this point are the nodal loads due to gravity together with those due to the normal pressure acting on the element and the stress matrix. The subroutine then returns to the calling subroutine **"ELFILE"** where the feasibility of the stiffness matrix that emerged from the subroutine **"SHELL"** is checked to see if any of the diagonal terms are zero or negative. In either case an error message is printed and the run terminated. If there are no such errors in the stiffness matrix, the matrix is transferred to file on back-up tape storage together with the array containing the nodal loads due to the forces acting on the element. This process is repeated, until all the stiffness matrices, stress matrices and the load vectors for all the elements in the job have been created and stored on back-up tape storage. The subroutine then returns to the main segment **"FINITE"** for the next subroutine **"FRONT"** to be called, which is the equation solving subroutine, using the "Frontal" technique. The "Frontal" solution technique is described in detail in the section 5.6.2.

The subroutine **"FRONT"** assembles the global stiffness matrix and load vector and solves the resulting equations.

Having calculated the nodal deflections, the subroutine calls the subroutine **"STRESS"** which calculates the tensions and the bending moments per unit width. The principal values and their directions are calculated by calling the subroutine **"PRINPL"**. After completing the back substitution process the subroutine continues to print the reactions at the nodes with prescribed values.

5.6.2. THE FRONTAL SOLUTION TECHNIQUE

The method adopted for equation solution is a major factor influencing the efficiency of any finite element program. Several options are open to the programmer ranging from iterative methods such as Gauss-Seidel technique [35] to the direct Gaussian elimination algorithms and the frontal solution technique. The frontal equation solution technique was originated by Irons [36] and has earned the reputation of being easy to use.

The frontal method can be considered as a particular technique for first assembling finite element stiffnesses and nodal forces into a global stiffness matrix and load vector and then solving for the unknown displacements by means of a Gaussian elimination and back substitution process. It is designed to minimise core storage requirements, the number of arithmetic operations and the use of peripheral equipment (i.e. backing storage onto magnetic tapes).

The main idea of the frontal solution is to assemble the equations and eliminate the variables at the same time. As soon as the coefficients of an equation are completely assembled from the contributions of all the relevant elements, the corresponding variables can be eliminated. Therefore the complete structural stiffness matrix is never formed as such, since after elimination the reduced equation is immediately transferred to back-up tape storage. The core contains, at any given instant, the upper triangular part of a square matrix containing the equations which are being formed at that particular time. These equations, their corresponding nodes and degrees of freedom are termed the front. The number of unknowns in the front is the front width, this length generally changes continually during the assembly/reduction process. The maximum size of problem which can be solved is governed by the maximum front width. The equations, nodes and degrees of freedom belonging to the front are termed active, those which are yet to be considered are inactive, and those which have passed through the front and have been eliminated are said to be deactivated.

During the assembly/elimination process the elements are considered each in turn according to a prescribed order. Whenever a new element is called in, its stiffness coefficients are read from back-up tape storage and summed either to existing equations, if the nodes are already active, or into new equations which have to be included in the front if the nodes are being activated for the first time. If some nodes are appearing for the last time (i.e. they have a negative sign in front of them from the subroutine "MATRON"), the corresponding equations can be eliminated and stored on back-up tape storage and are thus deactivated. In doing so they free space in the front which can be used during assembly of the next element.

When a node which has its displacement prescribed (i.e. the boundary conditions) is to be eliminated, the process is straight forward. The right hand sides of the system of equations are modified and the corresponding column of the front matrix (except the diagonal term) is set to zero. The modification to the right hand side is that in addition to the nodal force (F), which is the sum of the contributions coming from all elements connected by the node, there must be a concentrated force (R) applied at the node to produce the prescribed displacement, the value of which can only be found after the backsubstitution phase.

The backsubstitution process can best be understood as the frontal process applied in reverse order. Although the elimination process has been preformed in an untraditional way i.e. in the ordinary Gaussian process variables are eliminated in the order in which they are encountered going down the matrix, whereas in the frontal technique the order of elimination is governed by the available space in the front, the reduced matrix which is transferred to back-up tape storage, can be treated in the usual way for backsubstitution. Moving upwards from the last equation, each new equation considered introduces only one unknown quantity which can be directly calculated.

When an element has been re-processed, all its nodal displacements are known, since either they have appeared for the last time with this element in the assembly process and been eliminated and hence recovered on re-processing of the element, or they have appeared later in the assembly process and have already been recovered when the elements are taken in reverse order for backsubstitution. The stresses are also calculated at this stage.

In the case where nodal displacements are prescribed, the nodal reaction (R) which is the force required to balance the force contribution from the neighbouring loaded elements and to produce the imposed displacement is calculated immediately after the backsubstitution process.

5.6.3. THE ADVANTAGES OF THE FRONTAL SOLUTION TECHNIQUE

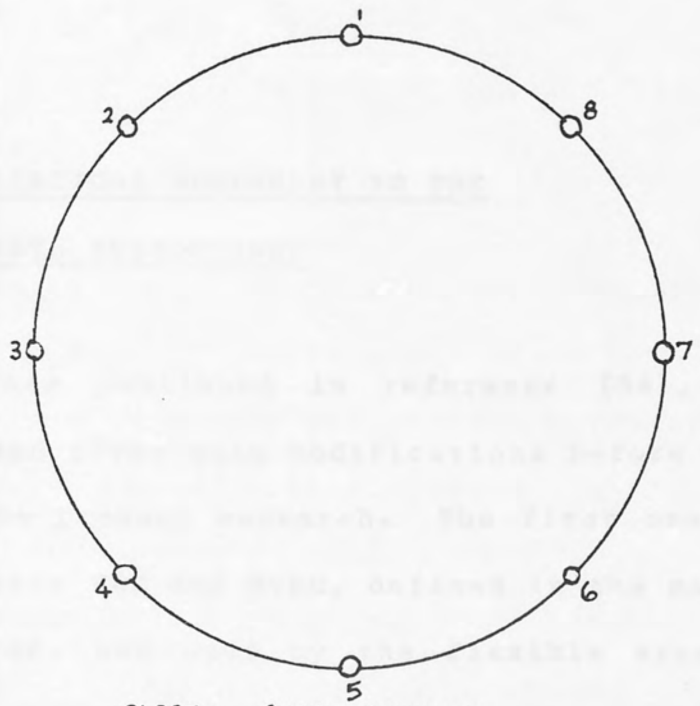
The frontal solution is a very efficient direct solution process. Its main attraction is that variables are introduced at a later stage and eliminated earlier than in most other methods. This helps to reduce numerical inaccuracies due to round off. The active life of a node lasts from the time in which it first appears in an element to the time in which it last appears in an element. This has the following consequences:-

1). The ordering of elements is crucial and the ordering of nodal numbering is irrelevant. This is an opposite requirement to that for a banded solution and one easier to satisfy since, invariably, there are fewer elements than nodes, especially in three dimensional problems. Furthermore if a mesh is found to be too coarse in some region, its modification does not require extensive nodal point renumbering.

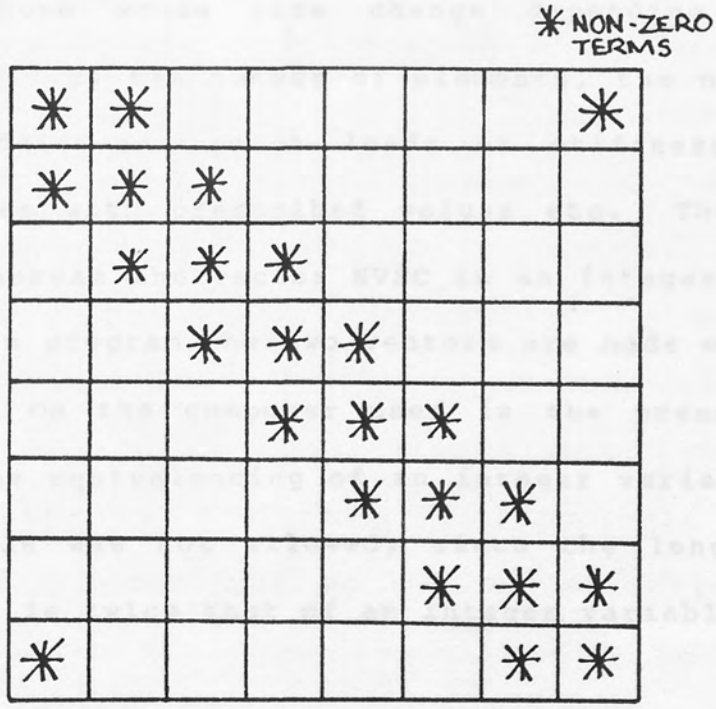
2). The core storage requirements are at most the same as those of a banded Gaussian solution. The core storage required is less for structures analysed by means of elements with midside nodes which give "a re-entrant band" in classical schemes. Several examples which justify this statement can be found in references [36] and [37]. One of the most striking is the problem of the closed ring illustrated in Fig. 5.5. With the nodal points numbered as shown an ordinary band solver leads to the half band width being equal to the total number of equations. This is effectively reduced to 3 by the frontal technique (of course, the bandwidth for a banded solution can also be reduced by judicious nodal renumbering).

3). Due to the compact nature of the front and because variables are eliminated as soon as possible, the operations on zero coefficients are minimised and the total arithmetic operations are fewer than with other methods. On the other hand, an elaborate housekeeping system is required for frontal solution. However, since it is entirely preformed with integer variables, little storage and computer time is used.

4). As any new equation occupies the first available space in the front, there is no need for a bodily shift of the



CLOSED RING PROBLEM



EXAMPLE OF A NON-BANDED EQUATION SYSTEM

Fig 5.5

in-core equations as in many other large capacity equation solvers.

5.6.4. MODIFICATIONS NECESSARY TO THE PUBLISHED SUBROUTINES

The subroutines published in reference [34], described earlier, needed three main modifications before they could be used in the present research. The first one concerned the long vectors **VEC** and **NVEC**, defined in the main segment of the program, and used by the flexible arrays in the subroutines called by the main program. The flexible arrays are those whose size change depending on the particular job i.e. the number of elements, the number of nodes with additional point loads or stiffnesses, the number of nodes with prescribed values etc. The vector **VEC** is real whereas the vector **NVEC** is an integer vector, and in the main program the two vectors are made equal (ie equivalenced). On the computer used in the present work (ICL 1904S) the equivalencing of an integer variable with a real variable was not allowed, since the length of a real variable, is twice that of an integer variable.

To overcome this difficulty the vector **NVEC** was declared as a real vector, which meant that the flexible arrays associated with this vector had to be declared real as well. This method had the advantage of involving the least modification to the existing subroutines, but had the disadvantage of increasing the main memory requirements since real variables require twice as much storage as integer variables.

With this modification and the two discussed below, several plate and shell problems for which known solutions exist were solved to verify the results obtained using the subroutines and these are discussed in Section 5.7. However, due to the slow turn around time experienced between submitting the job to the batch system and obtaining the results, it was suggested by the computer centre that the program files be transferred onto the University of Manchester CDC Computer which had a much faster processing capability and hence would improve the turn around time for the jobs. However, the Manchester system had a memory limitation of 70 K bytes for each user, which was sufficient for problems with up to approximately 80 elements. In the case of the centrifugal fan this would result in a very coarse mesh. Additional memory (50 K bytes) was available, but this meant that the flexible array approach had to be abandoned, since any variables stored in the extra memory had to be of fixed size. This meant that as the size of the job changes, the size of the individual arrays would have to be changed manually each time, whereas previously changing the size of the vectors **NVEC** and **VEC** alone was sufficient.

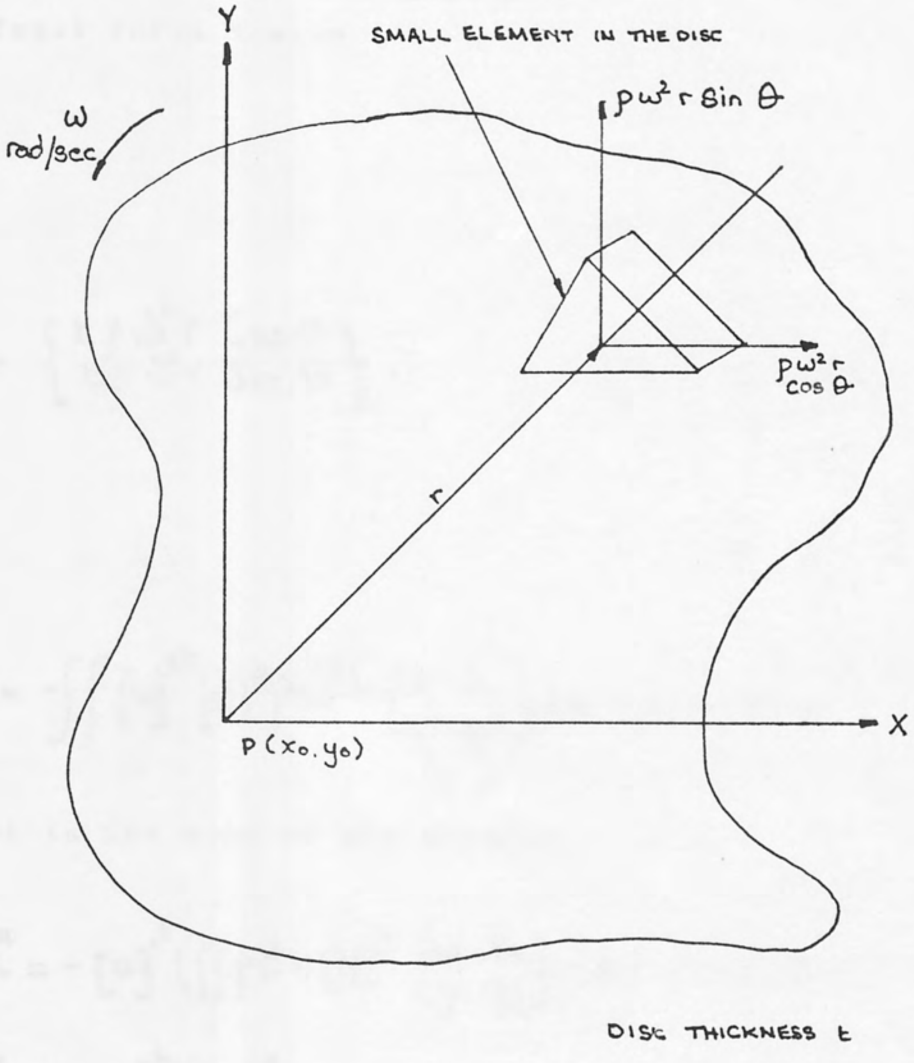
The subroutine **"STRESS"** as given in reference **[34]** printed the principal tensions and principal bending moments, per unit width. Since normally we are interested in the membrane and bending stresses, modification to achieve this was necessary. To achieve this the stress matrix in the subroutine **"SHELL"** was altered accordingly.

The third modification necessary to the subroutines was due to the fact that the subroutine **"SHELL"**, the Semi-Loof shell element subroutine catered for uniformly pressure loading and point loads at particular nodes only. In the case of centrifugal fans, the main loading is as a result of the centrifugal force due to rotation. Since this type of loading was not being catered for, the load vector due to this type of force had to be calculated. The load vector due to the centrifugal force was calculated as follows:-

Consider a plane disc spinning about the z axis at the point (x_o, y_o) as shown in Fig. 5.6. The potential energy is given by:-

$$\Omega = \sum \Omega^e \qquad \text{..................5.21}$$

where



PLANE DISC ROTATING ABOUT THE Z AXIS (OUT OF THE PAPER) AT THE POINT (X_0, y_0)

Fig 5.6

$$\Omega^e = -\iint \{u\}^t \{F\}\, dx\, dy \quad \ldots\ldots\ldots\ldots\ldots\ldots 5.22$$

where $\{F\}$ is a vector of x and y components of the centrifugal force i.e.:-

$$\{F\} = \begin{Bmatrix} t\rho\omega^2 r \cos\theta \\ t\rho\omega^2 r \sin\theta \end{Bmatrix} \quad \ldots\ldots\ldots\ldots\ldots\ldots 5.23$$

Then

$$\Omega^e = -\iint \{u\}^{et} [N]^t \rho\omega^2 t \begin{Bmatrix} x - x_o \\ y - y_o \end{Bmatrix} dA \quad \ldots\ldots 5.24$$

where dA is the area of the element.

i.e. $\Omega^e = -\{u\}^{et} \left(\iint \rho\omega^2 t [N]^t \begin{Bmatrix} x - x_o \\ y - y_o \end{Bmatrix} dA \right) \quad \ldots\ldots 5.25$

or $\Omega^e = -\{u\}^{et} \{P\}^e \quad \ldots\ldots\ldots\ldots\ldots\ldots 5.26$

where

$$\{P\}^e = \iint \rho\omega^2 t [N]^t \begin{Bmatrix} x - x_o \\ y - y_o \end{Bmatrix} dA \quad \ldots\ldots 5.27$$

Integrating equation 5.27 gives the desired load vector due to the centrifugal force. Since the subroutine

"SHELL" carries out the integration at the Gauss points of the element, to reduce computer processing time, it was decided to carry out the above integration at the same points, i.e. using the values of x_G , y_G for x and y in **[N]** and $\begin{Bmatrix} x - x_0 \\ y - y_0 \end{Bmatrix}$. This modification was incorporated in the subroutine "SHELL". The example of a rotating solid disc was solved to verify the results obtained due to this modification. The results obtained are discussed in the following section.

5.7. TEST EXAMPLES

To test the set of finite element subroutines given in reference **[34]** and the modifications necessary to them, and also to gain experience in analysing the output produced by the program, several plate and shell examples were solved and the results compared with known solutions. The finite element analysis of the centrifugal fan impeller will be discussed in Chapter 6.

The first class of problem to be solved was that of a square plate with a uniformly distributed load of 1×10^8 N/M^2 intensity with the following edge conditions:-

1). Clamped along all the edges

2). Simply supported along all the edges.

The results are shown in Figs. 5.7 and Fig. 5.8 respectively, and shows that the finite element solution converges to the theoretical solution rapidly as more degrees of freedom are used.

As the fan impeller has sharp junctions, i.e. the blade/backplate and blade/conesheet junctions, the next problem solved was that of an "L" shaped plate as shown in Fig. 5.9. to assess the ability of the Semi-Loof shell element in dealing with structures with sharp junctions. As no known solution to this type of problem exists, deflections were measured experimentally and compared with those obtained using the finite element method. The results obtained are shown in Table 5.1. The finite element results obtained show that they are in very good agreement with those obtained experimentally bearing in mind that a coarse mesh with only 8 elements was used.

The third example to be solved was that of a cylindrical shell roof loaded by its own weight. The details and dimensions of this problem are shown in Fig. 5.10. The straight edges of the shell are free whilst the curved edges are supported by diaphragms which are assumed to be infinitely rigid in their plane and infinitely flexible out of it. The problem was used to test the program because membrane and bending effects are of importance in the solution and also it has been used by many finite element investigators to test the efficiency of their elements and in the development of Finite Element software packages. To assess the rate of convergence of this problem, various triangular (Fig. 5.11) and rectangular (Fig. 5.12) meshes were used and the results were compared with the shallow shell solution of Scordalis and LO [38] and the deep shell solution of Forsberg [39]. The results obtained for the vertical displacement at the centre of the free edge and the vertical displacement along the mid-section are shown in Fig 5.13 and Fig 5.14 respectively. The results show that the triangular element solutions converge much slower than the rectangular element solutions, which converge very quickly to the deep shell solution of reference 39.

Having gained experience in analysing and verifying the output produced by the program, the next problem solved was that of a rotating solid disc. This problem was solved to verify the modification necessary to the subroutines given in reference [34] as the load vector generated did not cater for loads arising as a result of centrifugal forces acting on the element due to rotation about the z axis. The results obtained are shown in Fig. 5.15 and show that they are in very good agreement with the theoretical results. The above examples have shown that very good results are obtained with coarse meshes when using the Semi-Loof shell element and gives confidence that the choice in selecting this element for the presents work was correct.

Having verified the output produced by the finite element program using test examples for which solutions are known, the finite element analysis of the centrifugal impeller was attempted. This is described in chapter 6.

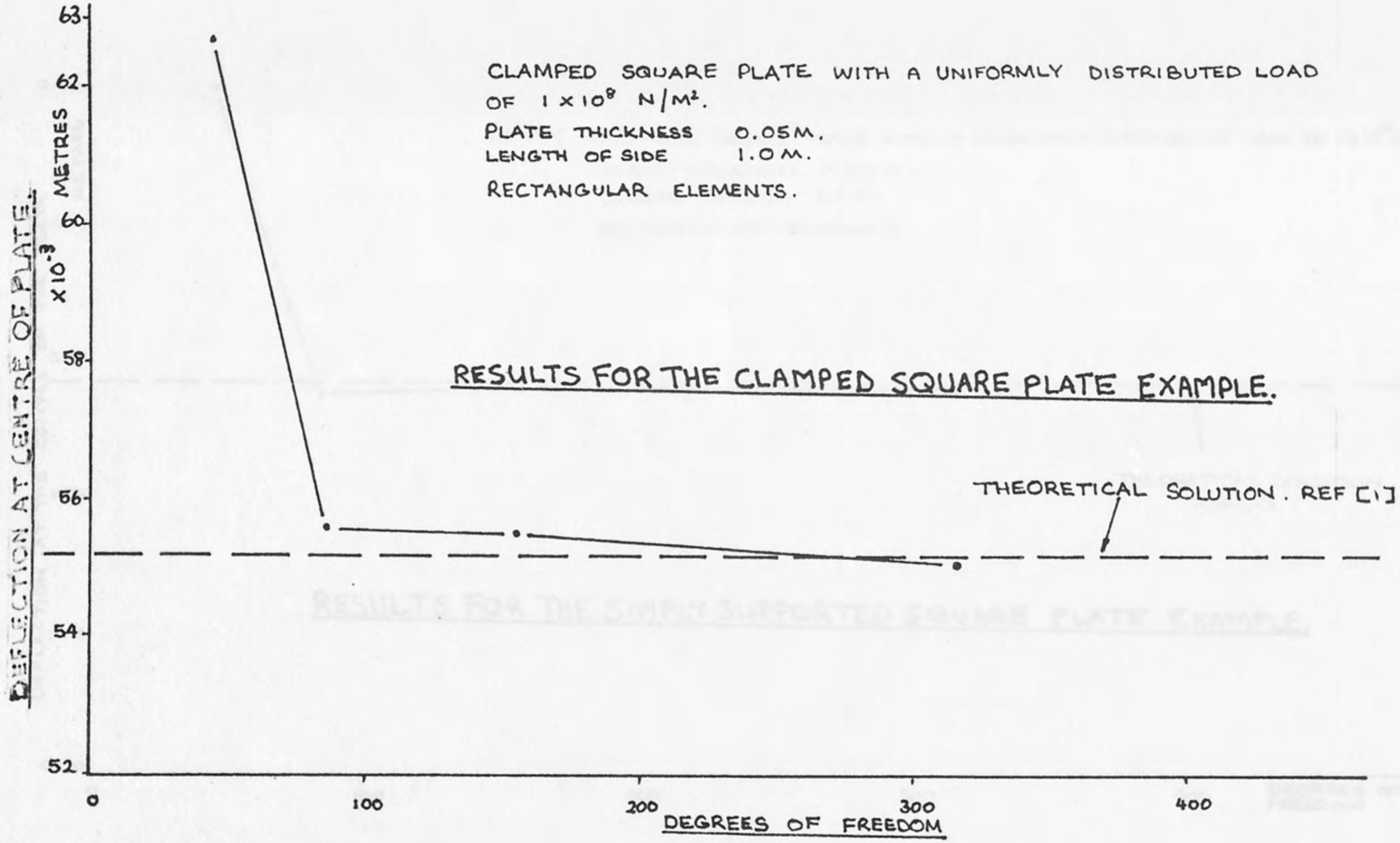
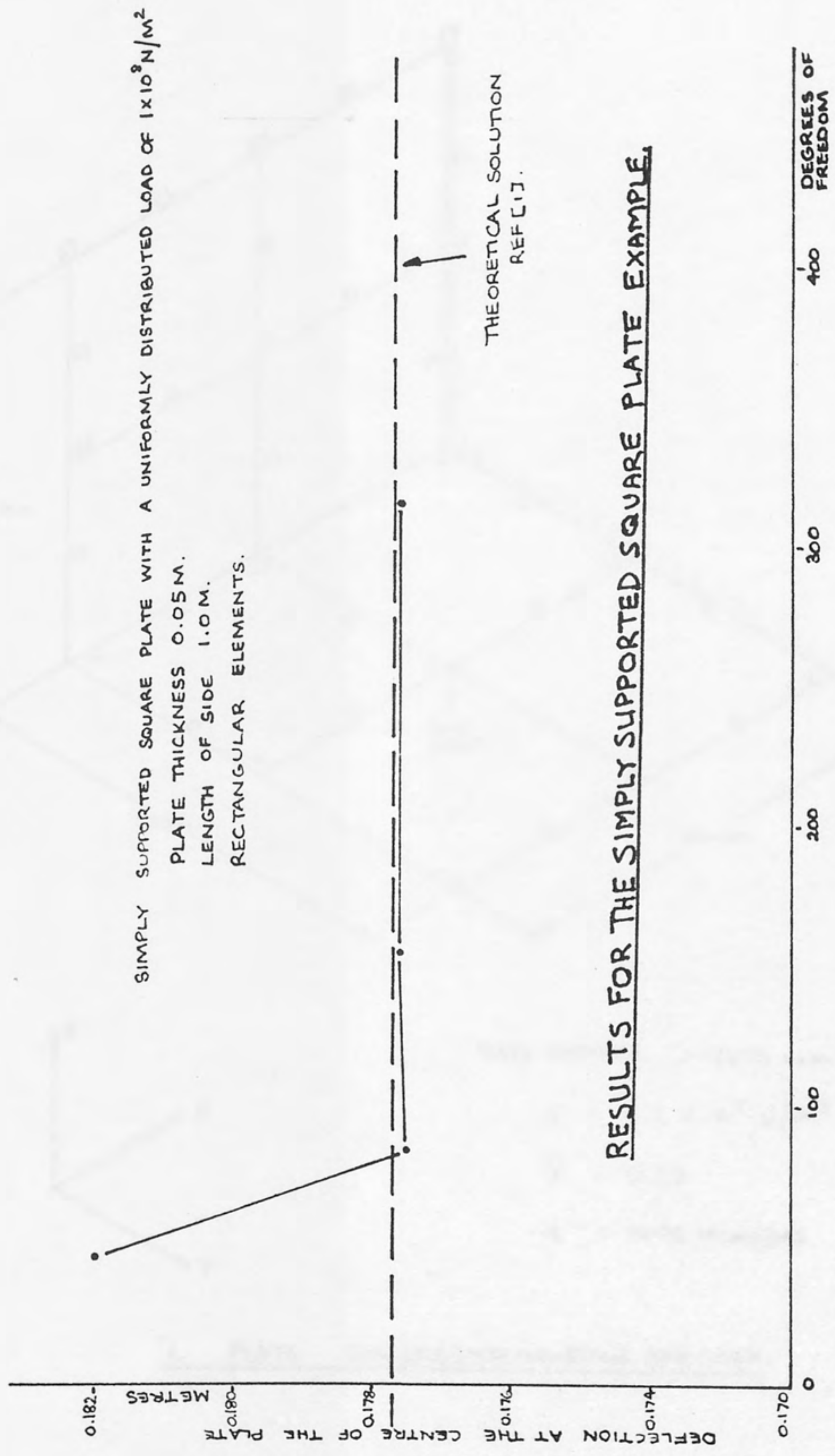
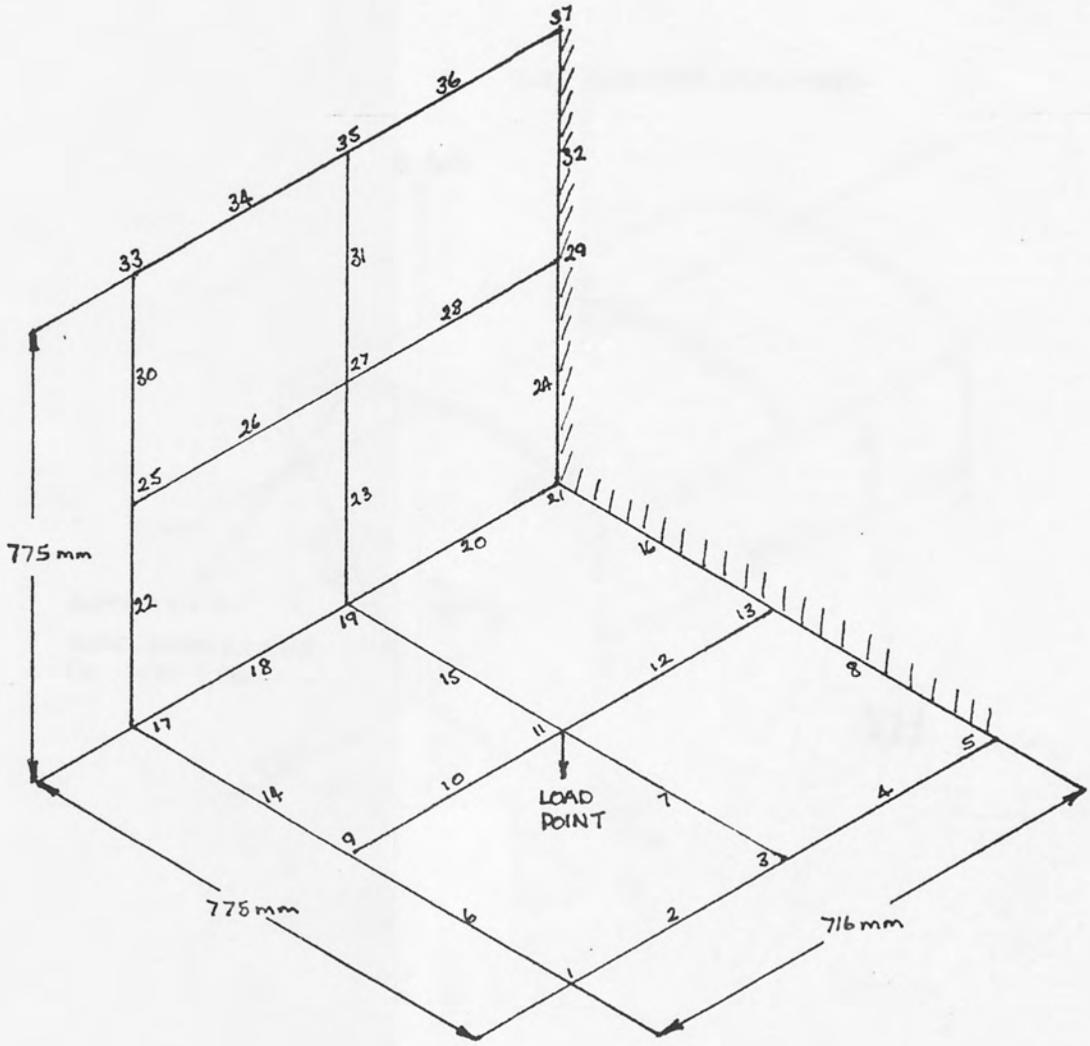

CLAMPED SQUARE PLATE WITH A UNIFORMLY DISTRIBUTED LOAD
OF 1 x 10^8 N/M^2.
PLATE THICKNESS 0.05M.
LENGTH OF SIDE 1.0M.
RECTANGULAR ELEMENTS.
RESULTS FOR THE CLAMPED SQUARE PLATE EXAMPLE.
THEORETICAL SOLUTION. REF [1]
DEFLECTION AT CENTRE OF PLATE.
x10^-3 METRES
63
62
60
58
56
54
52
0
100
200
300
400
DEGREES OF FREEDOM

Fig 5.7



RESULTS FOR THE SIMPLY SUPPORTED SQUARE PLATE EXAMPLE

fig 5.8



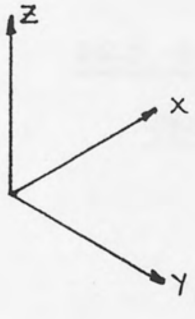


PLATE THICKNESS 1.5875 mm

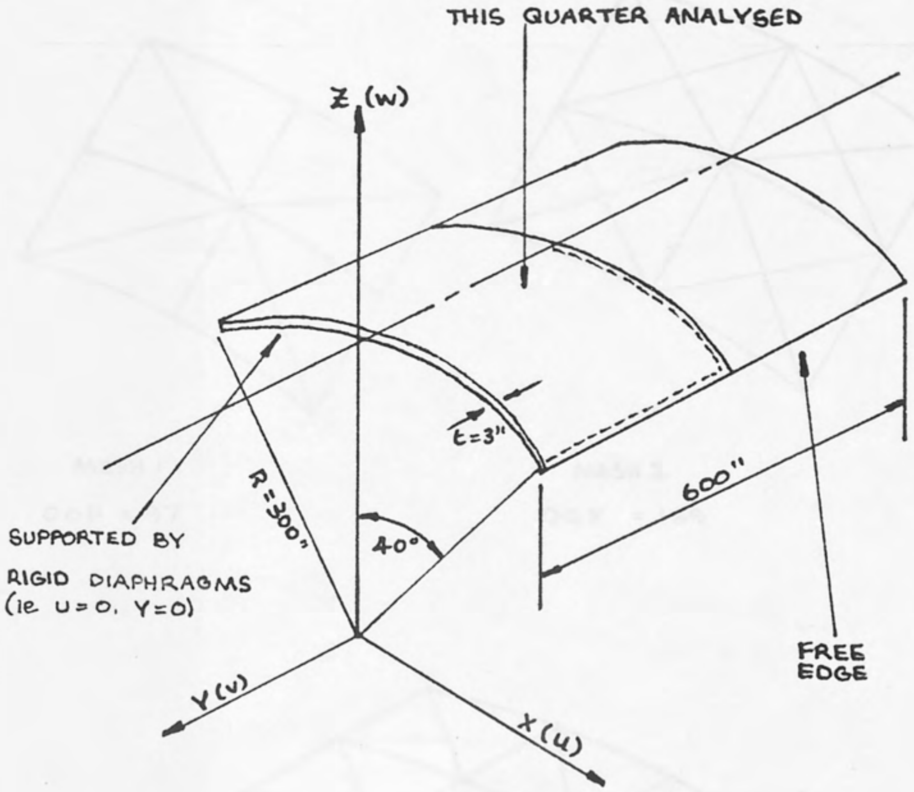
E = 2.1 × 10^7 N/CM^2

ν = 0.29

N = NODE NUMBERS

"L" PLATE EXAMPLE:- DIMENSIONS AND MESH.

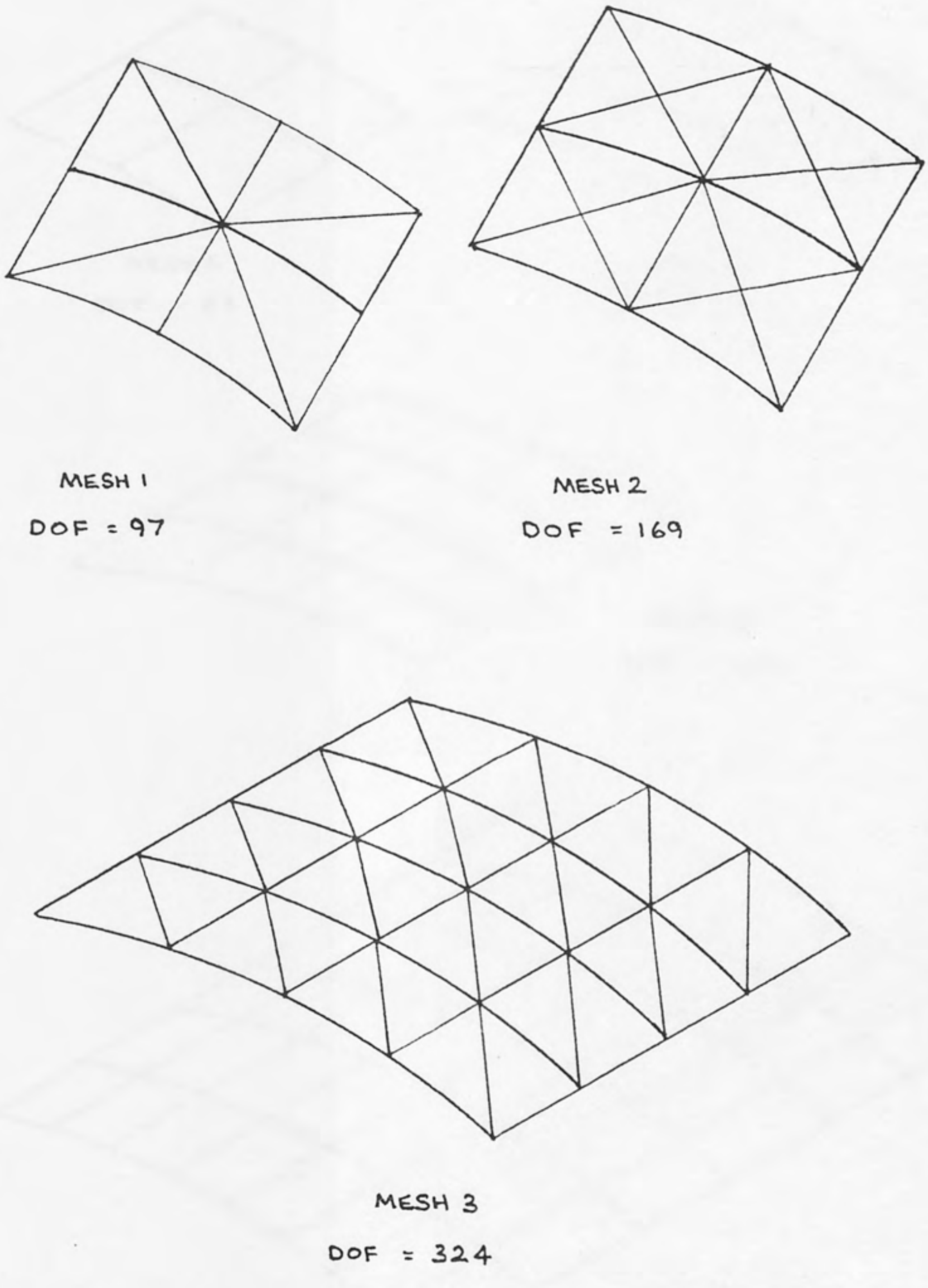
Fig 5.9



SELF - LOADED CYLINDRICAL SHELL ROOF

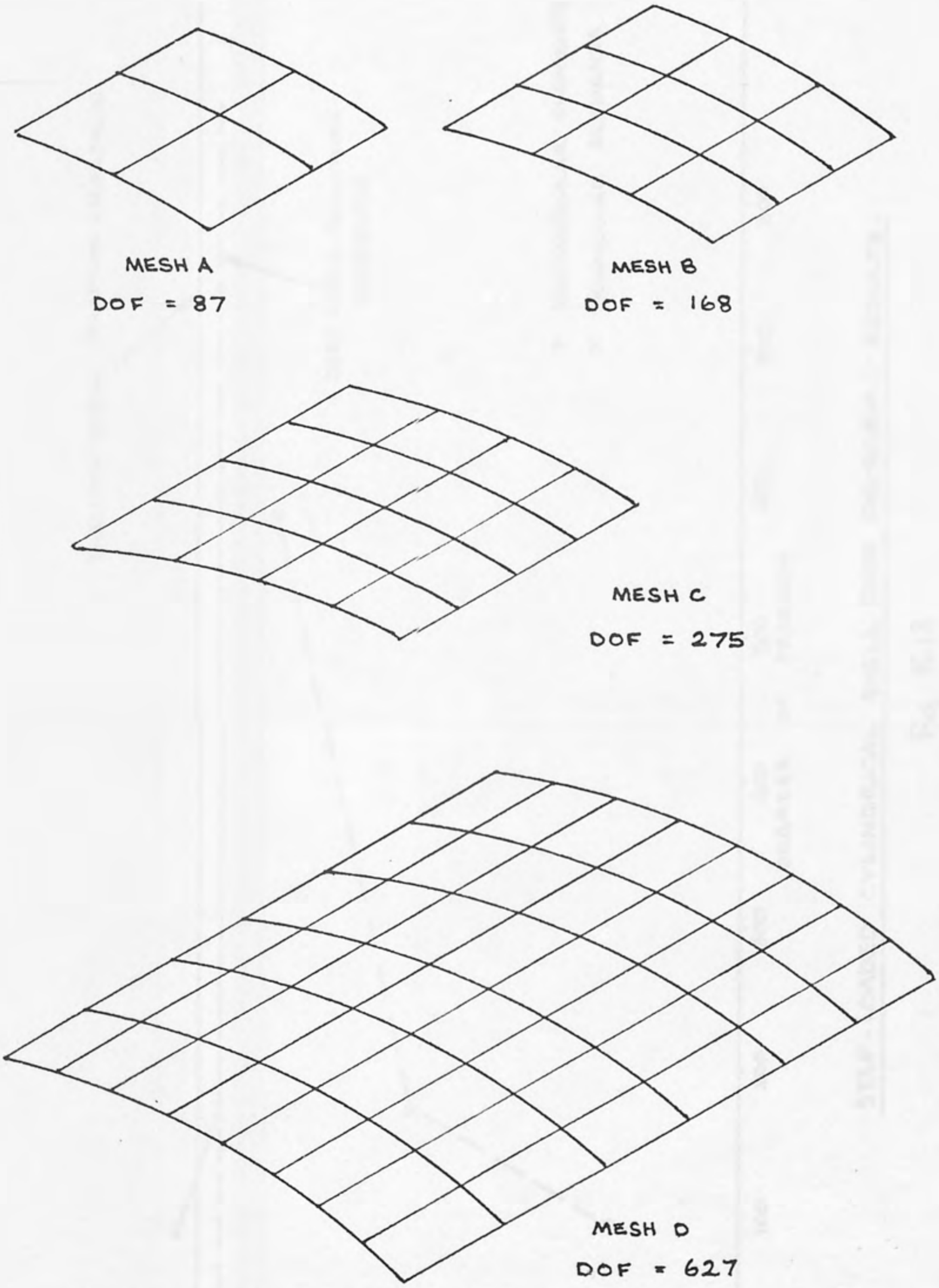
PROBLEM :- DIMENSIONS.

Fig 5.10



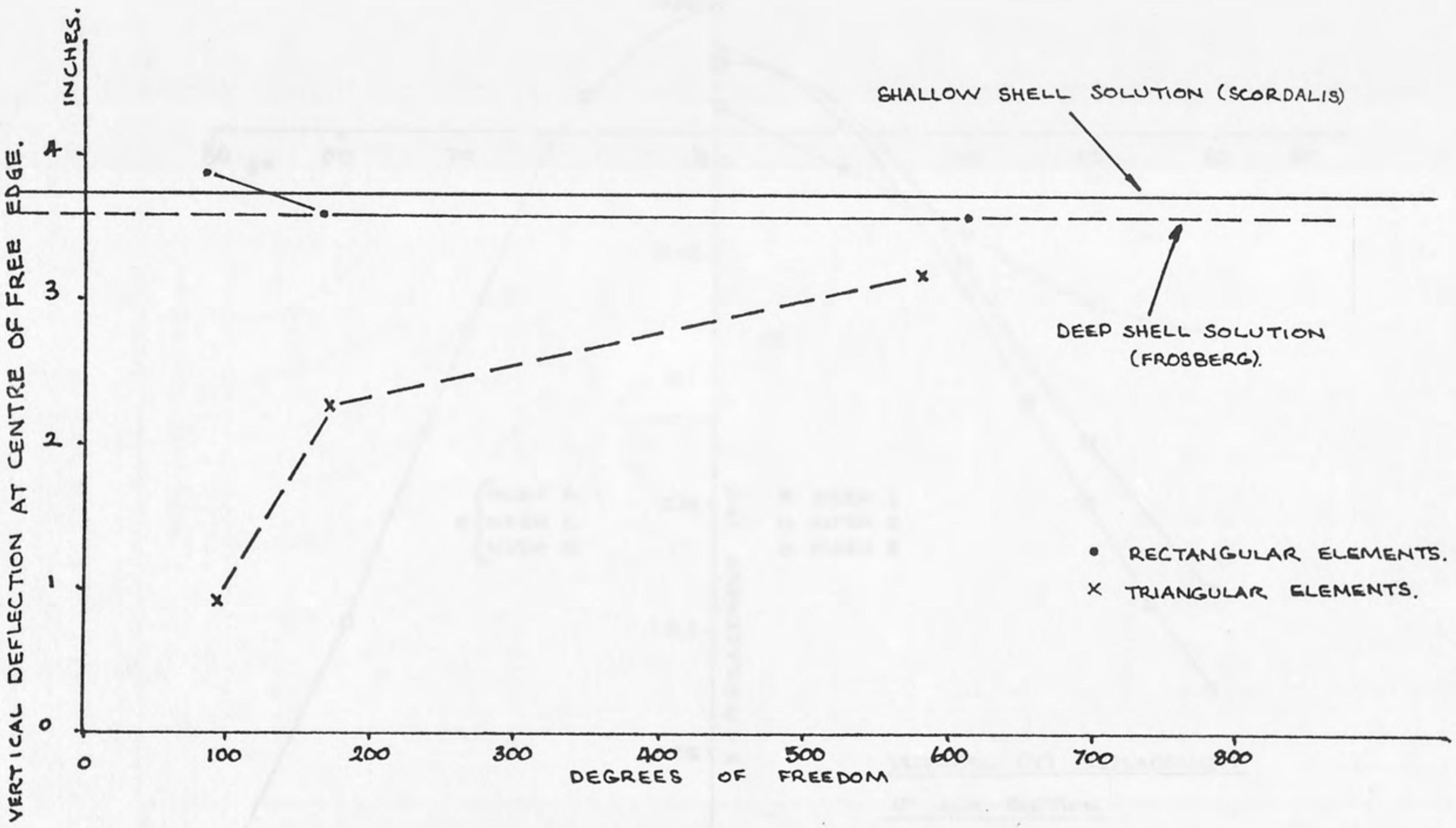
TRIANGULAR MESHES USED IN THE CYLINDRICAL SHELL ROOF PROBLEM.

Fig 5.11



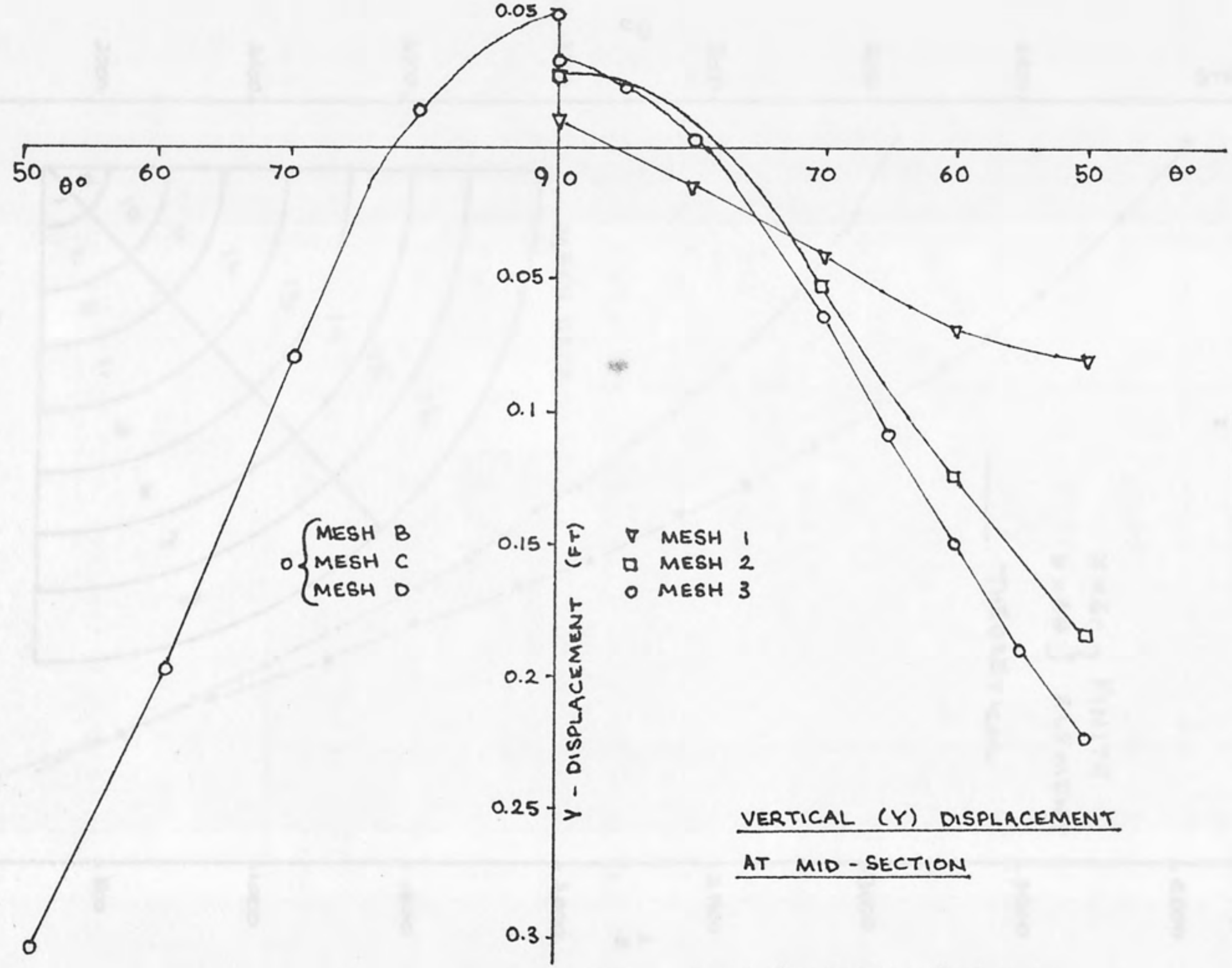
RECTANGULAR MESHES USED IN THE CYLINDRICAL SHELL ROOF PROBLEM.

Fig 5.12



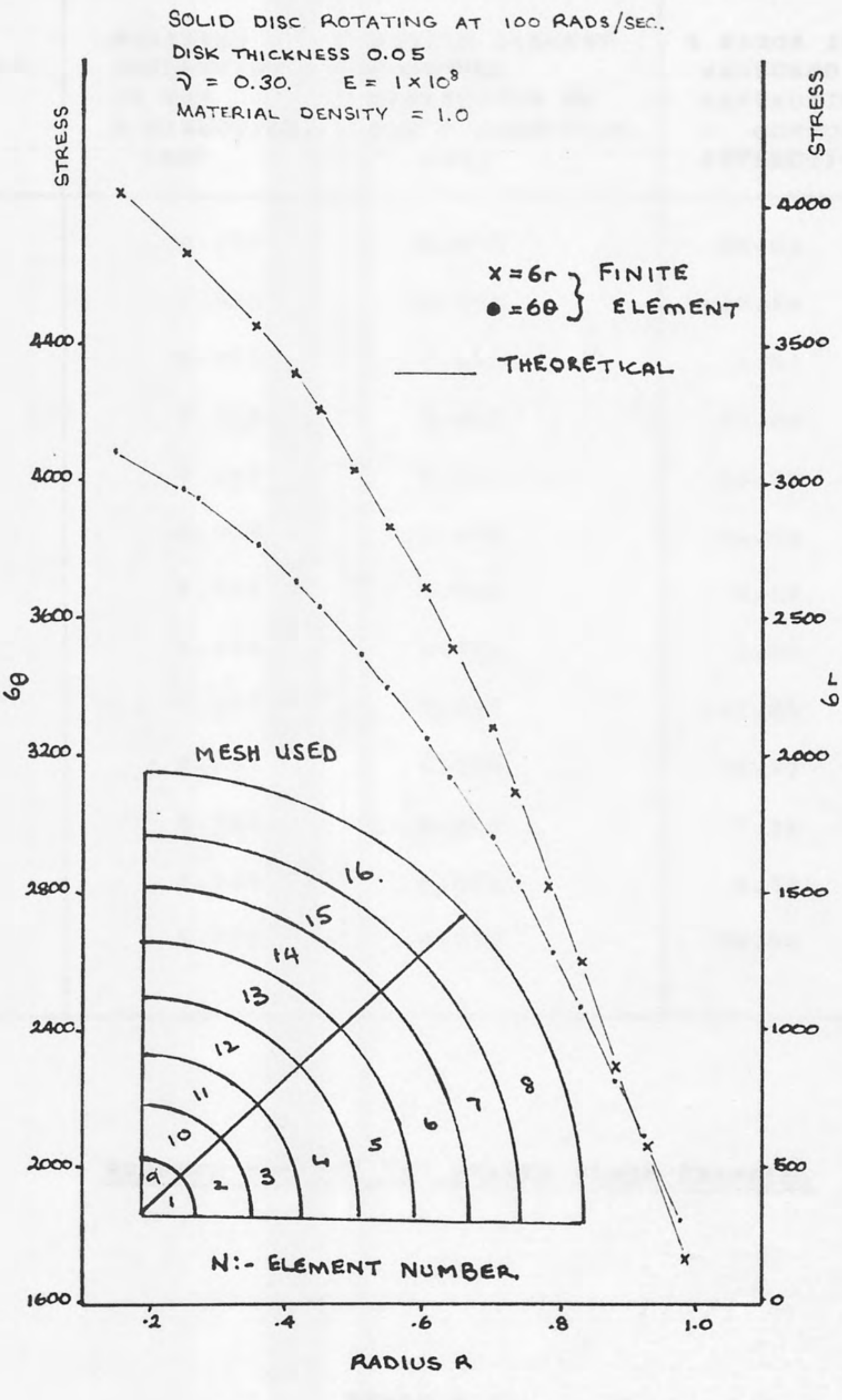
SELF-LOADED CYLINDRICAL SHELL ROOF PROBLEM :- RESULTS.

Fig 5.13



SELF - LOADED CYLINDRICAL SHELL ROOF PROBLEM:- RESULTS

Fig 5.14



ROTATING DISC EXAMPLE :- RESULTS

Fig 5.15

NODE NUMBER.	MEASURED DEFLECTION IN THE Z DIRECTION. (mm)	FINITE ELEMENT COMPUTED DEFLECTION IN THE Z DIRECTION. (mm)	% ERROR IE MEASURED DEFLECTION - COMPUTED DEFLECTION.
1	0.098	0.073	26.02
2	0.068	0.082	-20.59
3	0.057	0.055	3.51
4	0.020	0.023	15.00
7	0.047	0.041	12.77
9	0.088	0.076	13.64
10	0.073	0.066	9.59
11	0.044	0.042	4.55
12	0.017	0.020	-17.65
17	0.061	0.064	-4.92
18	0.068	0.063	7.35
19	0.046	0.042	8.70
20	0.020	0.018	10.00

RESULTS FOR THE "L" SHAPED PLATE EXAMPLE.

TABLE 5.1

CHAPTER SIX

NUMERICAL STRESS ANALYSIS OF THE CENTRIFUGAL FAN IMPELLER

INTRODUCTION

This chapter presents the results obtained in the stress analysis of the centrifugal fan impeller using the finite element method, incorporated in the computer program described in Chapter 5.

Before attempting the finite element solution of the actual centrifugal impeller, to gain insight into the complex stress distribution in it, four simplified models of the impeller were solved for and the results are presented.

Having gained insight using the simple models, the finite element results obtained for the actual impeller are then presented.

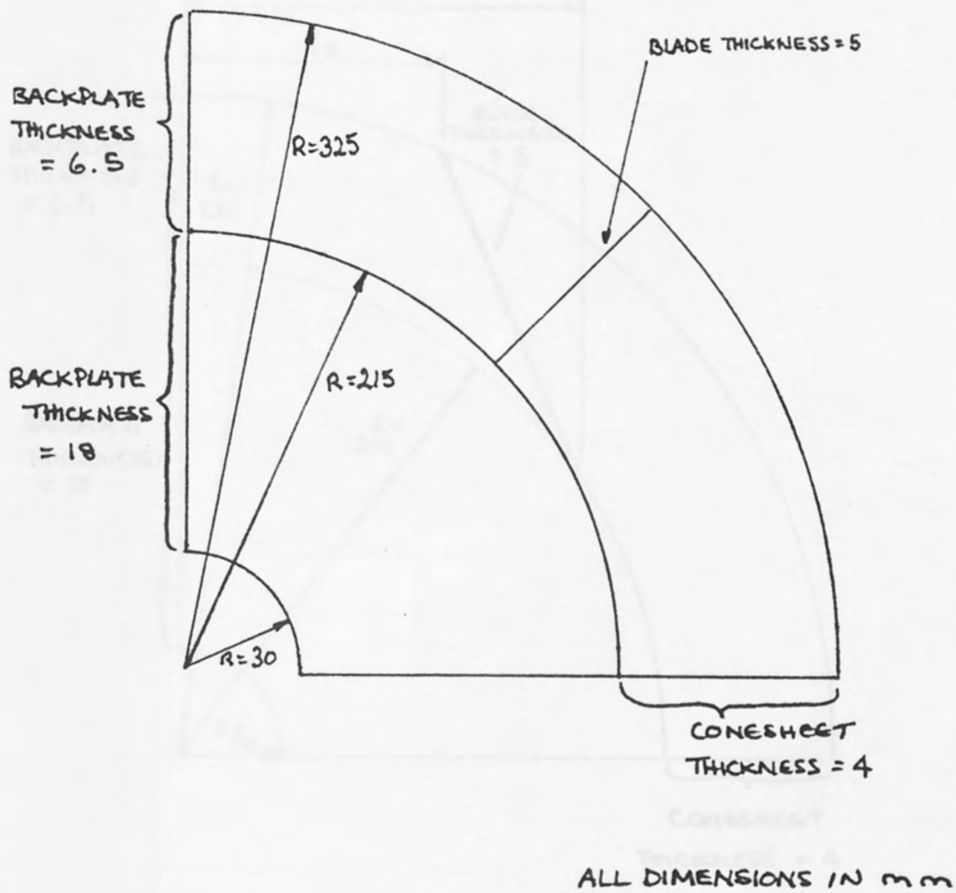
6.1. DESCRIPTIONS OF THE SIMPLE MODELS OF THE IMPELLER

Before attempting the numerical stress analysis of the actual impeller used in the experimental aspect of the research (discussed in Chapter 7), using the computer program developed (discussed in Chapter 5), it was decided to gain some insight into the complex stress distribution in the centrifugal fan impeller using simple models. To achieve this an impeller with a blade in the radial direction as shown in Fig. 6.1. and an impeller with the blade inclined at the same angle as the actual impeller, as shown in Fig. 6.2. were used. In these two models the following assumptions were made:-

1). There are only 4 blades in the impeller and that the impeller is symmetrical about the X and Y axis. This is not so in the case of the actual impeller as there is no symmetry about any axis in the type considered. This assumption was made so that symmetry could be exploited, to reduce the scale of the model that needed to be considered.

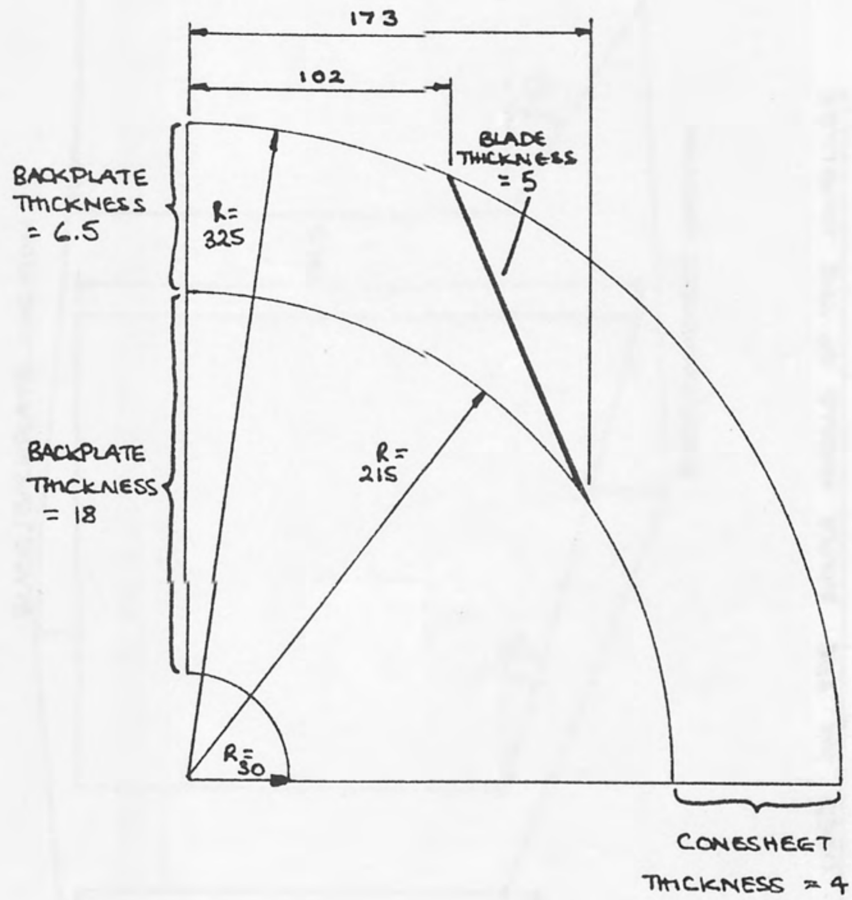
2). The dimensions of the three components, namely the blades, conesheet and backplate are the same as those of the actual impeller.

To study the effect of the conesheet on the other two components, impellers as described above were analysed with and without a conesheet. The angle of the conesheet was also varied by keeping the width of the blade at tip constant and increasing the width of the blade at the root. The blade widths considered are shown in Fig. 6.3. The meshes used for the radial bladed impeller are shown in Fig. 6.4. and Fig.6.5., while those for the inclined bladed impeller are shown in Fig. 6.6. and Fig.6.7. In all cases the blade mesh consisted of 8 elements as shown in Fig. 6.8. The results obtained for the above models for an impeller speed of 1550 r.p.m. are presented in the following sections. The strains referred to in the backplate and conesheet are the maximum principal strains at the Gauss points of the element, while those for the blades are the strains in the direction parallel to the blade tip and root. In the case of the backplate and conesheet due to the coarse meshes used, no attempt has been made to plot the stress distribution in these components, due to so few points.



DIMENSIONS AND SEGMENT USED FOR THE RADIAL BLADED MODEL OF THE CENTRIFUGAL FAN IMPELLER

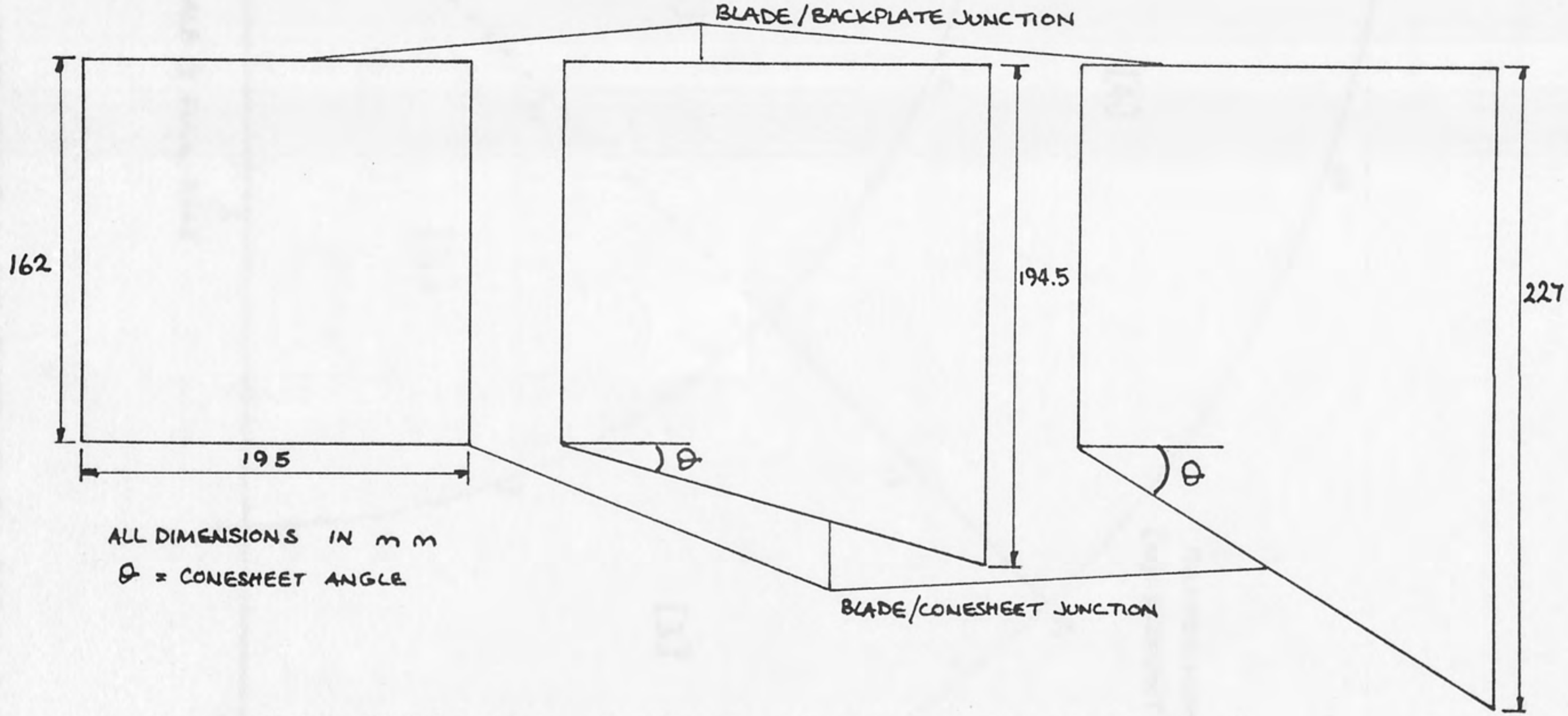
Fig 6.1



ALL DIMENSIONS IN mm

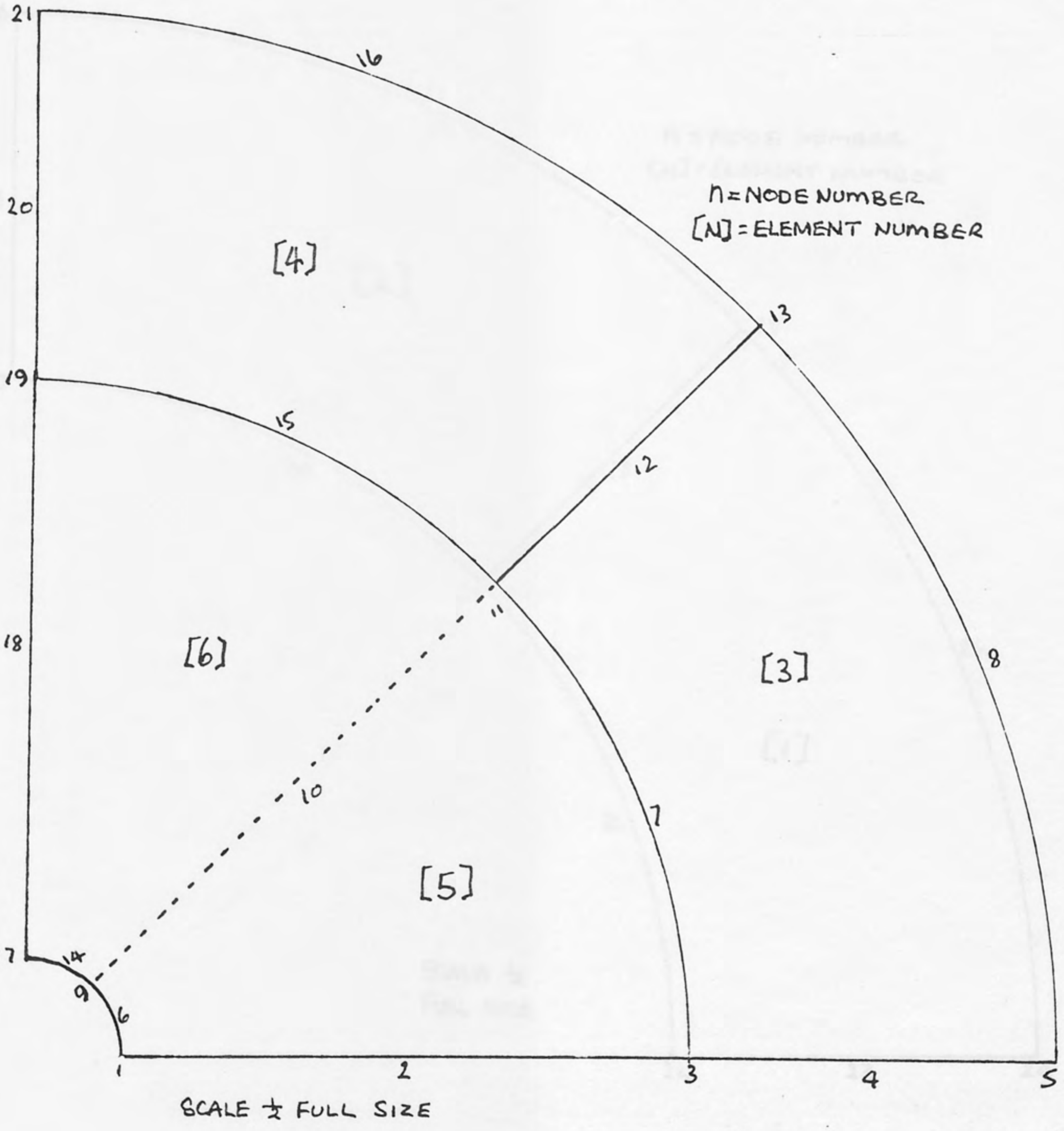
DIMENSIONS AND SEGMENT USED FOR THE INCLINED BLADED MODEL OF THE CENTRIFUGAL FAN IMPELLER

Fig 6.2



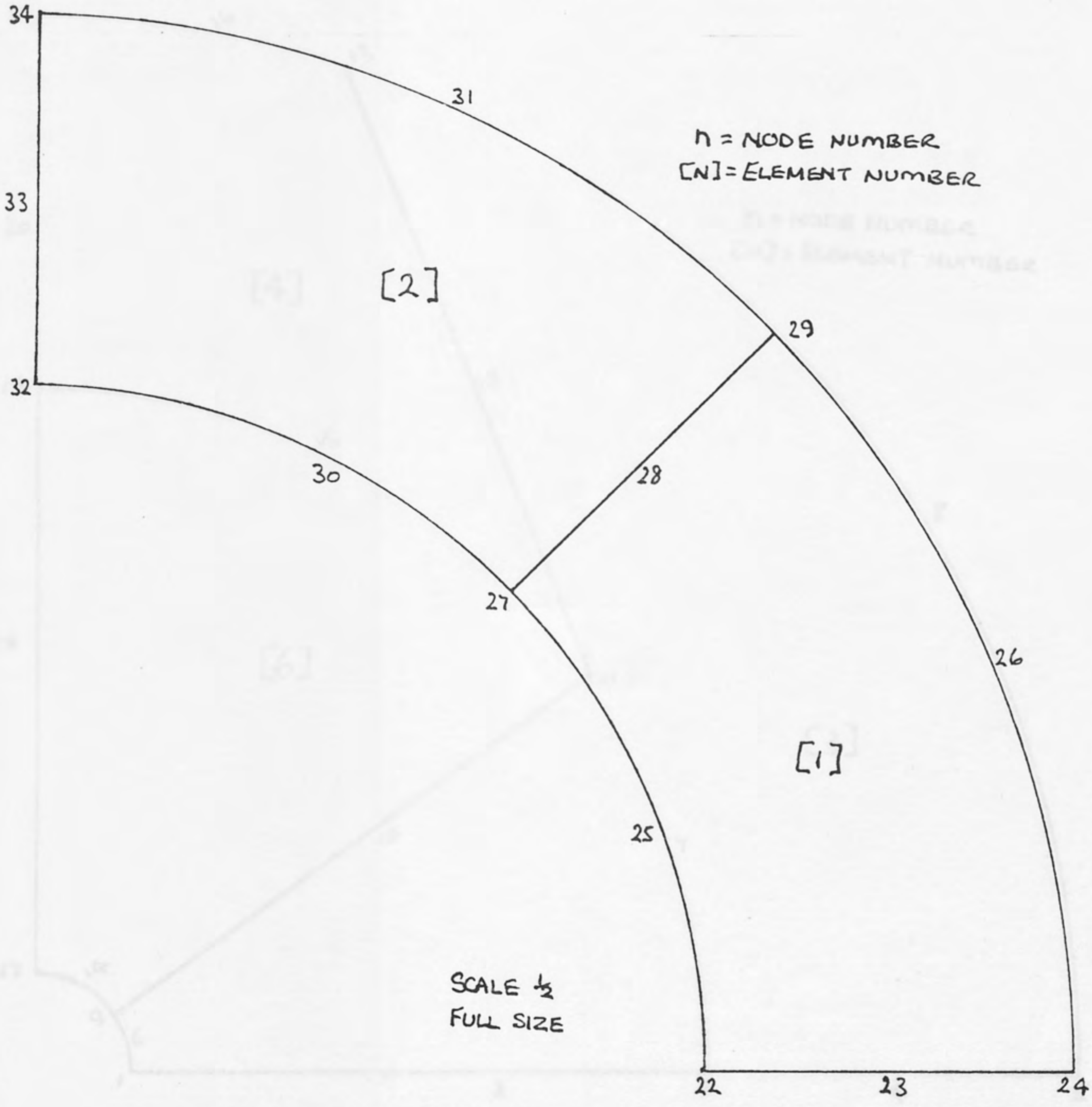
BLADE SHAPES USED IN THE SIMPLE MODELS OF THE IMPELLER.

Fig 6.3



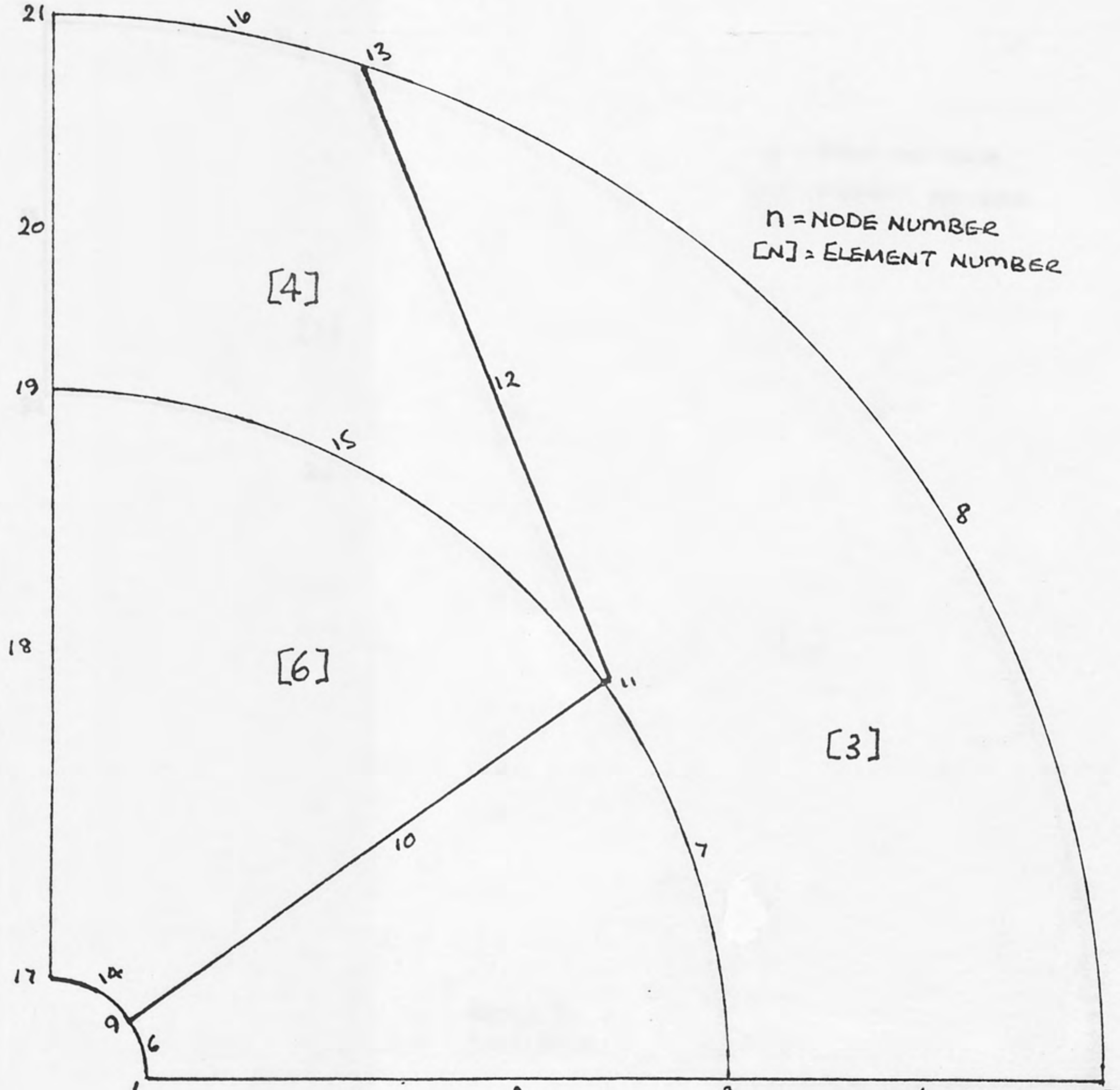
MESH USED FOR THE BACKPLATE IN THE RADIAL BLADED MODELS OF THE IMPELLER.

Fig 6.4



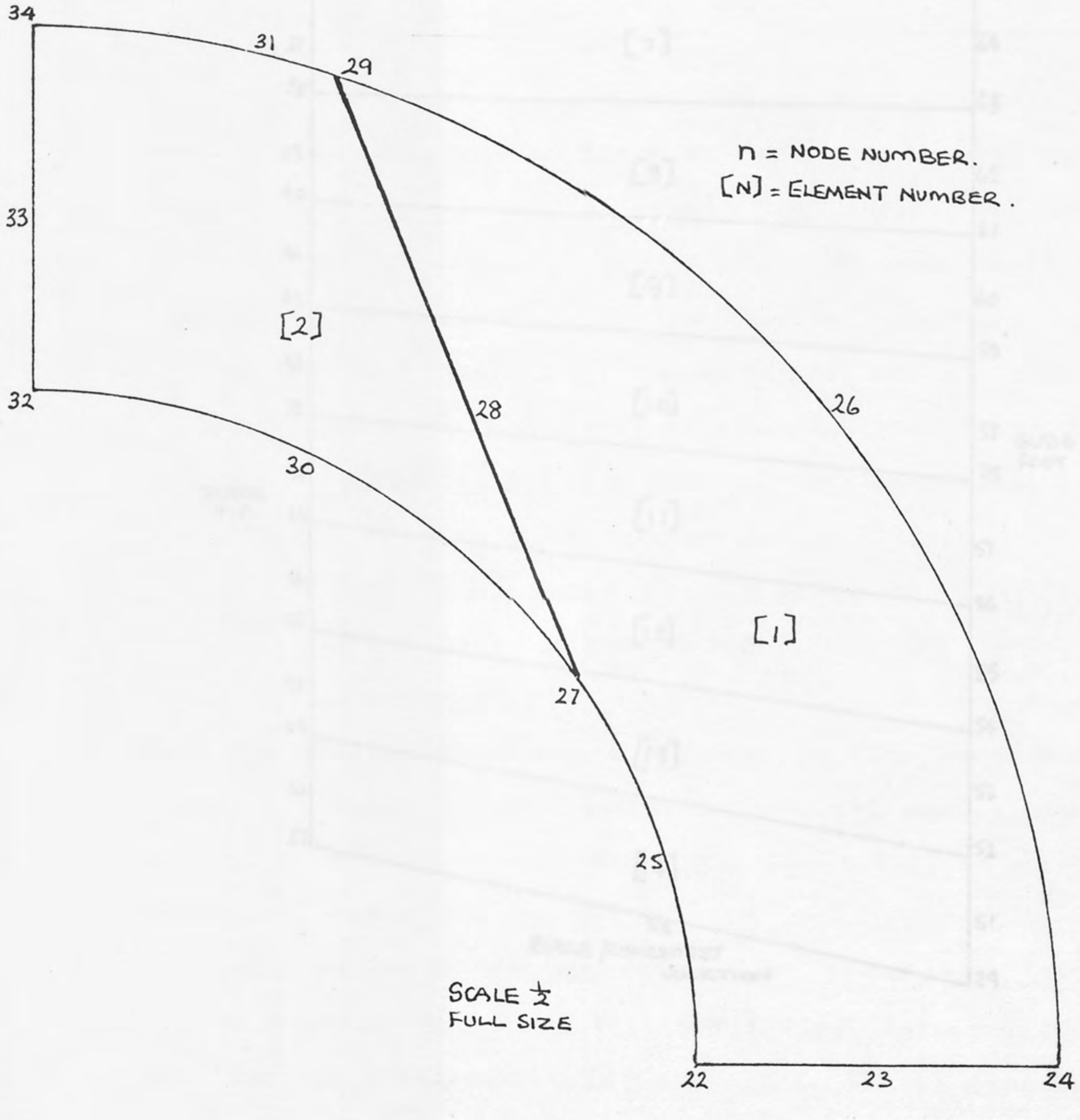
MESH USED FOR THE CONESHEET IN THE RADIAL BLADED MODELS OF THE IMPELLER.

Fig 6.5



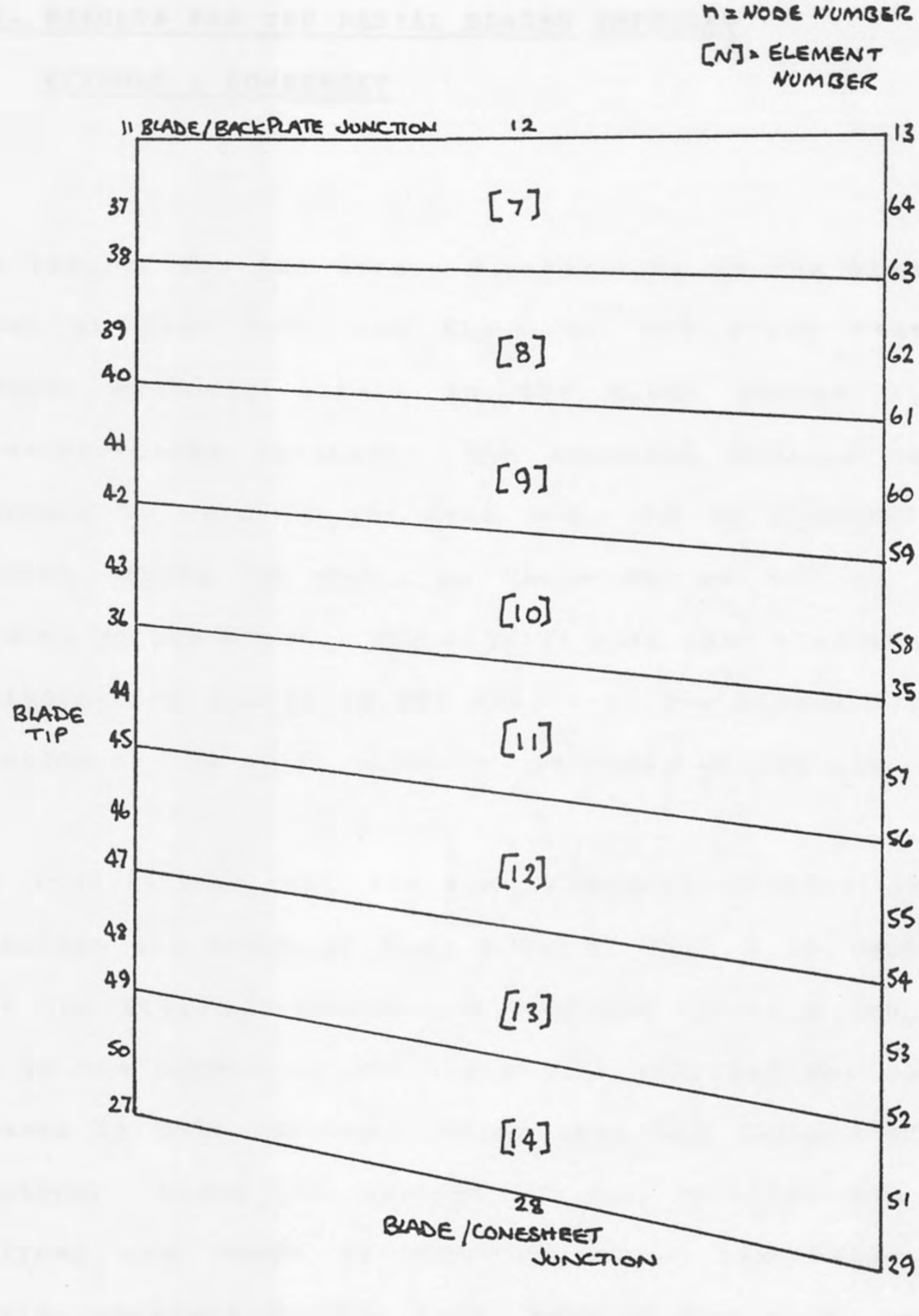
MESH USED FOR THE BACKPLATE IN THE INCLINED BLADED MODELS OF THE IMPELLER.

Fig 6.6



MESH USED FOR THE CONESHEET IN THE INCLINED BLADED MODELS OF THE IMPELLER.

Fig 6.7



MESH USED FOR THE BLADE IN THE SIMPLE MODELS OF THE IMPELLER

Fig 6.8

6.2. RESULTS FOR THE RADIAL BLADED IMPELLER WITHOUT A CONESHEET

The results for the strain distribution in the blade are shown in Fig. 6.9. and Fig.6.10. and shows that the maximum membrane strain in the blade occurs at the blade/backplate junction. The membrane strains rapidly decrease to zero at the free end, and as expected, the bending strain is zero, as there is no out of plane loading on the blade. The results also show that there is insignificant change in the strain at the blade/backplate junction as the blade width is increased at its tip.

The results for the maximum principal strains in the backplate are shown in Fig. 6.11. to Fig. 6.13. and show that the bending strains are dominant in this component due to the affect of the blade. As expected the maximum strains in this component occur near the backplate/blade junction. Since one quarter of the impeller has been analysed and there is symmetry about the blade, the results obtained confirm this, thus giving confidence in the results obtained. As with the blade itself, increasing the blade width at its tip has insignificant effect on the membrane strains in the backplate, but the bending strains increase.

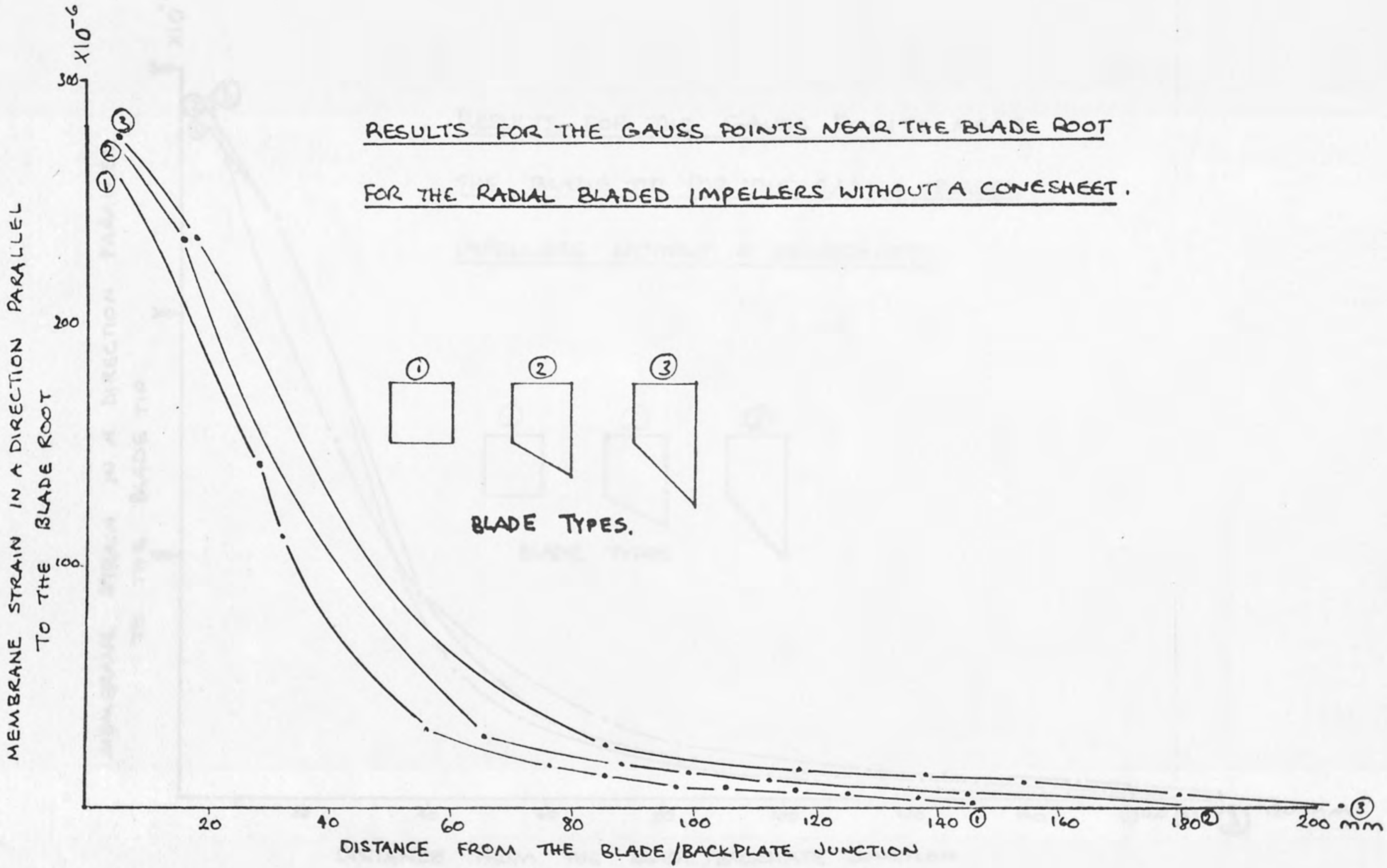

RESULTS FOR THE GAUSS POINTS NEAR THE BLADE ROOT
FOR THE RADIAL BLADED IMPELLERS WITHOUT A CONESHEET.
BLADE TYPES.
1
2
3
MEMBRANE STRAIN IN A DIRECTION PARALLEL TO THE BLADE ROOT
$\times 10^{-6}$
300
200
100
20
40
60
80
100
120
140
160
180
200 mm
DISTANCE FROM THE BLADE/BACKPLATE JUNCTION

Fig 6.9

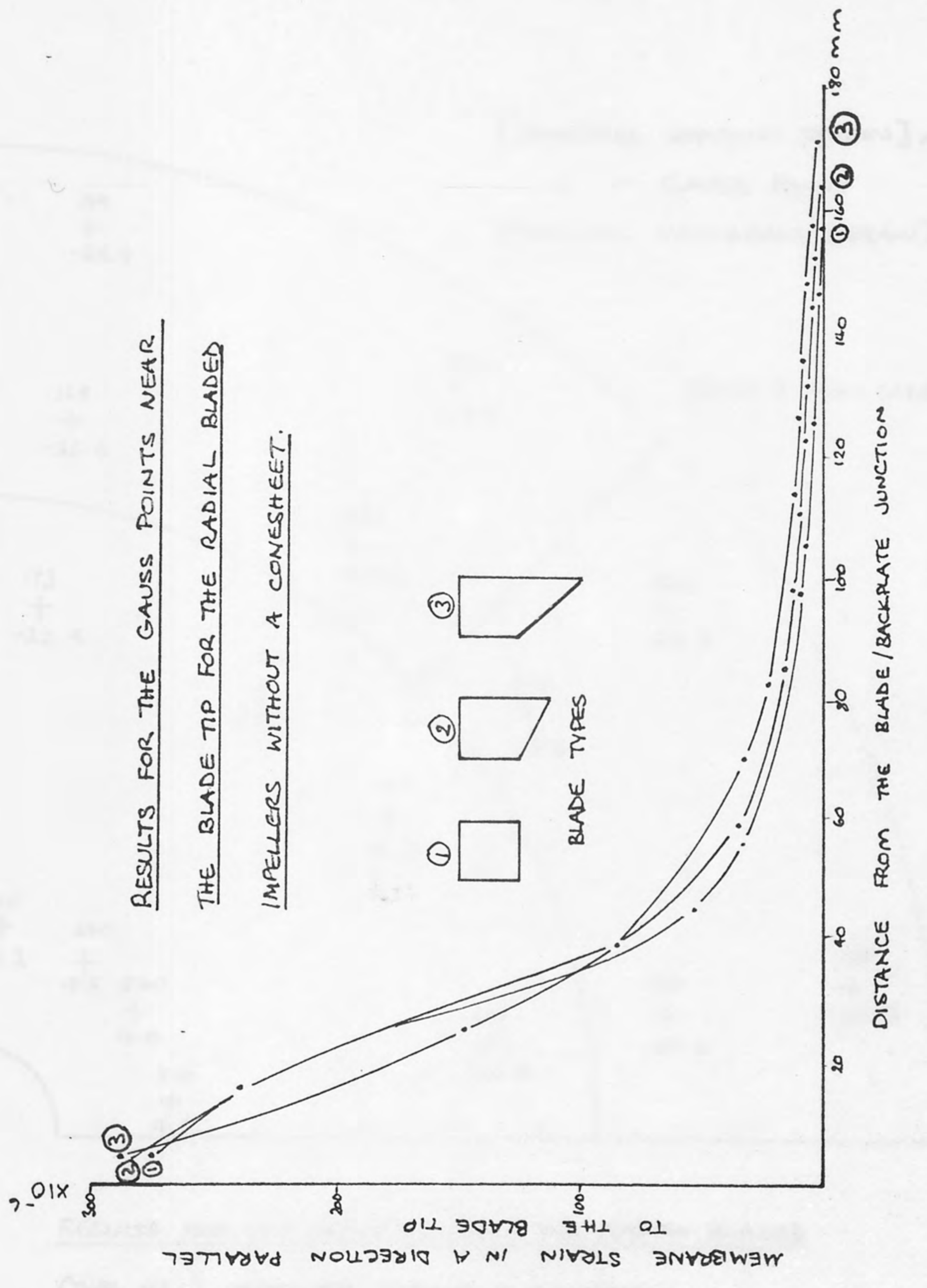
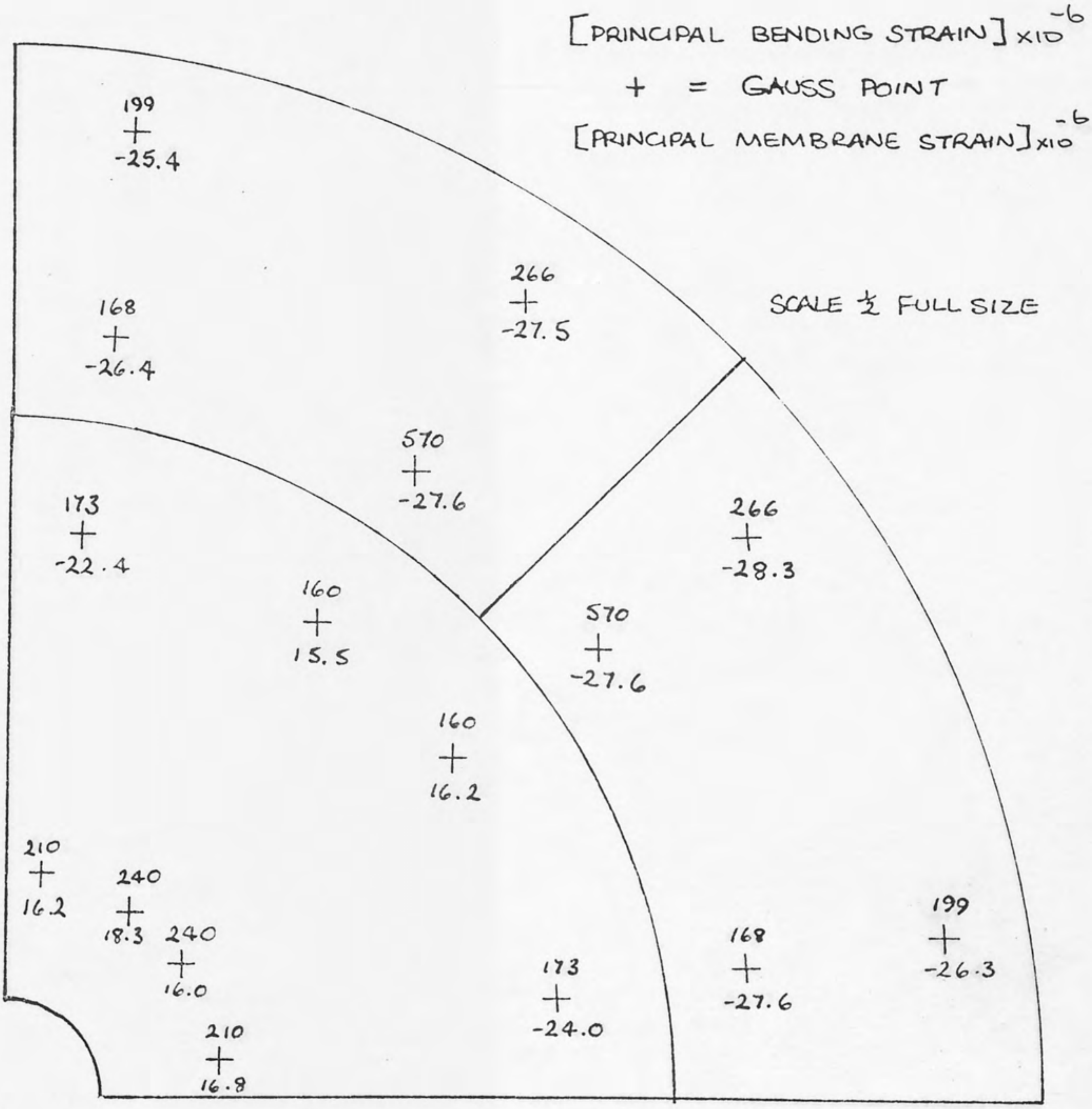

RESULTS FOR THE GAUSS POINTS NEAR
THE BLADE TIP FOR THE RADIAL BLADED
IMPELLERS WITHOUT A CONESHEET.
BLADE TYPES
1
2
3
MEMBRANE STRAIN IN A DIRECTION PARALLEL
TO THE BLADE TIP
x10⁻⁶
300
200
100
DISTANCE FROM THE BLADE/BACKPLATE JUNCTION
20
40
60
80
100
120
140
160
180 mm

Fig 6.10



RESULTS FOR THE BACKPLATE WITH THE RADIAL BLADED (TYPE No 1) IMPELLER WITHOUT A CONESHEET

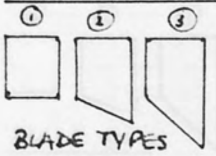
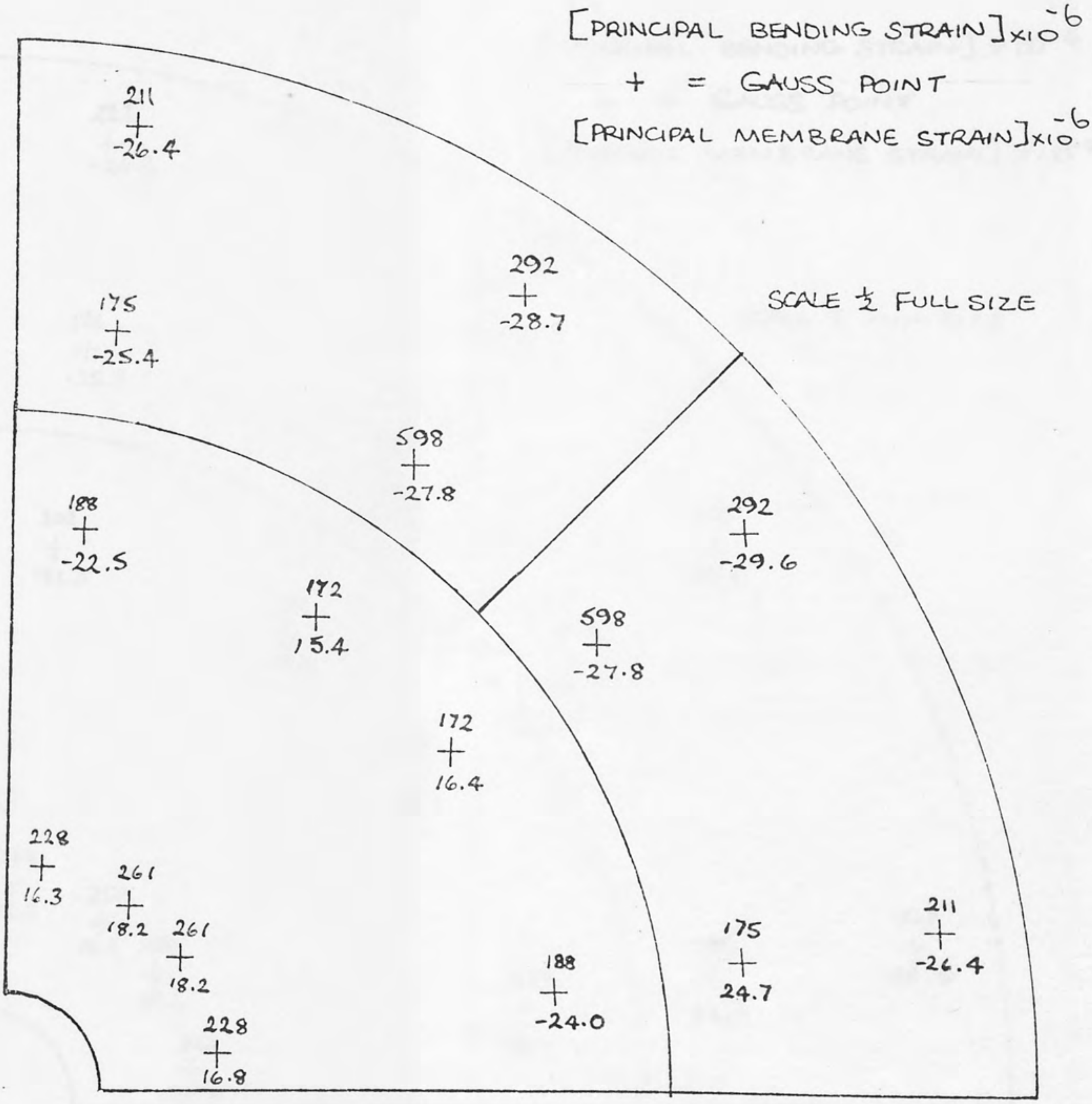


Fig 6.11



RESULTS FOR THE BACKPLATE WITH THE RADIAL BLADED (TYPE Nº2) IMPELLER WITHOUT A CONESHEET

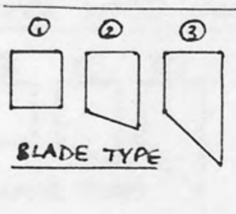
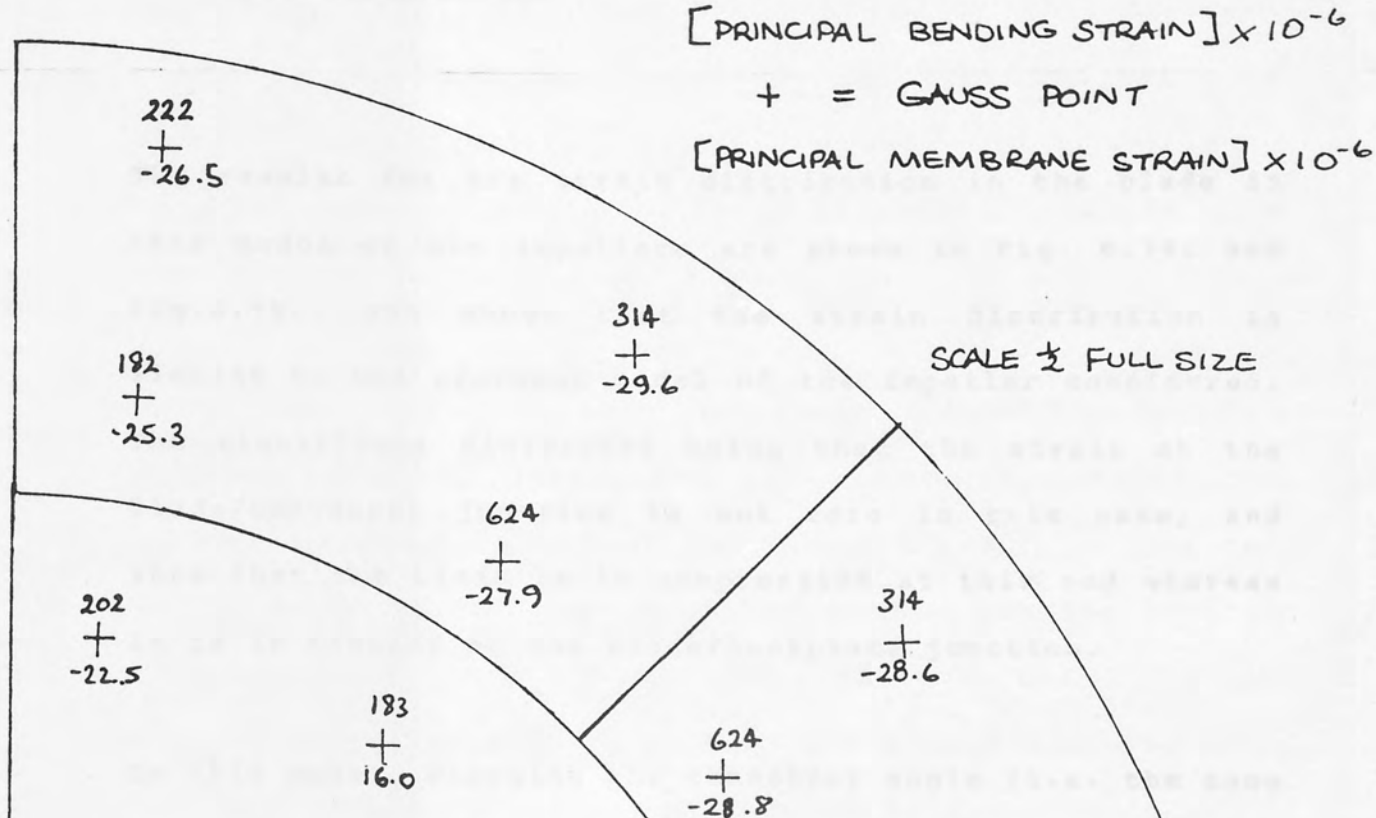


Fig 6.12



RESULTS FOR THE BACKPLATE WITH THE RADIAL BLADED (TYPE No 3) IMPELLER WITHOUT A CONESHEET.

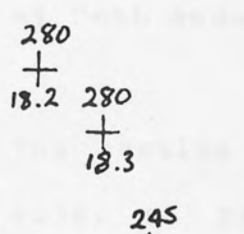


Fig 6.13

6.3. RESULTS FOR THE RADIAL BLADED IMPELLER WITH A CONESHEET

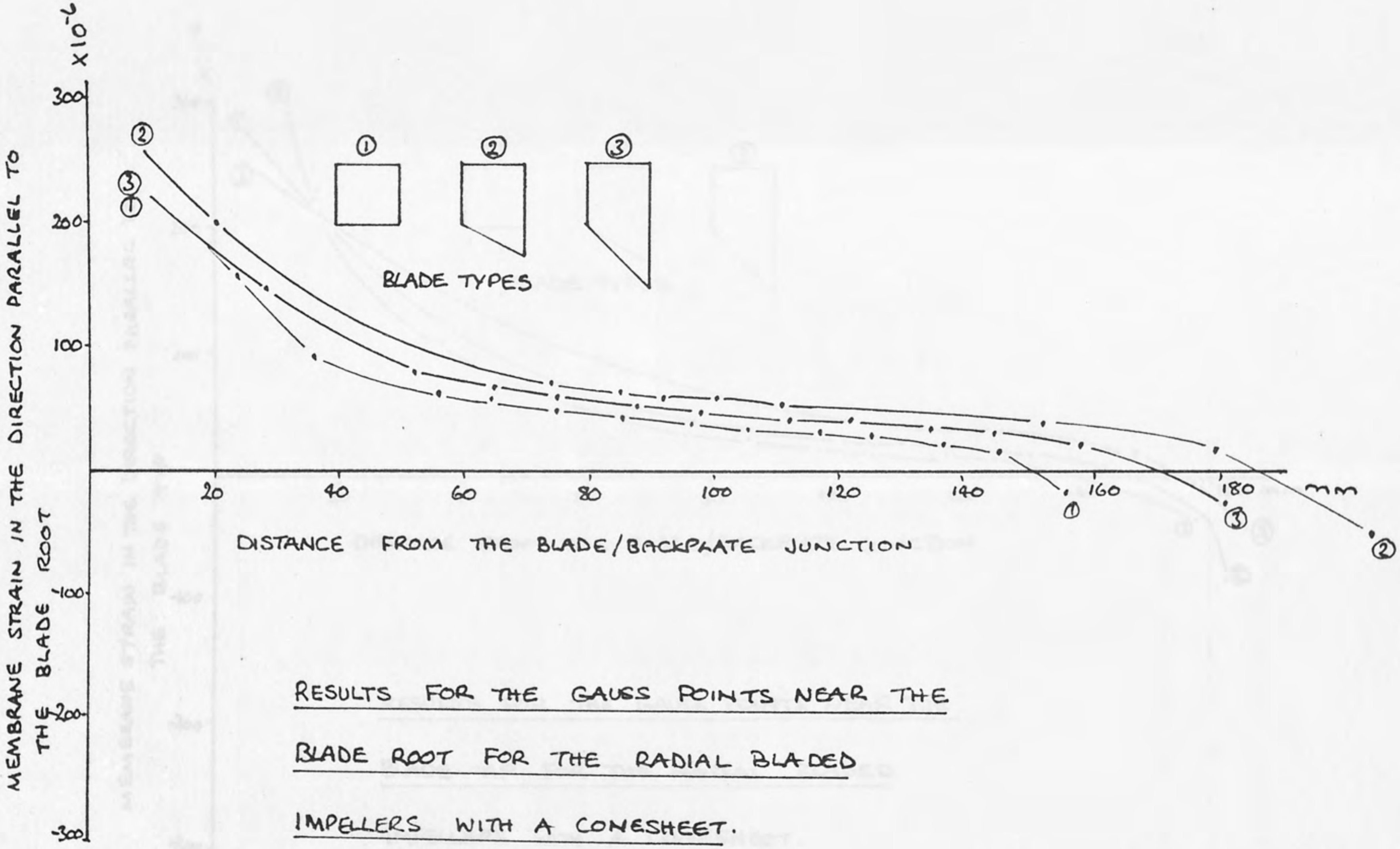
The results for the strain distribution in the blade in this model of the impellers are shown in Fig. 6.14. and Fig.6.15., and shows that the strain distribution is similar to the previous model of the impeller considered. The significant difference being that the strain at the blade/conesheet junction is not zero in this case, and show that the blade is in compression at this end whereas it is in tension at the blade/backplate junction.

In this model, changing the conesheet angle (i.e. the same effect as varying the blade width at the blade tip) first increases the strain at both ends of the blade, but as the angle is further increased, the strain is reduced, again at both ends of the blade.

The results obtained for the backplate are shown in Fig. 6.16. to Fig. 6.18. In this case the strain in the backplate is much less than in the previous example, as the load is now shared by the backplate and conesheet: it shows that even though the bending strain is greater than the membrane strain, the latter is no longer negligible compared with the former. Again, as expected the maximum strain in the backplate occurs near the backplate/blade junction at the blade root. The strain distribution is symmetrical about the blade, and there is insignificant change in the membrane strains as the conesheet angle is changed. The bending strains, however, first increase, but then decrease as the angle is increased, just as with the membrane strains in the blade.

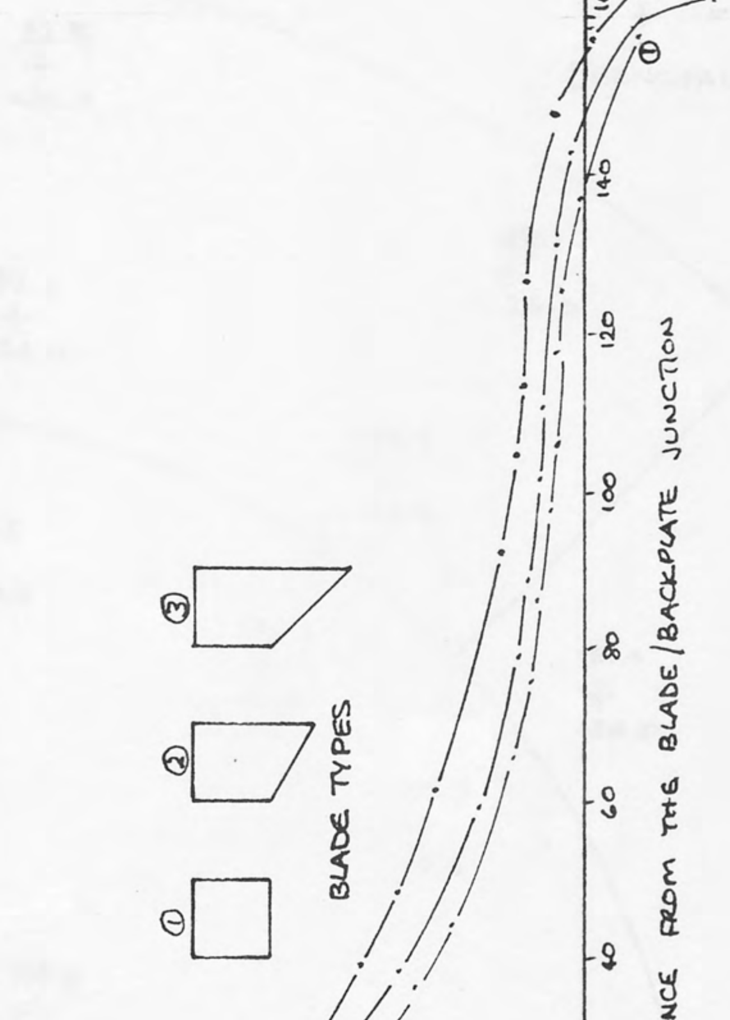
It is interesting to note that even though the bending strains in the backplate are reduced, there is insignificant change in the membrane strains in the two models considered. The reduction in the bending strains is more significant in the outer part of the backplate (i.e. the area most influenced by the blade) compared with the inner part which is some distance from the blade.

The results for the conesheet are shown in Fig. 6.19. to Fig. 6.21. Figure 6.19. shows that when the conesheet angle is zero, the strain in the conesheet is mainly membrane, but as the angle is increased the bending strains become dominant, the magnitude of the bending strain, however, decreases as the angle is further increased. As with the backplate, the results are symmetrical about the blade. The maximum strains again occur near the conesheet/blade junction. The results show that in the case of the impeller with a conesheet, the conesheet is the most highly strained component, whereas it is the backplate in the case of the impeller without a conesheet.



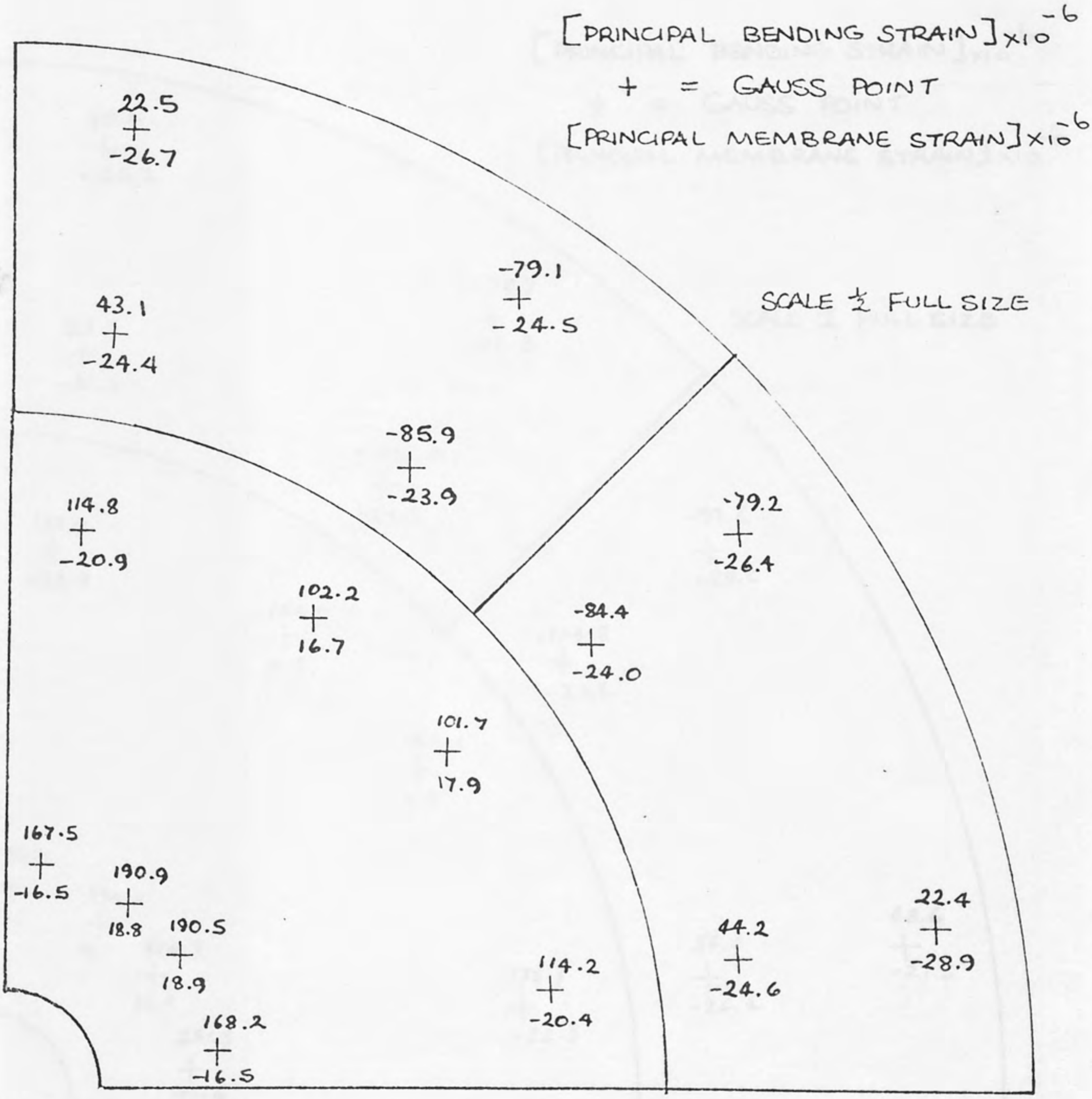
RESULTS FOR THE GAUSS POINTS NEAR THE BLADE ROOT FOR THE RADIAL BLADED IMPELLERS WITH A CONESHEET.

Fig 6.14



RESULTS FOR THE GAUSS POINTS NEAR THE BLADE TIP FOR THE RADIAL BLADED IMPELLERS WITH A CONESHEET.

fig 6.15



RESULTS FOR THE BACKPLATE WITH THE RADIAL BLADED (TYPE No1) IMPELLER WITH A CONESHEET

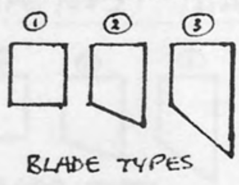
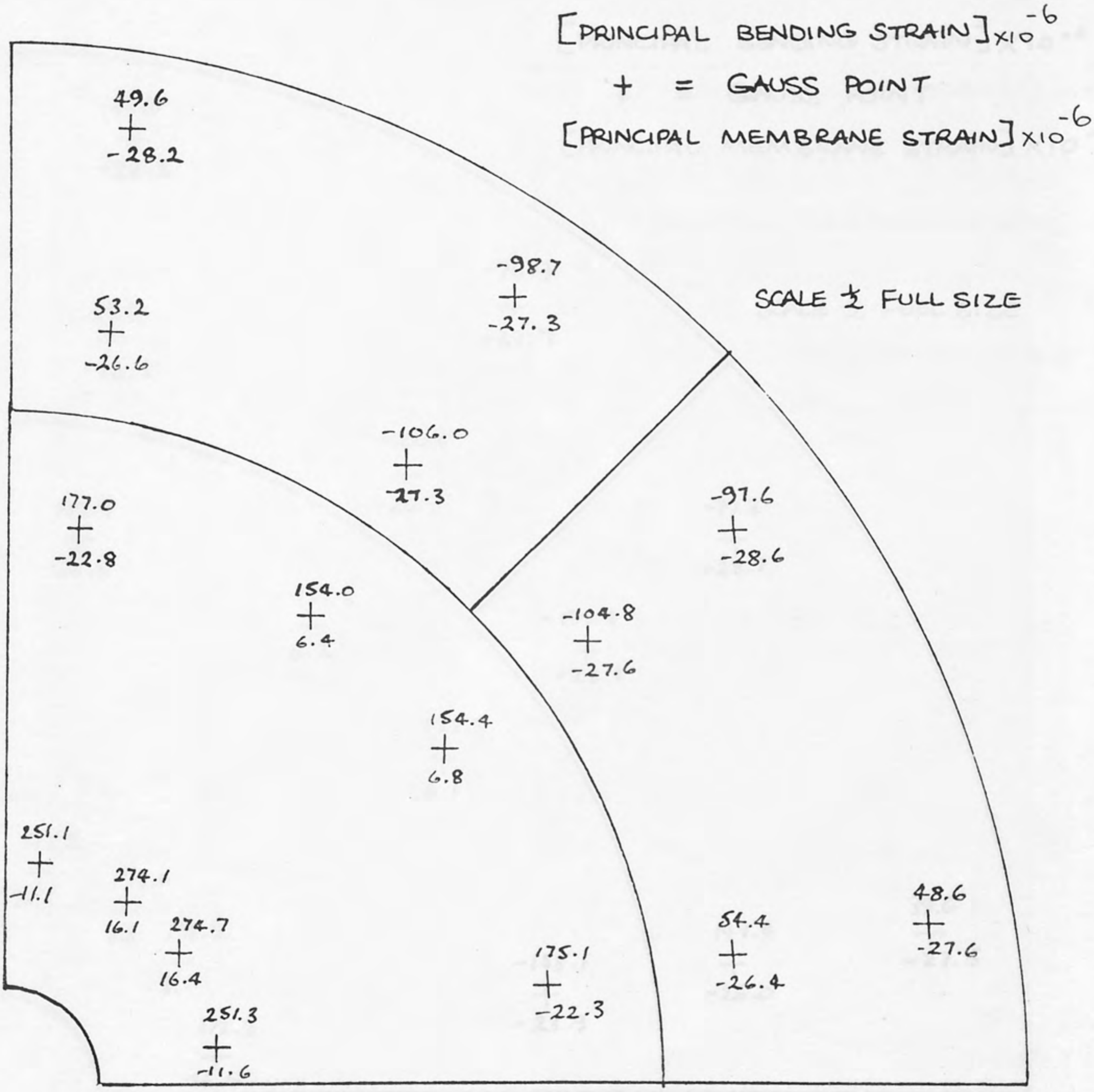


Fig 6.16



RESULTS FOR THE BACKPLATE WITH THE RADIAL BLADED (TYPE Nº 2) IMPELLER WITH A CONESHEET

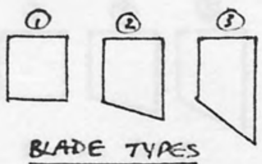
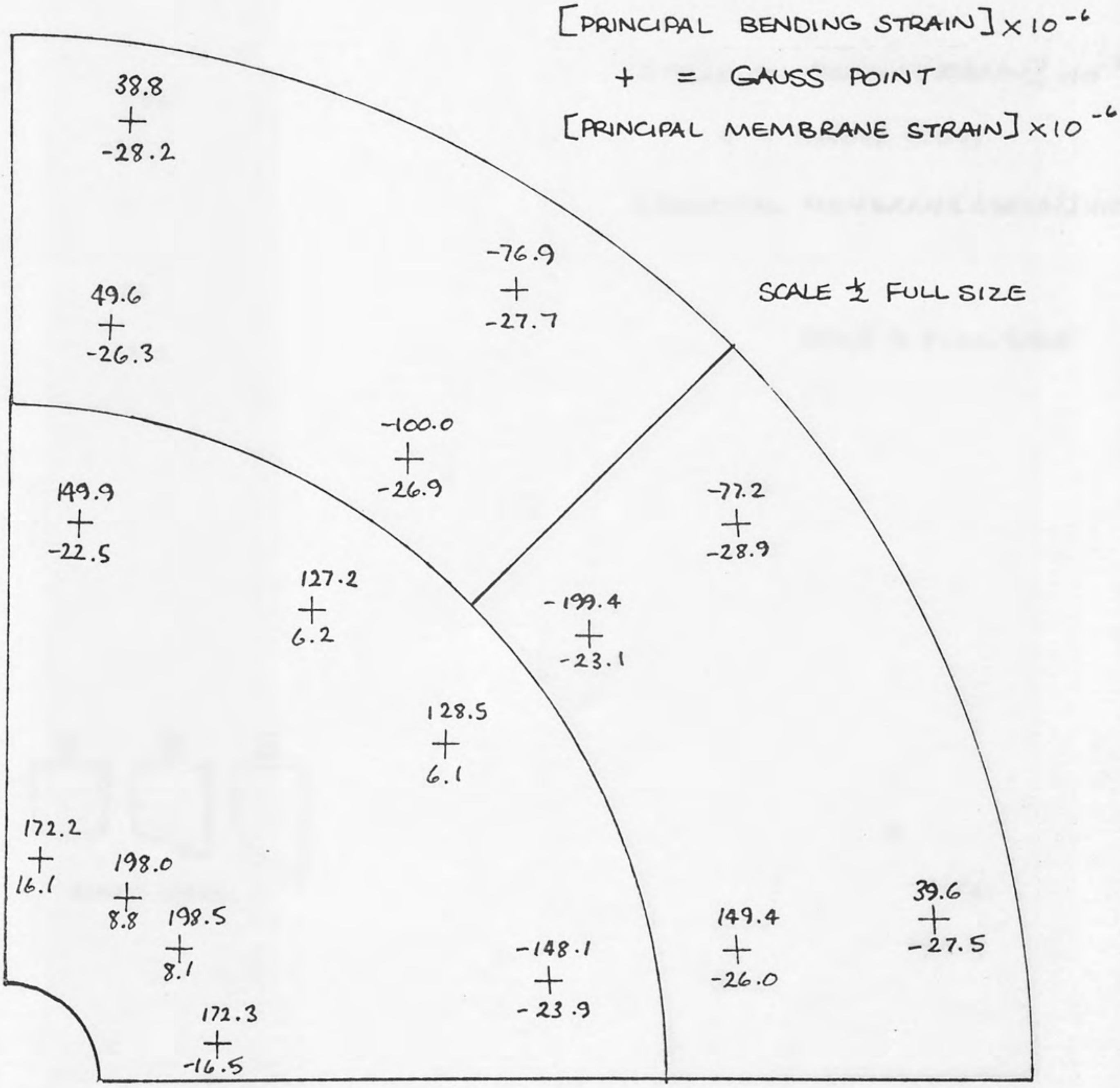


Fig 6.17



RESULTS FOR THE BACKPLATE WITH THE RADIAL BLADED (TYPE Nº 3) IMPELLER WITH A CONESHEET.

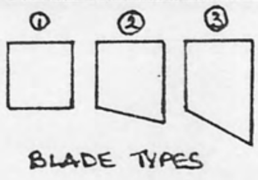
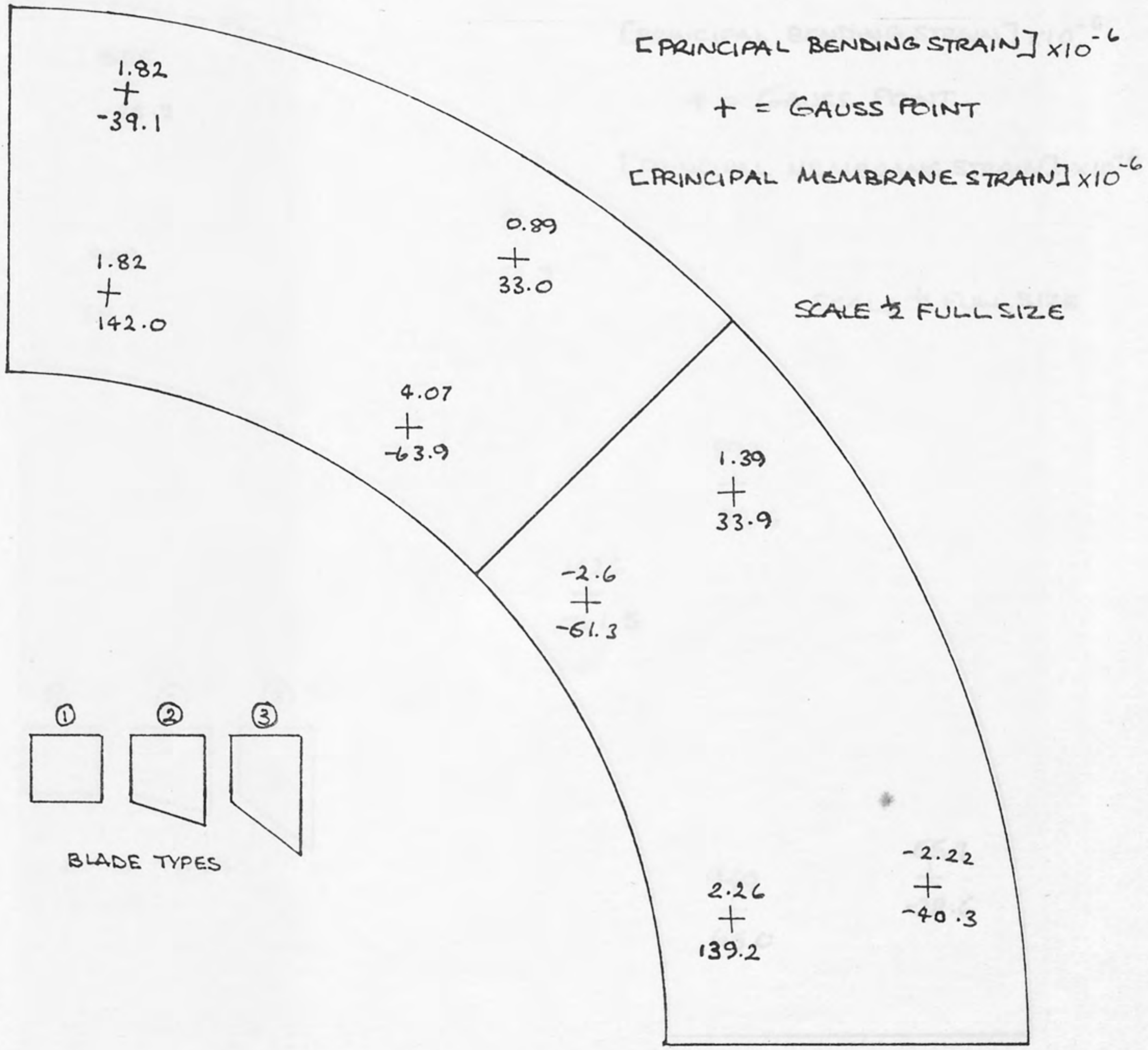
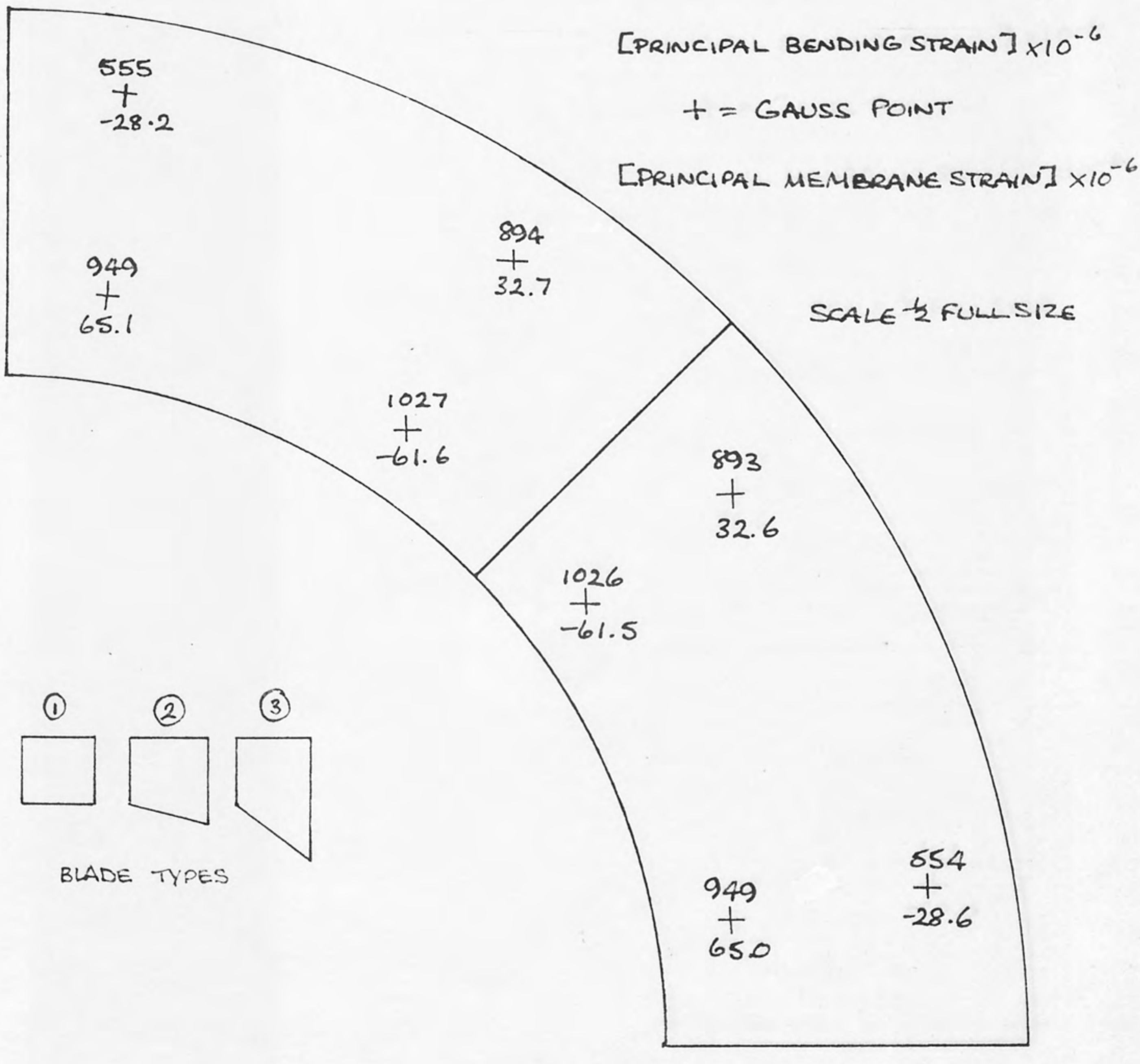


Fig 6.18



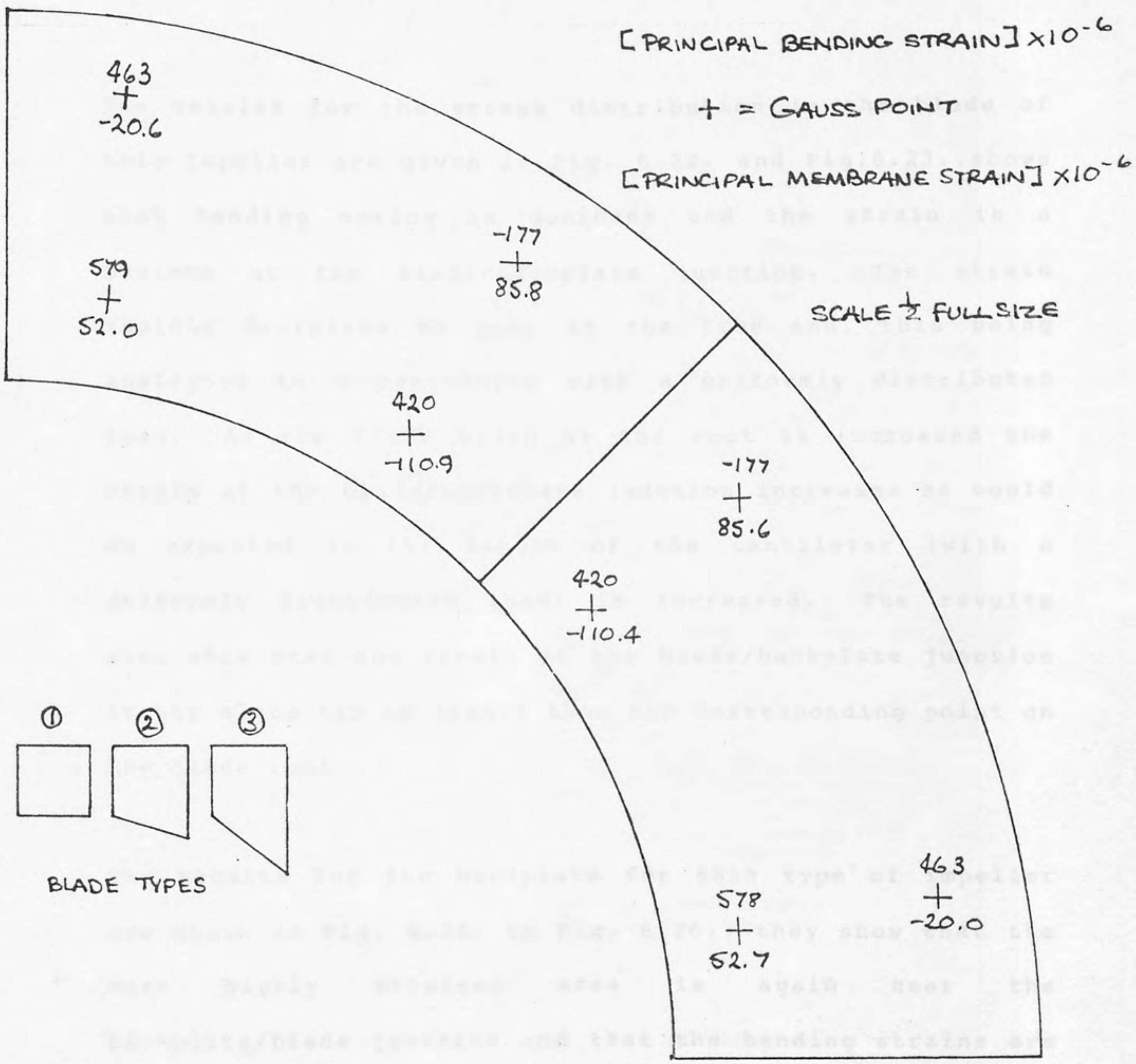
RESULTS FOR THE CONESHEET WITH THE RADIAL BLADED (TYPE Nº1) IMPELLER.

Fig 6.19



RESULTS FOR THE CONESHEET WITH THE RADIAL BLADED (TYPE Nº 2) IMPELLER.

Fig 6.20



RESULTS FOR THE CONESHEET WITH THE RADIAL BLADED (TYPE No 3) IMPELLER.

Fig 6.21

6.4. RESULTS FOR THE INCLINED BLADED IMPELLER WITHOUT A CONESHEET

The results for the stress distribution in the blade of this impeller are given in Fig. 6.22. and Fig.6.23.:shows that bending action is dominant and the strain is a maximum at the blade/backplate junction. The strain rapidly decreases to zero at the free end, this being analogous to a cantilever with a uniformly distributed load. As the blade width at the root is increased the strain at the blade/backplate junction increases as would be expected if the length of the cantilever (with a uniformly distributed load) is increased. The results also show that the strain at the blade/backplate junction at the blade tip is higher than the corresponding point on the blade root.

The results for the backplate for this type of impeller are shown in Fig. 6.24. to Fig. 6.26.; they show that the most highly strained area is again near the backplate/blade junction and that the bending strains are dominant, with the membrane strains negligible in comparison. The bending strain in the backplate is higher in the area just below the backplate/blade junction compared with the area above the junction, the maximum being near the blade tip. As with the two previous models, varying the blade width at the root has insignificant effect on the membrane strains.

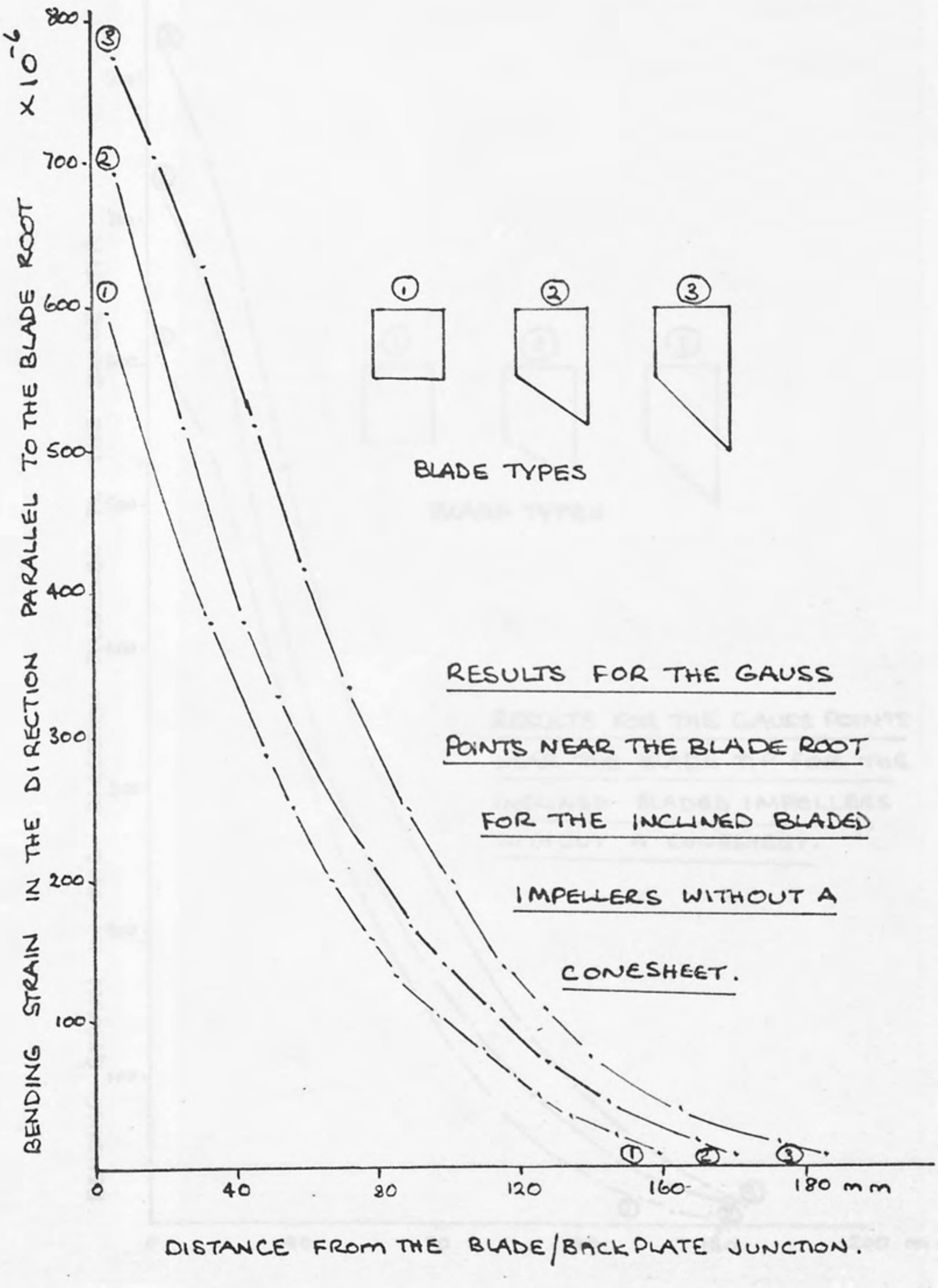

BENDING STRAIN IN THE DIRECTION PARALLEL TO THE BLADE ROOT $\times 10^{-6}$
800
700
600
500
400
300
200
100
0
40
80
120
160
180 mm
DISTANCE FROM THE BLADE/BACKPLATE JUNCTION.
1
2
3
BLADE TYPES
RESULTS FOR THE GAUSS POINTS NEAR THE BLADE ROOT FOR THE INCLINED BLADED IMPELLERS WITHOUT A CONESHEET.

Fig 6.22

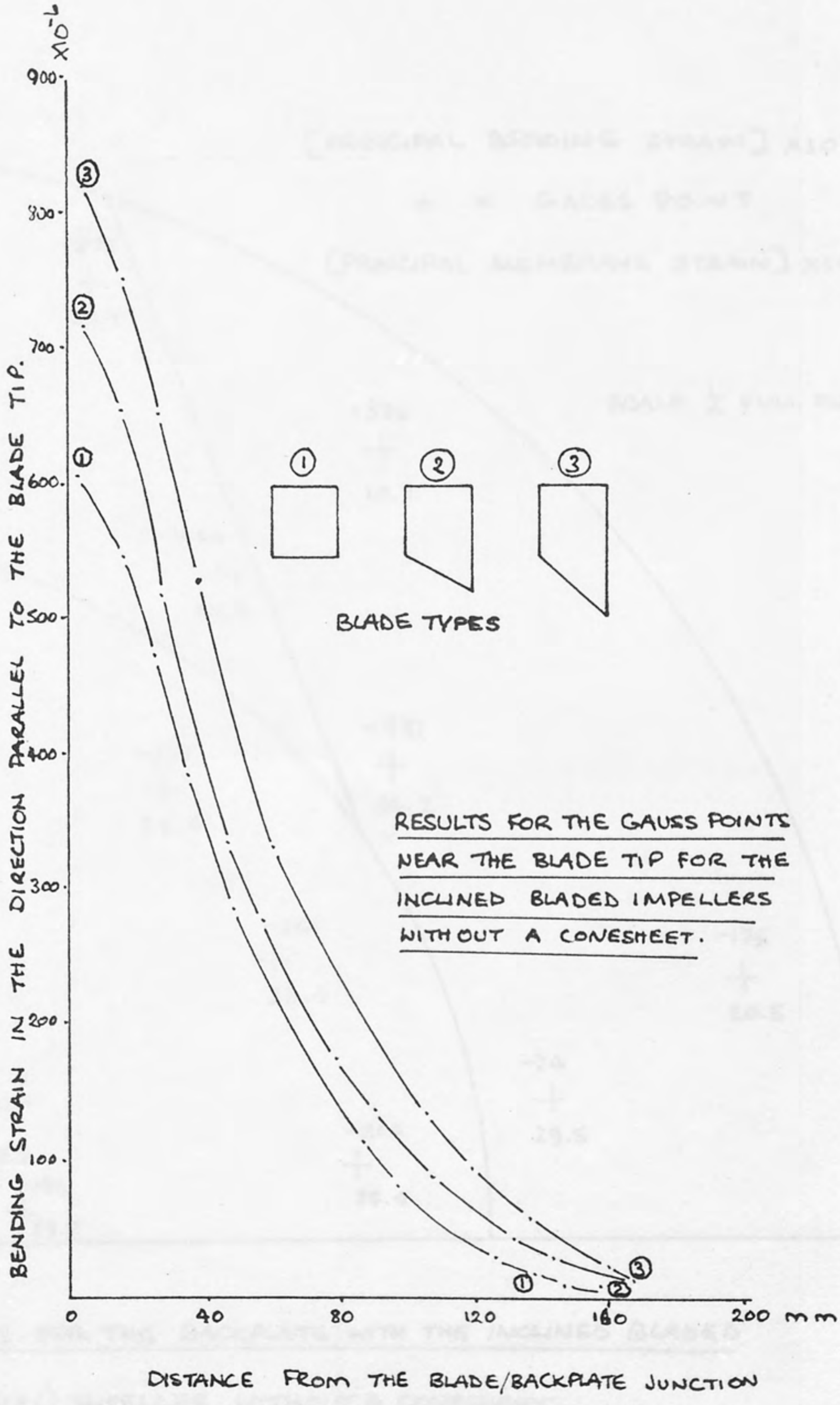
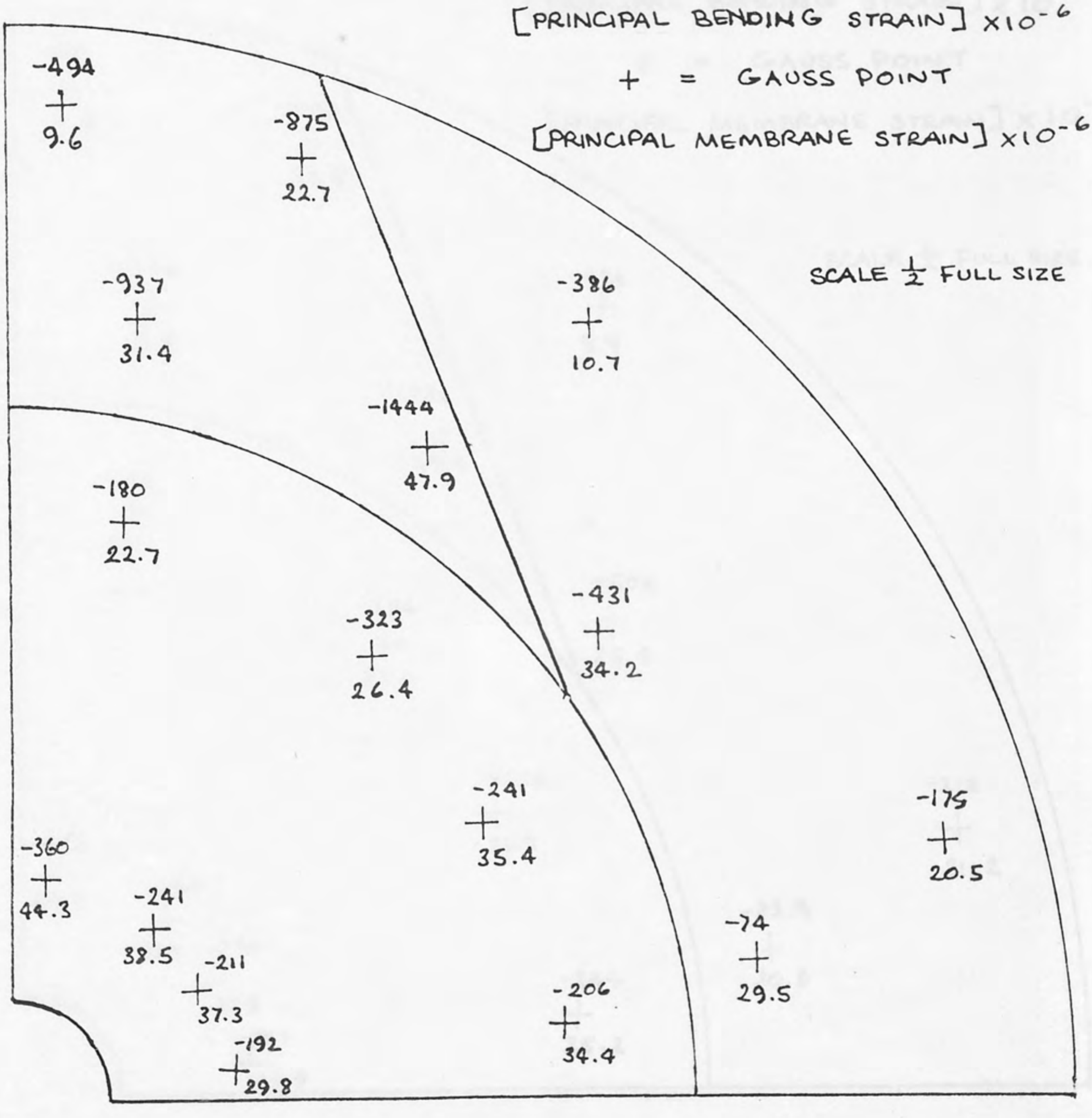

$\times 10^{-6}$
900
800
700
600
500
400
300
200
100
BENDING STRAIN IN THE DIRECTION PARALLEL TO THE BLADE TIP.
0
40
80
120
160
200 mm
DISTANCE FROM THE BLADE/BACKPLATE JUNCTION
1
2
3
BLADE TYPES
RESULTS FOR THE GAUSS POINTS NEAR THE BLADE TIP FOR THE INCLINED BLADED IMPELLERS WITHOUT A CONESHEET.

fig 6.23



RESULTS FOR THE BACKPLATE WITH THE INCLINED BLADED (TYPE No 1) IMPELLER WITHOUT A CONESHEET.

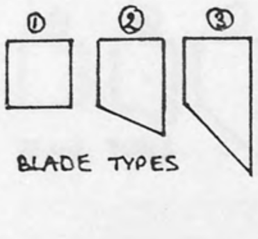
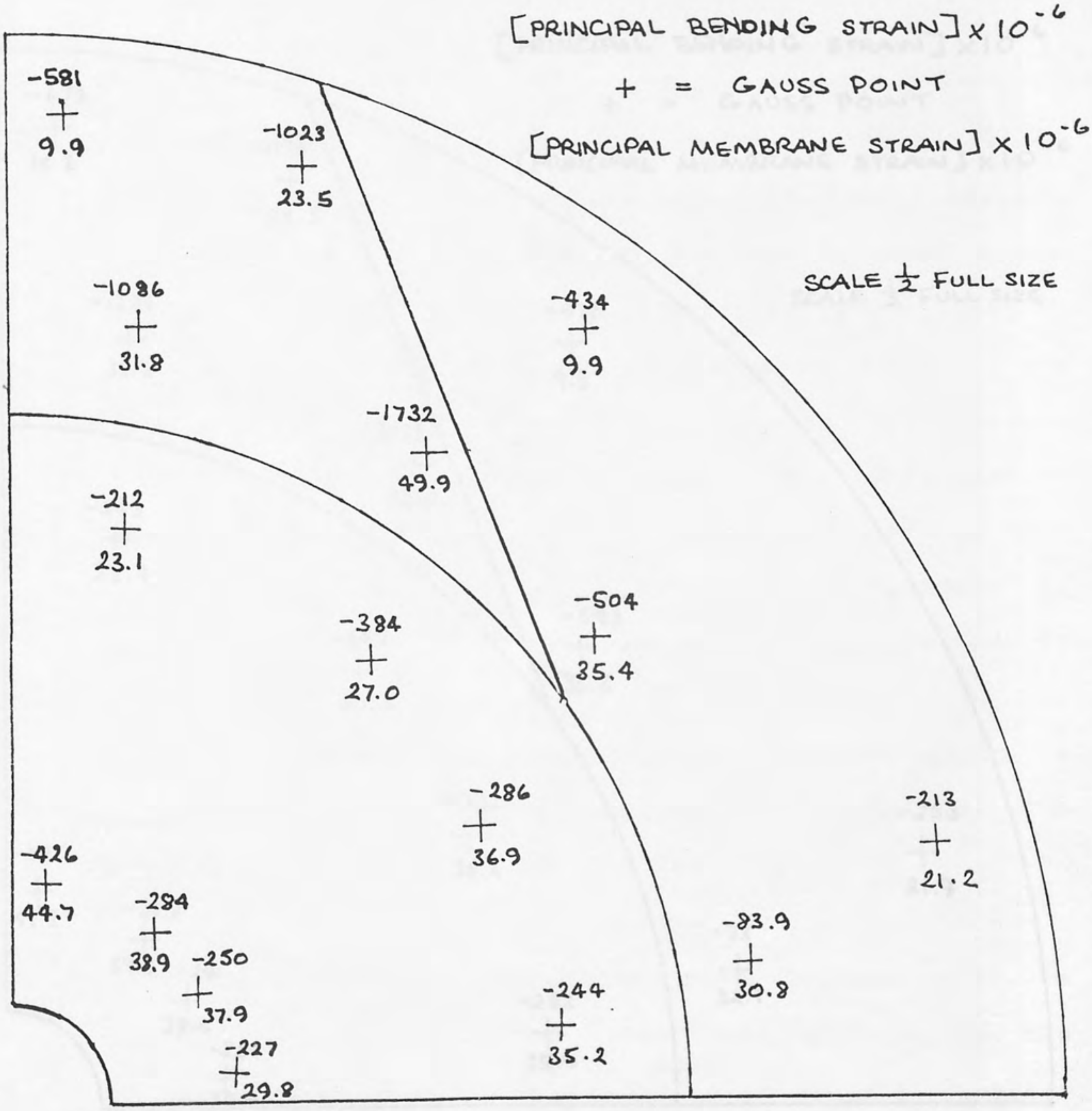


Fig 6.24



RESULTS FOR THE BACKPLATE WITH THE INCLINED BLADED (TYPE Nº 2) IMPELLER WITHOUT A CONESHEET.

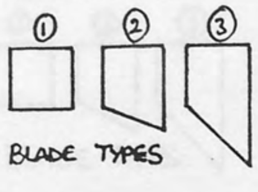
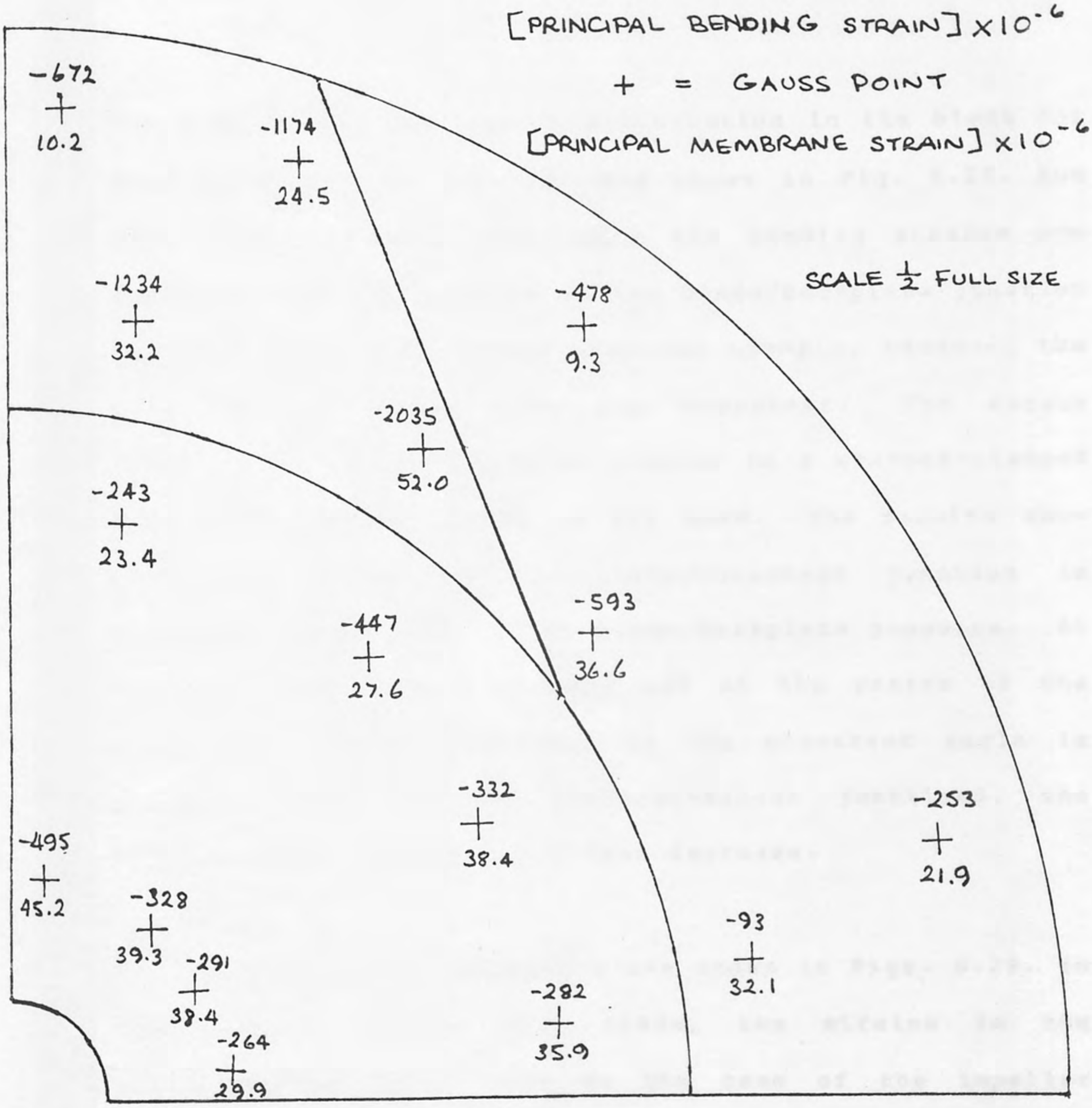


Fig 6.25



RESULTS FOR THE BACKPLATE WITH THE INCLINED BLADED (TYPE N° 3) IMPELLER WITHOUT A CONESHEET.

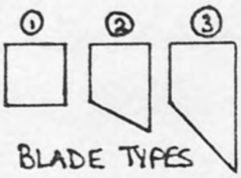


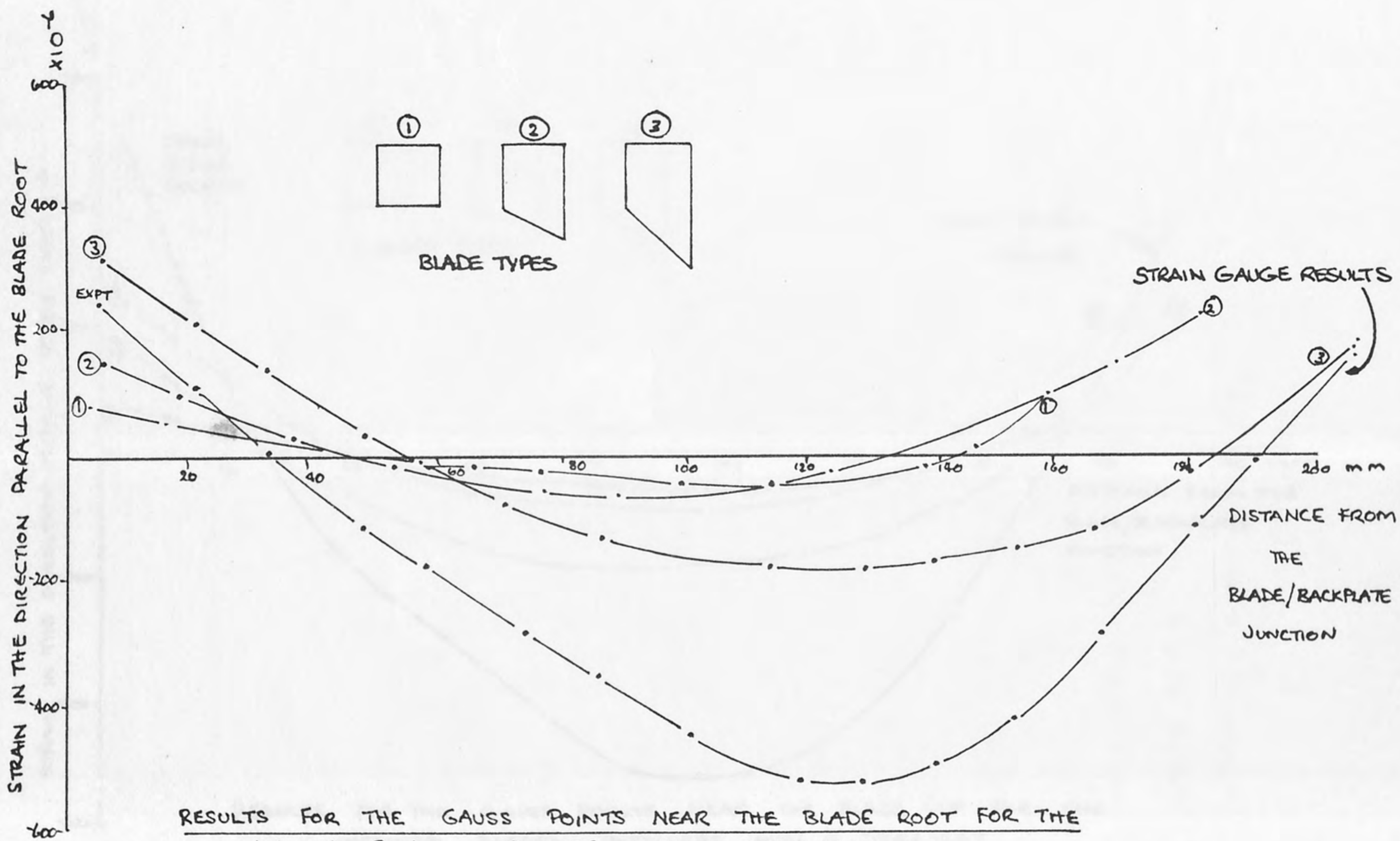
Fig 6.26

6.5. RESULTS FOR THE INCLINED BLADED IMPELLER WITH A CONESHEET

The results for the stress distribution in the blade for this model of the impeller are shown in Fig. 6.27. and Fig. 6.28. In this case again the bending strains are dominant, and the strains at the blade/backplate junction are much lower than in the previous example, because, the load is now shared with the conesheet. The strain distribution in the blade is similar to a clamped-clamped beam with loading normal to the beam. The results show that the strain at the blade/conesheet junction is slightly higher than at the blade/backplate junction. At the blade/backplate junction, and at the centre of the blade the strains increase, as the conesheet angle is changed, while at the blade/conesheet junctions, the strains first increase, but then decrease.

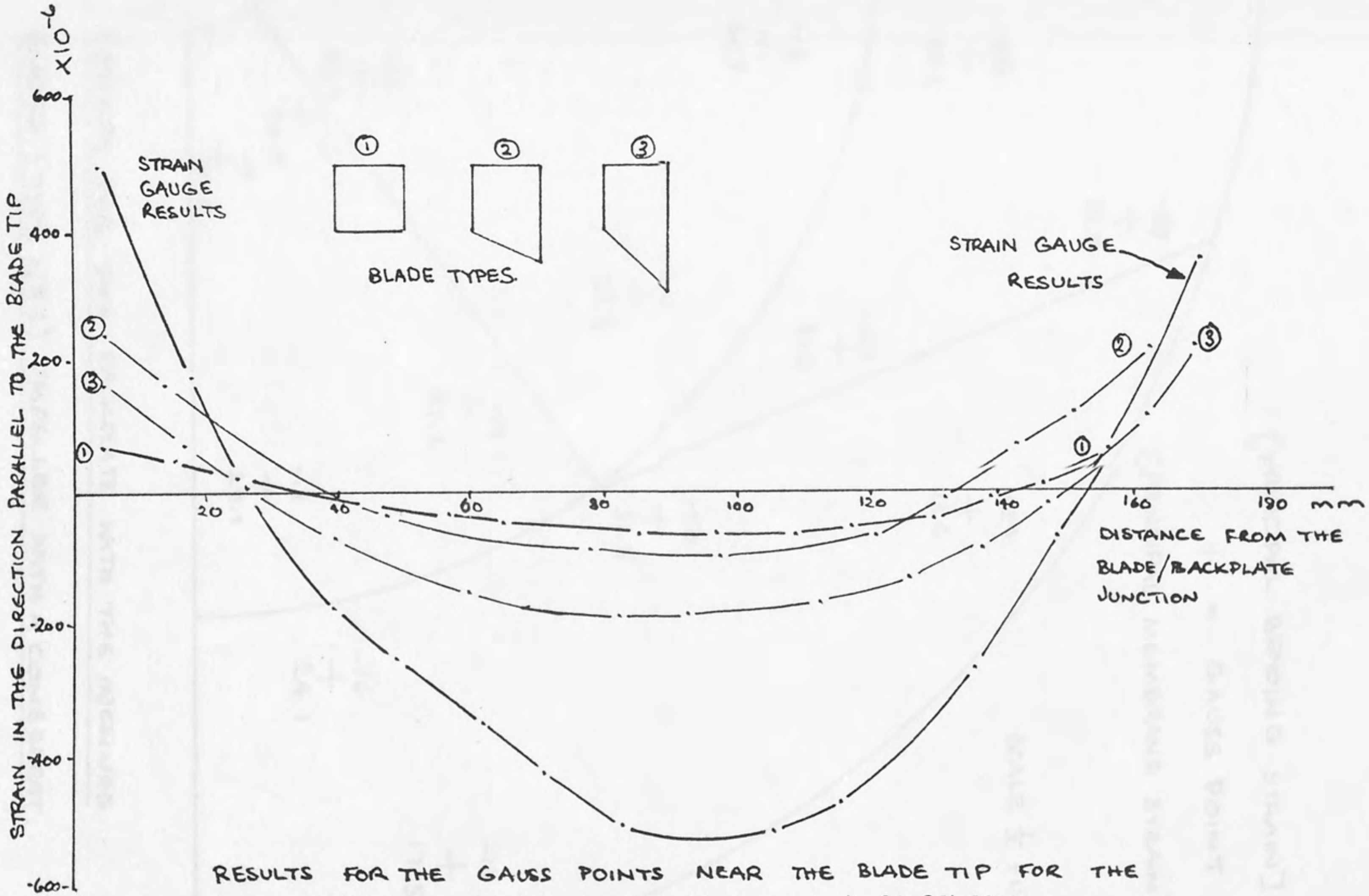
The results for the backplate are shown in Figs. 6.29. to Fig. 6.31., and as with blade, the strains in the backplate are lower than in the case of the impeller without a conesheet. Again, the bending strains are dominant and the maximum occurs just below the backplate/blade junction at the blade root. As in the case of the blade at the blade/conesheet junction, in the backplate, there appears to be no clear pattern in the strain distribution as the conesheet angle is changed. However, the strains near the backplate/blade junction (the maximum) increase, while the membrane strains as before are unaffected.

The results for the conesheet are shown in Figs. 6.32 to Fig. 6.34.; as with the backplate the bending strains are dominant and the maximum strain occurs just below the conesheet/blade junction at the blade root. Again, the strain distribution in conesheet is complex, and no clear pattern emerges as the conesheet angle is varied. In most areas of the conesheet, the strains first increase, but then decreases as the angle is increased. Once again, there is insignificant change in the membrane strains. In areas where the bending strain is of the order of 100 micro strain, the membrane strains cannot be neglected. Indeed when the conesheet angle is 0°, the membrane strains are dominant in all areas of the conesheet while in the backplate in this case, bending strains are mainly dominant.



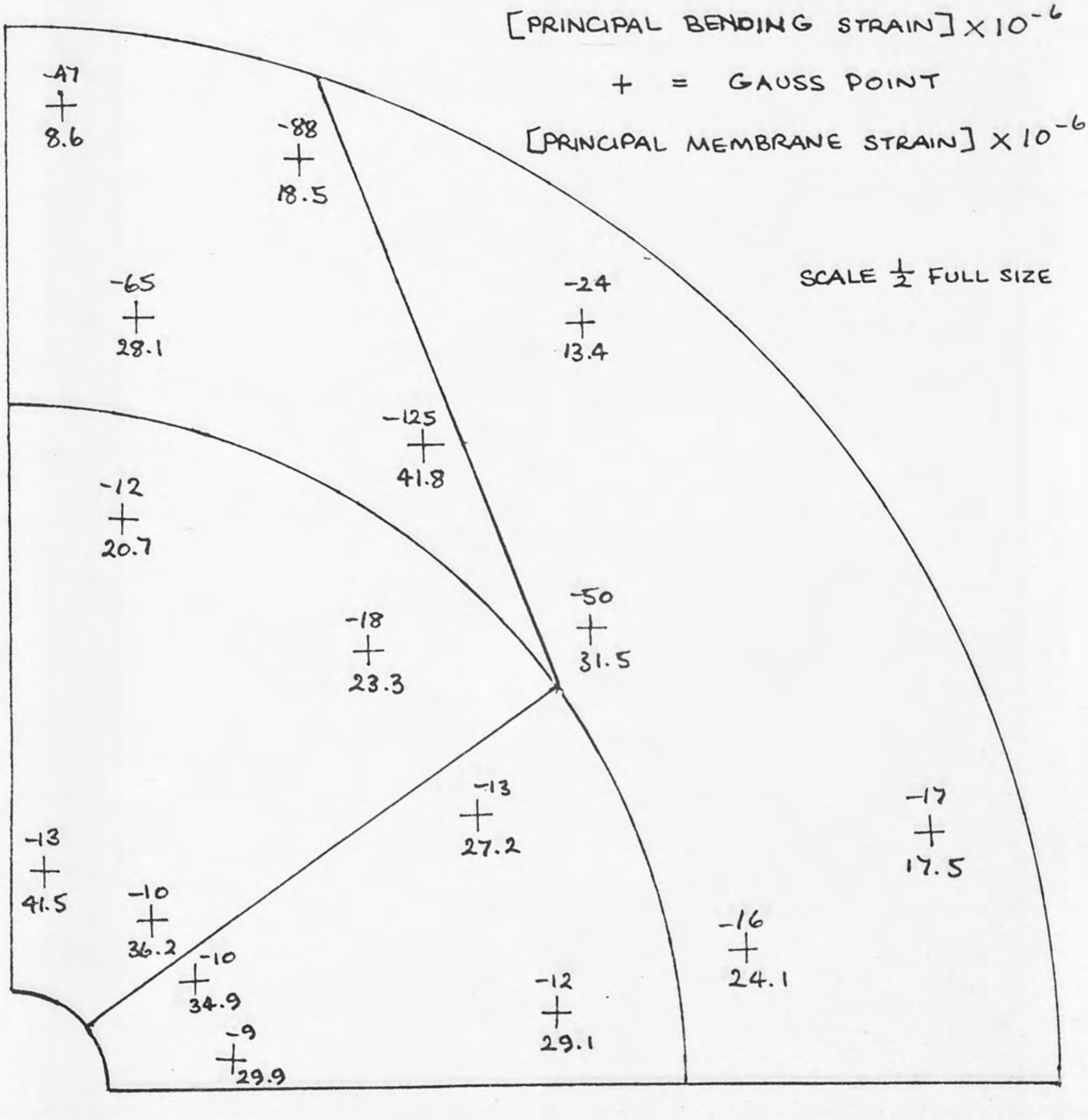
RESULTS FOR THE GAUSS POINTS NEAR THE BLADE ROOT FOR THE INCLINED BLADED IMPELLERS WITH A CONESHEET

Fig 6.27



RESULTS FOR THE GAUSS POINTS NEAR THE BLADE TIP FOR THE INCLINED BLADED IMPELLERS WITH A CONESHEET

Fig 6.28



RESULTS FOR THE BACKPLATE WITH THE INCLINED BLADED (TYPE Nº 1) IMPELLER WITH A CONESHEET.

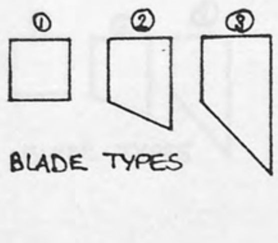
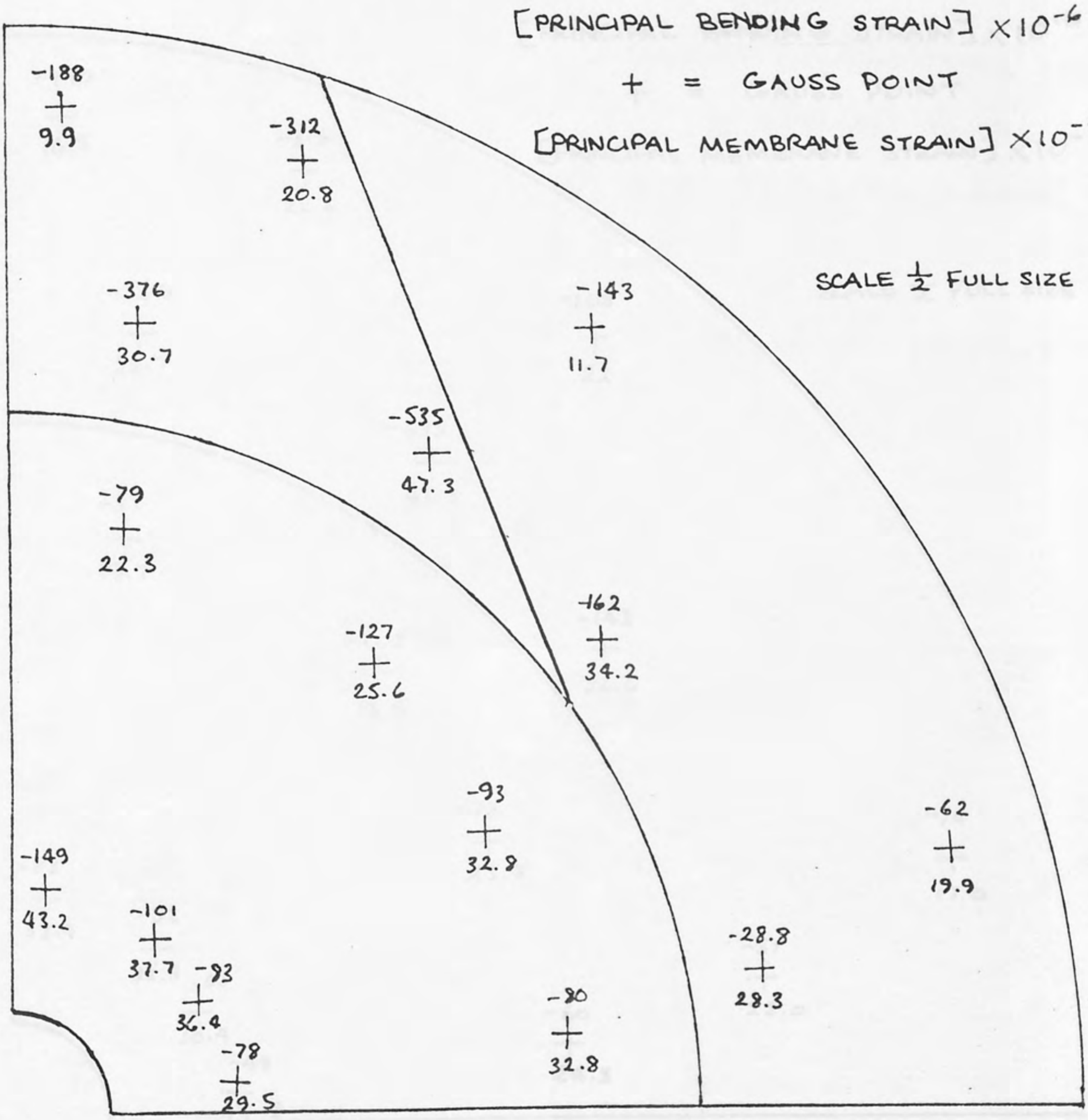
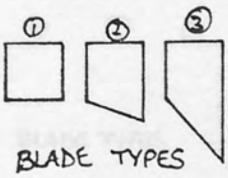


Fig 6.29

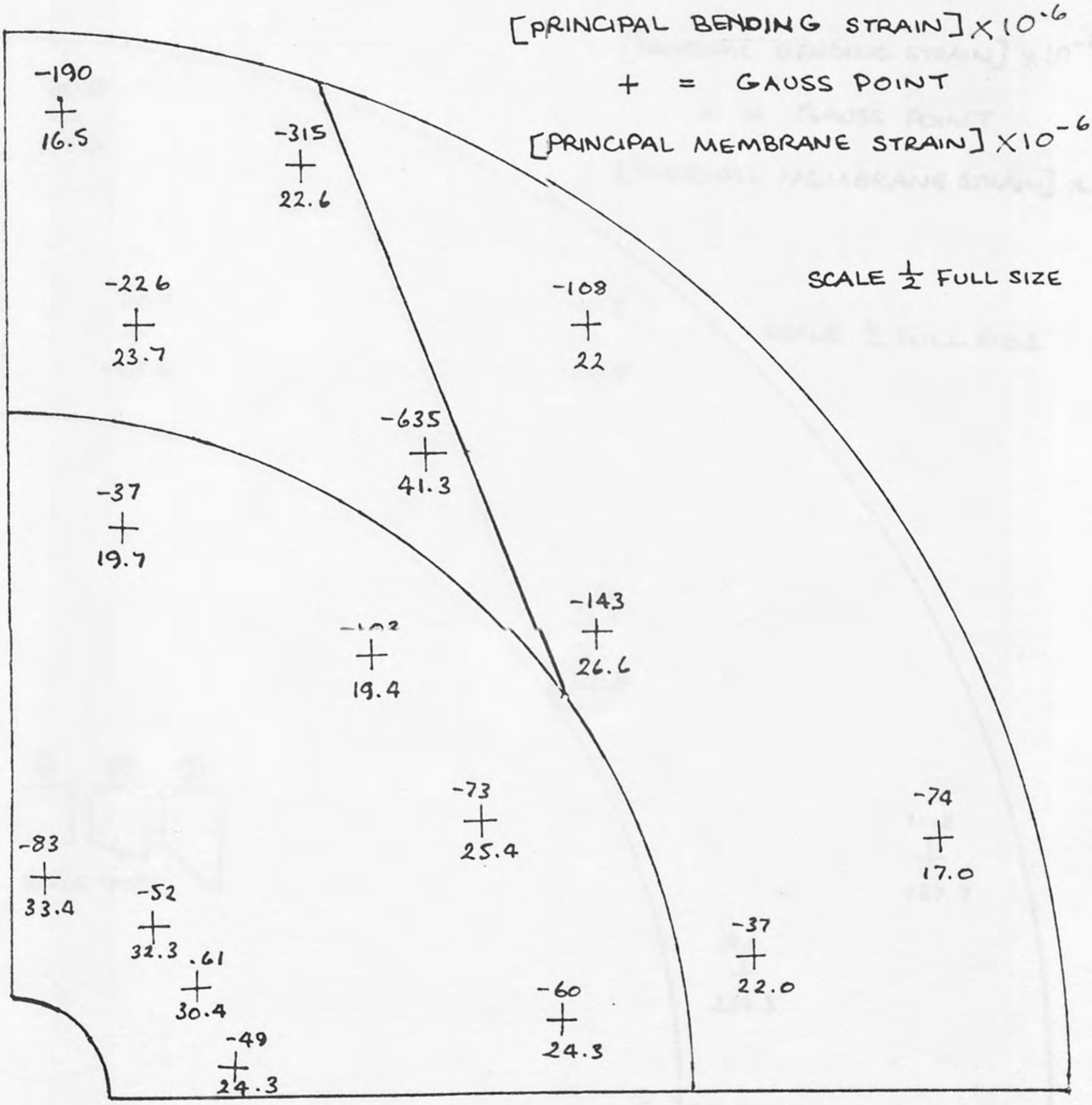


RESULTS FOR THE BACKPLATE WITH THE INCLINED BLADED (TYPE N° 2) IMPELLER WITH A CONESHEET.

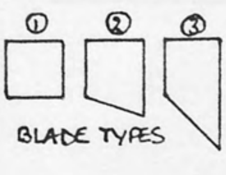


BLADE TYPES

Fig 6.30

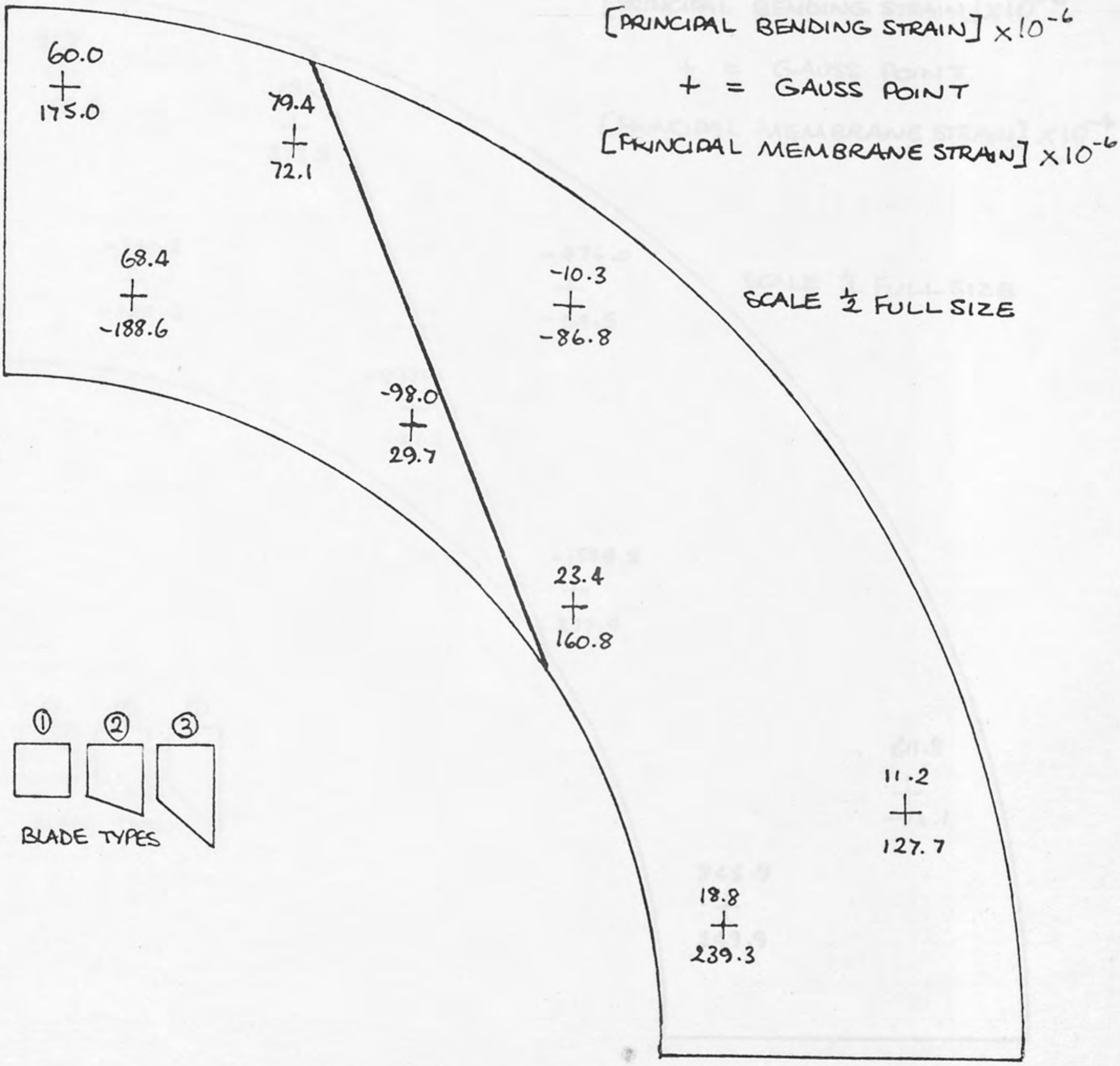


RESULTS FOR THE BACKPLATE WITH THE INCLINED BLADED (TYPE Nº 3) IMPELLER WITH A CONESHEET.



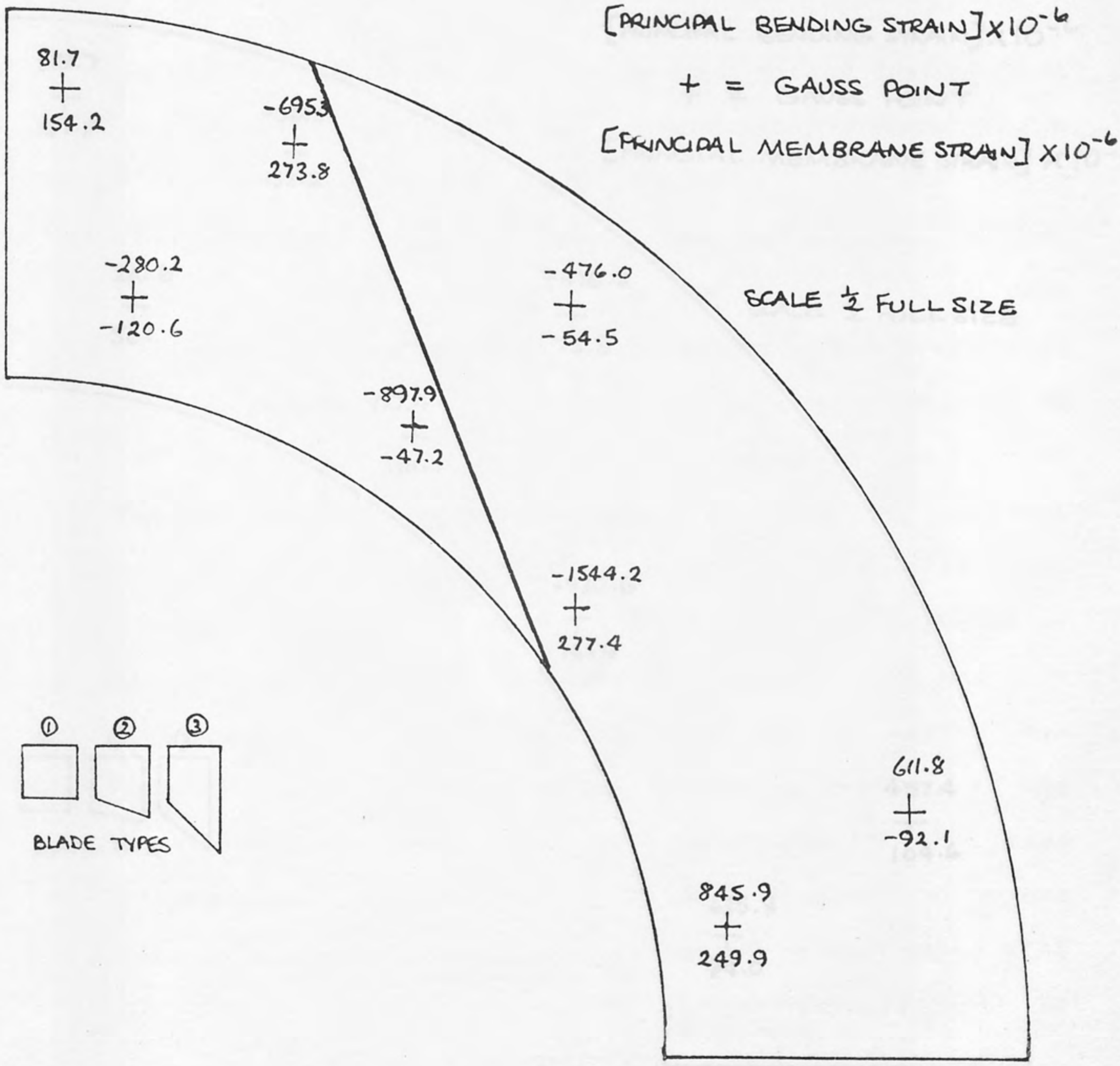
BLADE TYPES

Fig 6.31



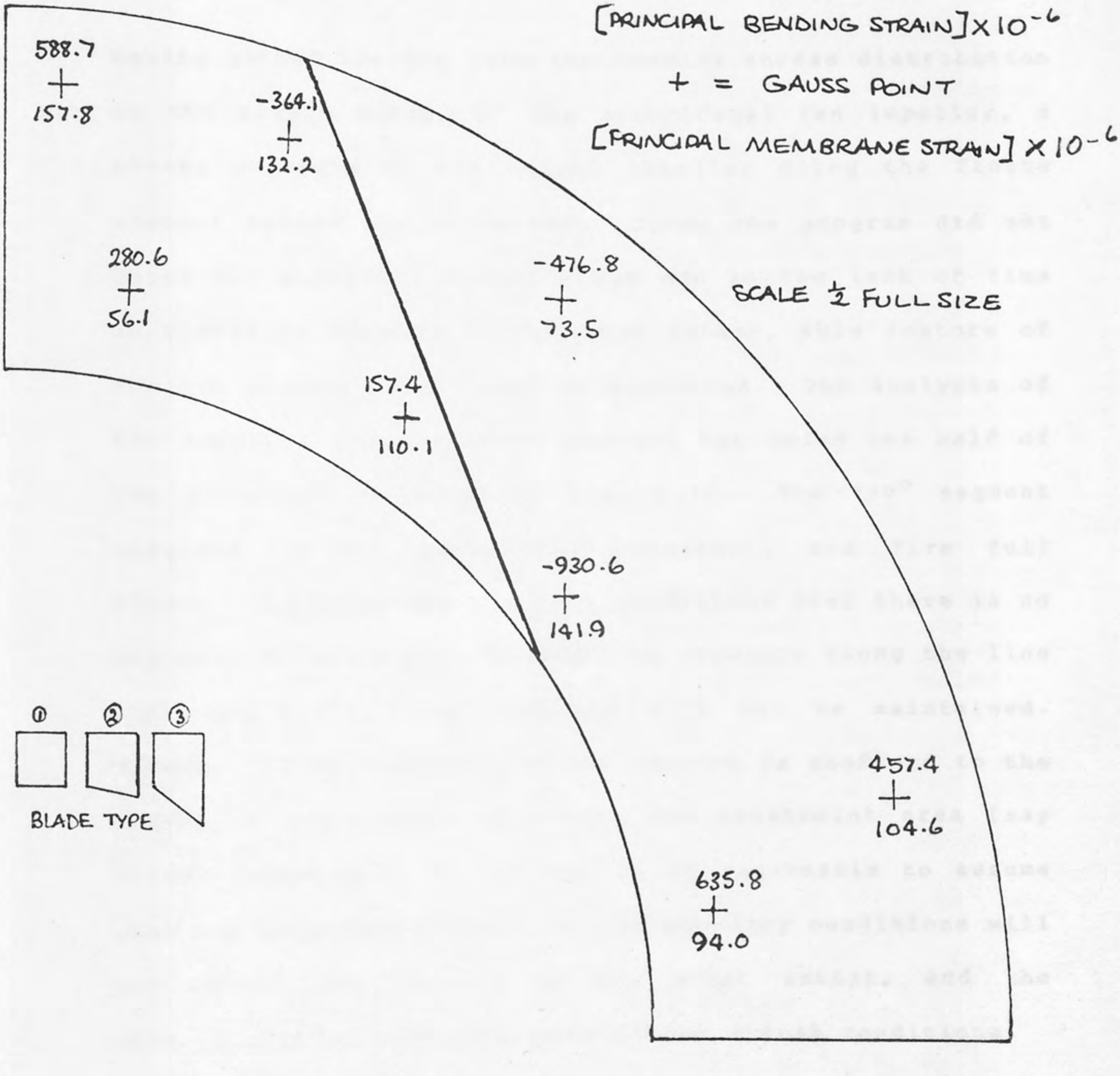
RESULTS FOR THE CONESHEET WITH THE INCLINED BLADED (TYPE Nº1) IMPELLER.

Fig 6.32



RESULTS FOR THE CONESHEET WITH THE INCLINED BLADED (TYPE No 2) IMPELLER.

Fig 6.33



RESULTS FOR THE CONESHEET WITH THE INCLINED BLADED (TYPE No 3) IMPELLER.

Fig 6.34

6.6. NUMERICAL STRESS ANALYSIS RESULTS FOR THE CENTRIFUGAL FAN IMPELLER

Having gained insight into the complex stress distribution in the simple models of the centrifugal fan impeller, a stress analysis of the actual impeller using the finite element method was attempted. Since the program did not cater for sectorial symmetry and due to the lack of time available to develop the program futher, this feature of the fan geometry could not be exploited. The analysis of the impeller was therefore carried out using one half of the structure as shown in Fig. 6.35. The 180° segment consists of the backplate, conesheet, and five full blades. Applying the boundary conditions that there is no movement in the Y direction and no rotation along the line A-B (Fig 6.35), true symmetry will not be maintained. However, if the analysis of the results is confined to the region of the blades away from the constraint area (say blades numbered 2, 3, and 4), it is reasonable to assume that the localised effects of the boundary conditions will not affect the results to any great extent, and the results will be representative of the actual conditions.

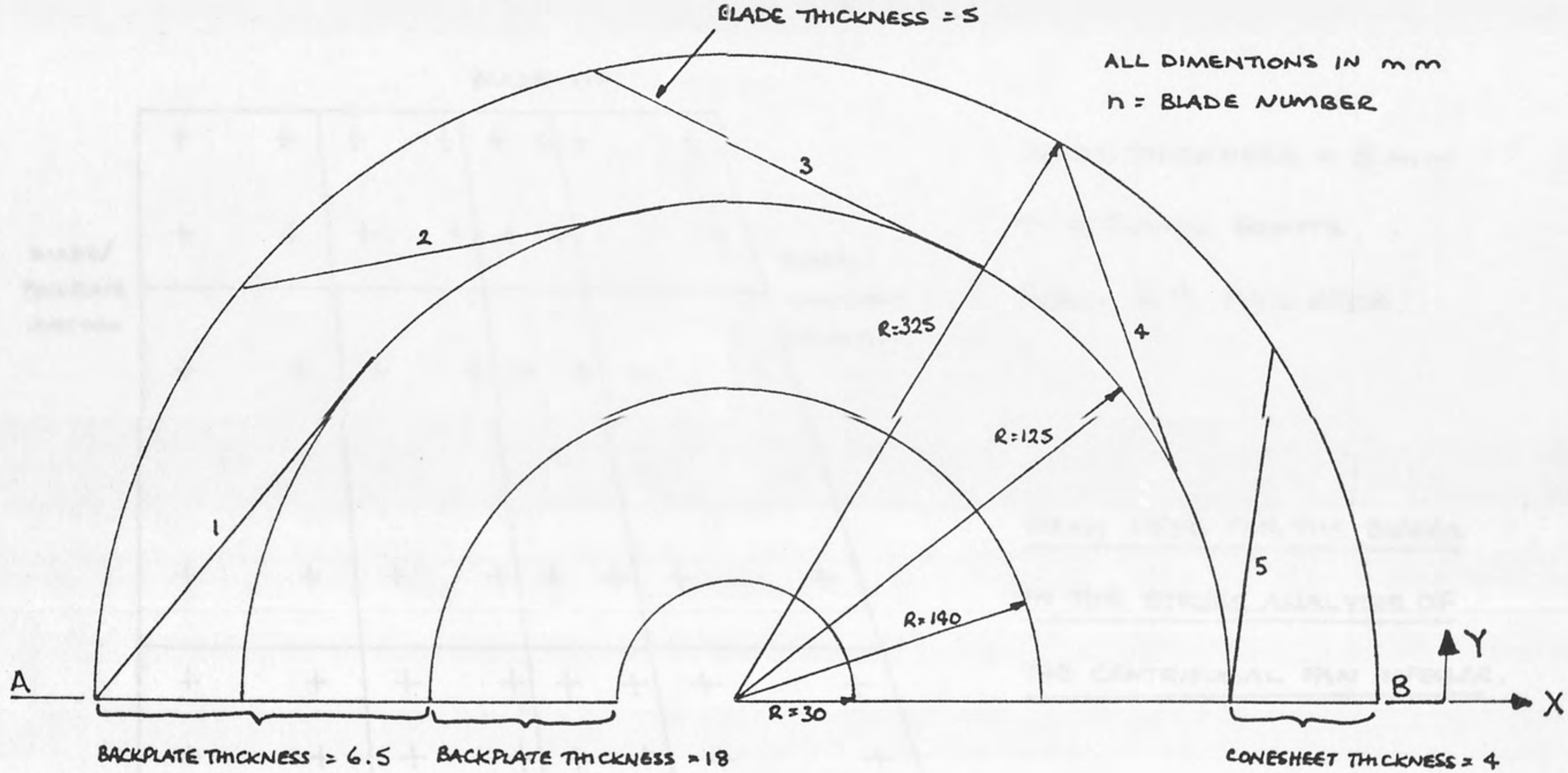
The structure was discretized into 60, 30, and 60 elements in the backplate, conesheet and the blades respectively. This representation of the impeller gave a structure with 2016 degrees of freedom of which 155 were constrained as applications of the boundary conditions. As before, the results obtained for an impeller speed of 1550 r.p.m. will be discussed in terms of the maximum principal strain for the backplate and conesheet; the strain in the direction parallel to the blade tip and root for the blade.

The results obtained for the finite element analysis of the blades at the Gauss points (Fig. 6.36) are shown in Fig. 6.37. The results show that at the blade/backplate junction, the strain decreases by 25% from the blade root to the blade tip. Near the centre of the blade, the strain decreases from the blade root to the blade tip, but at some distance from the blade tip (approximately 25% of the blade length) the strain then increases towards the blade tip. At the blade/conesheet junction, there appears to be no clear pattern in the strain distribution from the blade root to the blade tip. This is because the conesheet is much more flexible than the thicker section backplate, and therefore influenced considerably by the action of the blade.

The results for the backplate are shown in Figs. 6.38 to Fig. 6.40. It was decided to present the results in the form shown because of the difficulty in drawing curves of strain (or stress) variation in a very rapidly varying and complex stress field. The results for the shaft hub (Fig. 6.38) area shows that the membrane strains are dominant, although the bending strains cannot be neglected. Due to the distance from the blade and the thickness of this part of the backplate, this area of the backplate is unaffected by the action of the blade. Due to sectorial symmetry, the results obtained are symmetrical, thus giving confidence in the results obtained. Both the membrane and bending strains decrease, as the radius increases. The results for the backplate from the outer radius of the hub to the root radius of the blade are shown in Fig. 6.39. There is a sharp increase in the strain in this area (membrane strain increases from -23 to -53 micro strain and the bending from -12 to -149 micro strain) due to the change in thickness between the hub and the backplate (Fig. 6.35) and also due to the influence of the blades. In this area of the backplate bending strain is dominant, although membrane strains cannot be ignored. As the distance to the blade decreases, the strain as expected increases due to the influence of the blade.

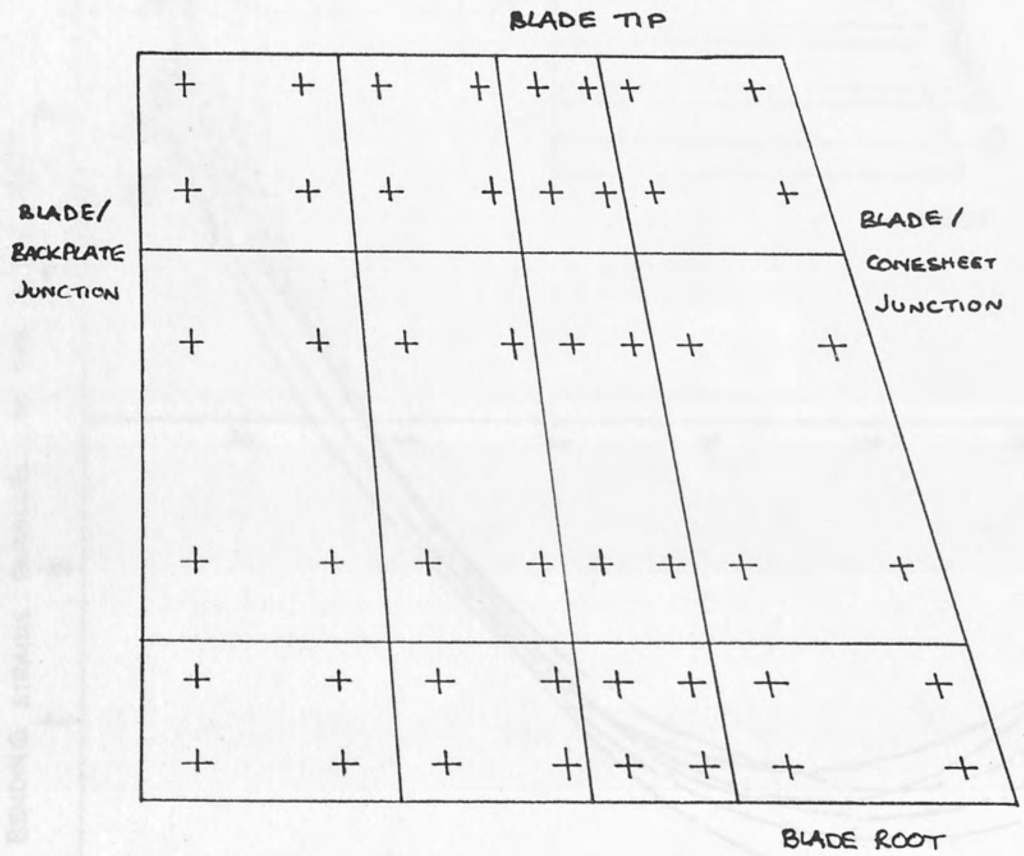
The area of the backplate most affected by the blade is shown in Fig. 6.40; it shows that the strain distribution is very complex, and that the bending strains are dominant, but in most of the area of this upper portion of the backplate, the membrane strains again cannot be ignored. In fact in some areas (towards the outer radius of the backplate) the membrane strains are of the same order. The results obtained in this area, agree well with corresponding points in adjacent sectors of the impeller, thus showing the consistency of the results. The area of high strain (as expected) is around the backplate/blade junction, with the higher strains occuring below this junction. The strains also increase from the blade tip to the blade root, with the maximum occuring at the blade root. Although the bending strains vary considerably, the membrane strains show insignificant change. The results show the complex strain (stress) distribution in this component, and confirms the need for a more modern approach to the design of the impeller.

The results for the strain analysis of the conesheet are shown in Fig. 6.41; they show that due the conesheet being more flexible than the backplate (i.e. thinner in section than the backplate), the strain distribution in this component is much more complex than in the backplate. The areas of high strain again are around the blade junction. Although bending action is predominant in the conesheet, membrane strains are significant, and in some areas (towards the outer radius) are of the same order or higher than the bending strain. Again, as expected, the strains increase from the blade tip to the blade root, with the highest strain above the blade tip. It is interesting to note that if the membrane strains are superimposed onto the bending strains, the inner radius of the conesheet is in tension (bending), while above the blade junction it is in compression (bending) and in the vicinity of the outer radius membrane (tension) is dominant, as shown in Fig. 6.42.



DIMENSIONS AND SEGMENT USED IN THE STRESS ANALYSIS OF THE CENTRIFUGAL FAN IMPELLER.

Fig 6.35



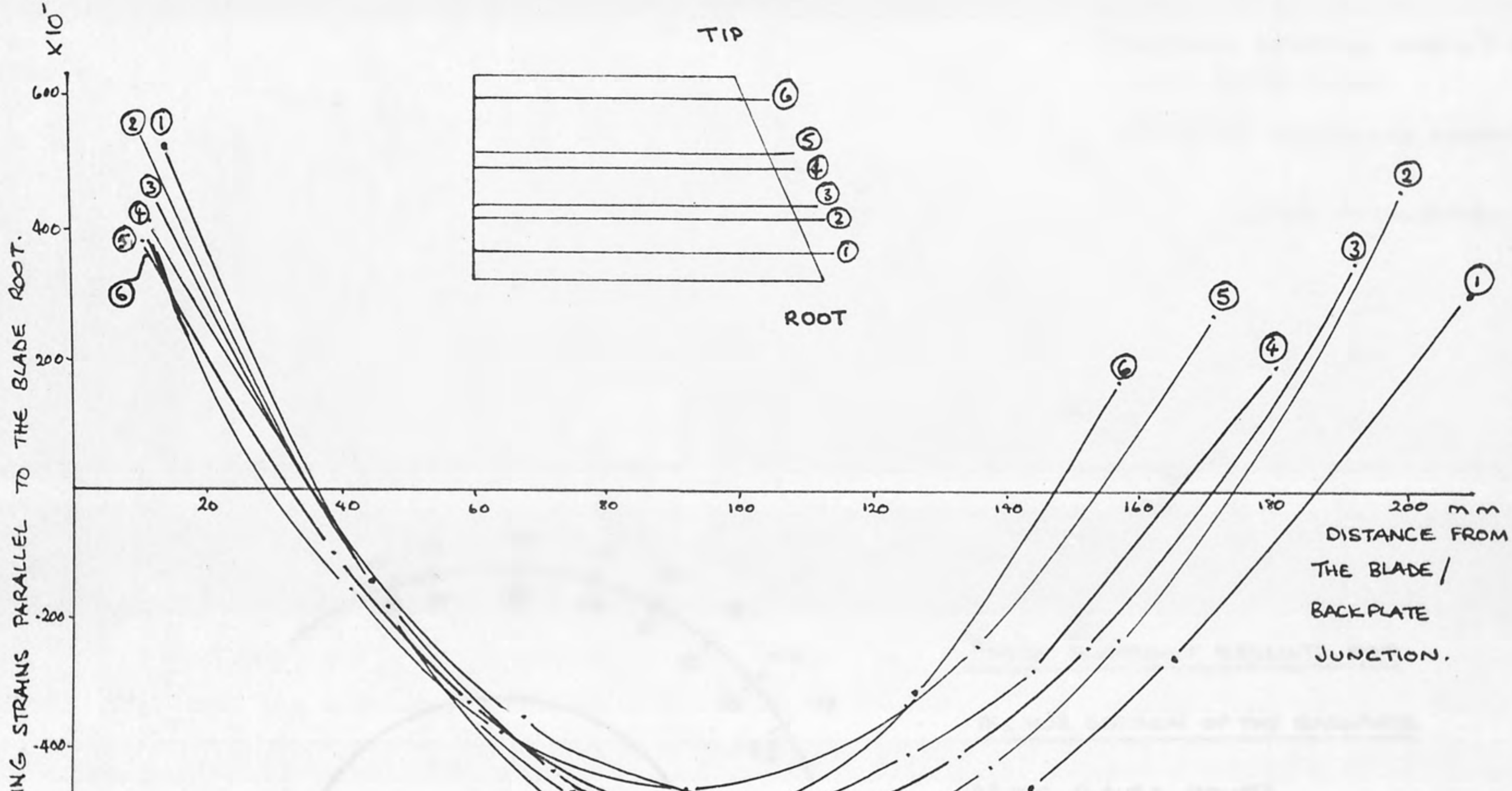
BLADE THICKNESS = 5 mm.

+ = GAUSS POINTS.

SCALE = $\frac{1}{2}$ FULL SIZE.

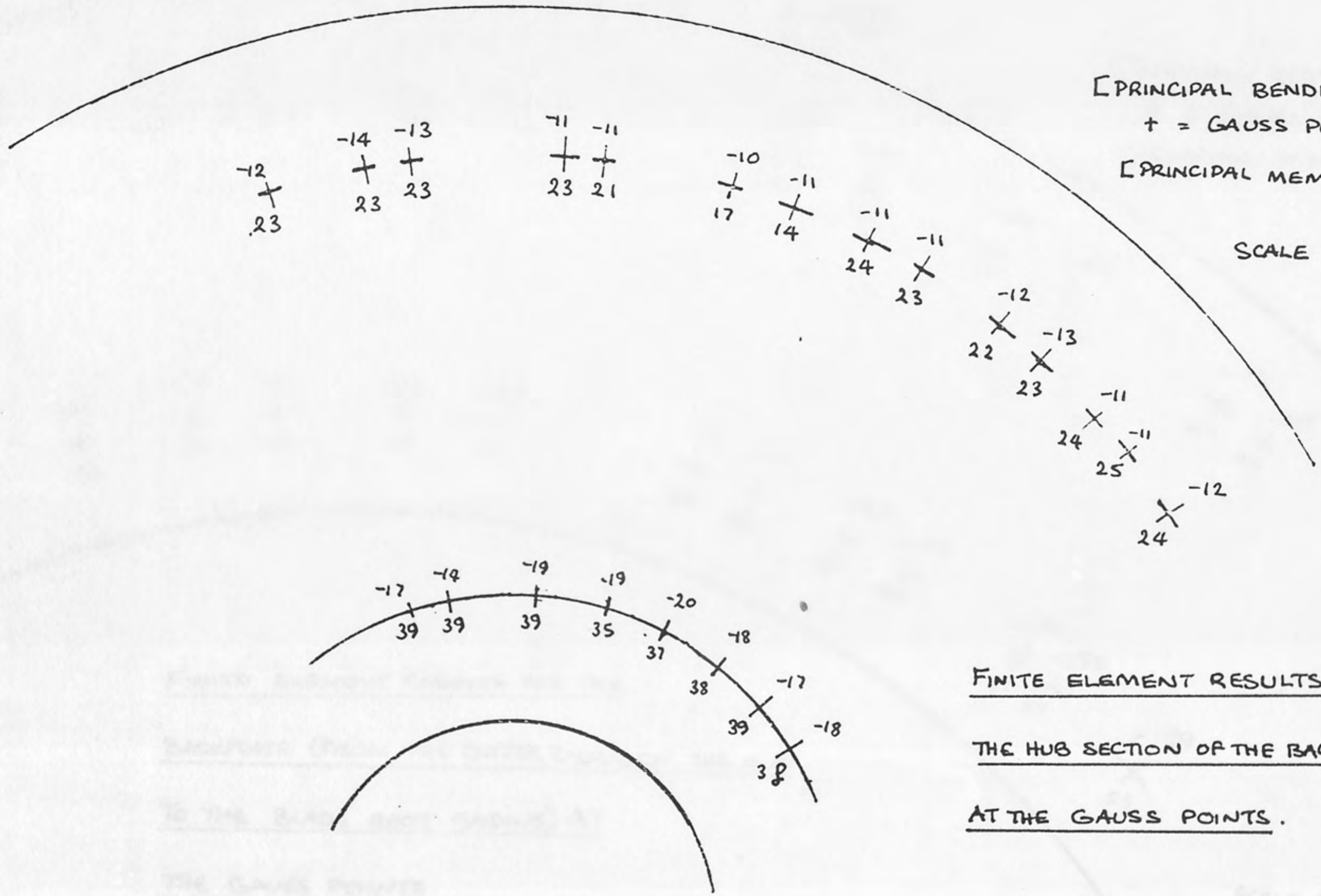
MESH USED FOR THE BLADES IN THE STRESS ANALYSIS OF THE CENTRIFUGAL FAN IMPELLER.

Fig 6.36



FINITE ELEMENT RESULTS FOR THE BLADE AT THE GAUSS POINTS

Fig 6.37



[PRINCIPAL BENDING STRAIN] X10^{-6}

+ = GAUSS POINT

[PRINCIPAL MEMBRANE STRAIN] X10^{-6}

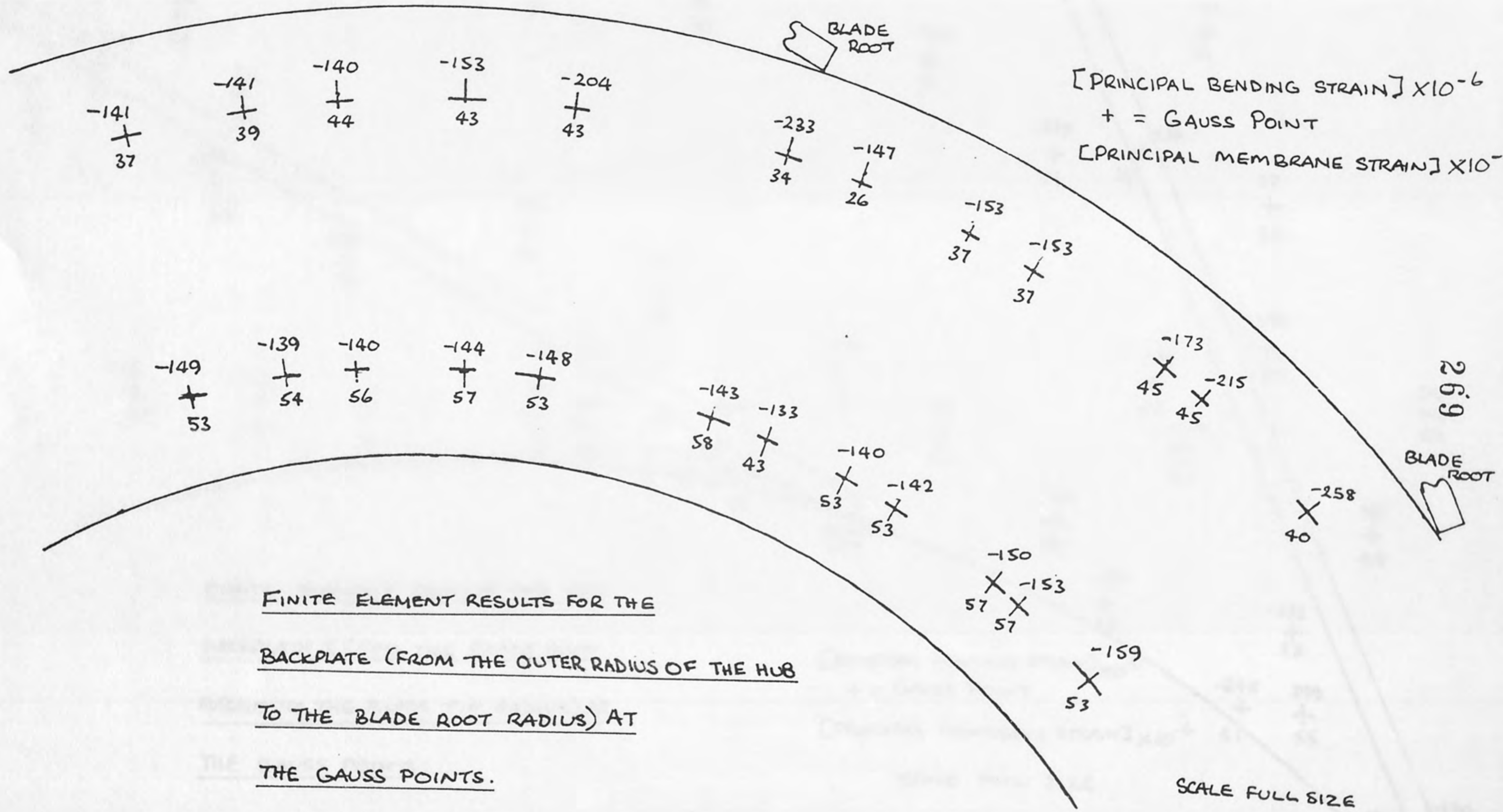
SCALE FULL SIZE.

FINITE ELEMENT RESULTS FOR

THE HUB SECTION OF THE BACKPLATE

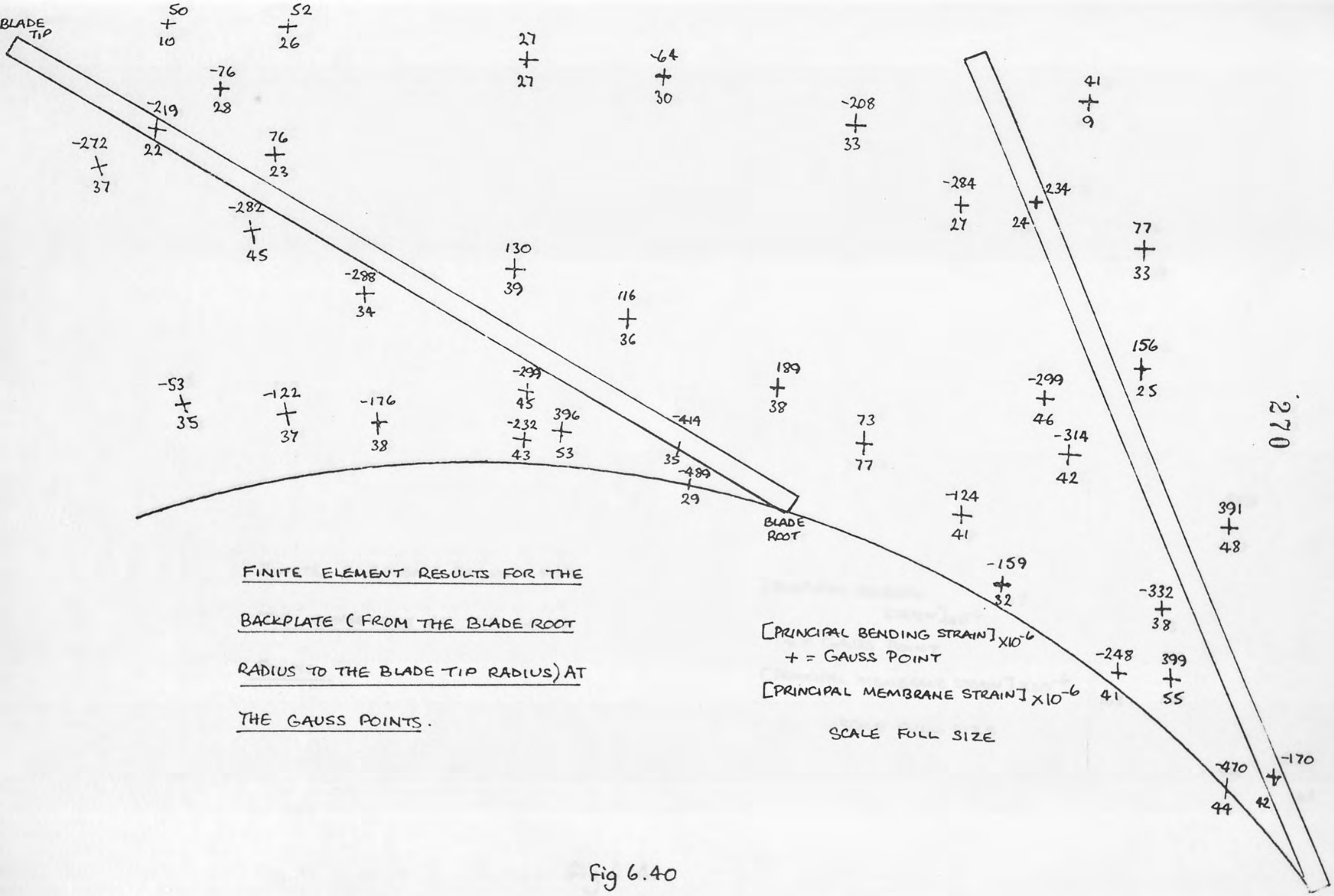
AT THE GAUSS POINTS.

fig 6.38



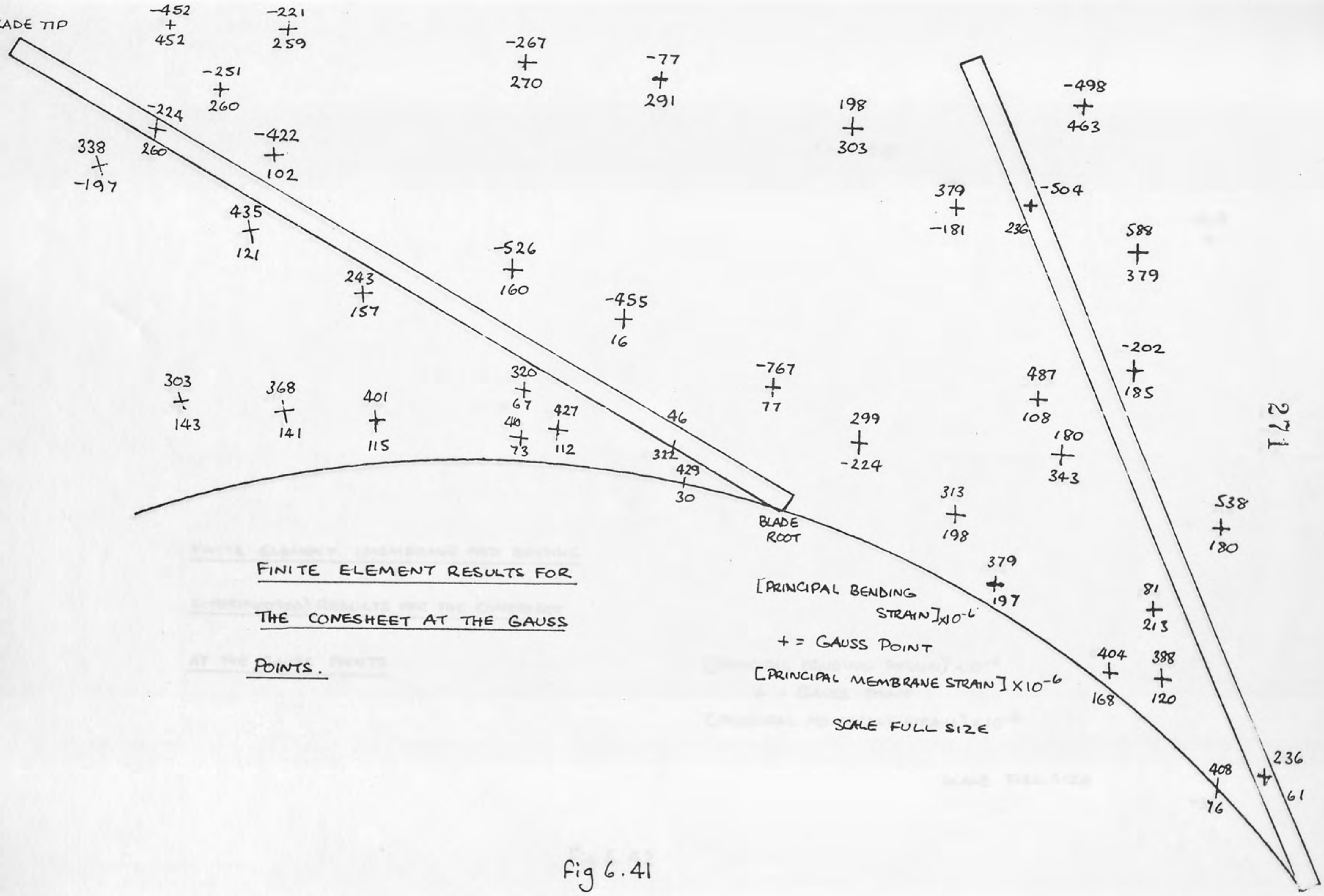
FINITE ELEMENT RESULTS FOR THE BACKPLATE (FROM THE OUTER RADIUS OF THE HUB TO THE BLADE ROOT RADIUS) AT THE GAUSS POINTS.

Fig 6.39



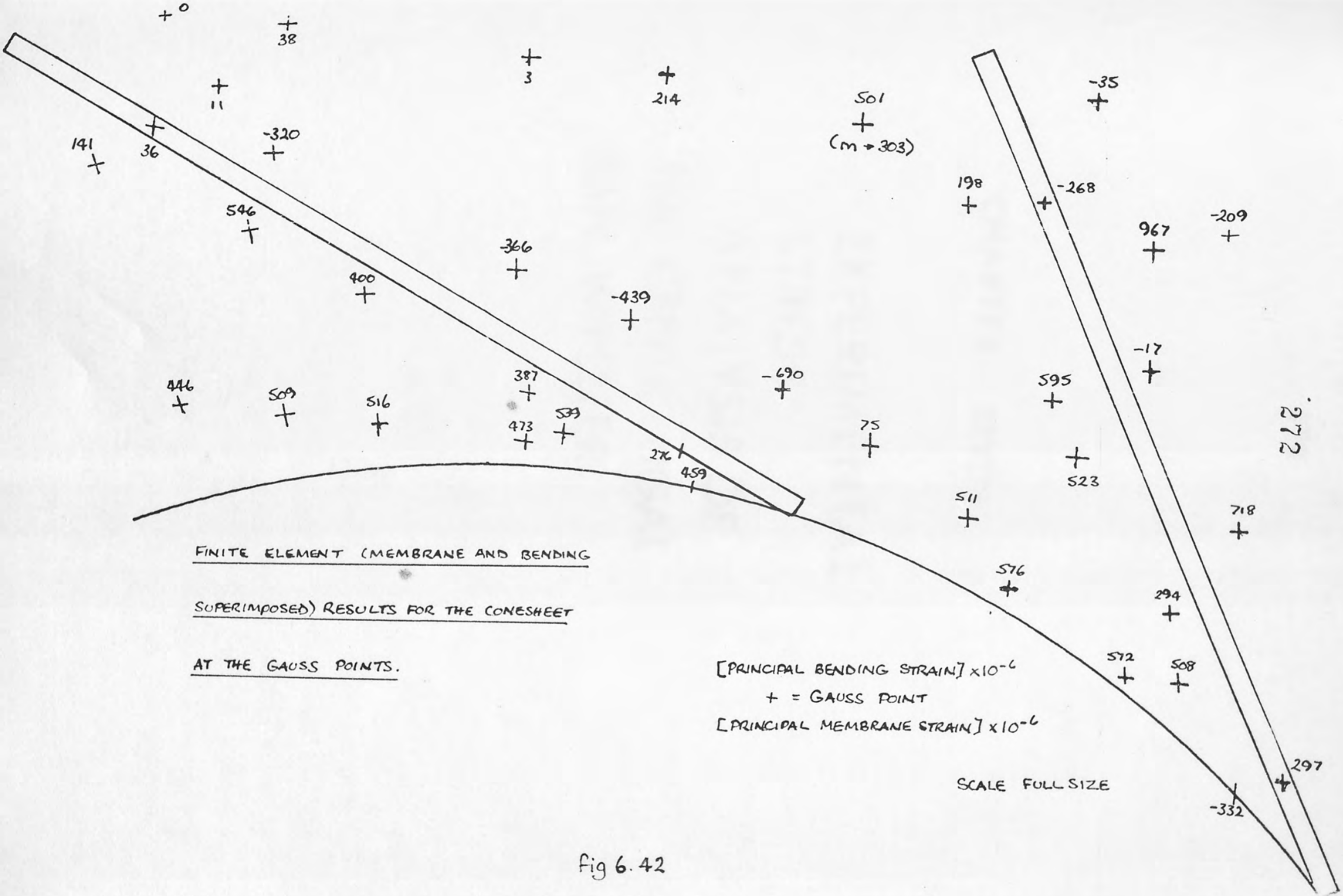
FINITE ELEMENT RESULTS FOR THE BACKPLATE (FROM THE BLADE ROOT RADIUS TO THE BLADE TIP RADIUS) AT THE GAUSS POINTS.

Fig 6.40



FINITE ELEMENT RESULTS FOR THE CONESHEET AT THE GAUSS POINTS.

Fig 6.41



FINITE ELEMENT (MEMBRANE AND BENDING SUPERIMPOSED) RESULTS FOR THE CONESHEET AT THE GAUSS POINTS.

fig 6.42

CHAPTER SEVEN

EXPERIMENTAL STRESS ANALYSIS OF THE CENTRIFUGAL FAN IMPELLER

INTRODUCTION

Chapter 7 discusses the experimental stress analysis of the centrifugal fan impeller; it begins with a brief discussion on the importance of experimental stress analysis as part of the design process. It leads onto a discussion of the various methods considered and the objectives of the present work.

Having decided the method to be used to be strain gauges, the electrical circuit to be used in measuring the strain in the centrifugal fan impeller is then described.

The test procedure adopted and the results obtained are then presented.

7.1. THE IMPORTANCE OF EXPERIMENTAL STRESS ANALYSIS IN THE DESIGN PROCESS

Many factors make the experimental approach indispensable, and is often the only means of access, in the investigation of problems of mechanical strength. It is remarkable how quickly we reach the limit of applicability of mathematical methods of stress analysis, and there is a multitude of comparatively simple, and in practice, frequently occuring, stress problems for which theoretical solutions cannot be obtained easily. In addition to this, theoretical considerations are usually based on simplifying assumptions which imply certain detachment from reality, and it can be decided only by experimentation, whether such idealisation has not resulted in any undue distortion of the essential features of the problem. No such doubt, however, enters experimental stress analysis, especially if it is done under actual service conditions, where all factors due to the properties of the employed materials, the methods of manufacture, and the conditions of operation are fully represented. The advantage of the experimental approach becomes especially obvious if we consider that it is possible to determine experimentally the stress distribution in a structure (or machine part) under actual operating conditions without knowing the nature of the forces acting on the structure under these circumstances, such reasoning which cannot be applied to any theoretical method of analysis.

Having obtained data experimentally, the adequacy of a theoretical model can be assessed (with all its built in assumptions) and the assumptions modified as necessary.

7.2. EXPERIMENTAL STRESS ANALYSIS METHODS CONSIDERED FOR THE CENTRIFUGAL FAN IMPELLER

Of the various experimental techniques available to the stress analyst in analysing the stress distribution in the centrifugal fan impeller, three methods were considered, namely photoelasticity, brittle lacquer and strain gauges. After considering the possibility of applying a photoelastic method, it was decided that it could not be reasonably attempted, because an appropriate scaling down of the overall diametral and axial dimensions to obtain a suitable size model, would result in the sheet thickness becoming so small as to make the whole model grossly deformable at the softening temperature for stress freezing.

The use of brittle lacquer for the experimental stress analysis of rotating machinery presents certain difficulties and drawbacks not usually encountered in normal applications. Firstly 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 highlight areas of compressive strains. When the fan is run, the rise in temperature reduces the strain sensitivity of the brittle lacquer by approximately 55 micro strain for every $1^{o}C$ rise in temperature. The disadvantage with this technique is that is very difficult to ensure that the temperature of the calibration bar is the same as that of the impeller. Another major disadvantage with this technique is that only qualitative results are obtained and would therefore be of limited use, since quantitative results of the magnitude and direction of the actual stresses in the impeller are required for the present work.

Strain gauging of rotating parts also presents several problems. The major one, in bringing the leads from the rotating assembly to the measuring instrument, has now been overcome with modern slip-ring units, in which electrical noise is not only small but also, and which is most important, constant in each channel. This means that it can be compensated for by arranging the electrical bridge in the appropriate way (which is discussed in section 7.4). With temperature compensated gauges it is now possible to dispense with the dummy gauge(s) near the impeller, which never could be relied upon to be at the same temperature as the active gauge. The main advantage of using strain gauges is that quantitative results can be obtained in the areas of interest easily. In view of the advantages of using strain gauges, it was decided to use this method in the present work.

7.3. OBJECTIVES OF THE EXPERIMENTAL STRESS ANALYSIS

1). Obtain the stress distribution in the centrifugal fan impeller operating under normal conditions.

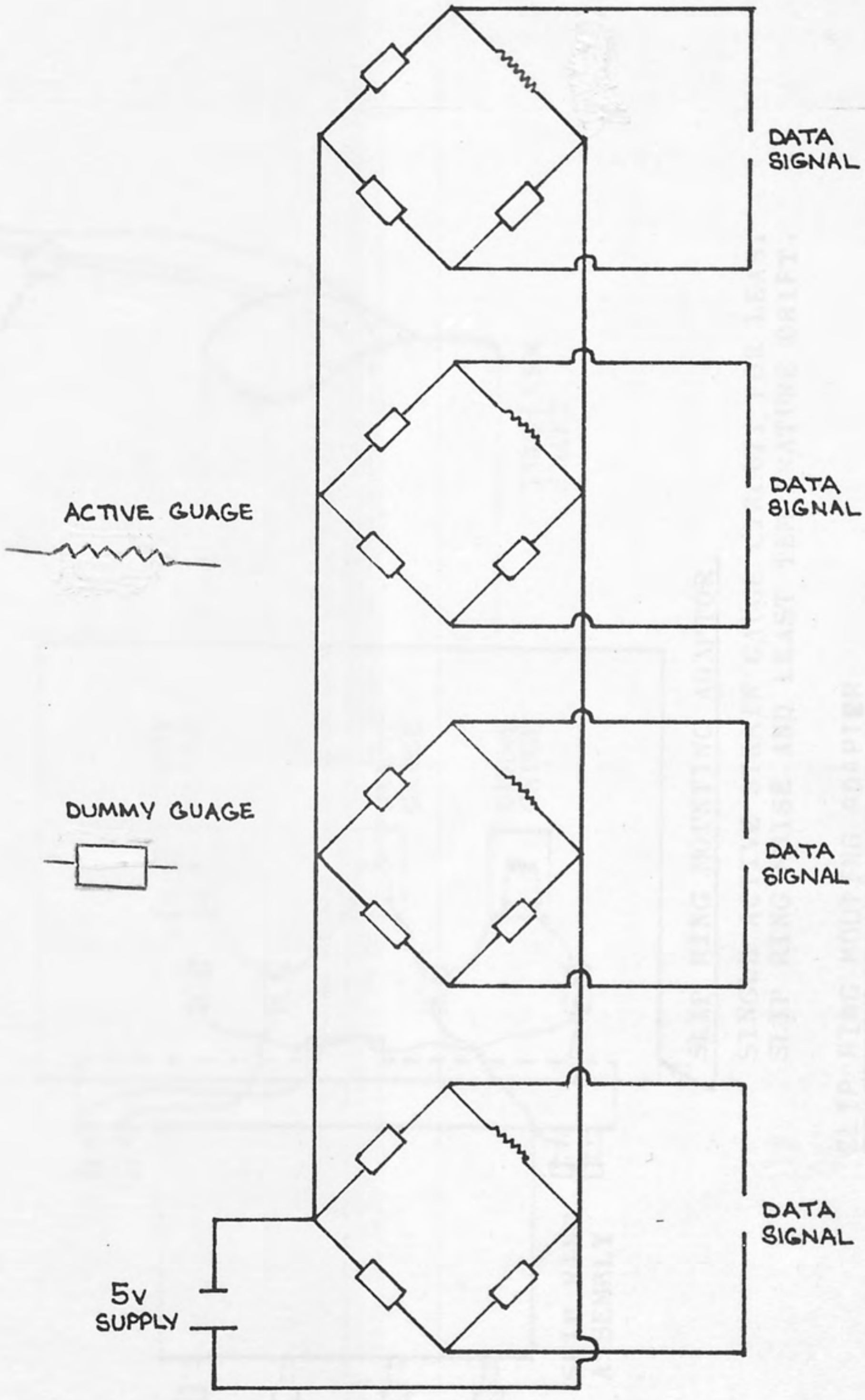
2). Assess the adequacy of the finite element model of the centrifugal fan impeller by comparing the analytical results with those obtained experimentally.

7.4. ELECTRICAL CIRCUIT USED IN THE MEASUREMENT OF THE STRAIN IN THE CENTRIFUGAL IMPELLER

Gall [40] describes and discusses the advantages and disadvantages of various circuits used in the measurement of strain in rotating structures. Slip ring noise which is caused by the slight variation in the ring connection resistance and the rubbing contacts, is the main problem when using slip ring assemblies. The important thing to note according to Gall is to use a full bridge circuit on the rotating side and only to bring out the current and potential connections through the slip rings. In this case, since the rings and the rubbing contacts which cause the noise, are outside the bridge circuit it cannot affect the bridge balance, and since the bridge output is a function of the bridge unbalance these small resistance changes will not affect the signal. In well designed slip ring assemblies, the peak noise attributable to the slip ring contacts is usually less than 0.1% of the signal.

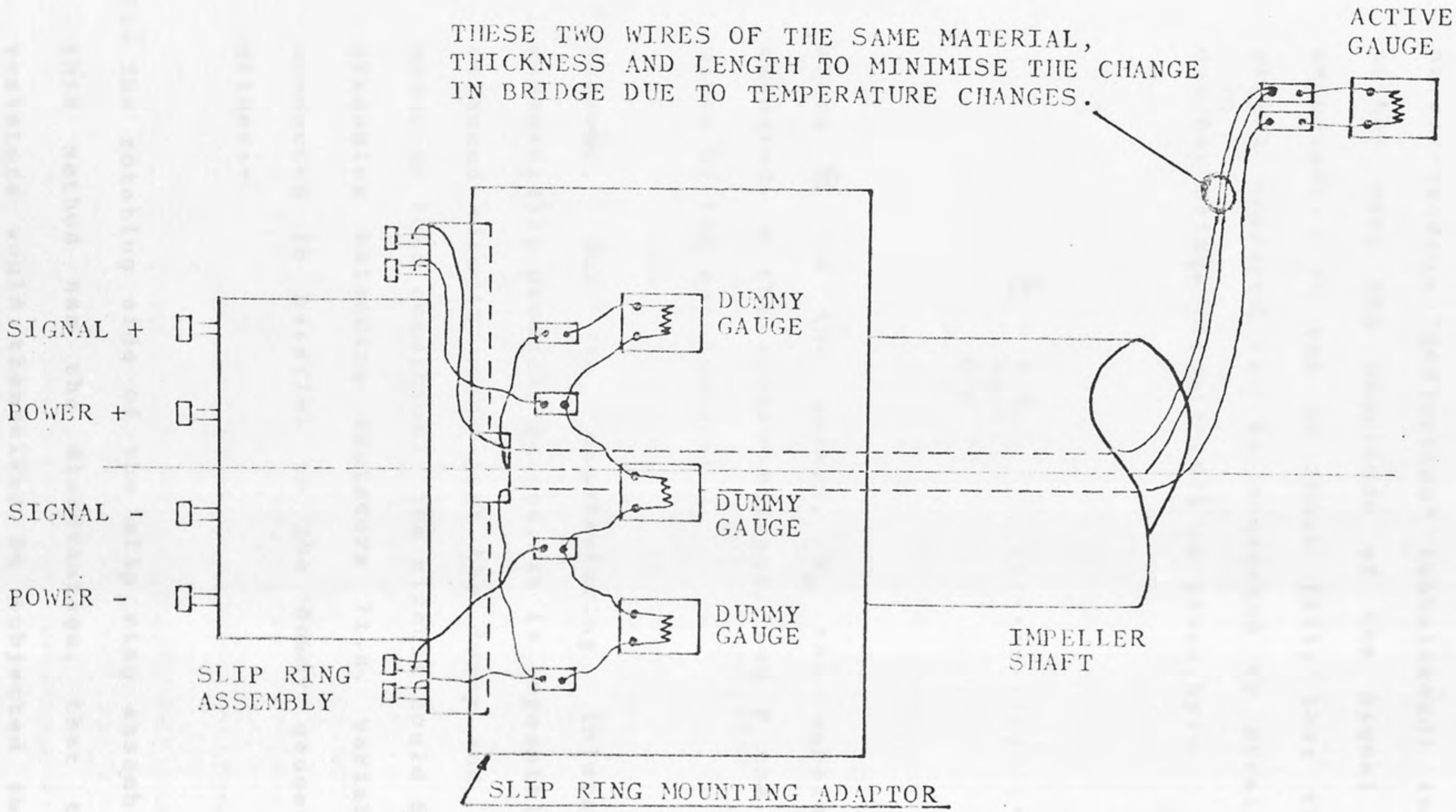
The circuit which consistently produces a high quality signal and the one used in this instance, is shown in Fig. 7.1. With this arrangement, using a 10 channel slip ring assembly, 4 channels of data using a 5 volt d.c. supply are obtained. The dummy gauges used to complete the full bridge are arranged on an adapter fixed onto the end of the shaft as shown in Fig. 7.2 (for clarity only one data circuit is shown). The leads from the active gauge are brought to the slip ring assembly via the hole in the shaft and adapter and out through the slot cut in the slip ring adapter. The advantage of this design is that it gives insignificant strain at the dummy gauge location, and when a quarter bridge circuit is used (as in the present work), the insignificant strain on the three dummy gauges in each circuit is the same, and thus has no effect on the signal measured due to the active gauge. That is, the signal measured is due solely to the change in resistance of the active gauge. The circuit shown also has the advantage of least slip ring noise and least temperature drift than any other circuit which could be used.

The Wheatstone bridge circuit theory is well known and used in a variety of strain measurement applications. The Wheatstone bridge can be used in either of two distinct



4 DATA SIGNAL STRAIN MEASURING CIRCUIT

Fig 7.1



SINGLE ACTIVE STRAIN GAUGE CIRCUIT FOR LEAST SLIP RING NOISE AND LEAST TEMPERATURE DRIFT.

SLIP RING MOUTING ADAPTER

FIG 7.2

modes of operation: as a null (balance) system or as a direct-reading "deflection" (unbalanced) instrument. In either case the magnitude of the signal (voltage) is important. It can be shown **[41]**, that the unbalanced signal measured can be converted to strain and, for a quarter bridge circuit this is given by:-

$$\varepsilon = \frac{4\ V_o}{V\ F} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(7.1)$$

where ε is the strain, V_o the unbalanced voltage measured, **V** the supply voltage, and **F** the gauge factor given by the manufacturer.

However, due to manufacturing tolerances, using commercially produced gauges, it is impossible to obtain a balanced circuit using just the dummy and active gauges under no load condition. The circuit could be balanced by arranging balancing resistors (i.e. variable resistors connected in parallel to the dummy gauges) to be on either:-

1). The rotating side of the slip ring assembly. However, this method has the disadvantage, that the balancing resistors would themselves be subjected to strain when rotating which would affect the signal being measured, and no compensation for this could be made.

2). The stationary side of the slip ring. The disadvantage of this arrangement is the same as having the dummy gauges on the stationary side of the slip ring assembly, which has already been discussed.

Due to these difficulties, it was decided not to use any balancing resistors at all. That is, the unbalanced signal with the fan stationary and with the fan rotating are measured. The difference between these two signals, gives the signal due to change in resistance of the active gauge, as a result of the forces acting on it. The corresponding strain can then be calculated using equation 7.1.

7.5. AREAS OF INTEREST

In order to understand the complex stress distribution in the impeller, the aim of the experimental stress analysis was to strain gauge the three components of the impeller extensively. On the backplate and conesheet the areas of interest, are at the blade junction with the backsheet and conesheet and the area between two successive blades on these two components. As the magnitude and direction of the principal stresses are not known and are of interest, rosette type strain gauges were required in these areas. Where possible it was hoped to install gauges on the outer and inner surfaces so that the bending effect on the component can be analysed.

For the blades, the aim was to again extensively strain this component of the impeller. In this case linear strain gauges are to be used in the transverse direction parallel to the blade tip.

Due to the time left to complete the project, extensive strain gauging of all the components of the impeller could not be accomplished. It was therefore decided to concentrate this work on the area of the blades and to obtain an accurate and detailed stress distribution in this component. This limited information would be of considerable importance to the sponsoring company as they have no such information, and most in service failures are due to blade failures.

The position of the strain gauges installed on the blade are shown in Fig. 7.3, in total 20 strain gauges were installed.

The gauges at the junction of the blade with the backplate and conesheet were placed as close to the junction as physically possible to obtain the stress at the junction

SCALE ½ FULL SIZE

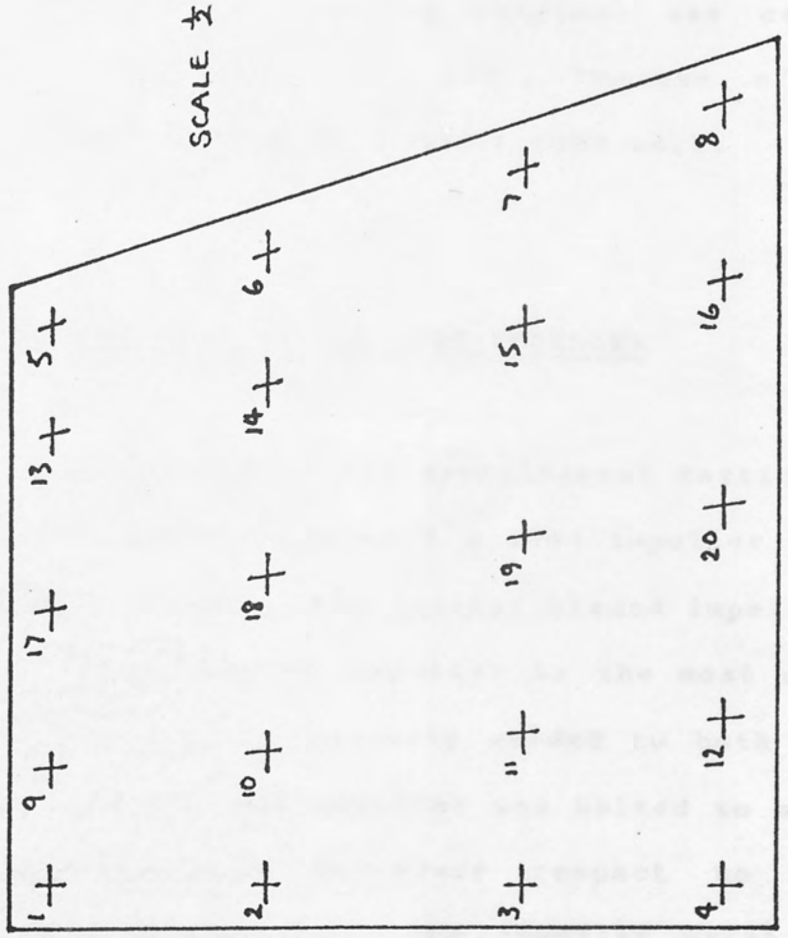


Fig 7.3

LOCATION AND NUMBERS OF THE STRAIN GAUGES ON THE BLADE and thus obtain information regarding the interaction of the two components at the junction. Gauges were also placed along the width of the blade so that the stress distribution along the width of the blade could be obtained.

The experimental stress analysis aspect of the project (described later in this chapter) was carried out at Alldays, Peacock & Co. Ltd., because of the testing facilities they had to conduct such work.

7.6. DESCRIPTION OF THE TEST IMPELLER

The impeller used in the experimental testing was a 650mm diameter laminar, backward bladed impeller which had ten equispaced blades. The laminar bladed impeller was chosen because this type of impeller is the most commonly used. The blades are continuously welded to both the backsheet and conesheet. The impeller was bolted to a cast hub and is representative in every respect to the fans in commercial production. The impeller which was balanced statically and dynamically to reduce the vibrations, was mounted on a shaft which was itself supported in bearings on a base plate. The baseplate was clamped onto a test bed and drive through a Vee belt was provided by a variable speed motor.

7.7. TEST PROCEDURE

The strain gauges were installed in the places of interest using the procedure recommended by the strain gauge manufacturer. The leads were soldered onto the strain gauges and securely attached to the impeller surface, so that they did not fly off at speed. The circuits were then connected to a constant 5 volt supply and the unbalanced voltage in each of the 4 data circuits measured using a Sintron digital volt meter, and noted. A digital voltage meter was used to measure the signal instead of a chart recorder, since an initial test had shown that with the fan running at a constant speed, the measured signal was constant and as such the digital volt meter would be suitable in measuring the signal. This constant signal can be attributed to the good design of the measuring circuit, and showed that slip ring noise was insignificant.

The fan speed was then increased to the desired value, and when the speed had been constant for a few minutes, the unbalanced signal in each data circuit measured again. This was repeated for a series of increments of speed until the readings for the maximum speed had been measured. The procedure was repeated with the fan speed decreasing from the maximum. The unbalanced signal was again measured with the fan stationary.

The leads soldered onto the active gauge were removed and soldered onto another gauge and the procedure repeated. Table 7.1 shows the readings taken on test. As there was no significant difference in the two readings measured, (ie. with the fan speed increasing and with the fan speed decreasing), the readings given in Table 7.1 are the average of those measured.

7.8. CALCULATION AND DISCUSSION OF THE EXPERIMENTAL STRESS ANALYSIS RESULTS

From the measured unbalanced voltages, the corresponding strains were calculated using equation 7.1. These results are shown in Table 7.2 to Table 7.4.

In order to check the behaviour of the gauges at different speeds, $\log_e$(strain) was plotted against $\log_e$(speed) for all the strain gauges (Fig. 7.4 and Fig.7.5). The gradient was calculated in each case, and the average gradient obtained was 2.0, showing that the strain is proportional to the speed squared. This is to be expected, since the body force (and hence strain) due to rotation is proportional to the speed squared. These

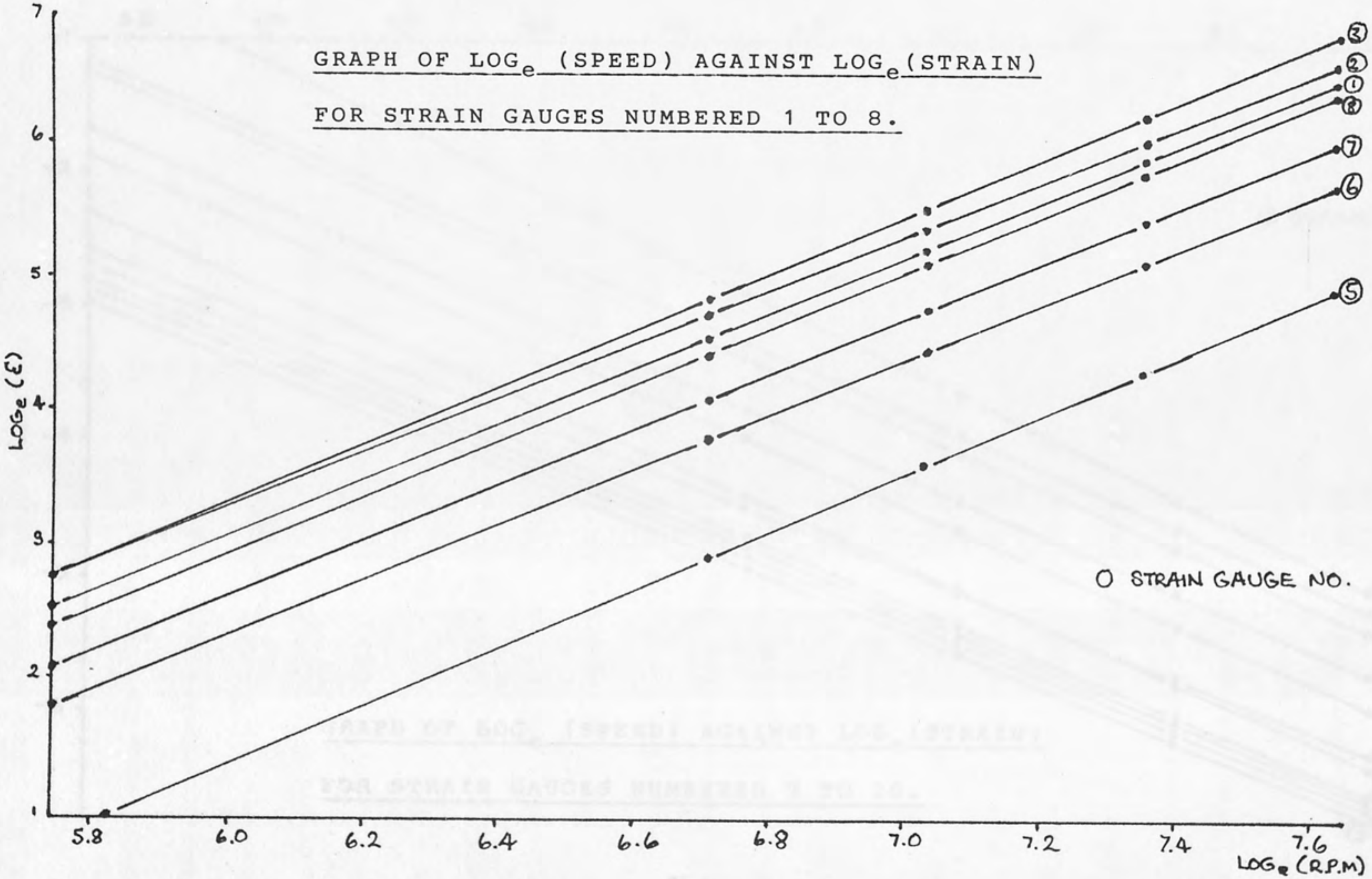
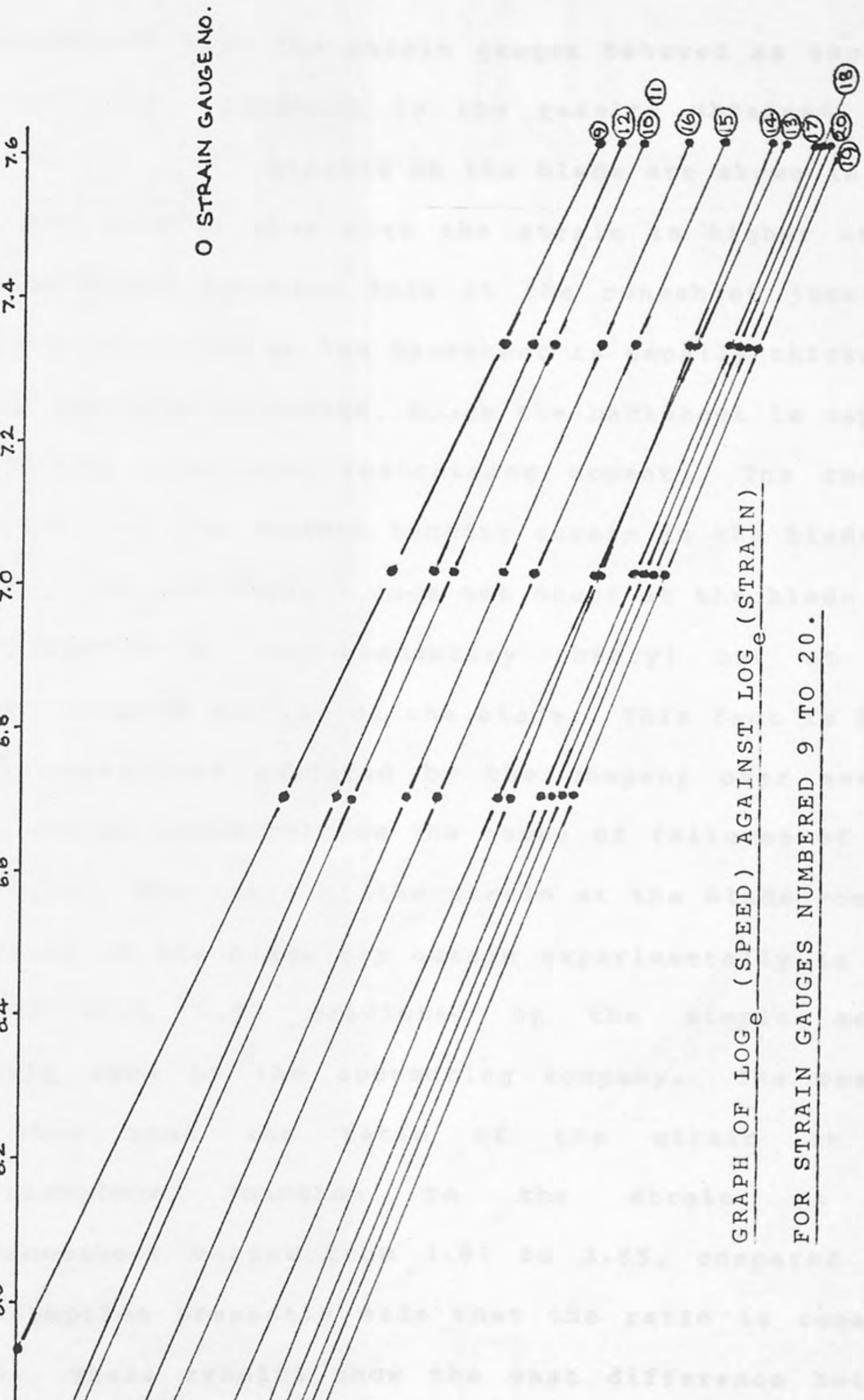

GRAPH OF LOG_e (SPEED) AGAINST LOG_e(STRAIN)
FOR STRAIN GAUGES NUMBERED 1 TO 8.
O STRAIN GAUGE NO.
LOG_e (ε)
LOG_e (R.P.M)
5.8
6.0
6.2
6.4
6.6
6.8
7.0
7.2
7.4
7.6
1
2
3
4
5
6
7

Fig 7.4

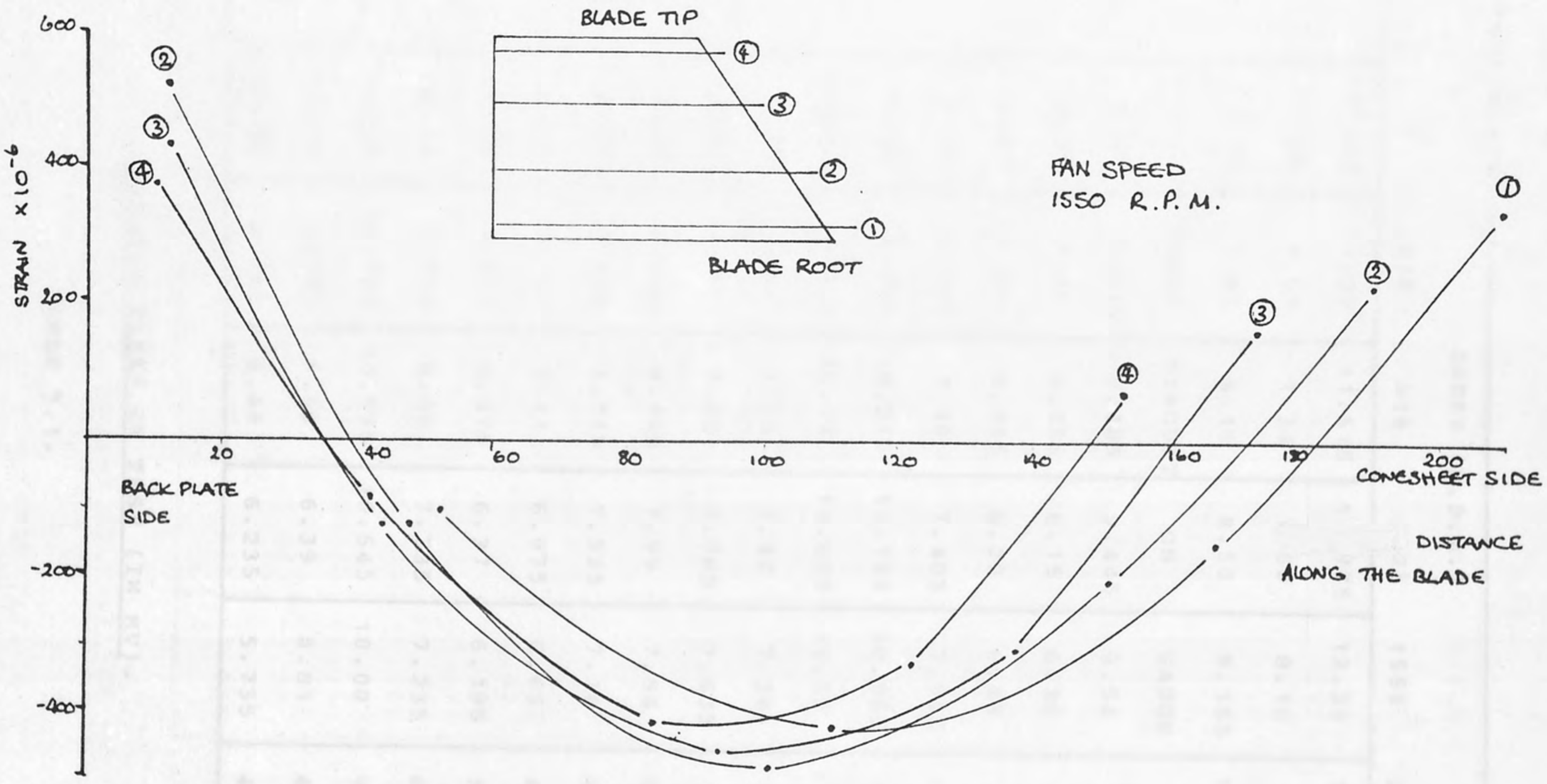


GRAPH OF $\log_e$ (SPEED) AGAINST $\log_e$ (STRAIN)

FOR STRAIN GAUGES NUMBERED 9 TO 20.

fig 7.5

results showed that the strain gauges behaved as expected and thus gave confidence in the results obtained. The measured strains on the blade are shown in Fig. 7.6. The results show that the strain is higher at the backsheet/blade junction than at the conesheet junction. This is predictable as the backsheet is usually thicker in section than the conesheet, hence the backsheet is capable of applying a greater restraining moment. The results also show that the maximum bending strain in the blade (at the centre of the blade) does not occur at the blade root (as predicted by the elementary theory) but at some distance towards the tip of the blade. This fact is borne out by experience collated by the company over several years, whilst investigating the cause of failures of fans in service. The ratio of the strain at the blade root to the strain at the blade tip obtain experimentally is 1.07 compared with 1.92 predicted by the simple method presently used by the sponsoring company. The results also show that the ratio of the strain at the blade/backplate junction to the strain at the blade/conesheet varies from 1.61 to 3.65, compared with the assumption presently made that the ratio is constant at 1.0. These results show the vast difference between the experimental results and those predicted by the simple theory used presently, and shows why large factors of safety have to be used to accomodate uncertainty in the analysis and why a more modern approach is necessary.



MEASURED STRAINS ALONG THE WIDTH OF THE BLADE

Fig 7.6

GAUGE NO.	SPEED (R.P.M.)					
	0	310	818	1125	1550	2050 RPM
1	11.28	11.29	11.515	11.775	12.25	12.92
2	7.05	7.09	7.34	7.62	8.16	8.98
3	7.84	7.85	8.16	8.50	9.165	10.195
4		SHORT	CIRCUIT	ON	GAUGE	
5	9.35	9.315	9.405	9.445	9.54	9.70
6	5.93	5.96	6.055	6.15	6.36	6.69
7	5.89	5.91	6.045	6.20	6.48	6.94
8	7.08	7.105	7.30	7.485	7.955	8.575
9	10.26	10.255	10.215	10.135	10.055	9.935
10	13.82	13.815	13.74	13.665	13.52	13.29
11	7.58	7.57	7.50	7.42	7.28	7.045
12	7.89	7.89	7.82	7.765	7.635	7.43
13	8.36	8.335	8.155	7.95	7.54	6.96
14	7.94	7.895	7.715	7.525	7.16	6.59
15	7.17	7.18	7.11	6.975	6.65	6.275
16	6.58	6.565	6.475	6.37	6.195	5.915
17	8.28	8.245	8.00	7.735	7.235	6.475
18	11.15	11.125	10.835	10.545	10.00	9.13
19	7.03	6.995	6.69	6.39	5.81	4.895
20	6.79	6.735	6.49	6.235	5.735	4.915

READINGS TAKEN ON TEST (IN MV).

TABLE 7.1.

GAUGE NO.	SPEED (R.P.M.)				
	310	818	1125	1550	2050
1	0.01	0.235	0.495	0.97	1.64
2	0.04	0.29	0.57	1.11	1.93
3	0.01	0.32	0.66	1.325	2.355
4	SHORT	CIRCUIT	ON	GAUGE	
5	0.025	0.055	0.095	0.19	0.35
6	0.03	0.125	0.22	0.42	0.76
7	0.02	0.155	0.31	0.59	1.05
8	0.025	0.22	0.405	0.875	1.495
9	-0.005	-0.045	-0.125	-0.205	-0.325
10	-0.005	-0.08	-0.165	-0.30	-0.53
11	-0.01	-0.08	-0.16	-0.30	-0.535
12	0	-0.07	-0.125	-0.255	-0.46
13	-0.025	-0.205	-0.41	-0.82	-1.40
14	-0.045	-0.225	-0.415	-0.78	-1.35
15	0.01	-0.06	-0.195	-0.52	-0.895
16	-0.015	-0.105	-0.21	-0.385	-0.665
17	-0.035	-0.28	-0.545	-1.405	-1.805
18	-0.025	-0.315	-0.605	-1.15	-2.02
19	-0.035	-0.34	-0.64	-1.22	-2.135
20	-0.055	-0.30	-0.555	-1.055	-1.875

CHANGES IN VOLTAGES (MV)

TABLE 7.2.

GAUGE NO.	SPEED (R.P.M.)				
	310	818	1125	1550	2050
1	3.94	92.59	195.0	382.2	646.2
2	15.8	114.3	224.6	437.3	760.4
3	3.94	126.1	260.0	522.1	927.9
4	SHORT	CIRCUIT	ON	GAUGE	
5	9.6	21.67	37.4	74.9	137.9
6	11.8	49.3	86.7	165.5	299.4
7	7.9	61.1	122.1	232.5	413.7
8	9.9	86.7	159.6	344.6	589.0
9	-2.0	-17.7	-49.3	-80.8	-128.1
10	-2.0	-31.5	-65.0	-118.2	-208.8
11	-3.94	-31.5	-63.0	-118.2	-210.8
12	-0.0	-27.6	-49.3	-100.5	-181.2
13	-9.9	-80.8	-161.5	-323.1	-551.6
14	-17.7	-88.7	-163.5	-307.3	-531.9
15	-3.94	-23.6	-76.8	-204.9	-352.6
16	-5.9	-41.4	-82.7	-151.7	-262.0
17	-13.8	-110.3	-214.7	-411.8	-711.2
18	-9.9	-124.1	-238.4	-453.1	-795.9
19	-13.8	-134.0	-252.2	-480.7	-841.2
20	-21.7	-118.2	-218.6	-415.7	-738.8

STRAINS $X10^{-6}$

TABLE 7.3.

GAUGE NO	LOGe (SPEED (R.P.M.)) 5.74	6.71	7.03	7.35	7.63
1	1.37	4.53	5.27	5.95	6.47
2	2.76	4.74	5.41	6.08	6.63
3	1.37	4.84	5.56	6.26	6.83
4	SHORT	CIRCUIT	ON	GAUGE	
5	2.26	3.08	3.62	4.32	4.93
6	2.47	6.20	4.46	5.11	5.70
7	2.07	4.11	4.80	5.45	6.03
8	2.29	4.46	5.07	5.84	6.38
9	-0.69	-2.87	-3.89	-4.39	-4.85
10	-0.69	-3.45	-4.17	-4.77	-5.34
11	-1.37	-3.45	-4.14	-4.77	-5.35
12	$-\infty$	-3.32	-3.90	-4.61	-5.20
13	-2.29	-4.39	-5.08	-5.78	-6.31
14	-2.87	-4.49	-5.10	-5.73	-6.28
15	-1.37	-3.16	-4.34	-5.32	-5.87
16	-1.77	-3.72	-4.42	-5.02	-5.57
17	-2.62	-4.70	-5.37	-6.02	-6.57
18	-2.29	-4.82	-5.47	-6.12	-6.68
19	-2.62	-4.90	-5.53	-6.18	-6.73
20	-3.08	-4.77	-5.39	-6.03	-6.61

LOG_e(STRAINS X10^6).

TABLE 7.4.

CHAPTER EIGHT

DISCUSSION ON THE STRESS ANALYSIS RESULTS

INTRODUCTION

Centrifugal fans have been designed and used for many years. The reason that these fans have been designed for so long without apparent failures is that large safety factors were incorporated in the approximate design procedures used. However, due to more stringent demands now being placed on centrifugal fans, in performance and economy of materials, the design stresses have steadily increased to such an extent that the once pessimistic design stresses are approaching the values which actually do exist in these structures in practice. The need for more sophisticated theoretical methods of stress analysis for these impellers is therefore necessary.

The dilemma of the designer of centrifugal fans is to reach a compromise between designing a component for infinite life, which would result in a structure which is extremely heavy, and designing for a specified finite life. In designing a centrifugal fan for a finite life it is essential that the overall "static" stress (or strain) distribution due to rotation is known accurately, before the effects of stress concentrations, weld fatigue, and extreme temperature are considered in the design.

This aspect of the project has therefore been concerned with the investigation of the stresses in a centrifugal fan impeller caused by rotation. In this chapter the finite element results obtained for the simple models are discussed first. The simple models gave siginficant insight into the strain distribution in the various components of the impeller and showed that a more refined finite element model of the impeller would be justified. The finite element results obtained for the blades of the actual impeller are compared with those obtained experimentally (discussed in Chapter 7). Finally the finite element results obtained for the conesheet and the backplate are briefly discussed.

8.1. DISCUSSION ON THE RESULTS FOR THE SIMPLIFIED MODELS OF THE IMPELLER (RADIAL AND INCLINED BLADED) WITHOUT A CONESHEET

The results obtained for the simplified models of the centrifugal fan impeller provided useful insight into the complex stress distribution in the impeller. In these models, the impeller consisted only of the backplate and blades (radially or inclined) and the results showed that as the blade length is increased at the blade root, whilst keeping the length of the blade at the tip constant, the strain in both the backplate and blades increases due to the increased loading. When the blade is in the radial direction, the strain in the blades is purely membrane, since there is no loading normal to the blade surface. When the blade is inclined at some angle, since, there is now a significant component of inertia loading normal to the blade surface, bending stress become dominant in the blade. In both cases, the maximum stress occurs at the blade/backplate junction and rapidly decreases to zero at the free end.

In the backplate, irrespective of the inclination of the blade, bending action is dominant, and the membrane stresses can be neglected in comparison. The bending stresses in the backplate increase but there is insignificant change in the membrane strains as the blade length is increased. The strains are higher in the models where the blade is inclined at an angle, compared with the radially bladed models. In both types, as expected, the area of high stress is in the region of the blade junction.

8.2. DISCUSSION ON THE RESULTS FOR THE SIMPLIFIED MODELS OF THE IMPELLER (RADIAL AND INCLINED BLADED) WITH A CONESHEET

In these models, the impeller consisted of a backplate, blades (radially or inclined) and a conesheet. The results, when compared with the models without a conesheet, show the expected considerable influence the conesheet has on the stress distribution in the other two components. In the backplate there is a significant reduction in the strains in the area most influenced by the blade, due to the fact that some of the load is now carried by the conesheet. Due to the conesheet being thinner than the backplate, it is the most highly stressed component of the three. In these models, it is interesting to note, that as the conesheet angle is increased, the strains in all the three components first increase, but then decrease. Again, the area of high strain in the backplate and conesheet is in the region of the blade junction.

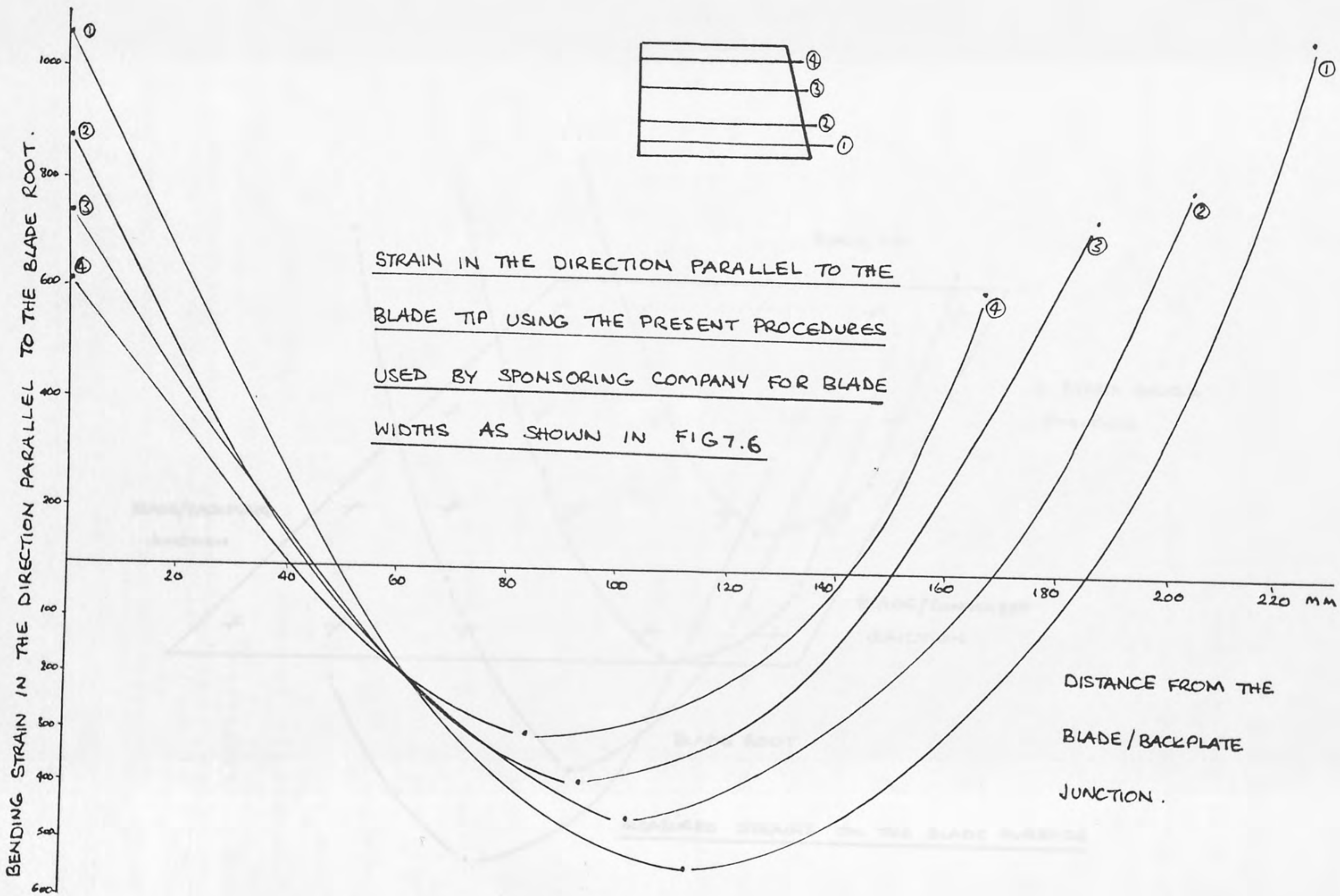
The model of the impeller with the blade (type No. 3) inclined at an angle, is the closest representation of a common actual impeller. The strain distribution in the blade (Fig. 6.27 and Fig. 6.28) when compared with the measured strains in the actual impeller, show that the finite element results are much lower. This is due to the fact an assumption of the model is that there are four blades in the impeller, whereas actually there are ten blades. The results show the considerable influence the blades have not only on the backplate and conesheet but on each other as well. This would be high-lighted even more if the blades overlapped each other, which does not occur in the impeller used in this work, but can occur in practice where there is a large number of blades in the impeller.

It was hoped to examine the change in the stress distribution in the impeller using these simple models, by varying the thickness of each component in turn. However, due to insufficient time, this was not possible. The results obtained, however, show that it is very difficult to predict the change in the stress distribution in the impeller when the geometry of one of the components is changed. Considerable benefit could occur from a more detailed study of parameter design changes, possibly leading to design codes. However, to reduce computing costs, refinements in the program to exploit sectorial symmetry would be necessary.

8.3. COMPARISON OF THE NUMERICAL AND EXPERIMENTAL RESULTS FOR THE ACTUAL IMPELLER

The procedures used in the stress analysis of impeller presently used by the sponsoring company has already been described in Chapter 2. Using these procedures for the impeller used in this work, the strain distribution in the blade is shown in Fig. 8.1. It shows that the strain at the blade/backplate and blade/conesheet junction is the same and the strain at the centre of the blade is half the value of the strain at the ends of the blade. The experimental results obtained (Fig.7.6), however, show that the strain at blade/backplate junction is higher than at the blade/conesheet junction, indicating that the backplate applies a greater restraining moment than the flexible conesheet. The strain gauge results also show that the strain increases from the blade tip and reaches a maximum value some distance from the blade root and then decreases, a fact borne out by experience gained by the company whilst investigating failures of impellers in service over a considerable period. An important fact not high-lighted by the method currently used.

As the Gauss points of the elements in the mesh used for the blade analysis, did not coincide with the positions of the strain gauges on the blade, a plot of the strains measured over the blade surface (Fig. 8.2) was produced,



STRAIN IN THE DIRECTION PARALLEL TO THE BLADE TIP USING THE PRESENT PROCEDURES USED BY SPONSORING COMPANY FOR BLADE WIDTHS AS SHOWN IN FIG7.6
BENDING STRAIN IN THE DIRECTION PARALLEL TO THE BLADE ROOT.
DISTANCE FROM THE BLADE/BACKPLATE JUNCTION.
1000
800
600
400
200
100
200
300
400
500
600
20
40
60
80
100
120
140
160
180
200
220 MM
①
②
③
④

fig 8.1

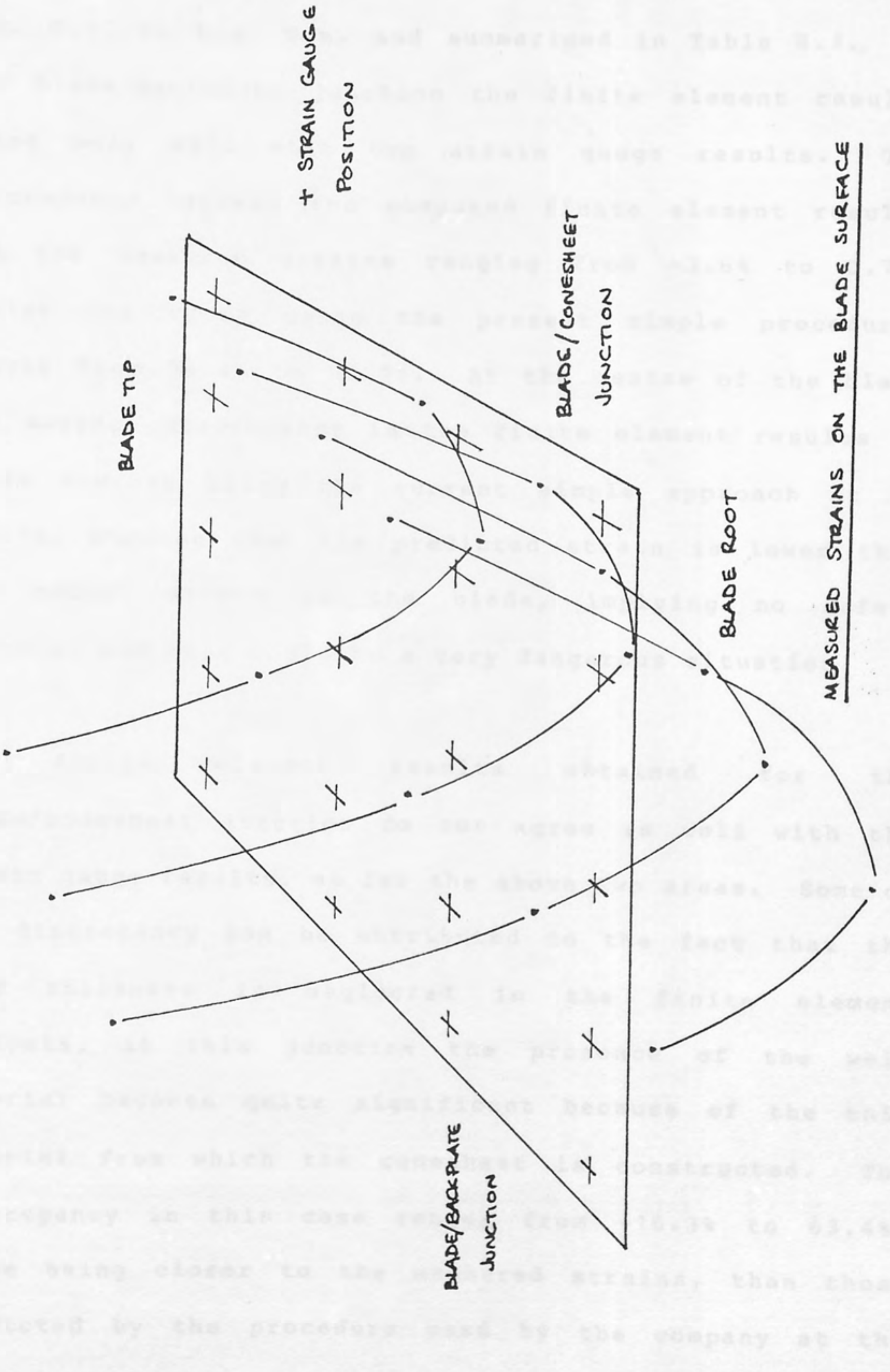

+ STRAIN GAUGE POSITION
BLADE TIP
BLADE/CONESHEET JUNCTION
BLADE ROOT
MEASURED STRAINS ON THE BLADE SURFACE
BLADE/BACKPLATE JUNCTION

Fig 8.2

and the corresponding strain at the Gauss points of the elements interpolated. The results obtained are shown in Fig. 8.3. to Fig. 8.6. and summarised in Table 8.1. At the blade/backplate junction the finite element results agree very well with the strain gauge results. The discrepancy between the computed finite element results and the measured strains ranging from -3.6% to 0.7%, whilst the error using the present simple procedures ranges from 56.4% to 80.0%. At the centre of the blade the maximum discrepancy in the finite element results is 22.4% whereas using the current simple approach it is -35.1%, showing that the predicted strain is lower than the actual strain in the blade, implying no safety factors, and thus could be a very dangerous situation.

The finite element results obtained for the blade/conesheet junction do not agree as well with the strain gauge results, as for the above two areas. Some of the discrepancy can be attributed to the fact that the weld thickness is neglected in the finite element analysis, at this junction the presence of the weld material becomes quite significant because of the thin material from which the conesheet is constructed. The discrepancy in this case ranges from -16.3% to 63.4%, these being closer to the measured strains, than those predicted by the procedure used by the company at the present, where the errors range from 190.3% to 543.5%.

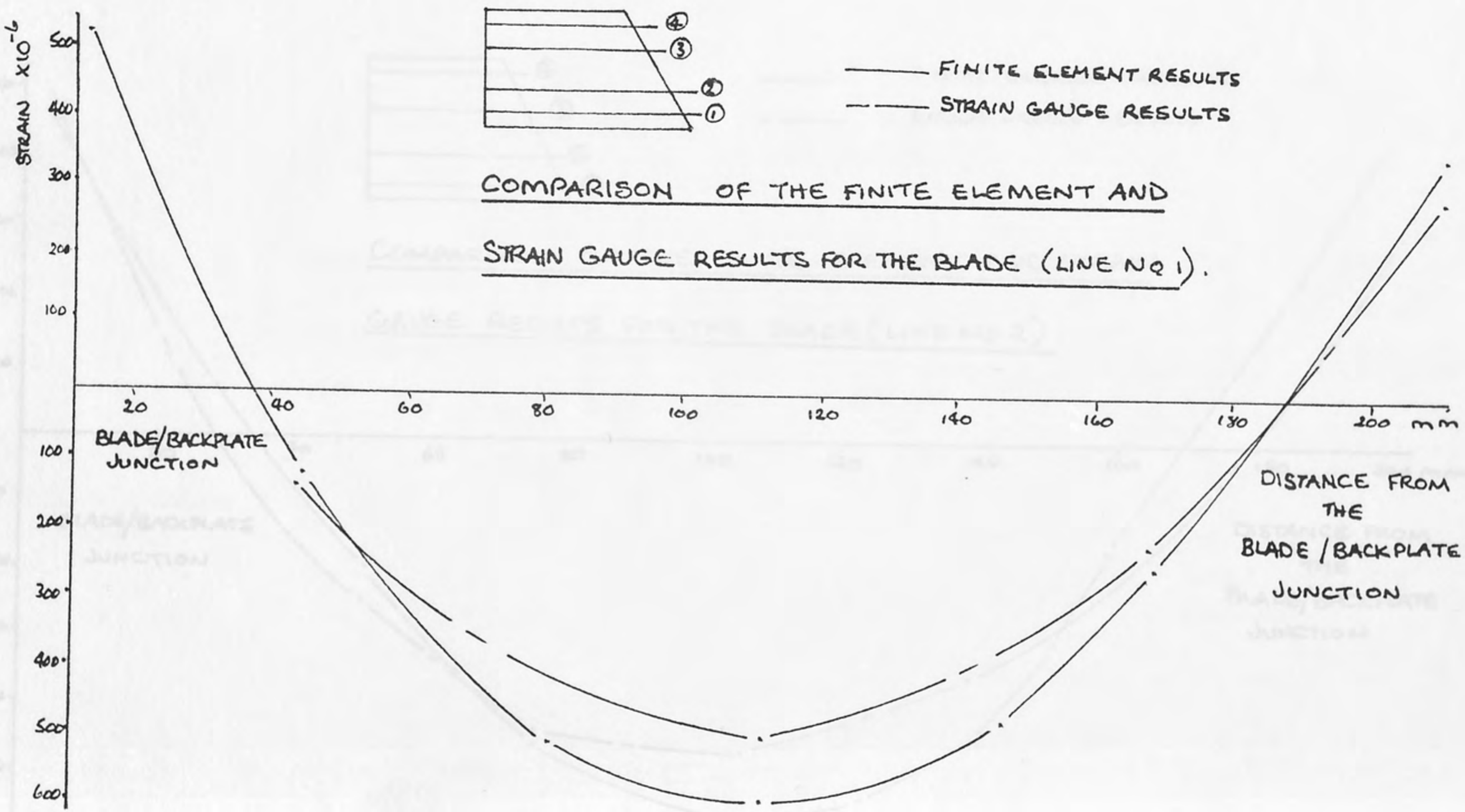
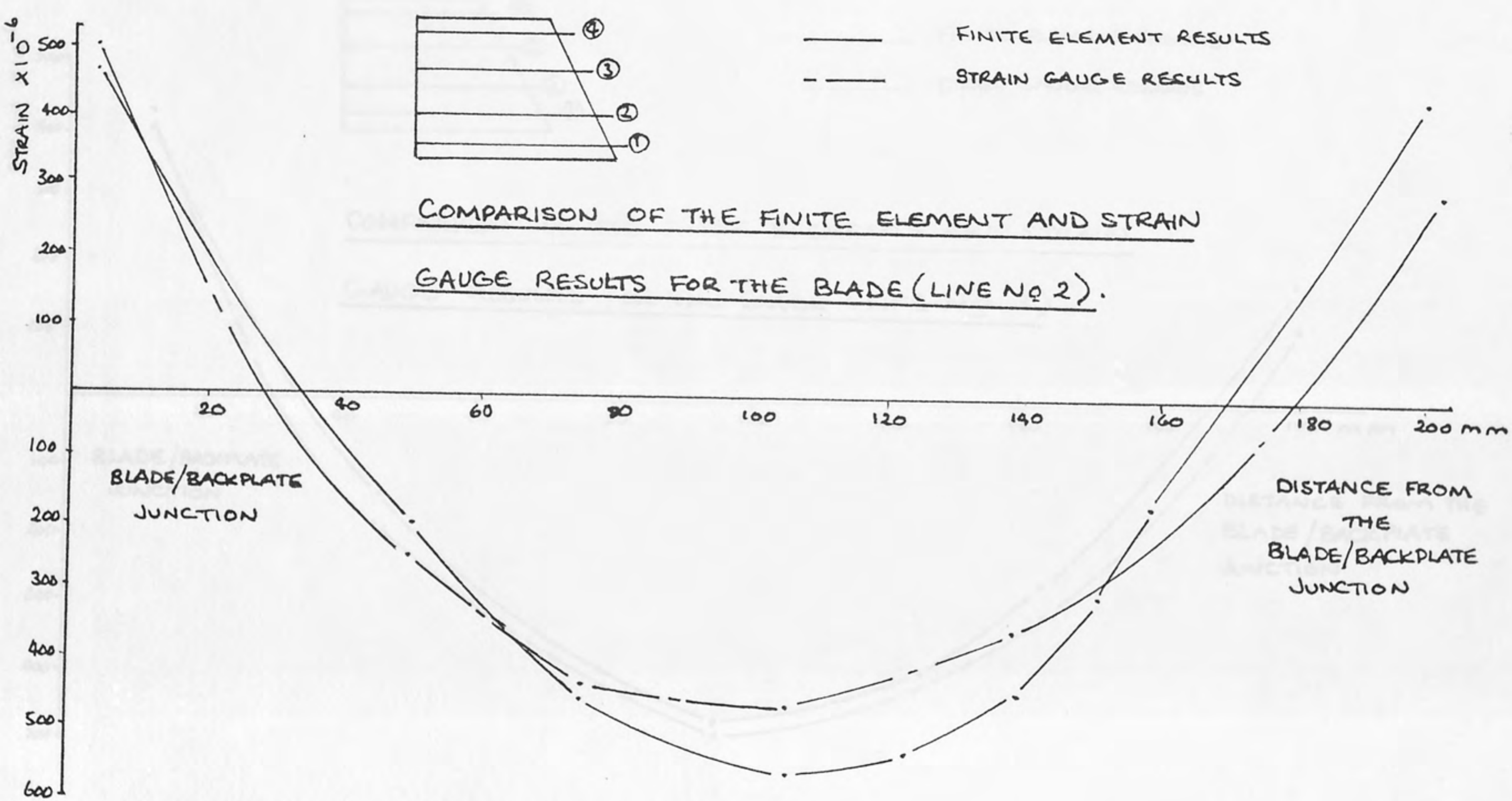

STRAIN X10⁻⁶
500
400
300
200
100
100
200
300
400
500
600
20
40
60
80
100
120
140
160
180
200 mm
BLADE/BACKPLATE JUNCTION
DISTANCE FROM THE BLADE / BACKPLATE JUNCTION
4
3
2
1
FINITE ELEMENT RESULTS
STRAIN GAUGE RESULTS
COMPARISON OF THE FINITE ELEMENT AND
STRAIN GAUGE RESULTS FOR THE BLADE (LINE No 1).

fig 8.3



STRAIN x10^-6
500
400
300
200
100
100
200
300
400
500
600
20
40
60
80
100
120
140
160
180
200 mm
① ② ③ ④
FINITE ELEMENT RESULTS
STRAIN GAUGE RESULTS
COMPARISON OF THE FINITE ELEMENT AND STRAIN
GAUGE RESULTS FOR THE BLADE (LINE No 2).
BLADE/BACKPLATE JUNCTION
DISTANCE FROM THE BLADE/BACKPLATE JUNCTION

Fig 8.4

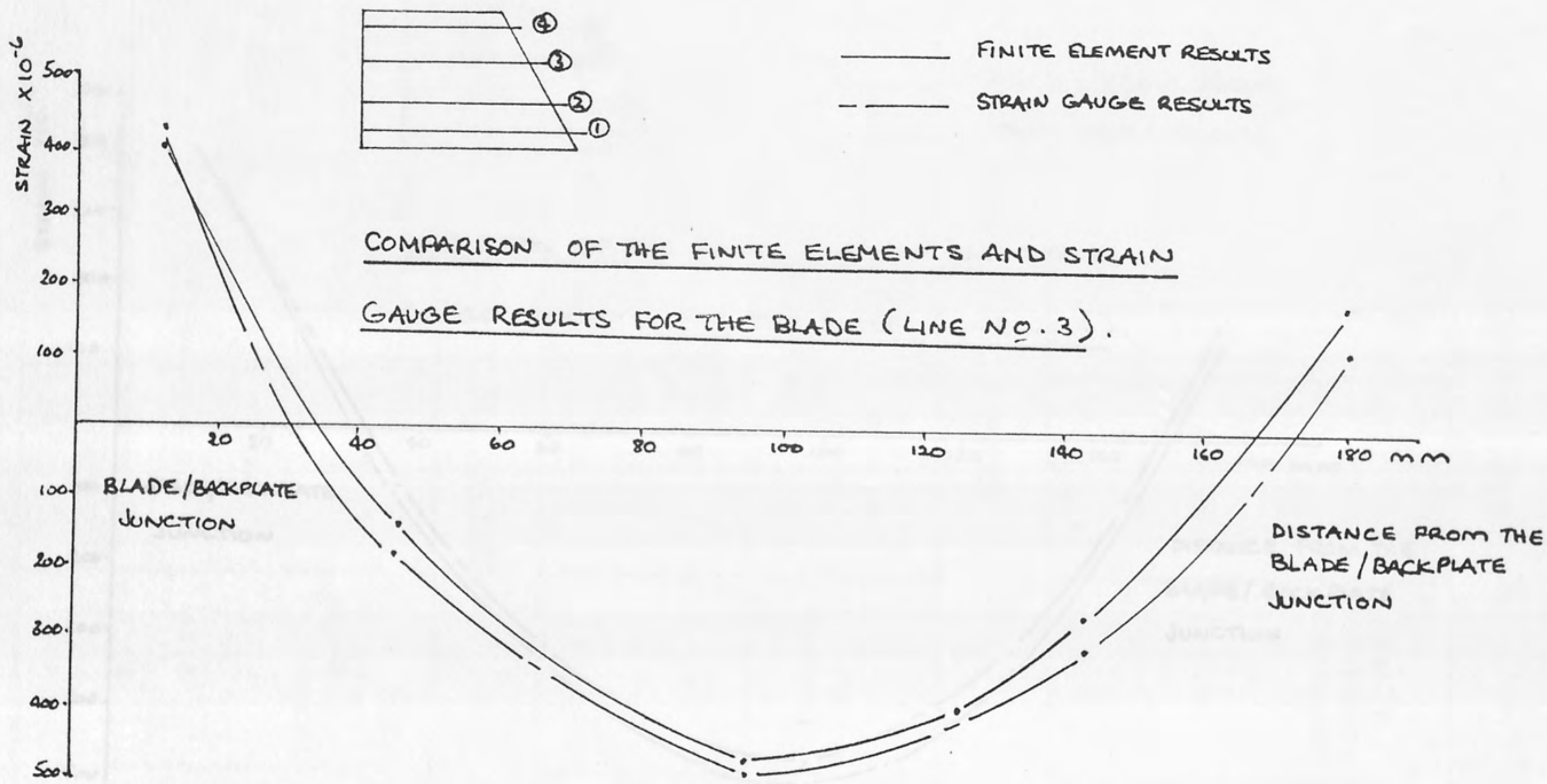
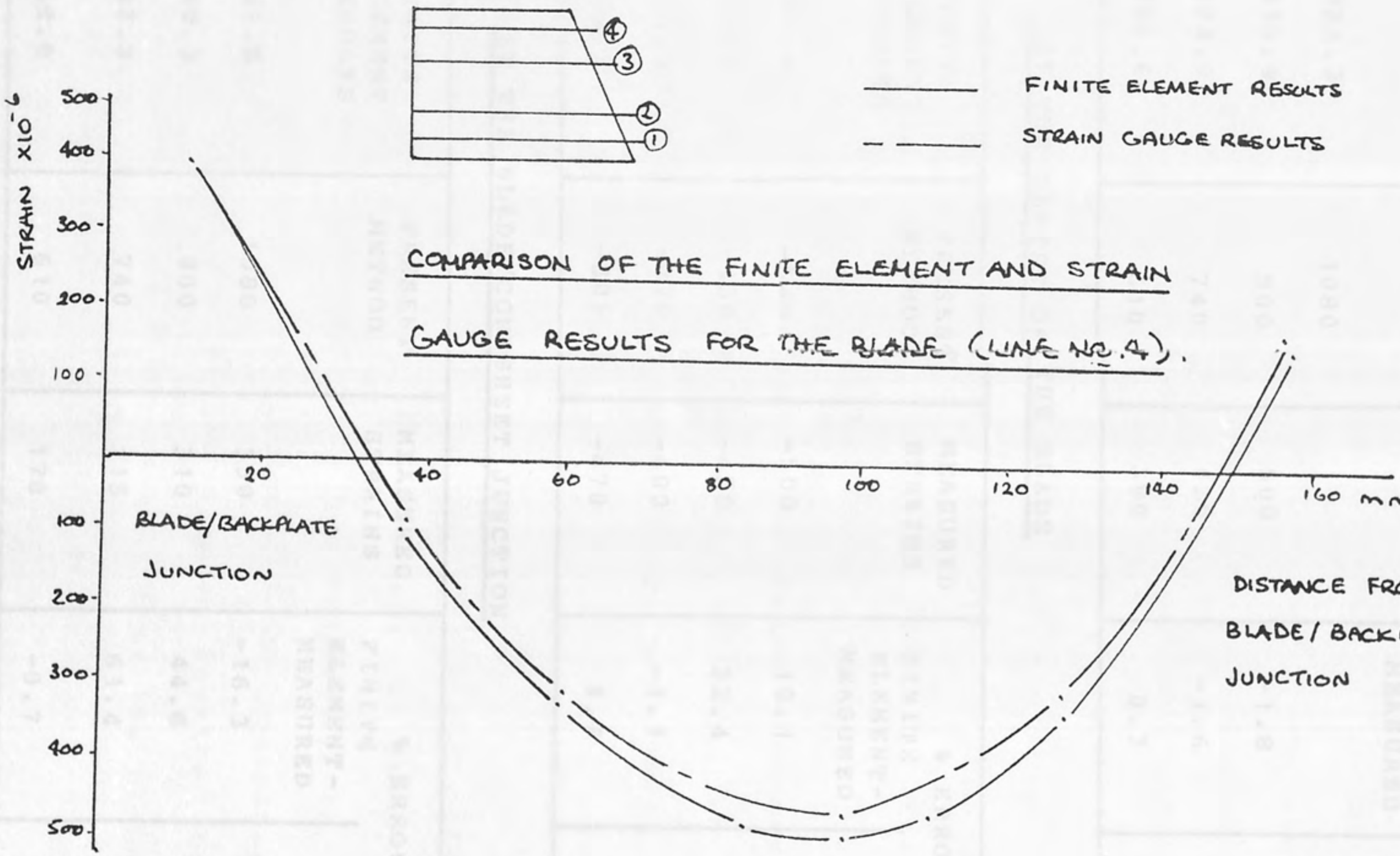

COMPARISON OF THE FINITE ELEMENTS AND STRAIN
GAUGE RESULTS FOR THE BLADE (LINE NO. 3).
FINITE ELEMENT RESULTS
STRAIN GAUGE RESULTS
STRAIN X10^{-6}
500
400
300
200
100
100
200
300
400
500
20
40
60
80
100
120
140
160
180 mm
BLADE/BACKPLATE
JUNCTION
DISTANCE FROM THE
BLADE/BACKPLATE
JUNCTION
1
2
3
4

Fig 8.5



COMPARISON OF THE FINITE ELEMENT AND STRAIN GAUGE RESULTS FOR ~~THE BLADE~~ (LINE NO. 4).

Fig 8.6

AT THE BLADE/BACKPLATE JUNCTION

LINE NO.	FINITE ELEMENT RESULTS	PRESENT METHOD	MEASURED STRAINS	% ERROR FINITE ELEMENT-MEASURED	PRESENT METHOD-MEASURED
1	524.2	1080			
2	490.9	900	500	-1.8	80.0
3	404.9	740	420	-3.6	76.2
4	392.6	610	390	0.7	56.4

AT THE CENTRE OF THE BLADE

LINE NO.	FINITE ELEMENT RESULTS	PRESENT METHOD	MEASURED STRAINS	% ERROR FINITE ELEMENT-MEASURED	PRESENT METHOD-MEASURED
1	-595.4	-540	-500	19.1	8.0
2	-550.6	-450	-450	22.4	0.
3	-474.5	-370	-480	-1.1	-22.9
4	-492.2	-305	-470	4.7	-35.1

AT THE BLADE/CONESHEET JUNCTION

LINE NO	FINITE ELEMENT RESULTS	PRESENT METHOD	MEASURED STRAINS	% ERROR FINITE ELEMENT-MEASURED	PRESENT METHOD-MEASURED
1	293.0	1080	350	-16.3	208.6
2	448.3	900	310	44.6	190.3
3	187.9	740	115	63.4	543.5
4	168.8	610	170	-0.7	258.8

ALL STRAINS = STRAINS x 10^{-6}

SUMMARY OF THE STRESS ANALYSIS RESULTS

TABLE 8.1.

The above discussion show the considerable variance in the computed results, using the present method adopted by the company compared with those obtained experimentally and why a more sophisticated approach is needed in the stress analysis of these structures. In some areas, the structure is highly over designed, whilst in others it is under designed. In view of the coarse mesh used in the finite element analysis of the blades, the results obtained are very good. Due to the variation in the strain at the three sections of the blade considered, for practical design and manufacturing purposes, the blade needs to be of constant thickness. In this case the maximum strain will determine the blade thickness, and this occurs at the blade/backplate junction, where the finite element results agree very well with those measured.

As it was not possible (due to insufficient time) to strain gauge the backplate and conesheet, no experimental results are available for these two components to assess the accuracy of the finite element and the presently used procedure results. The finite element results obtained for the backplate, some distance away from the blades are symmetrical, as expected due to geometrical symmetry. Even at this distance bending is dominant, showing the influence of blade action on the backplate, and this increases as the radius increases. In the vicinity of the blades, the strain distribution becomes more complex, the areas of high stress being in the area of the blade junction with the maximum occuring below the junction at the blade root. Another reason for assuming confidence in the numerical results obtained, is the fact that for the same points in sectors of adjacent blades the strains are approximately the same (see Fig. 6.38 to Fig.6.41).

The results obtained for the conesheet, shows the stress distribution in this component is even more complex than in the backplate. The area of high strain is in the region of the blade junction, again showing the great affect the blade has on the stress distribution in this component. The results show that there are regions in which bending action is dominant and others where membrane action is dominant. Superimposing the membrane and bending strains (Fig. 6.42) shows that bending action is dominant around the blade junction and at the inner radius of the conesheet, while at the outer radius, membrane action is dominant. The finite element results cannot even be compared with the results given by the procedure presently used by the company, because it does not provide such detailed information. In the simple method the approach is to regard the backplate and conesheet as a rotating disc. The centrifugal force due to the total mass of the blades acting at the mean of the two radii is calculated, and it is assumed that this inertia load is shared equally between the backplate and the conesheet. The hoop stress due to this load in each component is calculated and superimposed onto that due to rotation. In this type of analysis no account is taken of the fact that the load due the blades induces large bending stresses in the two components, as shown by the numerical results. Before summarising the results of stress analysis of the centrifugal impeller, it is worth pointing out the problem of round off errors experienced due the limitation on the size of the computer memory available.

Due to the complex interaction of the components of the impeller with each other, shown by the simple models, it became obvious that it would be very difficult to treat the three components separately by applying "suitable" boundary conditions on each component. The three components of the impeller had to be treated as an integral unit. A major disadvantage is that in the computer program developed, sectorial symmetry cannot be exploited, which is the reason why such a large proportion of the impeller had to be analysed. This also had the disadvantage that a fine mesh could not be used, due to the limitation in the computer memory (120 K bytes) available to each user.

With the memory limitation imposed, and the size of the problem considered (150 elements) errors as a result of round off, due to the number of significant figures retained by computer during calculations, and the number of operations performed on them, round off errors became a major problem, with the results becoming progressively worse. This was confirmed by the out-put of diagnostic error message number 60 (see Appendix No. 4) at the end of the results. To futher confirm that the deterioration in the results was due to round off errors, and not due to some error in generating the load vector due to the centrifugal force loading on the structure, the modification carried out (mentioned in chapter 5) was checked, and no errors were found. A very coarse mesh of half the impeller using 35 elements was analysed. The results for this analysis were comparable with those obtained for the simple inclined bladed models of the impeller and the error message was not printed in this case, confirming that round off errors had not affected the results. The results due to round off errors can be improved by using more precision in the number of significant figures retained for calculations, but this would require more main memory which was not available.

The problem was resolved by using an advantage of the Frontal solution technique: that of the ordering of the elements being important. The finite element results for the impeller was obtained in two attempts. In the first case, the element numbers were arranged such that the deflections and strains in the blades were determined before the backplate and conesheet. In the second case, using the same mesh, the element numbers were re-arranged so that the results for the conesheet and backplate were obtained before the blades. Solving the problem in this way gave good results for the three components before errors due to round off became a major concern. It was also hoped to calculate the residual error in the force vector by calculating it, using the computed deflections and comparing it with the original force vector used to calculate the deflections. However, this again required extra memory which was not available.

It is worth mentioning that this problem of round off errors will be of no concern when the computer program is implemented on the Prime computer at the sponsoring company. This is because the Prime computer operates on the "virtual" memory principle, which means that programs requiring up to 32 M bytes of main memory can be executed, irrespective of the size of the actual main memory of the computer(which in this case is 1/2 M bytes). Double precision, with an accuracy of 64 significant figures is available, so that analysing the stress distribution in the impeller, using an even finer mesh (with double precision accuracy) could be analysed, without having to be concerned about round off errors.

8.4. CONCLUSIONS ON THE STRESS ANALYSIS OF THE IMPELLER

Although it was not possible (due to lack of time) to carry out an extensive stress analysis of the impeller using strain gauges as originally intended. The limited experimental work carried out in this research has provided the company with a valuable insight into the true stress distribution in the blade, which it did not previously enjoy. In particular the experimental results obtained show the vast difference in the actual strain in the blade compared with those predicted by the simple procedures currently used.

The finite element results obtained agreed well with the strain gauge values. Although, the results obtained for the backplate and conesheet cannot be compared with experimentally produced data as in the case of the blades, the results obtained are as expected. In conclusion, it can be said that this aspect of the project has achieved its original objectives, that of providing more detailed and accurate information on the stress distribution in the centrifugal impeller, than was currently available. However, as shown by the simple models of the impeller, no "hard or fast" rule or trend can be used when moving from one design to another. For example the effect of a change in one more of the dimensions of a component, would have on the overall stress distribution in the impeller assembly is not obvious because of the complex interaction between the conesheet, blades and the backplate. Therefore, it would appear that there is a case for a detailed study of parameteric design changes, which would show the effects of any such modifications before the production stage, thus eliminating the need for extensive experimental tests which are costly in money, man power and time.

CHAPTER NINE

EVALUATION OF THE PROJECT

INTRODUCTION

In this chapter an evaluation of the project and a discussion of the benefits to the sponsoring company are discussed. Finally a list of recommendations for further work is provided.

9.1. EVALUATION OF THE PROJECT

There is a considerable amount of literature about the initial stages of innovative design, whereby a completely new product is conceived, and developed. In industry, however, the continuing success of a company is more often achieved by improving or developing existing designs to maintain their marketability, a process known as evolutionary design. Evolutionary design changes, by definition do not have a major impact on the product, but that is not to say that they should not be made. In fact such changes maintain the marketability of the product, thereby generating resources for further research into evolutionary or innovative designs. This research has been concerned with the evolutionary design process as applied to centrifugal fan impellers, and the benefits to the sponsoring company as a result of it are mainly long term. This is why no attempt will be made in the following discussion to quantify them, because of the difficulty in predicting accurately the factors which have a significant bearing on the final results over a long period.

Centrifugal fans have been designed and used for many years. These fans have been designed adequately for so long because the design method (discussed in Chapter 2) applies large safety factors in very approximate design procedures which over-estimate the actual stresses in the impeller. Today, more accurate methods of stress analysis of engineering components have become necessary due to demands for higher performance, economy of material, reduction of weight and improved standards of safety. More sophisticated theoretical methods of analysis for the impeller are therefore necessary.

The sponsoring company achieves sales by tendering to enquiries which it receives. The selection of a suitable type and size of unit(s) to meet the customer's requirement involves a certain amount of tedious, time consuming calculations, resulting in a long time lapse between receipt of the equiry and the response to it. For the company to be successful in tendering to the enquiry, not only are the selection of a suitable type and size(s) and its mechanical design (stress analysis) important, so also is the time taken to respond. Selection and recommendation of a larger size than necessary could result in lost business as the capital and operating costs would be higher, whereas a smaller size than required would lead to a very short operating life for the unit, resulting in loss confidence in the company's product. Correct selection and recommendation of a suitable type and size of unit is thus very important.

The project has been concerned with the above two aspects of the evolutionary design process applied to the centrifugal fan impellers to maintain their marketability. The objectives of the project were therefore to:-

1). Develop a Computer Aided Selection system to enable the company to respond to the customer's enquiry quickly, utilising a "standard" range produced by the company.

2). Develop a computer program to analyse the stress distribution in the centrifugal fan impeller (due to centrifugal forces), thus giving more accurate and detailed information than is currently available.

The Computer Aided Selection system was developed with flexibility in mind, so that it could be implemented on whatever hardware acquired by the sponsoring company. This was acheived as shown, in that it was developed on a desk-top micro computer and implemented on a mini computer without any difficulty. The system was easy to use with very little (less than half an hour) training. The system was well received by the users as it relieved them from the routine and time consuming task of calculations, and left them with the more rewarding task of decision making in recommending a suitable size or sizes, thus satisfying an important criteria: that of "social acceptability".

In evaluating the performance of the Computer Aided Selection system, the overall enquiry response time before and after the implementation of the system was analysed. The reasons for such an evaluation was discussed in Section 7 of Chapter 4. It showed that for written enquiries there had been a significant improvement from 50.6% to 70.1% in the number of enquiries responded to in the first 5 working days of receiving them. In the case of verbal (telephone) enquiries the improvement was not that significant (from 96.0% to 98.1%) for the same period. An important point, however, not high-lighted by the evaluation in this case, was the fact that a large proportion of such enquiries were answered at the time of the enquiry, while the customer waited, with a written quotation following afterwards. This represented a substantial improvement in service provided compared with previously, where the customer's requirements were noted, and the customer informed of suitable size(s) later verbally with a written quotation following on.

Overall, there has been a significant improvement in the service provided to the customer, in that the selections are now discussed in detail with the customer, before recommending a suitable size(s), prior to this such discussion was not practicable in general and only occured in response to a customer's specific needs. The amount of information provided to the customer has also increased; now performance curves, sound spectra levels are provided as standard information in addition to engineering drawings giving details of the overall fan dimensions and the run-up time for the fan-motor combination. This improvement in service is not high-lighted by the type of evaluation carried out, but should be borne in mind when assessing the system, such improvements in service are difficult to quantify in the short term but they undoubtedly contribute to enhance business prospects. In conclusion, the limited but measurable improvement shown by the evaluation shows that the system will be of significant benefit to the company in the long term.

In the case of the stress analysis of the impeller, although it has not been possible to carry out as extensive a programme of strain gauging as desirable, the experimental data obtained for the blade, has been of considerable use to the company, as a majority of "in service failures" are due to blade failure. The experimental data obtained showed the vast difference in the stress distribution in the blade, compared with that predicted by the present simple procedure used by the company, and justified the need for a more sophisticated modern approach in the stress analysis of the impeller.

The results obtained for the blade analysis using the finite element method are very good at the blade/backplate junction and at the centre of the blade. The results at the blade/conesheet junction were however a little disappointing, but the results are closer to the measured strains than those predicted by the present method used by the company. Although the finite element results for the backplate and conesheet could not be compared against experimentally produced data, they were nevertheless as expected. The results obtained show that the choice of the Semi-Loof element as a suitable element in the stress analysis of the centrifugal fan impeller using the finite element technique was correct. However, the program requires futher refinements (discussed in section 9.3) before it could be used by the sponsoring company in the stress analysis of its fan range. The above discussion shows that the research work carried out has achieved the two original objectives of the project.

In conclusion, the overall result of this research has been the improvement of the stress analysis and selection procedures for centrifugal fan impellers, which will significantly help the company to retain/enhance their market share.

9.2. OTHER BENEFITS OF THE PROJECT

The benefits of Computer Aided approaches high-lighted has led the company considering computerisation of other aspects (finance, shop floor labour analysis, stock and production control etc) of its operations. The employees concerned also recognised the benefits of computerisation and did not oppose such moves by the company. As already mentioned in Chapter 4, the parent company noticing the benefits for the sponsoring company approved computerisation of another associate company within the Mitchell Cotts Engineering group, as a joint venture between the sponsoring company and the associate company. Since then, other companies within the Mitchell Cotts Engineering group have made moves towards computerisation of their activities. These benefits are a direct result of conducting the research work through the Interdisciplinary Higher Degree scheme. The research student works in close co-operation with sponsoring company and as the company operates it "real" time, the project has to be more responsive to changes in circumstances in the company due to factors within or outside its control. This has been high-lighted in this research, in the development and implementation of the Computer Aided Selection system, and the background to the decisions concerning the acquisition of the computer hardware discussed in Chapter 4. The Interdisciplinary Higher Degree scheme therefore, broadens the concept of traditional higher degree studies, which tend to restrict the researcher to a narrower but indepth type of research.

9.3. RECOMMENDATIONS FOR FURTHER WORK

As with any research project of this magnitude, there is usually insufficient time to accomplish everything that one would hope, also the results of the research may themselve reveal new areas which should be investigated. The recommendations for further work, in order that further benefits can be gained are, as follows:-

A). STRESS ANALYSIS OF THE IMPELLER

1). Measure the actual strain distribution in the backplate and conesheet using strain gauges, as originally intended.

2). Compare the strain gauge results obtained for the backplate and conesheet with those obtained using the finite element program developed in this research, to quantify the adequacy of the finite element results for these two components.

3). Modify the computer program, so that the geometrical property of sectorial symmetry can be exploited, to reduce the scale of the model that needs to be analysed, thus reducing the size of the computer main memory, and the processing time required.

4). To reduce the time taken to prepare the input data, and analyse the output produced, pre-processing of the input data (mesh generation) and post-processing of the output produced (stress contour plotting) subroutines are essential for the stress analysis of complicated structures such as centrifugal fan impellers. These subroutines will also remove the need for the user to know the finite element process before being able to use the program.

5). The program at present uses magnetic tapes for back-up storage, disc access is much faster and in common use these days. To cater for disc storage which is now available at the sponsoring company would require some program modification.

6). Carry out a detailed study of parameteric design changes using the simple models of the impeller, to see if any design codes can be established.

B. COMPUTER AIDED SELECTION SYSTEM

Futher develop the software (subroutines or programs) to:-

1). Prepare the tender to be sent to the customer in response to the initial enquiry, using the output produced by the fan selection program.

2). Calculate the density of a mixture of gases, at a specified temperature and pressure. Also to correct the density of the mixture to standard conditions (i.e. 20^{o}C and 760 mm Hg pressure) if necessary.

3). Fan selection limits to be over-ridden if necessary, so selections outside the "normal" limits can be made.

4). Calculation of the performance characteristics for a "non standard" size.

5). Plots of performance characteristics for a series of desired:-

a). Temperatures.

b). Speeds.

c). Densities.

6). Selection based on one of the following criteria:-

1). Smallest unit.

2). Lowest initial capital outlay.

3). Most efficient size.

7). Verification of the test data used a basis for the calculation of performance data for the sizes in the fan series using the fan laws. This is essential, as the original tests were carried out when the range was first developed, and there may have been minor design changes since which have resulted in changes in the performance of the fans.

APPENDIX Nº 1

C O S T B E N E F I T A N A L Y S I S

R E P O R T F O R T H E

C O M P U T E R A I D E D

S E L E C T I O N S Y S T E M

REPORT ON THE COST BENEFIT

ANALYSIS OF A COMPUTER AIDED SELECTION SYSTEM

FOR CENTRIFUGAL FANS.

OCTOBER 79

SUMMARY

Analysis of the number of enquiries received and the cost of operating the sales department for the year 1978 showed that the average cost of handling a sales enquiry for a centrifugal fan was approximately 17.50 pounds and that 889 enquiries received were not answered due to lack of time (this is equivalent to the workload of one and half extra persons). Analysis of the number of enquiries received per annum shows a gradual upward trend. To cater for this, extra staff can be taken on at a cost of 6,500 pounds per man, per annum (at 1978 figures), or finding a means of reducing the cost per enquiry, i.e. reducing the time spent on each enquiry, but still maintaining the quality of work necessary to generate more business.

Development of a computer aided selection system has shown that the average cost per enquiry could be reduced to approximately 10.30 pounds. Such a system can be implemented on a desk-top computer, such as the Hewlett Packard 9845 desk top machine.

RECOMMENDATION

The recommendation is therefore to acquire a Hewlett Packard 9845T desk top computer on which to implement the computer aided selection system software.

PRICE: 14,500 pounds plus VAT.

CONTENTS

APPENDICES

CHAPTER 1

DISCUSSION OF THE COST BENEFIT ANALYSIS

FOR THE PROPOSED COMPUTER AIDED

SELECTION SYSTEM

Analysis of the number of enquiries received and the cost of operating the technical sales department for the year 1978 showed that the average cost of handling a sales enquiry was approximately 17.50 pounds. Of the total enquiries received 889 (representing 18%) were not answered due to lack of time. This is equivalent to the work load of an extra one and half persons. There is a gradual upward trend in the number of enquiries received per annum (see Graph No. 1). Discussions with the technical sales staff, indicates that the prospective customer is requesting more information (e.g. fan performance curves, stress calculations, material specification, sound spectra levels, engineering drawings etc) than previously at the enquiry stage. This results in even more time being required to respond to each enquiry, leading to even less enquiries per man being handled. To cater for the increase in the number of enquiries received, and answer them all, there is the possibility of:-

a). Employing extra staff at a cost of 6,500 pounds per man per annum.

b). Finding a way of reducing the time spent on each enquiry, but still providing the prospective customer with all the information he requests.

The former has certain disadvantages (training, holidays, sickness, industrial relations etc) and therefore attention was given to the latter to finding a solution to the problem. The selection of a suitable size fan for a specific duty, involves a certain amount of routine but time consuming calculations. These could easily be processed by a digital computer. This method not only has the advantage of taking the laborious routine away from the individual, but also the time taken to obtain the results is reduced dramatically, and the accuracy improved. Computer aided selection seems an ideal way of reducing the time spent on the enquiry.

A computer aided selection system has been developed whereby the computer selects fan(s), to meet a specific duty within given selection parameters, from a range of standard sizes produced by the company. From the print-out of the suitable sizes, the technical salesman can recommend a particular size, bearing in mind the customer's requirements. A trial run of such a system was carried out in June 79, at Alldays, Peacock, using the Hewlett Packard 9845 desk-top computer system. The results showed that:-

a). Someone with no previous experience of using a desk-top computer selection system could use it efficiently within half an hour.

b). The average time taken to type in the necessary information and obtain a print-out of the suitable sizes was approximately five minutes. For an example of a typical computer print-out, refer to chapter 3 of this report.

c). Analysis of the print-out, produced by the computer aided selection system and other work necessary to answer the enquiry took approximately a further 45 minutes.

d). Even with this basic system, the average time taken to answer the enquiry was reduced from the present 2.9 hours to one hour. With further development of the program (e.g. fan-motor combination run-up times, producing the fan specification sheet etc.) the time taken could be reduced even further.

The advantages of a computer aided selection system are discussed in detail in chapter 3 of this report.

Having examined the advantages of computer aided selection, there are two ways in which the system could be implemented:-

1). Acquiring the necessary hardware.

2). Using a computer bureau service, i.e. storing the computer programs on their machine, and having access to the machine via a G.P.O. link between the computer and Alldays, Peacock & Co. Ltd. Cost analysis of the two methods (chapter 2 of this report) showed that the average cost per enquiry, using a bureau service would be approximately 12.60 pounds, compared with an average cost of approximately 10.30 pounds (including running and maintenance costs) by acquiring suitable hardware. The other disadvantages of implementing the system using the bureau service are discussed in Appendix No. 2 of this report. The main advantage with using the bureau service, is the comparatively small initial capital outlay, but this is out-weighted by the other disadvantages.

Having decided that the most suitable method of implementing the computer aided selection system was to acquire the hardware, a hardware specification to meet the requirements of the selection system was drawn up. A market survey of the desk-top computer range was then carried out to find the most suitable machine on which to implement the computer aided selection software. The survey included machines such as the Tektronix 4052, Tandy RS80, Apple, P.E.T. (by Commodore), Compucorp, Minic (by Digital Electrical Corporation), PCC 200 and Level 6 (by Honeywell) micro/mini computers.

The survey revealed the Hewlett Packard 9845T desk-top computer system as the one meeting the requirements of the hardware specification listed in Appendix 1 of this report.

CHAPTER 2

COST ANALYSIS

SECTION A :- SUMMARY OF COST ANALYSIS

SECTION B :- CALCULATION OF COSTINGS

METHOD OF SELECTION:-

	1).Fully Manual Selection	2).Computer Aided Selection using (1)Bought Hardware	2)Bureau Service
Average Costs (Pounds)			
Machine		2.24	3.58
Sales Labour Cost	10.87	2.80	3.75
Clerical Labour Cost	4.5	4.5	4.5
Telephone & Postage	0.80	0.80	0.80
Average Cost /Enquiry.	17.57	10.34	12.63
Typical range of cost per Enquiry	15 to 30	10 to 10.50	12.50 to 13.50
Typical annual cost (at 4000 Enquiries)	90,000	50,800	59,600
Cost of Implementation per annunm	20,400	8298	2979

B1. COST DETAILS: MANUAL SELECTION

Number of hours available /man year[i.e. 260 working days less holidays(20),Bank holidays (7), sickness (5) and 37 Hour Week		1687 HOURS
Number of men available for centrifugal fan selection		5 1/2 MEN
Total number of hours available for fan selection		9297 HOURS
Total number of centrifugal fan enquiries received in 1978		4051 ENQUIRIES
Less enquiries received but not answered in 1978		889 ENQUIRIES
Total number of enquiries answered in 1978		3162 ENQUIRIES
Average time taken to handle an enquiry	$\frac{9279}{3162}$ =	2.9 Hours
Total wages for technical sales sales office for 1978		76,000 POUNDS
Equivalent technical sales staff in sales office [ie Sales Manager =1.5 Technical sales man, Typing staff =0.6 Technical Salesman]		12 MEN
Average wage/annum of Technical salesman	=	6333 POUNDS
Average rate/hour for Techanical Salesman	$\frac{6333}{1687}$	3.75 POUNDS / HOUR
Average rate/hour for typing staff	= 0.6 x 3.75	3.25 POUNDS / HOUR
Telephone Cost for the sales office in 1978		6,000 POUNDS
Telephone Cost/Enquiry	$\frac{6000}{3162}$ =	1.90 POUNDS / ENQUIRY

Postage (say) 30p per enquiry

SUMMARY OF COST ANALYSIS FOR MANUAL SELECTION

Technical Salesman (i.e. 2.9 hours @ 3.75/hour)	10.87
Clerical Costs (i.e. 2 hours @ 2.25/hour)	4.50
[includes Logging Enquiries, Typing, Filing etc.]	
Telephone & Postage	2.20
	17.57

COST OF IMPLEMENTING A FULLY MANUAL SELECTION SYSTEM TO RESPOND TO ALL THE ENQUIRIES RECEIVED

To respond to all the enquiries received in the present year, and to cope with the expected increase of approximately 5%, the company needs to employ two extra technical salesmen. Assuming the average cost of employing an extra salesman is 6,333 pounds per annum, with an annual increase of 10% for the five years, the cost of employing two people would be 24,400 pounds per annum (in 5 years time). However each person will only be able to cope with an average of approximately 500 enquiries per annum. If the rate of number of enquiries received increases by more than the expected 5%, then there is the possibility that even more technical salesmen may be required, thereby further increasing the overheads.

B2. COMPUTER AIDED SELECTION (USING ACQUIRED HARDWARE)
E.G HEWLETT PACKARD 9845 MICRO COMPUTER

Initial cost of the hardware =	14,500 POUNDS

Assume useful life of the machine is 5 years.

Assume rate of interest that could be obtained by investing 14,500 pounds for five years is 10% per annum.

True cost of purchase =	23,352 POUNDS
Cost per annum = $\frac{28989}{5}$ =	4670 POUNDS
Operating cost per annum [i.e. maintenance @ 10% of initial cost, paper, power, etc]	2,500 POUNDS
Total cost per annum	7,170 POUNDS

Discussions with technical sales staff suggest that of the total enquiries received, approximately 80% of the required duties could be achieved by the standard sizes produced by the company. That is, 80% of the enquiries could be handled using the selection software developed (i.e. 80% of 4000 i.e. 3200 enquiries could be handled by the computer).

Average cost per enquiry $\frac{7170}{3200}$ =	2.24 POUNDS

Man hours to recommend a size from the choice and do other necessary work to complete the enquiry (e.g. motor-fan combination run-up time, sound spectra levels) say = 0.75 Hours.

Telephone charges say	0.50 POUNDS
Postage (as before)	0.30 POUNDS

B3. COMPUTER AIDED SELECTION (USING A BUREAU SERVICE)

Connection charge (say 0.1 hour per enquiry enquiry at 6 pounds/hour)	0.60 POUNDS
Computer Resource Units Charge (say 10 units units / enquiry @ 8p / unit) [i.e. Central Processor, Core Used, Disc Access etc.]	0.80 POUNDS
Program Storage Charges (say 500 Pounds/ year)	0.25 POUNDS
Printer (Rental @ 100 pounds per month including maintenance)	0.60 POUNDS
Graphics(20 Pounds/Hour plotter charges say 3 minutes/ enquiry	1.00 POUNDS
Telephone	0.10 POUNDS
Modem Rental (i.e. G.P.O. link to the computer @ 150 Pounds / annum	0.08 POUNDS
Postage (for graphics produce at Bureau Centre and sent via post)	0.15 POUNDS
Average cost per enquiry	3.58 POUNDS

COST OF IMPLEMENTATION USING ACQUIRED HARDWARE

Purchase cost per annum	5798 POUNDS
Operating cost per annum	2500 POUNDS

	8298

Note:-

These charges are fixed no matter how many enquiries are answered.

B3. COMPUTER AIDED SELECTION (USING A BUREAU SERVICE)

Connection charge (say 0.1 hour per enquiry enquiry at 6 pounds/hour)	0.60 POUNDS
Computer Resource Units Charge (say 10 units units / enquiry @ 8p / unit) [i.e. Central Processor, Core Used, Disc Access etc.]	0.80 POUNDS
Program Storage Charges (say 500 Pounds/ year)	0.25 POUNDS
Printer (Rental @ 100 pounds per month including maintenance)	0.60 POUNDS
Graphics(20 Pounds/Hour plotter charges say 3 minutes/ enquiry	1.00 POUNDS
Telephone	0.10 POUNDS
Modem Rental (i.e. G.P.O. link to the computer @ 150 Pounds / annum	0.08 POUNDS
Postage (for graphics produce at Bureau Centre and sent via post)	0.15 POUNDS
Average cost per enquiry	3.58 POUNDS

SUMMARY OF COST ANALYSIS FOR THE BUREAU SERVICE

METHOD OF IMPLEMENTATION OF THE COMPUTER AIDED SELECTION.

Bureau Service cost per enquiry	3.58 POUNDS
Technical Salesmen (1 hour @ 3.75/hour)	3.75 POUNDS
Clerical Cost (say 2 hours @ 2.25/hour)	4.50 POUNDS
Telephone & Postage	0.80 POUNDS
Average cost per enquiry	12.63 POUNDS

Note:-

a). Storage, printer, modem rental charges per enquiry are calculated on the basis of the amount paid per annum, being invested for 5 years, at an interest rate of 10%, and 3200 enquiries being answered using the bureau service.

b). Technical salesmen time is longer compared with an in-house implemented computer aided selection system, because of the procedure required when logging on to the bureau system, and also because the graphics produced will arrive by post later and the need to include it in with the rest of the enquiry.

COST OF IMPLEMENTATION THE SELECTION SYSTEM USING THE BUREAU

Fixed charges are those for:-

Program storage	500	POUNDS per annum
Printer rental	1,200	" " "
G.P.O. Link (modem unit)	150	" " "

	1,850	" " "

Note :-

Assuming that the above amount is invested for 5 years yielding an annual interest of 10%, the true value of the above sum becomes 2,979 pounds per annum. These are the fixed charges which will be incurred before any enquiry can be answered.

To answer each enquiry there is an additional computing cost of 2.65 pounds per enquiry. Assuming that the bureau costs increase annually by 10% for 5 years, this represents a true cost of 4.27 pounds per enquiry.

CHAPTER 3

THE ADVANTAGES OF COMPUTER AIDED SELECTION

The main advantages of computer aided selection are:-

1). Average cost per enquiry can be reduced from the present 17.50 pounds to 10.30 pounds using computer aided selection by acquiring suitable hardware. The average cost per enquiry would be further reduced as:-

a). The number of enquiries received increases (as shown by the trend) while operating costs remain fixed.

b). Further development of the selection program (e.g. run-up time for the fan-motor combination, sound level spectra, bearing selection etc.) would reduce the manual time spent on the enquiry even further.

2). More information obtained easily and quickly. Two trial runs in June and July 1979, using the Hewlett Packard 9845 and Tektronix 4052 desk top computers, at Alldays, Peacock, showed that someone with no previous experience in using such machines, could use them efficiently for this purpose within half an hour and that the time taken to obtain a typical print out (see end of this chapter) was approximately five minutes. Using the computer, on average, 2-3 suitable fan sizes can result for a specific duty, each giving details of:-

a). Type

b). Size

c). Running speed

d). Operating efficiency

e). Power required

f). Motor size required.

g). Price of the basic fan unit and ancillary equipment.

h). Performance curves of volume-pressure and volume-power at the operating conditions showing the duty point.

3). Improved service by providing:-

a). An instant response to most telephone enquiries (which accounted for 60% of the total for 1978) i.e. the enquiry could well be answered while the customer waits with a written quotation to follow to confirm the details given verbally. This will result not only in a better service, but could lead to a saving in telephone charges (e.g. up to say 50% i.e.approximately 3,000 pounds per annum) as the enquiry will be answered when the customer is paying for the telephone call. The present procedure is to obtain the requirements from the customer and to telephone him back with the details, in which case the company is paying for the telephone call.

b). Quicker response to all enquiries. Pie chart No. 5 (Appendix No. 3) shows the distribution of the time taken to answer a random same of 245 enquiries between 1978 and 1979. It shows that 47% of the enquiries were handled within the first 5 working days, and a further 21% took between 6-10 working days. With the computer aided selection system taking less time (e.g. with the present program say 1 hour compared with 2.9 hours with the present fully manual selection procedure) this could result in between 80% - 90% of the enquiries being handled within the first 5 working days. The remainder taking between 6-10 working days. This improvement in the response time is hoped will lead to an increase in the number of orders received.

4). As the time taken per enquiry is reduced, this will result in more enquiries per person being handled. Since the trend for the number of enquiries received is upwards, this means that the expected increase can be handled with the present level of staff.

5). Using the computer relieves the human from carrying out extensive "number crunching" releasing him for the more rewarding task of analysing the information and decision making. The program can be developed so that it gives the most suitable size, however, the resulting program would have to be very interactive with the user, requiring a larger size (memory) machine and resulting in longer processing time, because selecting a size from a range is an "art" rather than a "science". The other advantage of the developed computer aided selection system, compared with a fully computerised selection system (where the computer selects one size only) is that the user feels (and is) part of the system, and not isolated from it because the decision making task, has not been taken away from him. This means that there is likely to be less resentment in the introduction of such a system, thereby increasing the chance of the system being sucessful in its operation.

Example of a typical print-out from the

Hewlett Packard desktop micro computer

giving details of suitable selections

for a specific duty.

```
                    DUTY REQUIREMENTS
VOLUME= 14500 C.F.M.                    PRESSURE= 7 INS. WATER GAUGE
TEMPERATURE= 100.000 DEGREES C
ABOVE QUANTITIES ARE THE ACTUAL REQUIRED AT THE OPERATING TEMPERATURE
DENSITY = .944  KG / M ^ 3 AT 100.000 DEGREES C
INDIRECT COUPLED FAN
AT STANDARD TEMPERATURE (20 DEGREES C) AND PRESSURE(762 MM WATER GAUGE)
VOLUME= 24635 M ^ 3 / HR.               PRESSURE = 226.346 MM WATER GAUGE
DENSITY = 1.2015 KG / M ^ 3


SELECTION(S) THAT MEET THE ABOVE REQUIREMENTS
```

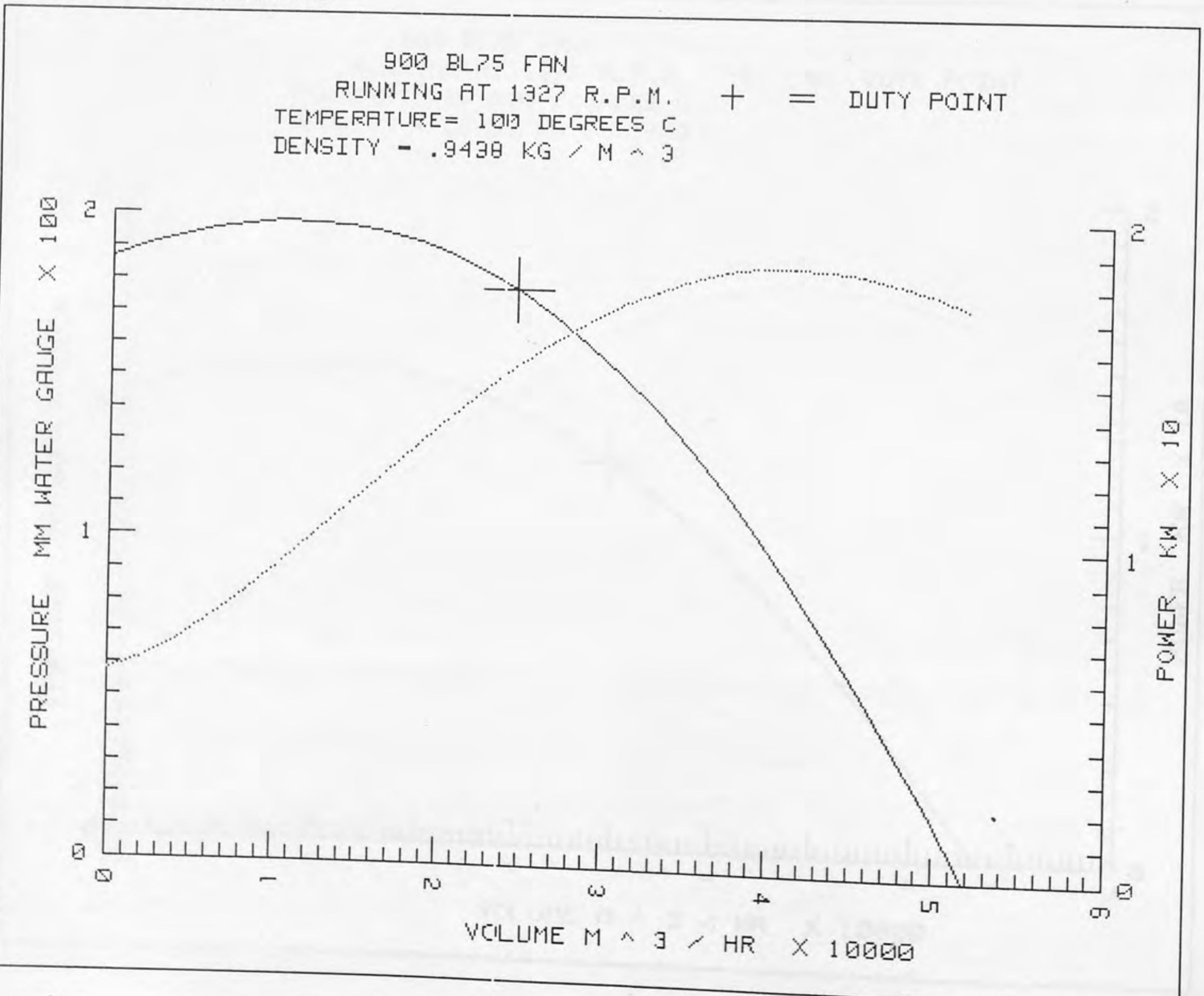
900 BL75 FAN

BASE CURVE VOLUME(IE VOLUME AT 1450 R.P.M) = 26900 M ^ 3 / HOUR
POWER REQUIRED AT 20 DEGREES C (OR 68 DEGREES F)= 19.59 KW
MOTOR SIZE REQUIRED = 30.00 KW
MOTOR FRAME SIZE = D.200L
POWER REQUIRED AT THE OPERATING TEMPERATURE 15.39 KW
MOTOR SIZE REQUIRED AT THE OPERATING TEMPERATURE = 18.50 KW
MOTOR FRAME SIZE = D.180M
EFFICENCY = 77.44 %
RUNNING SPEED = 1327 R.P.M
SOUND LEVEL SPECTRUM

FREQUENCY	63	125	250	500	1K	2K	4K	8K HZ
S P L SPECTRUM (dB)	112	105	97	110	97	86	110	101
	109	104	100	98	97	95	90	83

PRICE LIST (IN POUNDS) FOR ARRANGEMENT NO 3

BASIC FAN PRICE =	1033
FLANGED INLET =	43
INSPECTION DOOR =	36
DRAIN PLUG =	13
COOLING IMPELLER =	76
BASE FRAME =	314
SPECIAL MATERIALS =	
SPECIAL CONSIDERATIONS =	



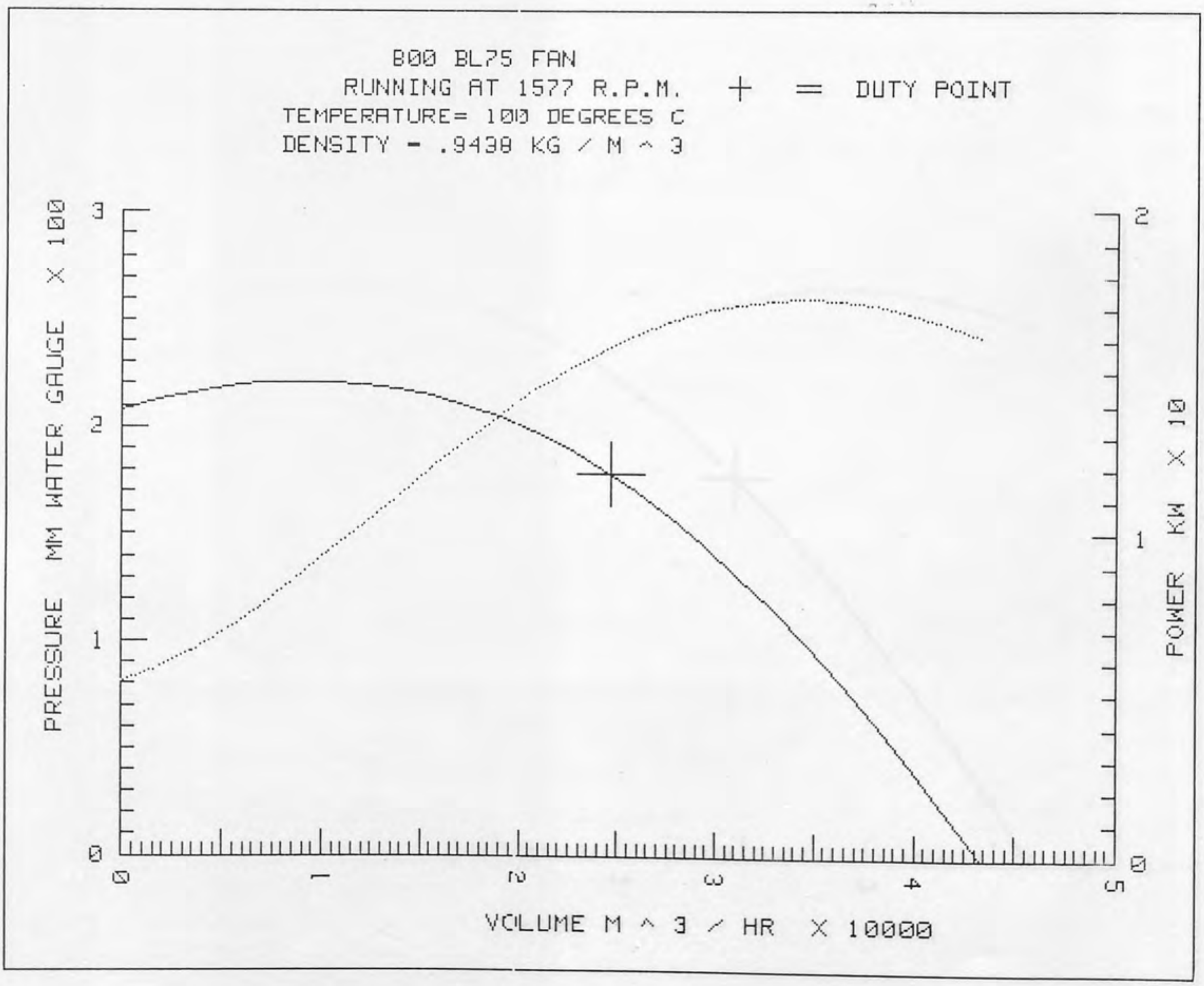
800 BL75 FAN

BASE CURVE VOLUME(IE VOLUME AT 1450 R.P.M) = 22640 M ^ 3 / HOUR
POWER REQUIRED AT 20 DEGREES C (OR 68 DEGREES F)= 20.12 KW
MOTOR SIZE REQUIRED = 30.00 KW
MOTOR FRAME SIZE = D.200L
POWER REQUIRED AT THE OPERATING TEMPERATURE 15.81 KW
MOTOR SIZE REQUIRED AT THE OPERATING TEMPERATURE = 18.50 KW
MOTOR FRAME SIZE = D.180M
EFFICENCY = 75.07 %
RUNNING SPEED = 1577 R.P.M
SOUND LEVEL SPECTRUM

FREQUENCY	63	125	250	500	1K	2K	4K	8K HZ
S P L SPECTRUM (dB)	112	105	97	110	97	86	110	101
	109	105	101	100	99	98	94	86

PRICE LIST (IN POUNDS) FOR ARRANGEMENT NO 3

BASIC FAN PRICE =	762
FLANGED INLET =	43
INSPECTION DOOR =	28.5
DRAIN PLUG =	13
COOLING IMPELLER =	59
BASE FRAME =	276
SPECIAL MATERIALS =	
SPECIAL CONSIDERATIONS =	



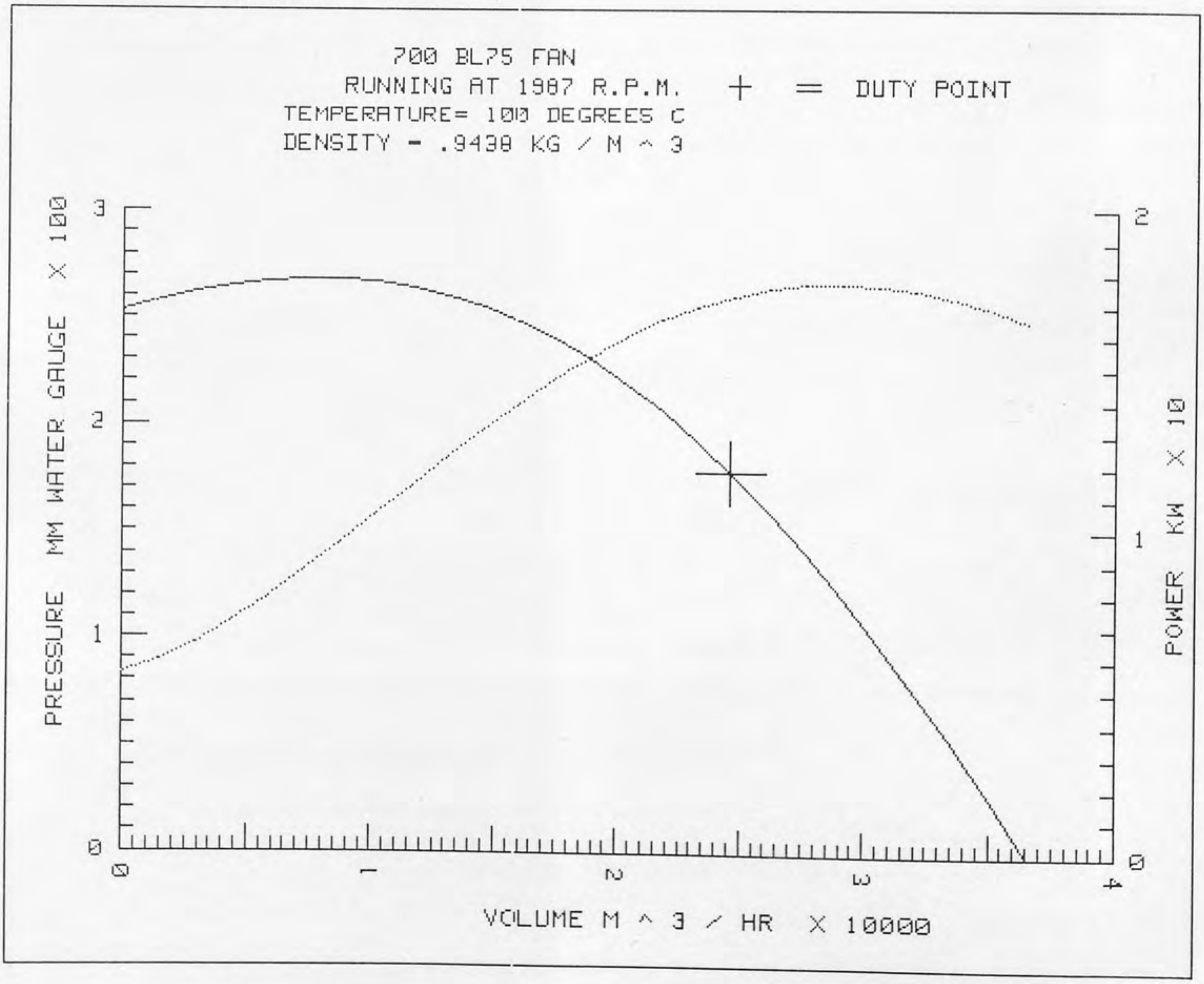
700 BL75 FAN

BASE CURVE VOLUME(IE VOLUME AT 1450 R.P.M) = 17980 M ^ 3 / HOUR
POWER REQUIRED AT 20 DEGREES C (OR 68 DEGREES F)= 22.03 KW
MOTOR SIZE REQUIRED = 30.00 KW
MOTOR FRAME SIZE = D.200L
POWER REQUIRED AT THE OPERATING TEMPERATURE 17.30 KW
MOTOR SIZE REQUIRED AT THE OPERATING TEMPERATURE = 22.00 KW
MOTOR FRAME SIZE = D.180L
EFFICENCY = 68.68 %
RUNNING SPEED = 1987 R.P.M
SOUND LEVEL SPECTRUM

FREQUENCY	63	125	250	500	1K	2K	4K	8K HZ
S P L SPECTRUM (dB)	112	105	97	110	97	86	110	101
	110	106	103	103	103	102	99	92

PRICE LIST (IN POUNDS) FOR ARRANGEMENT NO 3

BASIC FAN PRICE =	580
FLANGED INLET =	34
INSPECTION DOOR =	28.5
DRAIN PLUG =	13
COOLING IMPELLER =	59
BASE FRAME =	241
SPECIAL MATERIALS =	
SPECIAL CONSIDERATIONS =	



APPENDIX NO. 1.

HARDWARE SPECIFICATION.

1). Memory size to be a minimum of 64 k bytes. The size of the present program (including loading the necessary data into memory) is 40 k bytes, therefore the above size memory will allow for further development of the program.

2). Graphics capability to draw the fan performance characteristic curves.

3). Line printer (built in, or external units) to give hard copy of the suitable selections.

4). Plotter (drum or flat bed type). The performance curves are to be sent to the prospective customer and as such, need to be of high quality.

5). System must be flexible in expansion for future requirements e.g. increasing the memory size, addition of floppy and or hard disc drives, tape drives etc.

6). Easy to operate so no extensive training would be necessary.

7). Low operating and maintenance costs.

8). Good back-up service.

9). Operating system easy to use with attractive features for ease of program structuring, editing etc. i.e. implementing the system using a high level language (such as Basic, Fortran etc) suitable for development of interactive software.

APPENDIX NO. 2.

DISADVANTAGES OF USING A COMPUTER BUREAU SERVICE.

1). Average cost per enquiry of 12.60 pounds is higher than the 10.30 pounds using bought hardware (e.g. a Hewlett Packard desk top computer).

2). Performance curves cannot be produced at the terminal link in Alldays Peacock as the G.P.O. link is not capable of transferring graphic data. Curves will have to be produced on the flat bed plotter housed at the bureau's centre and sent by mail. Thus the Technical Salesman cannot visually see the duty point on the performance curve instantaneously to determine the suitability of the selection.

3). System is remote from the user and he feels he has no control over it. This could lead to some resentment in operating the system.

4). On maintenance days the computer cannot be used where as with a in-house machine maintenance can be planned. Also the maintenance time for a desk top computer is very short compared with a main frame computer.

5). Telephone charges will increase as the link to the computer is via a G.P.O. telephone line, and such charges are incurred each time access to the computer is required. The amount will depend on the amount used.

APPENDIX NO. 3

PIE CHARTS AND GRAPHS

PIE CHARTS AND GRAPHS

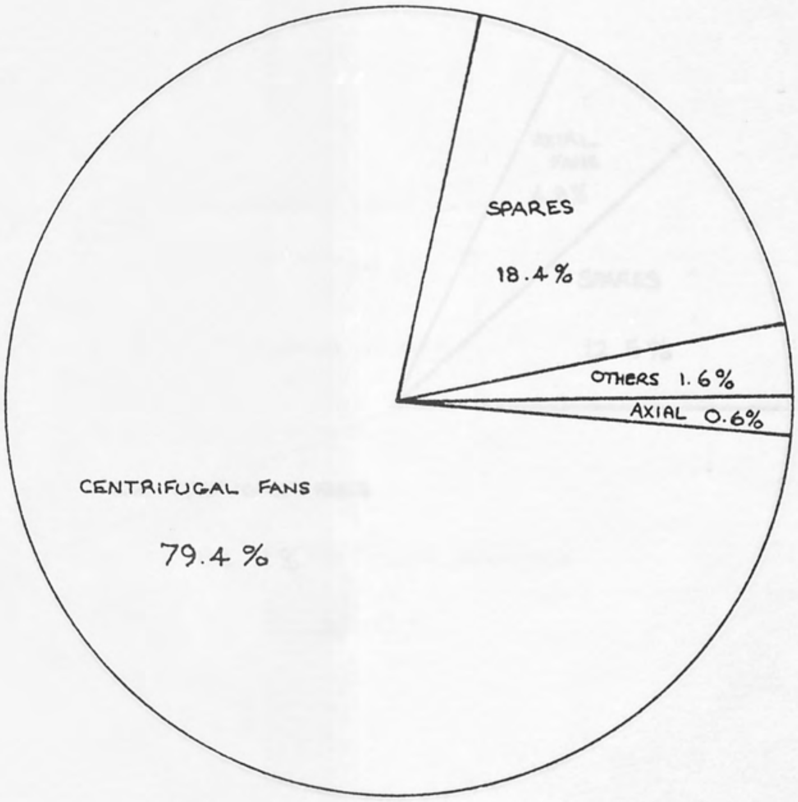
PIE CHARTS.

NUMBER	TITLE
1	Sales for the 12 months up to 30/6/1979 .
2	Type of fan enquiries received in 1978.
3	Centrifugal fan enquiries received in 1978.
4	Types of fans selected as suitable to meet the customer's specific requirements in 1978.
5	Number of working days taken to answer the enquiry. (random sample size of 245 enquiries)
6	Order to quote ratio (fans only).
7	Orders received :Number of working days taken to answer the enquiry (random sample size of 40 orders).
8	Orders received :Number of working days taken to answer the enquiry (random sample size of 130 orders).
9	% of total enquiries received but not answered due to insufficient time.

GRAPHS

1	Number of enquiries received per annum and the number of enuiries received but not answered per annum.

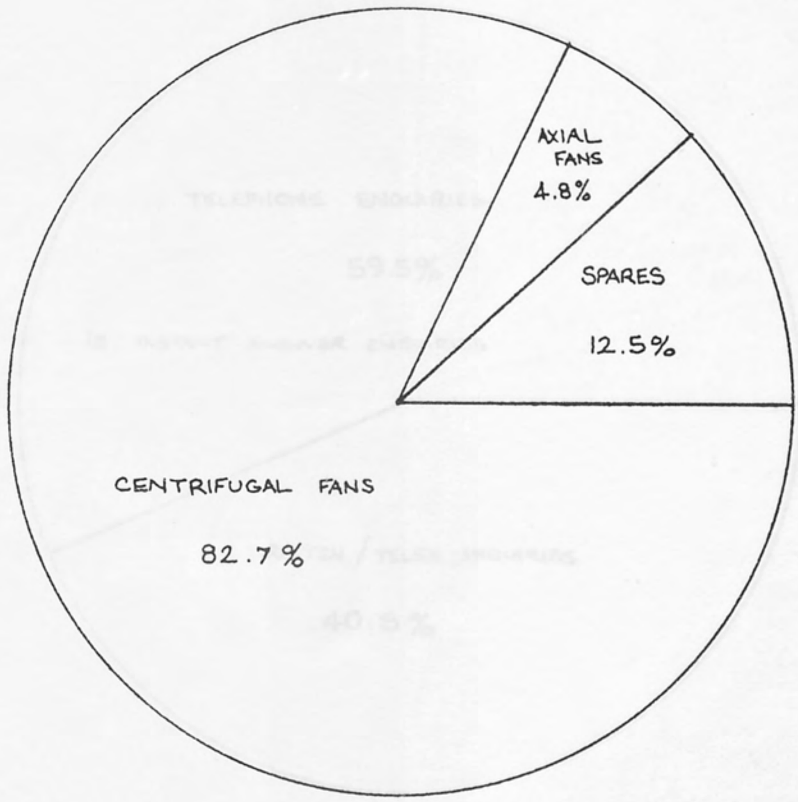
(1)



SALES FOR 12 MONTHS UP TO 30/6/78

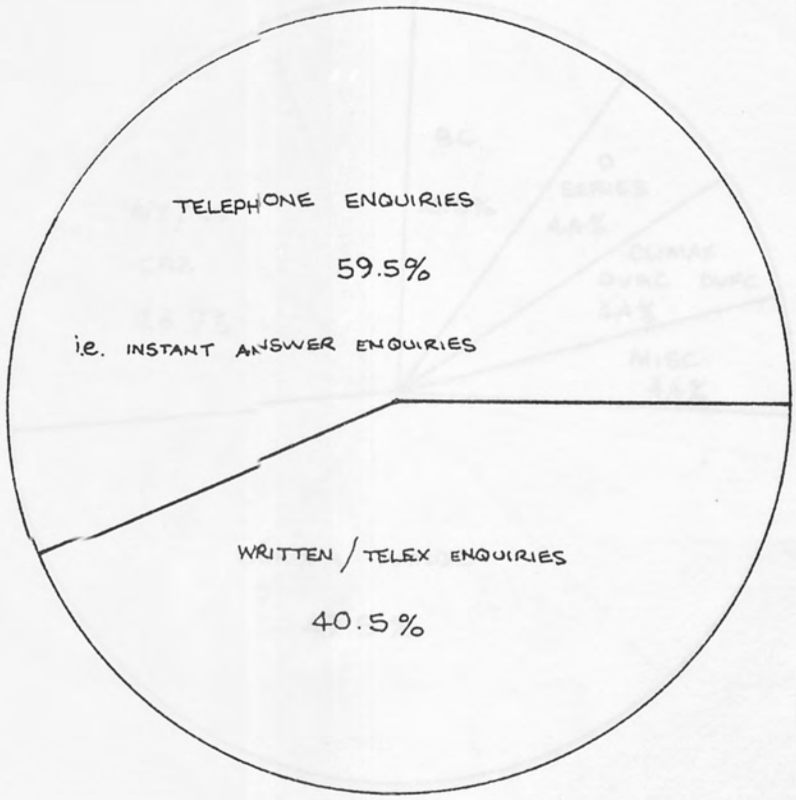
£2,548,764

(2)



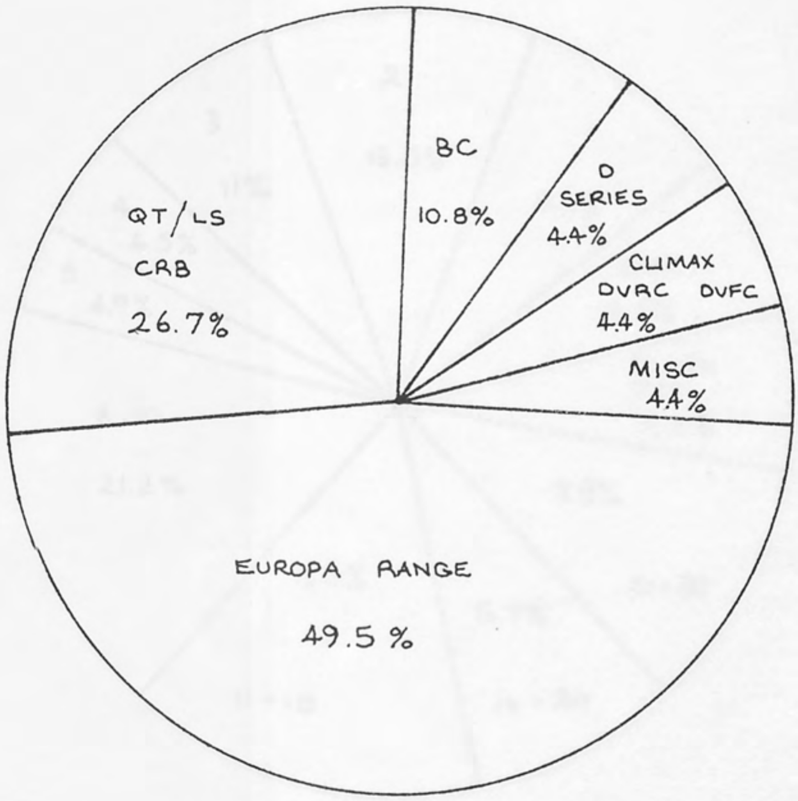
ENQUIRIES RECEIVED IN 1978 4,899

3



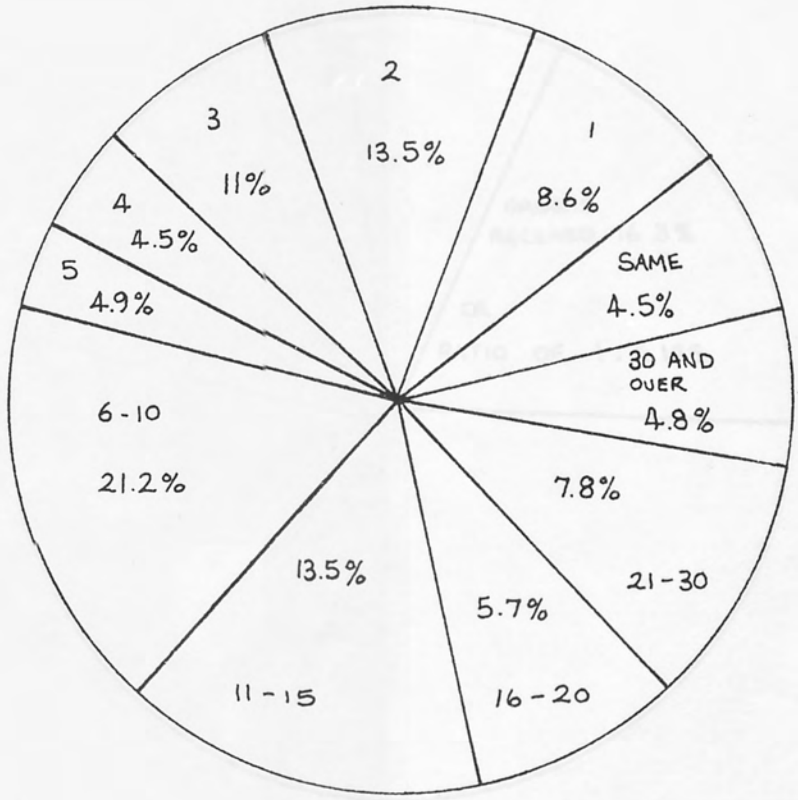
CENTRIFUGAL FAN ENQUIRIES FOR 1978 4,051

④



FANS SELECTED AS SUITABLE TO MEET CUSTOMER'S REQUIREMENTS FROM ENQUIRIES RECEIVED IN 1978 5,386

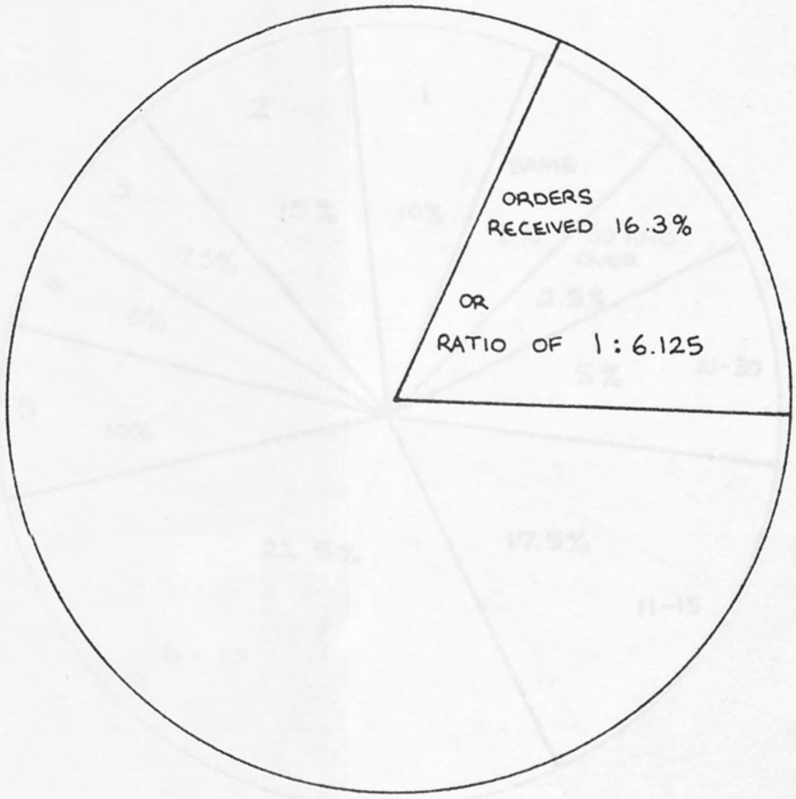
⑤



NUMBER OF WORKING DAYS TAKEN TO ANSWER AN ENQUIRY

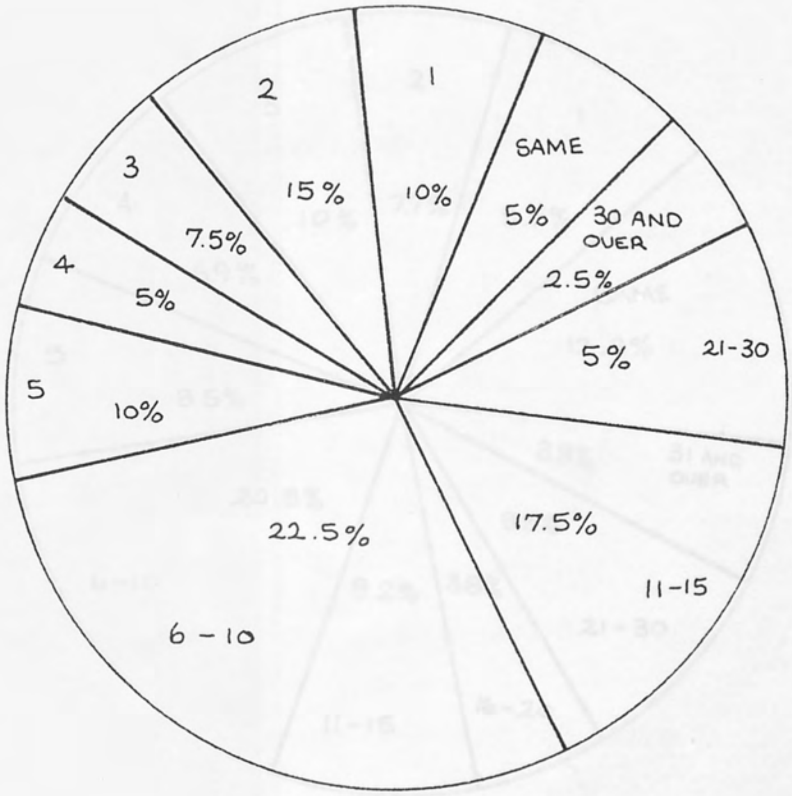
SAMPLE SIZE 245 ENQUIRIES.

⑥



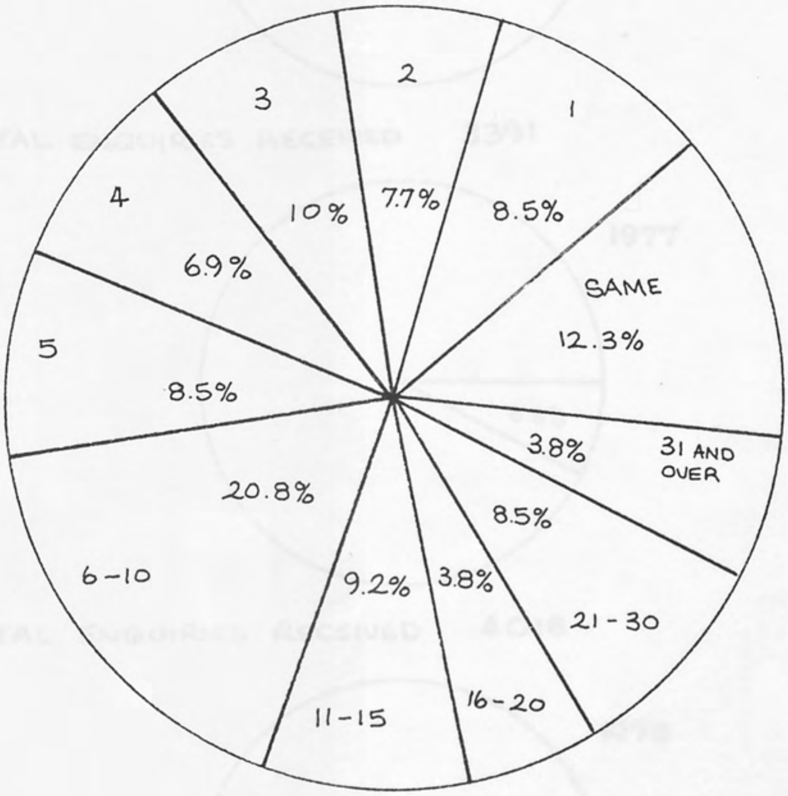
ORDER TO QUOTE RATIO (FANS ONLY)

7

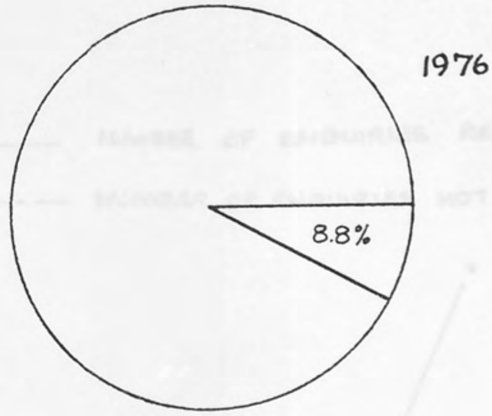


ORDERS RECEIVED: NUMBER OF DAYS TAKEN TO ANSWER THE ENQUIRY (NUMBERS NOT VALUES)
SAMPLE SIZE 40 ORDERS

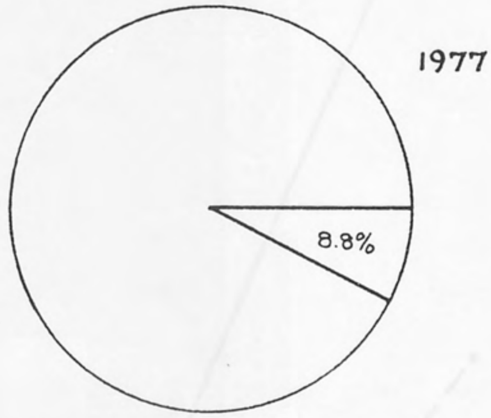
⑧



ORDERS RECEIVED ANALYSIS SHOWING NUMBER OF DAYS TAKEN TO ANSWER THE ENQUIRY (NUMBERS NOT VALUES)
SAMPLE SIZE 130 ORDERS.



TOTAL ENQUIRIES RECEIVED 3391

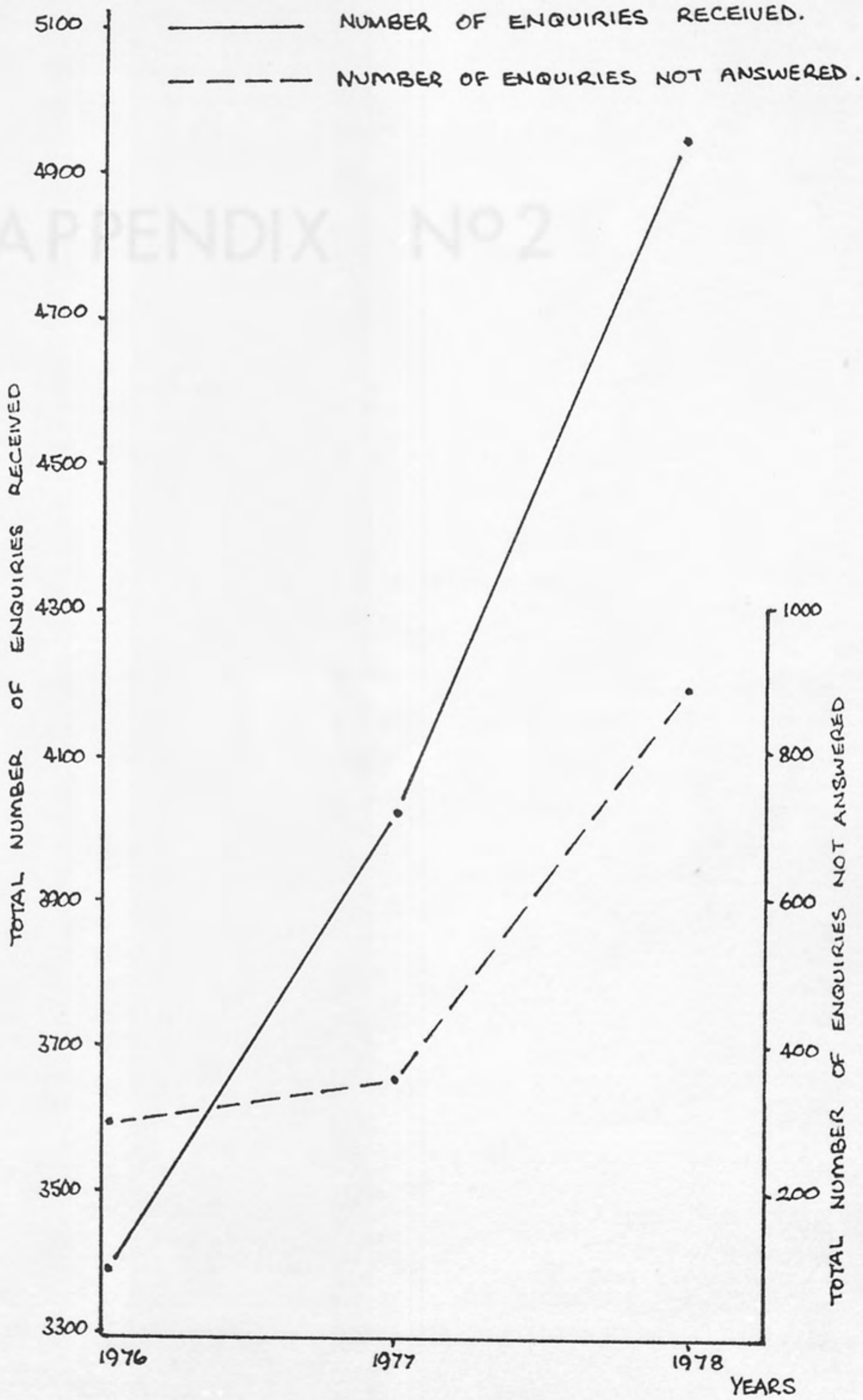


TOTAL ENQUIRIES RECEIVED 4018

TOTAL ENQUIRIES RECEIVED 4940

% OF TOTAL ENQUIRIES RECEIVED BUT NOT ANSWERED DUE TO INSUFFICIENT TIME.

APPENDIX N°2



GRAPH N° 1

APPENDIX Nº2

E N Q U I R Y   R E S P O N S E

T I M E   A N A L Y S I S

R E S U L T S

WRITTEN ENQUIRY RESPONSE TIME ANALYSIS.

DAYS TAKEN TO ANSWER THE ENQUIRY	ORDERS RECEIVED	NO ORDERS RECEIVED	TOTAL	ORDERS EXPECTED	ORDERS RECEIVED - ORDERS EXPECTED
0	5	11	16	3.7	+1.3
1	14	23	37	8.5	+5.5
2	20	25	45	10.3	+9.7
3	15	19	34	7.8	+7.2
4	12	22	34	7.8	+4.2
5	9	23	32	7.3	+1.7
6	4	19	23	5.3	-1.3
7	5	21	26	6.0	-1.0
8	8	33	41	9.4	-1.4
9	5	21	26	6.0	-1.0
10	1	10	11	2.5	-1.5
11	2	15	17	3.9	-1.9
12	3	11	14	3.2	-0.2
13	2	16	18	4.1	-2.1
14	4	12	16	3.7	+0.3
15	3	4	7	1.6	+1.4
16	0	9	9	2.1	-2.1
17	0	7	7	1.6	-1.6
18	2	11	13	3.0	-1.0
19	0	6	6	1.4	-1.4
20	0	6	6	1.4	-1.4
21	1	3	4	0.9	+0.1
22	0	9	9	2.1	-2.1
23	0	8	8	1.8	-1.8
24	1	3	4	0.9	+0.1
25	1	4	5	1.1	-0.1
26	1	6	7	1.6	-0.6
27	0	2	2	0.5	-0.5
28	0	2	2	0.5	-0.5
29	1	4	5	1.1	-0.1
30	0	4	4	0.9	-0.9
31-40	1	15	16	3.7	-2.7
41-50	0	10	10	2.3	-2.3
51+	1	13	14	3.2	-2.2
TOTALS	121	407	528	121	

$$\sum_{\substack{\text{all}\\ \text{categories.}}} \frac{(\text{ORDERS RECEIVED} - \text{ORDERS EXPECTED})^2}{\text{ORDERS EXPECTED}} = 42.0 \text{ HIGHLY SIGNIFICANT AT } 0.10\ \%.$$

As the results are highly significant and since we get more orders if the enquiry is answered quickly than expected, and less orders than expected if time taken is longer, we can conclude the quicker the enquiry is answered the more chance there is of turning the enquiry into an order.

VERBAL ENQUIRY RESPONSE TIME ANALYSIS.

DAYS TAKEN TO ANSWER THE ENQUIRY	NO. ORDERS RECEIVED	ORDERS RECEIVED	TOTAL	ORDERS EXPECTED	ORDERS RECEIVED - ORDERS EXPECTED
0	174	37	211	30.9	+6.1
1	213	39	252	37.0	+2.0
2	124	24	148	21.7	+2.3
3	102	13	115	16.9	-3.9
4	52	11	63	9.2	-2.2
5	35	7	42	6.2	+0.8
6	26	3	29	4.3	-1.3
7	25	2	27	4.0	-2.0
8	29	3	32	4.7	-1.7
9	17	1	18	2.6	-1.6
10	10	0	10	1.5	-1.5
11	5	1	6	0.9	+0.1
12	8	2	10	1.5	+0.5
13	16	1	17	2.5	-1.5
14	8	1	9	1.3	-0.3
15	5	0	5	0.7	-0.7
16-20	11	2	13	1.8	+0.2
21-30	9	3	12	1.8	+1.2
31-50	5	0	5	0.7	-0.7
50+	5	1	6	0.9	+0.1
TOTALS	870	151	1030	151	

$$\sum_{\text{all categories.}} \left(\frac{\text{ORDERS RECEIVED - ORDERS EXPECTED}}{\text{ORDERS EXPECTED}} \right)^2 = 7.51 \text{ SIGNIFICANT AT } 90\% .$$

Since the results are highly insignificant , one can conclude that there is no relationship between the time taken to answer the enquiry and whether it subsequently results in a firm order.

APPENDIX Nº3

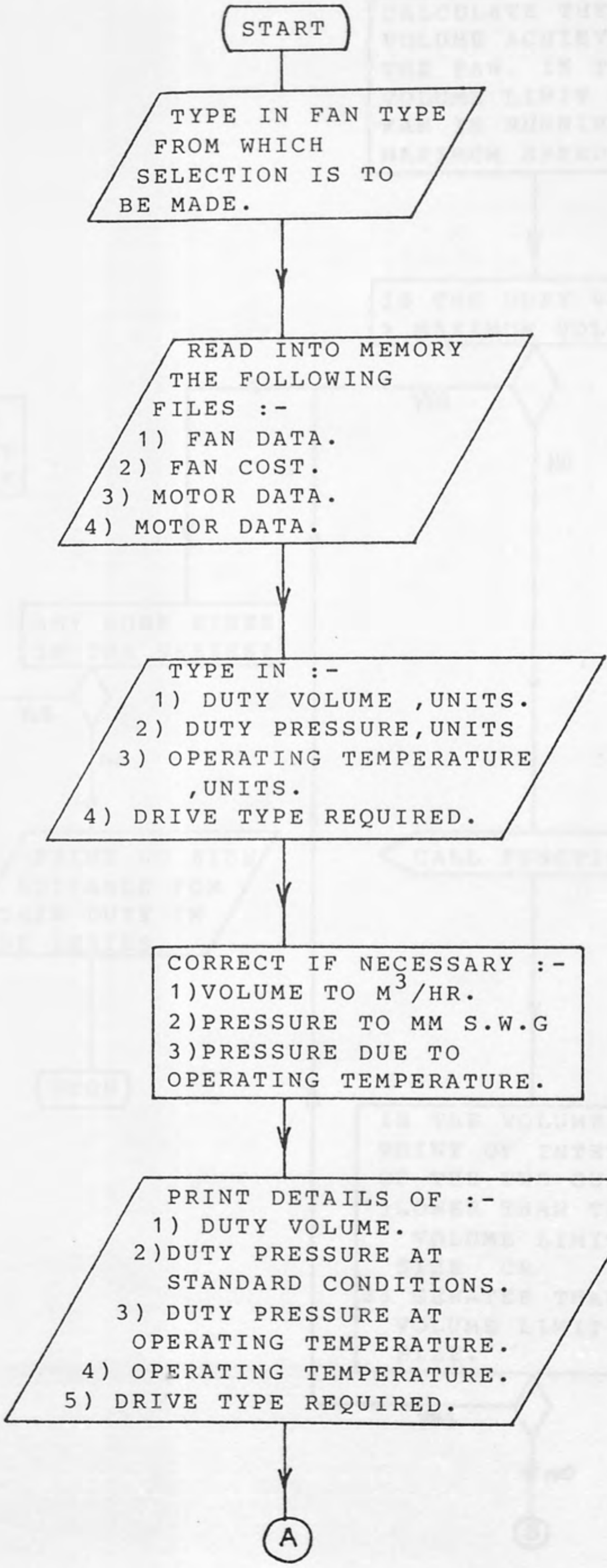
F L O W   C H A R T S   F O R

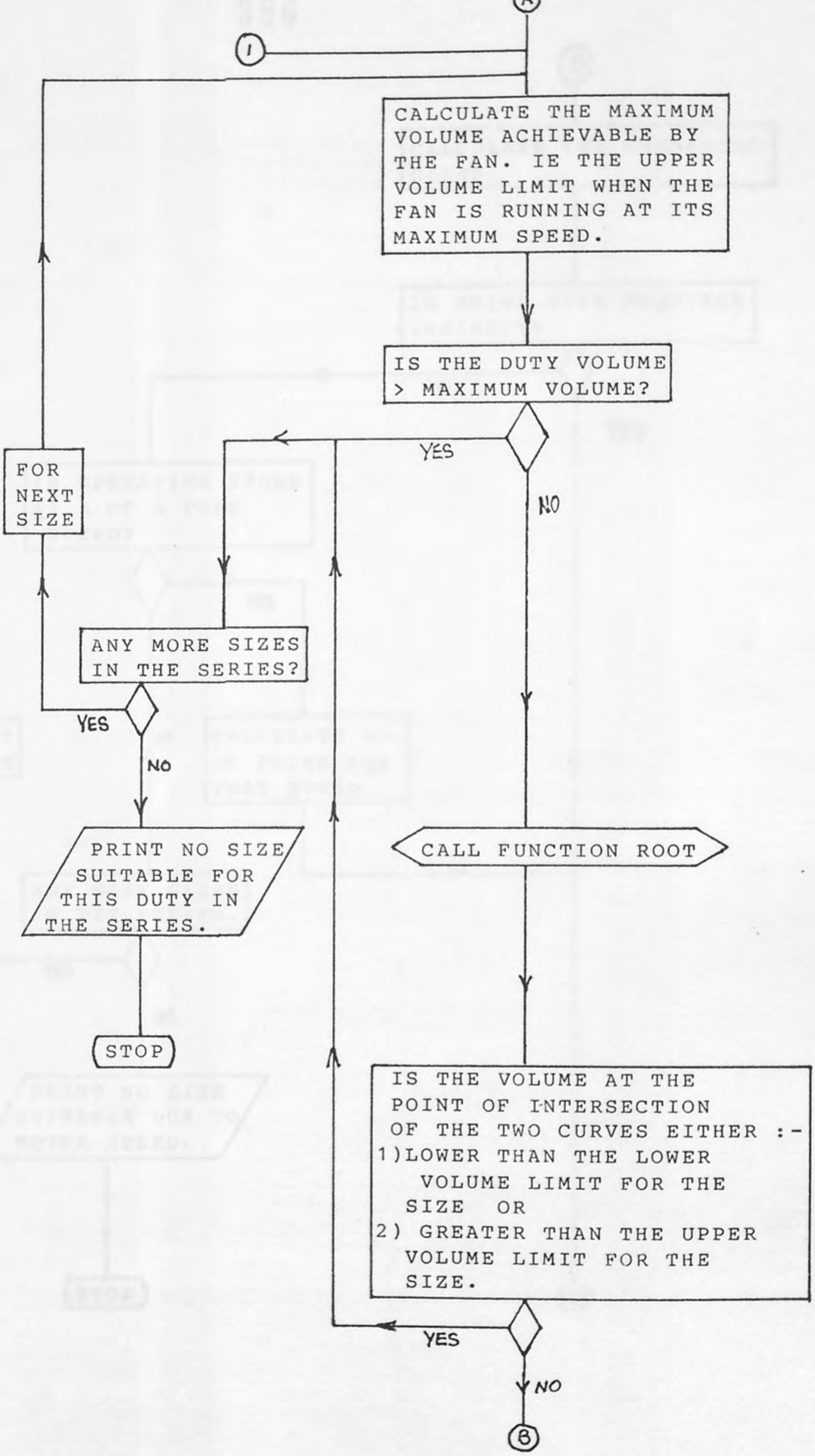
T H E   F A N   " S E L E C T I O N "   P R O G R A M

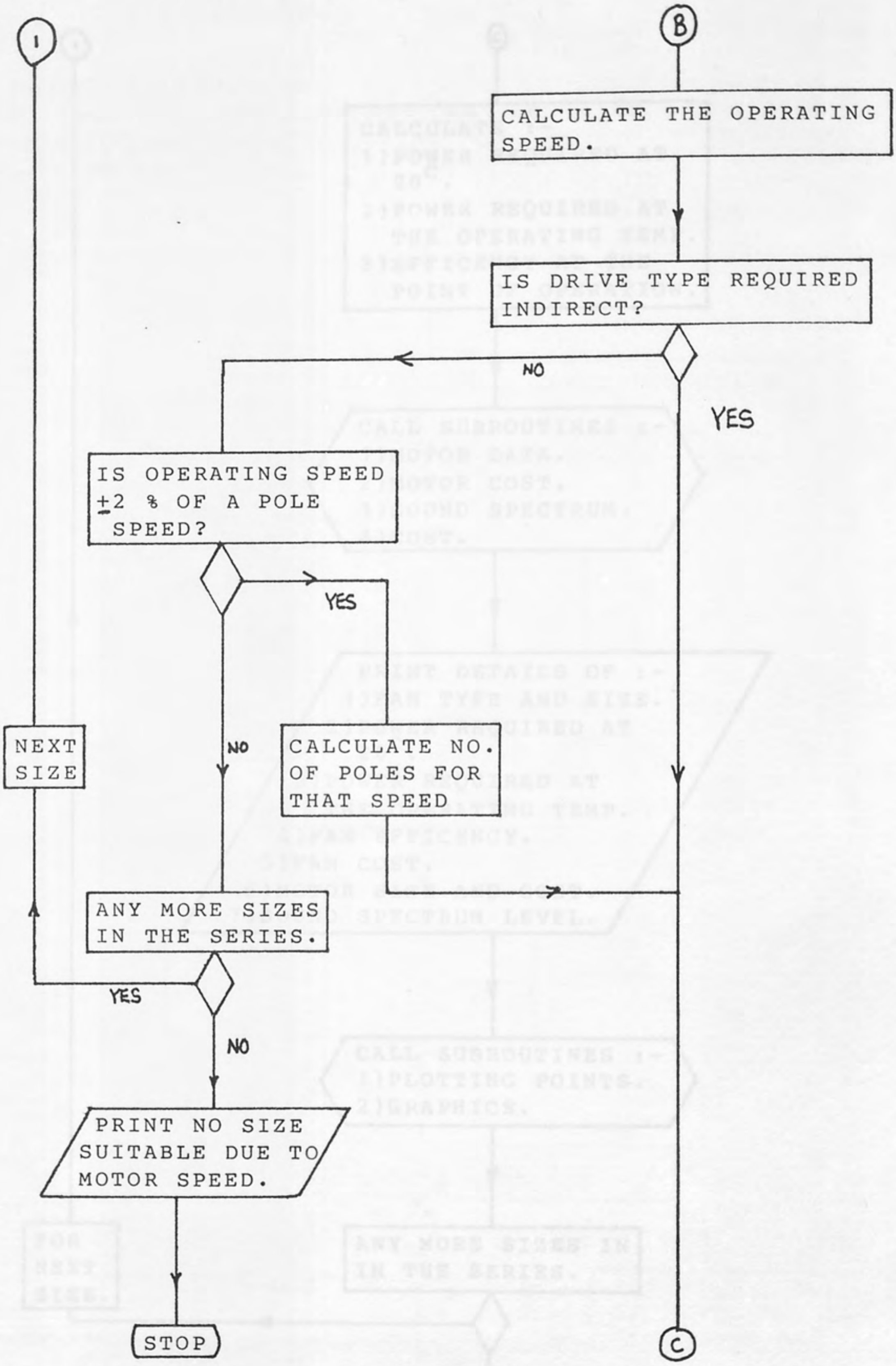
A N D   T H E   S U B R O U T I N E S

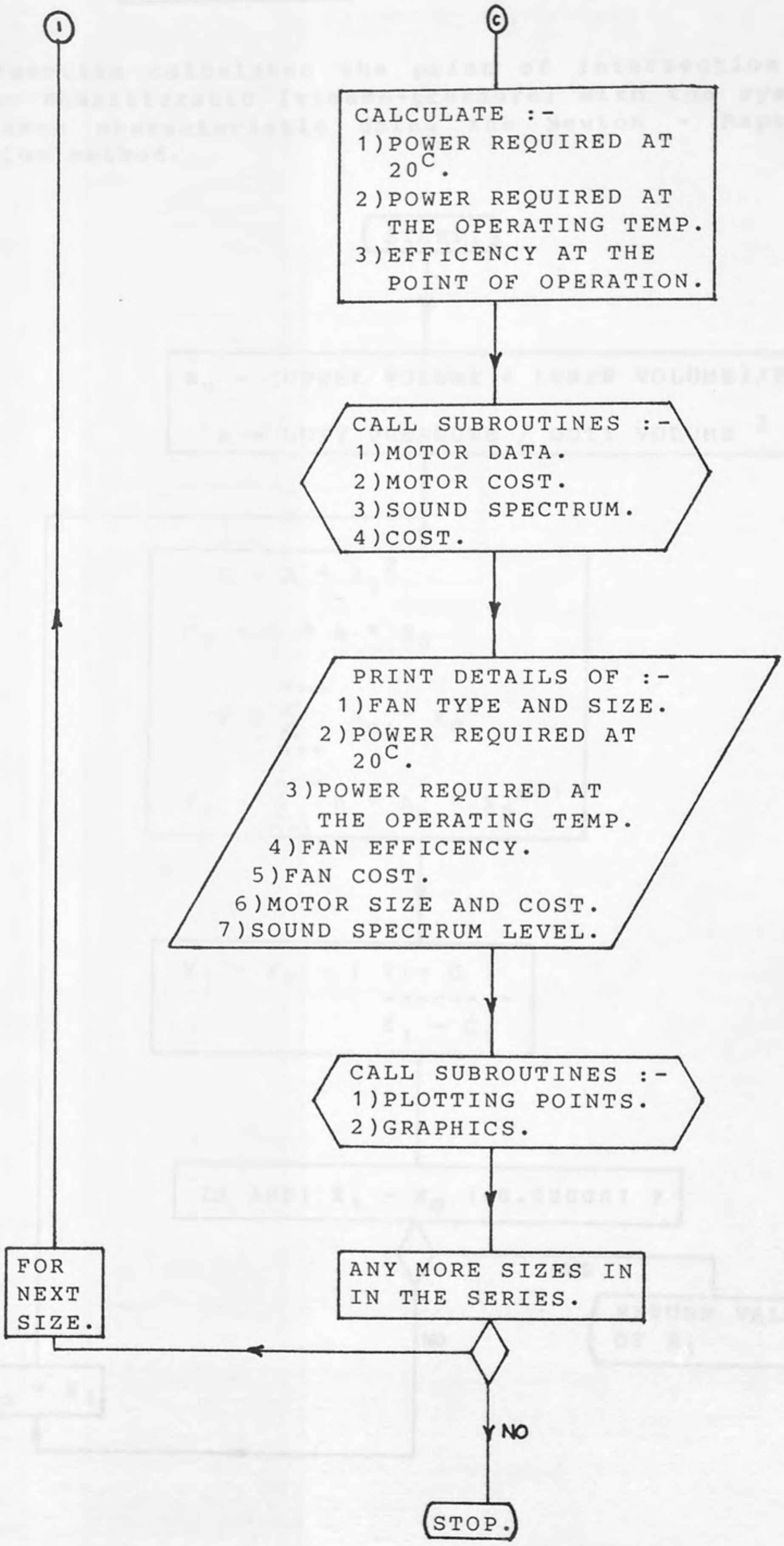
C A L L E D   B Y   I T

PROGRAM SELECT



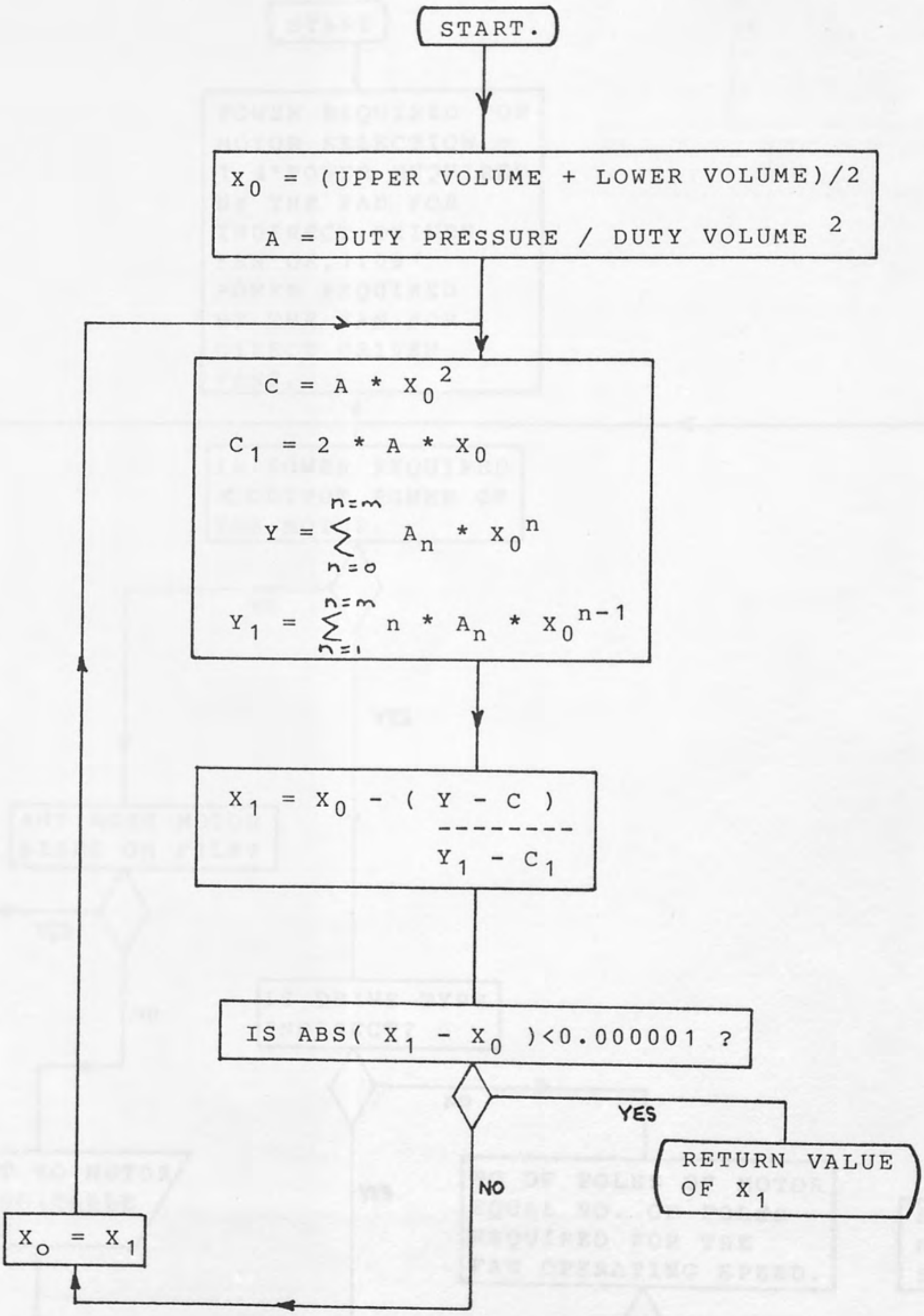

A
1
CALCULATE THE MAXIMUM VOLUME ACHIEVABLE BY THE FAN. IE THE UPPER VOLUME LIMIT WHEN THE FAN IS RUNNING AT ITS MAXIMUM SPEED.
IS THE DUTY VOLUME > MAXIMUM VOLUME?
YES
NO
FOR NEXT SIZE
ANY MORE SIZES IN THE SERIES?
YES
NO
PRINT NO SIZE SUITABLE FOR THIS DUTY IN THE SERIES.
STOP
CALL FUNCTION ROOT
IS THE VOLUME AT THE POINT OF INTERSECTION OF THE TWO CURVES EITHER :- 1) LOWER THAN THE LOWER VOLUME LIMIT FOR THE SIZE OR 2) GREATER THAN THE UPPER VOLUME LIMIT FOR THE SIZE.
YES
NO
B


1
B
CALCULATE THE OPERATING SPEED.
IS DRIVE TYPE REQUIRED INDIRECT?
NO
YES
IS OPERATING SPEED ±2 % OF A POLE SPEED?
YES
NO
CALCULATE NO. OF POLES FOR THAT SPEED
NEXT SIZE
ANY MORE SIZES IN THE SERIES.
YES
NO
PRINT NO SIZE SUITABLE DUE TO MOTOR SPEED.
STOP
C


I
C
CALCULATE :-
1)POWER REQUIRED AT 20C.
2)POWER REQUIRED AT THE OPERATING TEMP.
3)EFFICENCY AT THE POINT OF OPERATION.
CALL SUBROUTINES :-
1)MOTOR DATA.
2)MOTOR COST.
3)SOUND SPECTRUM.
4)COST.
PRINT DETAILS OF :-
1)FAN TYPE AND SIZE.
2)POWER REQUIRED AT 20C.
3)POWER REQUIRED AT THE OPERATING TEMP.
4)FAN EFFICENCY.
5)FAN COST.
6)MOTOR SIZE AND COST.
7)SOUND SPECTRUM LEVEL.
CALL SUBROUTINES :-
1)PLOTTING POINTS.
2)GRAPHICS.
ANY MORE SIZES IN IN THE SERIES.
FOR NEXT SIZE.
NO
STOP.

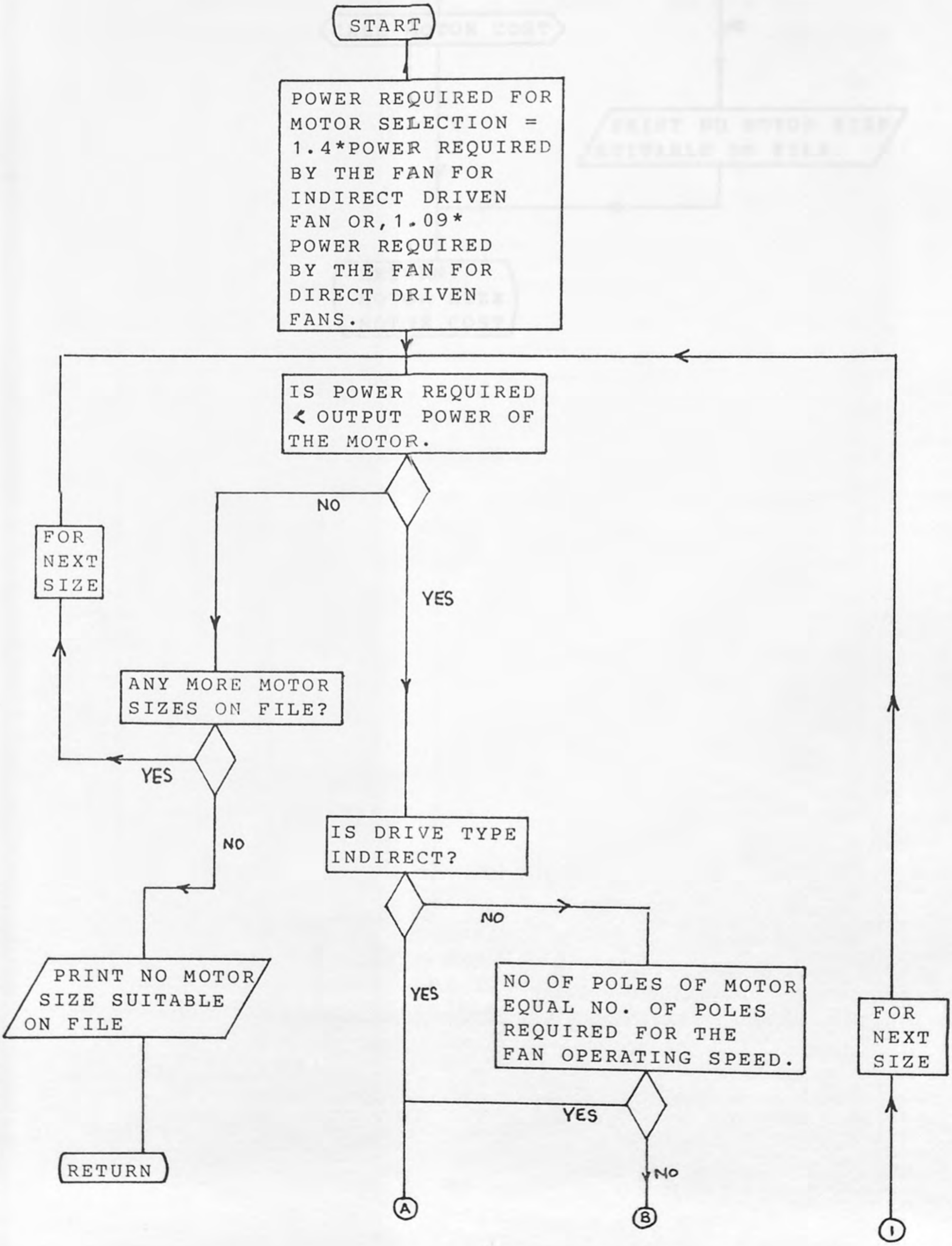
FUNCTION ROOT.

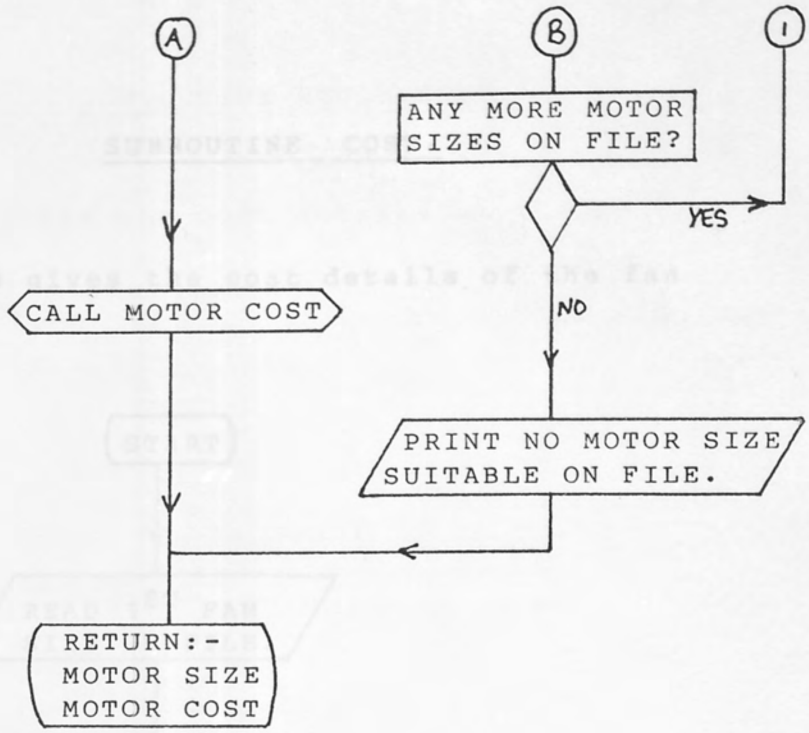
This function calculates the point of intersection of the fan characterstic (vloume-pressure) with the system resistance characteristic using the Newton - Raphson iteration method.



SUBROUTINE MOTOR DATA

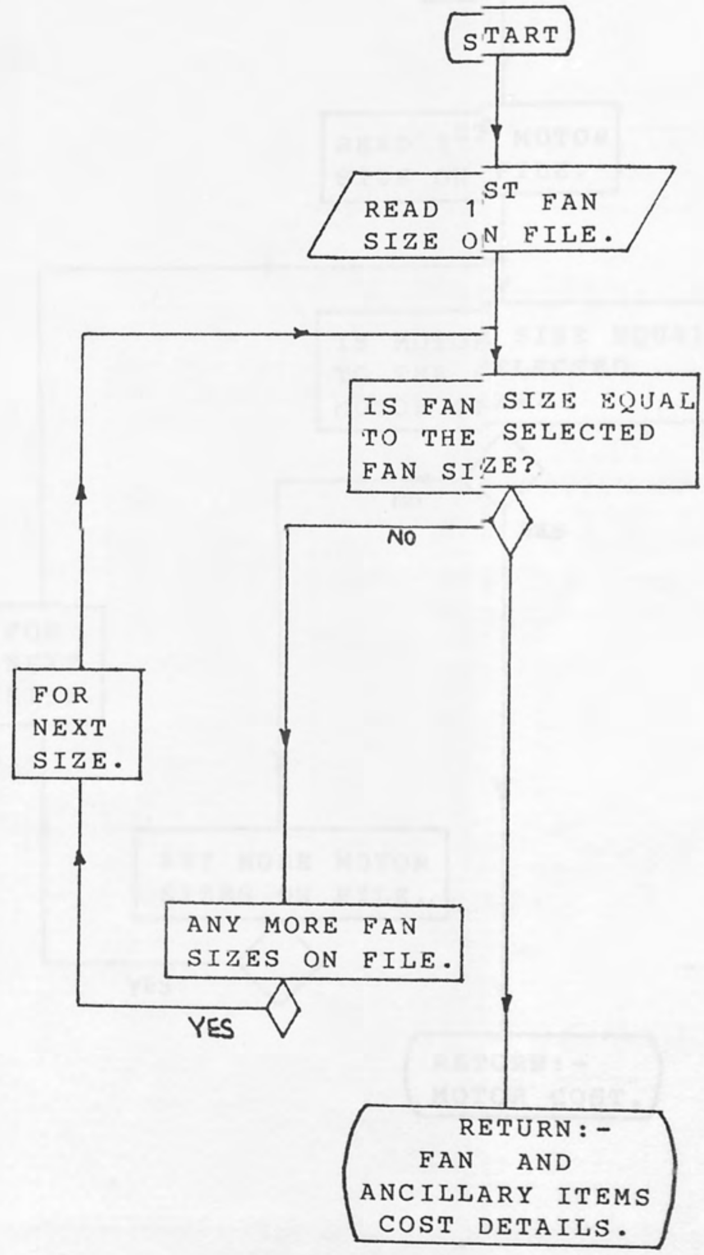
This subroutine finds the motor size for the selected fan size.




A
B
1
ANY MORE MOTOR SIZES ON FILE?
YES
NO
CALL MOTOR COST
PRINT NO MOTOR SIZE SUITABLE ON FILE.
RETURN:-
MOTOR SIZE
MOTOR COST

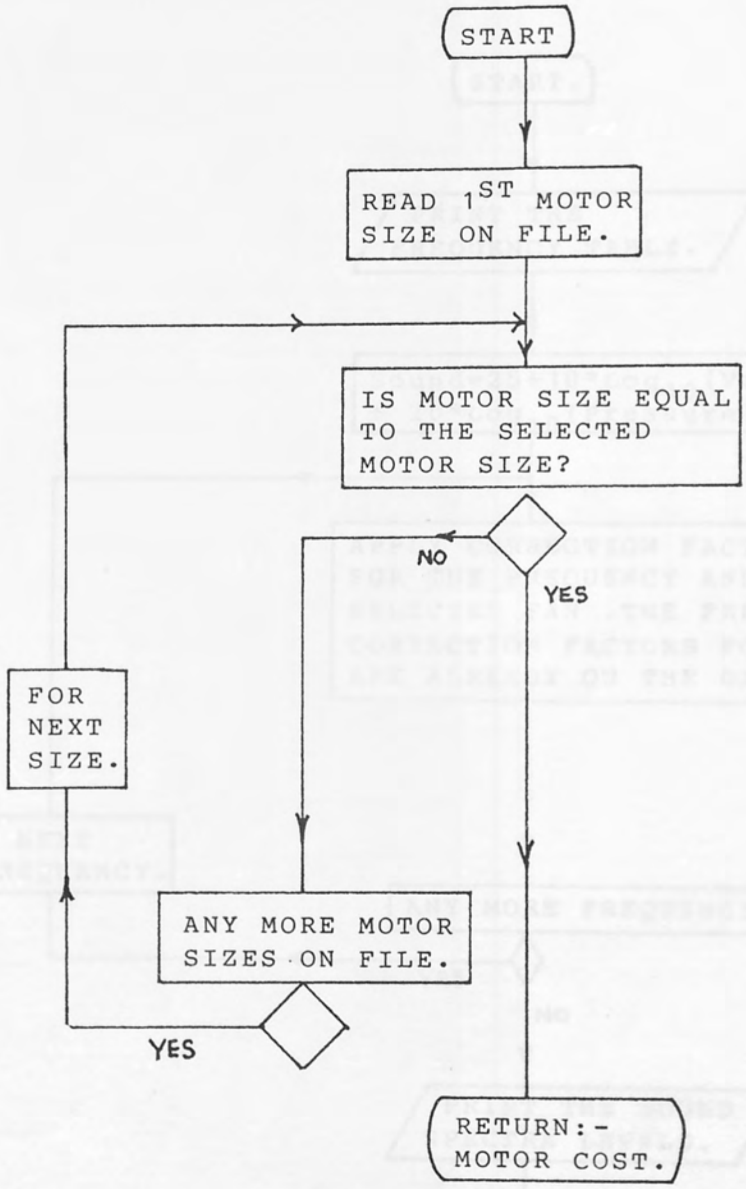
SUBROUTINE COST.

This subroutine gives the cost details of the fan size selected.



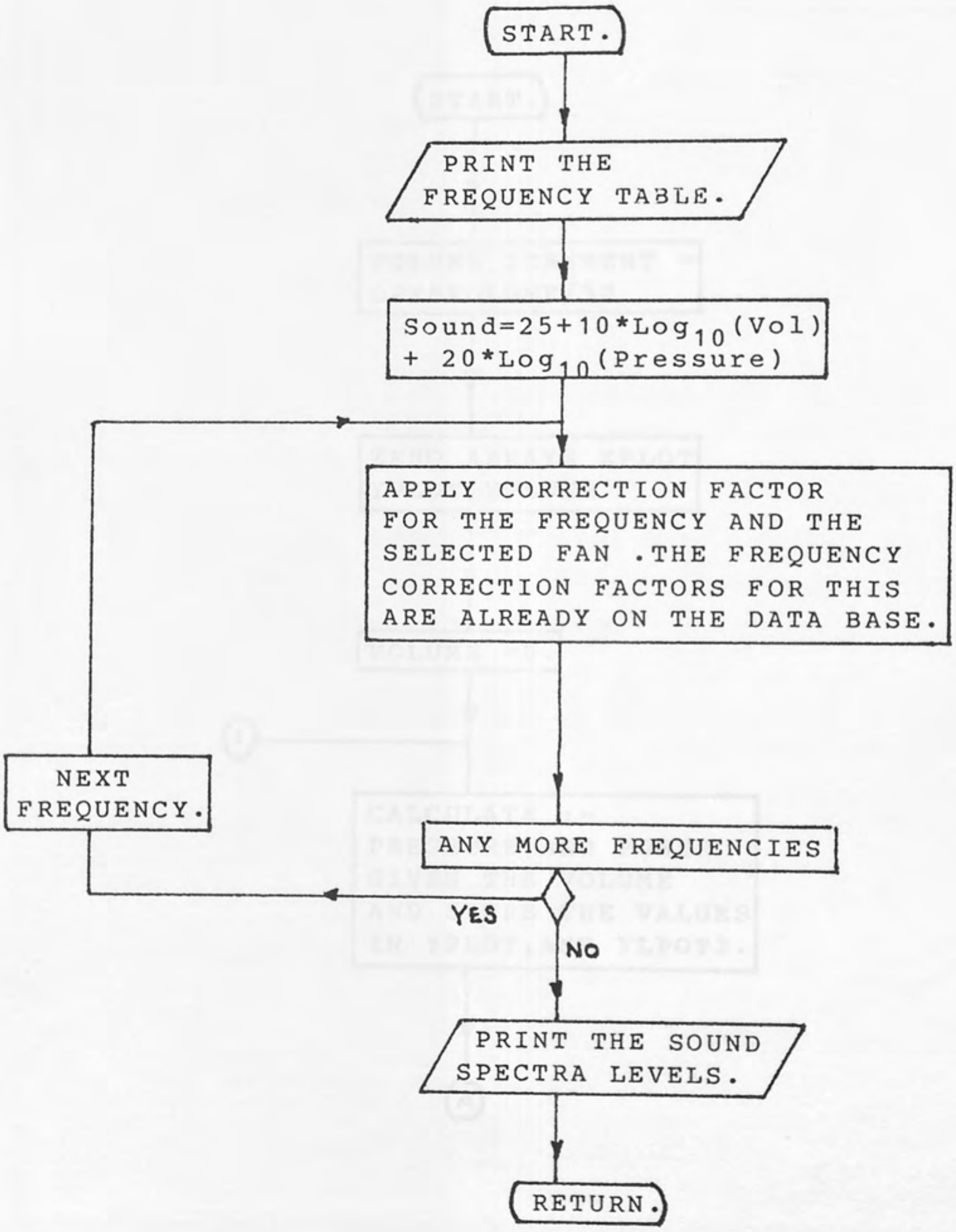
SUBROUTINE MOTOR COST.

This subroutine gives the cost details of the motor size selected.



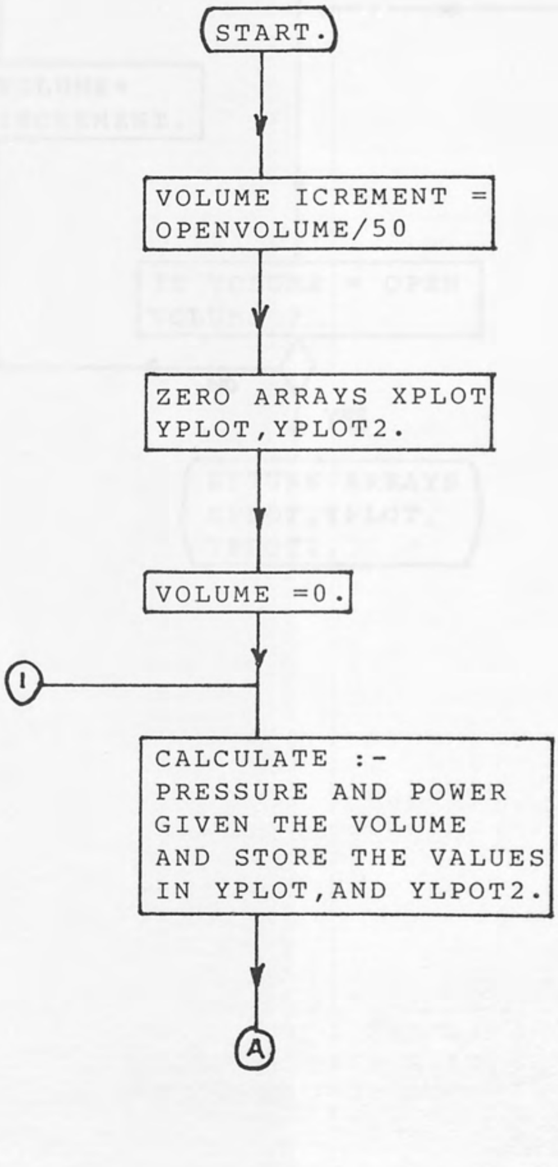
SUBROUTINE SOUND

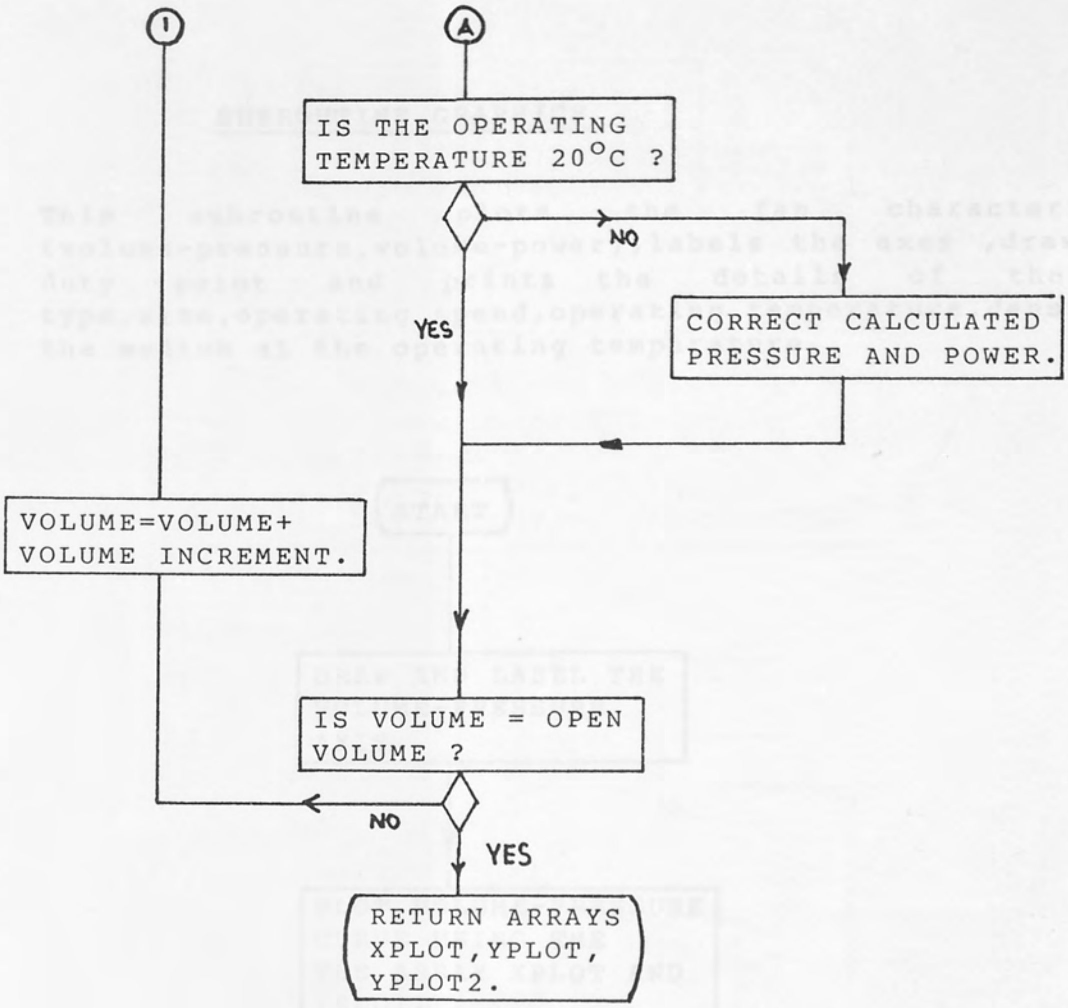
This subroutine calculates and prints the sound spectra levels at pre-determined fequencies for the selected fan size



SUBROUTINE PLOTTINGPOINTS.

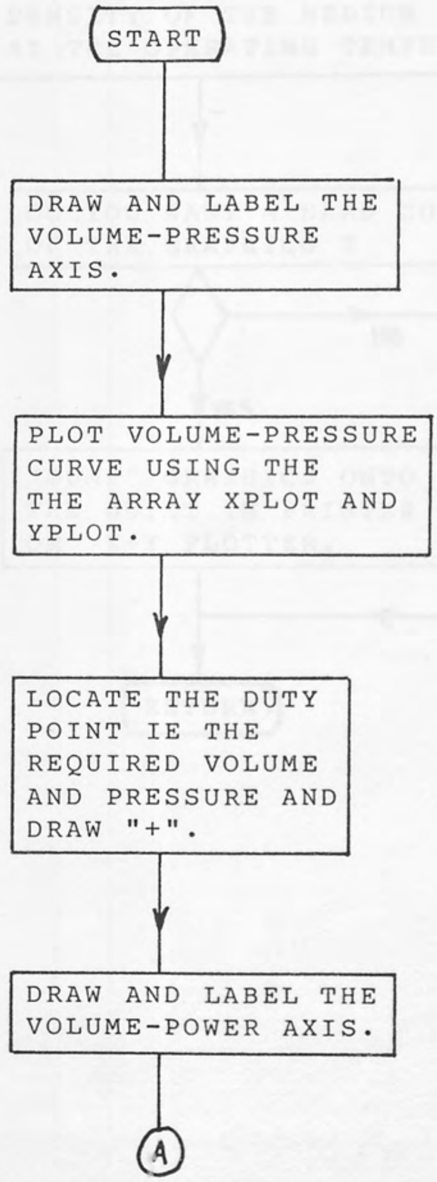
This subroutine calculates the points for the fan characteristics (volume-pressues,volume-power) at the operating temperature , and stores the values in the arrays xplot,yplot,yplot2, ready to be used by the subroutine 'GRAPHICS'.

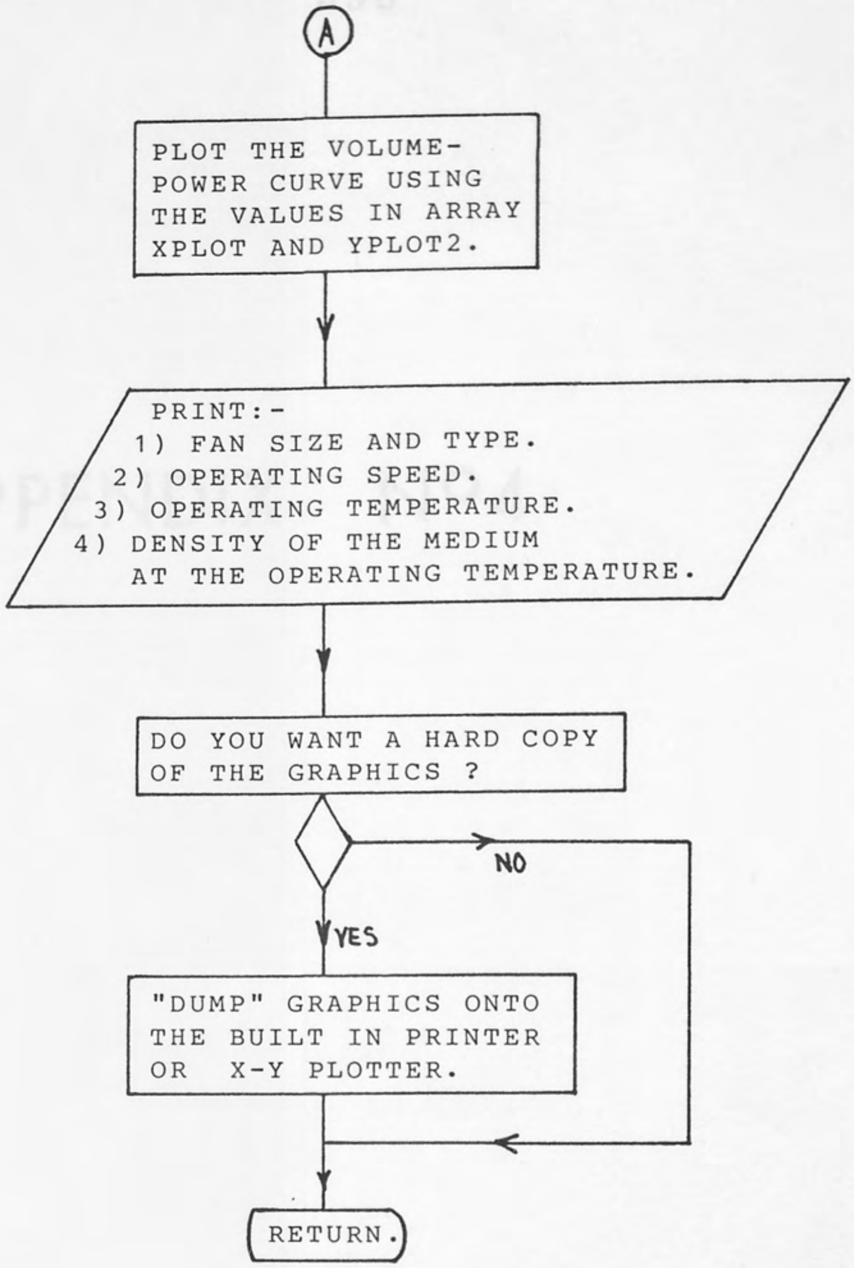



1
A
IS THE OPERATING TEMPERATURE 20°C ?
NO
YES
CORRECT CALCULATED PRESSURE AND POWER.
VOLUME=VOLUME+ VOLUME INCREMENT.
IS VOLUME = OPEN VOLUME ?
NO
YES
RETURN ARRAYS XPLOT, YPLOT, YPLOT2.

SUBROUTINE GRAPHICS

This subroutine plots the fan characteristics (volume-pressure,volume-power),labels the axes ,draws the duty point and prints the details of the fan type,size,operating speed,operating temperature,density of the medium at the operating temperature.




A
PLOT THE VOLUME-POWER CURVE USING THE VALUES IN ARRAY XPLOT AND YPLOT2.
PRINT:-
1) FAN SIZE AND TYPE.
2) OPERATING SPEED.
3) OPERATING TEMPERATURE.
4) DENSITY OF THE MEDIUM AT THE OPERATING TEMPERATURE.
DO YOU WANT A HARD COPY OF THE GRAPHICS ?
NO
YES
"DUMP" GRAPHICS ONTO THE BUILT IN PRINTER OR X-Y PLOTTER.
RETURN.

APPENDIX N°4

DIAGNOSTIC TEST

ERROR MESSAGES

Diagnostic test errors.

There are two kinds of errors, those that stop the machine immediately, because subroutine **DOCTOR** is called with a negative number (ie a fatal error) ; and those that allow the run to continue a little longer, in the hope of discovering further errors (ie a non fatal error).

FATAL ERRORS.

From subroutine **NURSE**:

1. There would appear to be no elements - **NELZ** is wrong.
2. The length of **NVEC**, as indicated by **LENVEC** and **INTEG**, is insufficient for subroutine **NURSE** even to read the element node numbers.
3. Something is negative on the first card.
4. The cards giving the element node numbers are jumbled, or one has perhaps been lost.
5. **LTYPE** cannot take this value (probably given by the associated number). There are not that many elements in the scheme.
6. **NDIM** may not be less than 1 or more than 4.
7. For some obscure reason **MAXRHS** is negative.
8. **NEWRHS** already exceeds the promised maximum, **MAXRHS**.

9. **NDF** appears to be zero - anyway, **NDFMAX** = 0.
10. There may be element property numbers, but there are no actual properties associated with them: **PROP** = 0.
11. **NFIX**, the number of nodes fixed in some way, cannot possibly exceed the total number of nodes.
12. Likewise, **NLOAD**, the number of loaded nodes, cannot ever exceed **NODMAX**.
13. Likewise, **NEXTIF**, the number of nodes with extra springs to earth, never exceeds **NODMAX**.
14. **LENVEC**, the dimension of **VEC**, has not been defined.
15. **INTEG** does not take the expected value of 1 or 2.
16. It is in fact zero, and the job aborts immediately.
17. There is not enough room in **VEC** to read the remaining data ,eq coordinates, fixities, etc.

From subroutine **DOCTOR**:

18. A sophisticated user has introduced a new and unacceptable error number. If the number is negative, the job aborts immediately.

From subroutine **INPUT**:

20. A card gives the coordinates for an inadmissible node number.
21. The coordinates for one of the nodes have been given twice.

22. The fixity cards are jumbled, or maybe one is lost.
23. The additional load cards are jumbled, etc.
24. The additional stiffness cards are jumbled, etc.
25. One of the additional stiffnesses is negative.
26. The element property cards are jumbled, etc.

From subroutine **MATRON**:

28. An element property number is inadmissible.
29. An element has an inadmissible node number.
30. An element has only zero nodes.
31. An element has a gap in its vector of node numbers.
32. A list of element properties may not be used for two different element types.
33. A vector **NDF** of the numbers of degrees of freedom at nodes, defined in **BLOCK DATA**, contains absurd or inadmissible values.
34. Or a vector of **NDF** contains a gap.
35. An element has no variables.
36. An unacceptable node is fixed.
37. The fixing code is negative.
38. The same node is fixed twice.
39. An unacceptable node has additional springs to earth.
40. We must not have two sets of extra springs for one node.
41. An unacceptable node has additional loads.
42. Only one set of loads can be applied to any one node.
43. The vector is not long enough even to accept the data.

44. The node has no variables. Presumably **LTYPE** is wrong.
45. A node has so many degrees of freedom at its first appearance and a different number later.
46. No right-hand sides.
47. Too many right-hand sides, in a later solution.
48. There is not enough core storage for the job to run. Increase **VEC** to the size given by the associated number, or much more is possible, and alter the assigned value of **LENVEC**. It may be found that runs with a very short buffer are prohibitively expensive. The user may choose to alter this diagnostic, depending on his system, so as to reject a nearly impossible job, or even to increase **LENVEC** automatically.

From subroutine **ELFILE**:

50. Some element requires more stresses at a point than were budgeted for in **MAXTRS**.
51. An element has a zero diagonal stiffness.
52. An element has a negative diagonal stiffness.

From the reduction in subroutine **FRONT**:

55. The code for fixing a node is too large.
56. The pivot is zero, and the arithmetic cannot continue.
57. The roundoff damage is unacceptable, even during the reduction.

From the back-substitution in subroutine **FRONT:**

60. At the end of the job, the roundoff damage makes the results worthless.

From subroutine **STRESS:**

62. When the data for stressing was retrieved, there was a contradiction in the number of stresses asked for.
63. As in 62, but the number of element variables is confused. If the number of stressing points **NBARLO** is inconsistent, it could explain both.

Non-fatal errors.

These are printed; then subroutine **DOCTOR** erases every trace, so that they will not be printed again, either during the same job or in any future job, in the same run.

From subroutine **NURSE:**

65. It is possible but unlikely that the biggest node number exceeds (number of elements) X (maximum number of nodes per element). For there might be large gaps in the sequence of nodes used.

66. The number of nodes per element is improbable. The ordinary beam in 3D would be an exception, with only 2 nodes.

67. We do not envisage more than 6 degrees of freedom at any node.

68. We expect 2D or 3D problems.

69. It is unlikely that a problem, apart from a large one using tetrahedra, will give a mean nodal valency of 10. This suggests that **NELZ** or **LNOMAX** is far too large, or **NODMAX** too small.

70. The number of physical properties for one element property number might exceed 10, but we prefer to call attention to it.

71. Even for a shell problem with elements of various thicknesses etc. 50 property numbers seems plenty.

72. In most 2D problems we must fix at least three variables, usually at two different nodes: in 3D, six variables at three different nodes.

73. Is **LENVEC** reasonable? **VEC** seems either absurdly small or large.

From subroutine **MATRON:**

80. Two nodes have identical coordinates.

81. A list of element properties was never used.

82. A fixing code number is zero: eccentric but admissible.

83. Repeated nodes in an element might cause trouble.

84. A node number is not used. This may well be intentional.

85. However, its coordinates are nonzero; is this a mistake or merely wasted effort?

86. An unused node is fixed: this is probably a mistake.

87. An unused node has an extra stiffness to earth.

88. An unused node has an extra point load.

From the reduction in subroutine **FRONT:**

90. A nonzero prescribed value was not used. Such rogue values appear as garbage amongst the reactions, but are not otherwise harmful.

91. It seems pointless to load a prescribed value. The extra load merely adds to the reaction.

92. It is equally strange to add a spring to earth, when the variable concerned is prescribed.

93. One of the pivots was negative.

94. Possible roundoff damage. The printed values may enable the user to identify variables that contribute overwhelmingly to the diagonal energy.

95. Frontwidth prematurely zero - two completely separate structures.

From the back-substitution in subroutine **FRONT:**

97. For some reason, all the non-prescribed deflections are zero.

98. The strain energy seems abnormally small.

99. It was worth continuing the run: the final roundoff damage is mild.

From subroutine **HALOOF**:

1. This is not a Semi-Loof shell. The number of nodes or of variables is wrong.
2. The thickness at a node cannot be negative or zero.
3. A node number is repeated in the same element.
4. Two nodes in the element have identical coordinates.
5. A midside node is not central enough.
6. A side is excessively curved.
7. The area Jacobian is zero.
8. ∂(position)$/\partial\xi$ or ∂(position)$/\partial\eta$ is zero.
9. The normals at the two Loof nodes are too divergent - the element is too curved.
10. The thickness at an integrating point is negative.
11. Subroutine **SFR** is not allowed to process a point lying outside the element.

APPENDIX Nº5

E Q U I P M E N T

U S E D

A

THE COMPUTER AIDED SELECTION SYSTEM.

1). For the development of the system:-

a). Hewlett Packard 9830 desk top micro-computer.

b). Hewlett Packard 9845 desk top micro-computer.

c). Hewlett Packard flat-bed X-Y pen plotter.

2). To assess the response of the Technical Sales staff to the developed system:-

a). Hewlett Packard 9845 desk top micro-computer.

b). Tektronix 4054 desk top micro-computer.

c). Tektronix hard copy unit.

3). Implementation of the Computer Aided Selection System at Alldays, Peacock & Co. Ltd.

a). Prime model 450 mini-computer with 1/2 M byte main memory and 64 M byte of disc storage.

b). GT2000 Graphics Terminal.

c). Calcomp model 81 X-Y 8 pen plotter.

d). Digital Electrical Corporation model LA34 printer terminal.

B EXPERIMENTAL STRESS ANALYSIS OF THE IMPELLER.

1). Commercially produced 650 mm diameter "BL" type centrifugal impeller.

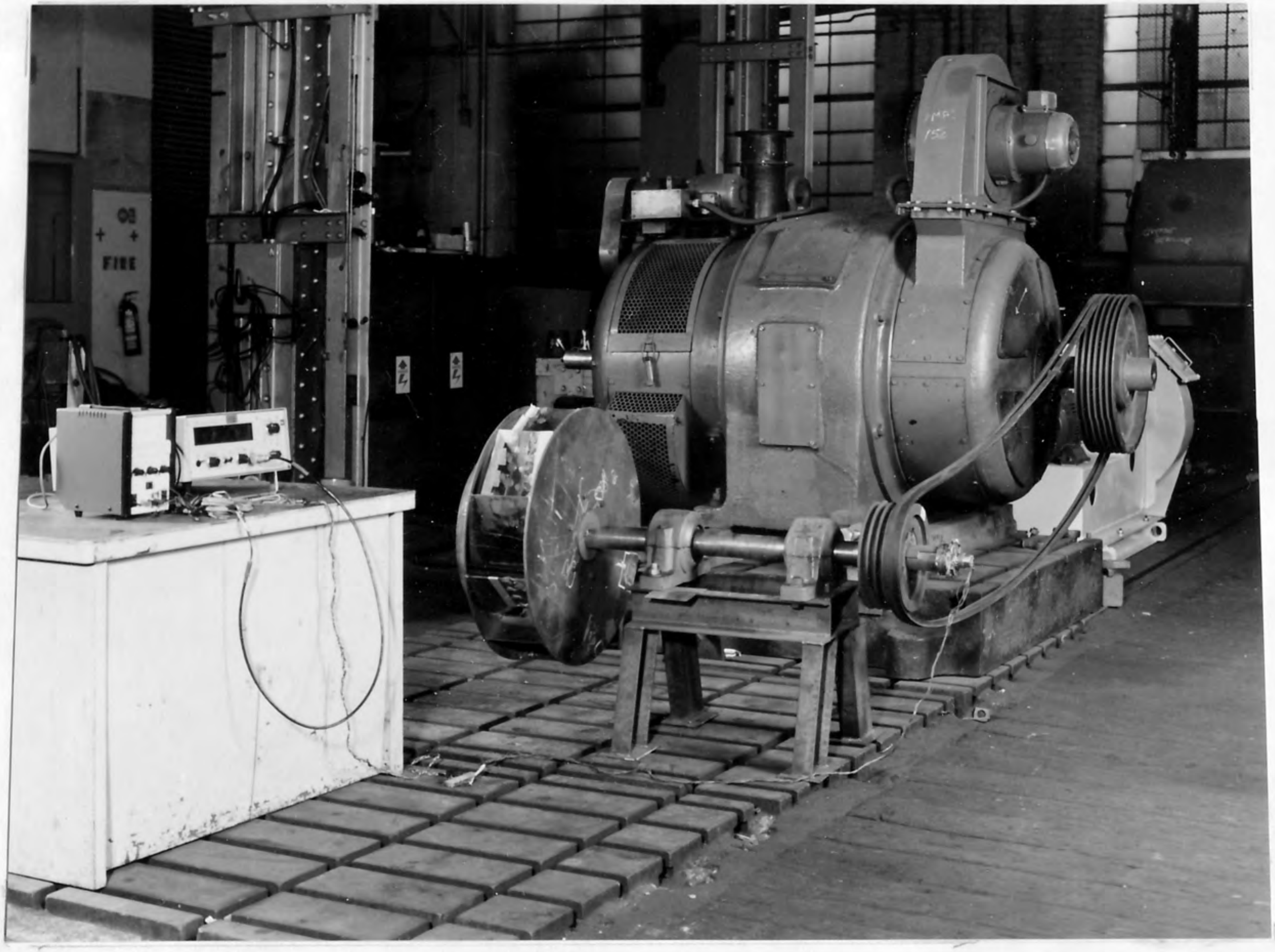
2). Thompson-Hudson 116 H.P. variable speed electric motor.

3). Michigan Scientific 10 way slip ring.

4). Micro-Measurements 6 mm 120 Ω linear strain gaugues.

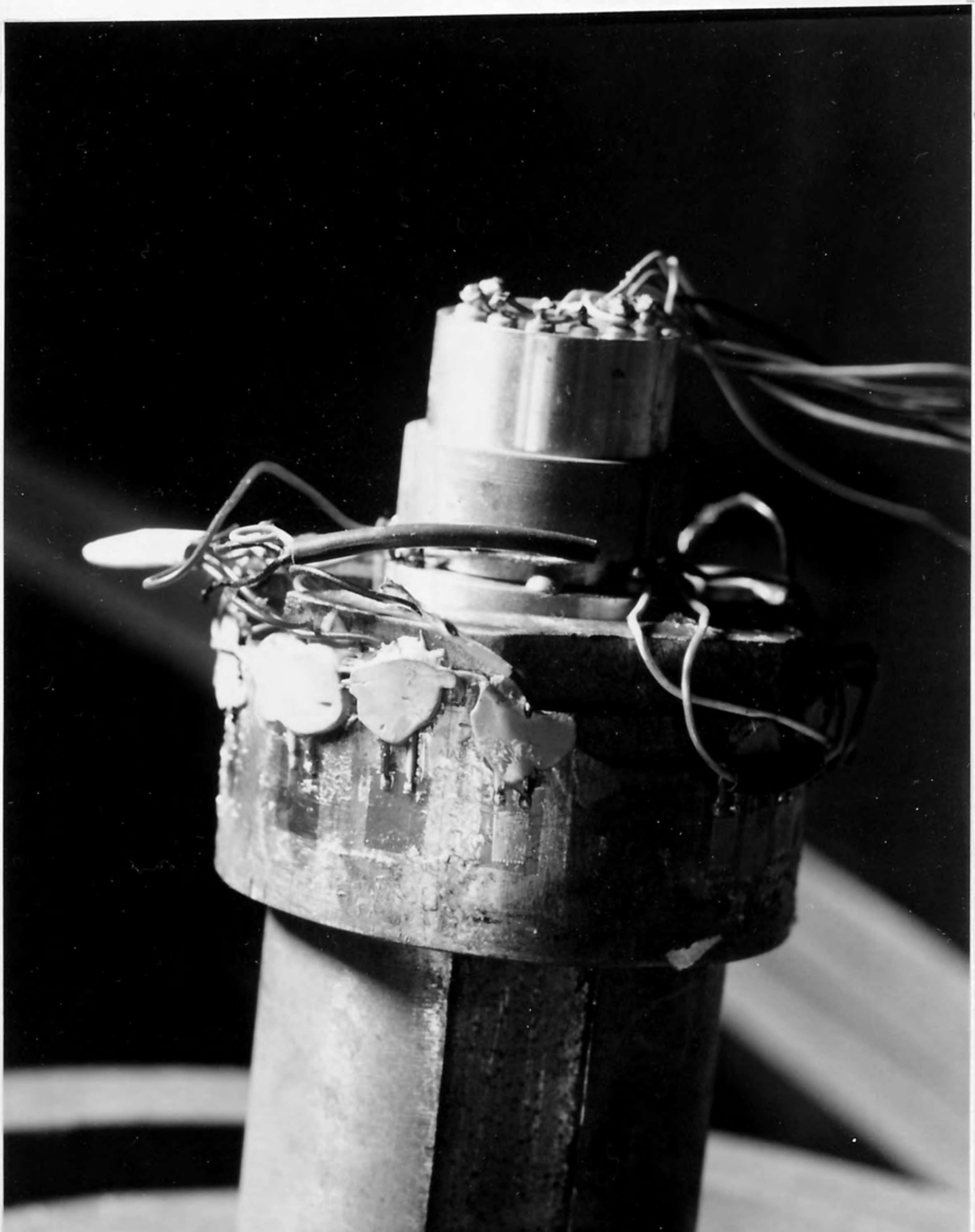
5). Farnell Instrument Ltd stablised dc power supply (0-50 volts, 0-.5 amps).

6). Solartron LM 14204 digital volt-meter.



EXPERIMENTAL RIG LAY OUT

LOCATION OF THE STRAIN GAUGES ON THE BLADES



DUMMY GAUGES ON THE SLIP RING ADAPTER

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