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T SPARKS

**OPERATIONAL CHARACTERISTICS OF
ROTOR/STATOR MIXERS**

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ABSTRACT

Fast chemical reactors require rapid mixing, and literature shows that rotor/stator mixers can provide this. Results have been published, at small scale, showing rotor/stator mixers to be promising as chemical reactors.

An industrial size rotor/stator mixer has been procured and fitted into a test loop. Instrumentation was used to measure shaft torque, shaft speed, differential pressure and differential temperature. Measurements were used to produce an energy balance for the mixer that could be used to assess its mixing performance, based upon general descriptions of mixing.

Residence time distribution (RTD) curves were obtained and it was shown that the mixer exhibited substantial backmixing. Modifications to the mixer are proposed that may produce a RTD that better approximates plug flow.

It is concluded that if the mixer could be made to better approach plug flow, as mixing rates would be very high, use of such a device could improve the product quality from a competitive chemical reaction.

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CONTENTS

CHAPTER 1.	INTRODUCTION	1
CHAPTER 2.	LITERATURE REVIEW AND OBJECTIVES	3
2.1	INTRODUCTION	3
2.2	IN-LINE ROTOR/STATOR MIXERS	4
2.2.1	Industrial applications	6
2.3	MIXING PROCESSES	8
2.3.1	Macromixing	8
2.3.2	Micromixing	9
2.3.3	Reactive mixing	11
2.4	FLOW PATTERN	13
2.5	RESIDENCE TIME AND RESIDENCE TIME DISTRIBUTION	16
2.6	ENERGY BALANCE IN A ROTOR/STATOR MIXER	19
2.6.1	P_{SHAFT} , shaft power input	21
2.6.2	P_{PUMP} , pumping power output	24
2.6.3	P_{DISK} , disk friction losses and P_{MECH} , mechanical losses	24
2.6.4	P_{LEAK} , leakage losses	25
2.6.5	$P_{\text{HYDRAULIC}}$, hydraulic losses	25
2.7	MIXING PROCESSES IN ROTOR/STATOR MIXERS	27
2.7.1	Macromixing	27
2.7.2	Micromixing	27
2.8	CONCLUSIONS	28
2.9	OBJECTIVES FOR THIS STUDY	30
CHAPTER 3.	EXPERIMENTAL DETAILS	31
3.1	INTRODUCTION	31
3.2	EXPERIMENTAL RIG	31
3.2.1	Band of Operation	35
3.3	ENERGY BALANCE - BACKGROUND	35
3.3.1	P_{SHAFT}	36
3.3.2	P_{PUMP}	36
3.3.3	P_{DISK}	37
3.3.4	P_{MECH}	37

3.3.5	P_{LEAK} and $P_{\text{HYDRAULIC}}$	38
3.4	RESIDENCE TIME DISTRIBUTIONS - BACKGROUND	38
3.5	DATA ACQUISITION AND ANALYSIS	39
3.6	INSTRUMENTATION	40
3.6.1	Shaft speed	40
3.6.2	Shaft Torque	42
3.6.3	Differential Pressure	42
3.6.4	Differential Temperature	43
3.6.5	Residence time distribution	43
3.7	EXPERIMENTAL METHOD	46
3.7.1	Energy balance	46
3.7.2	Residence time distribution	47
CHAPTER 4.	EXPERIMENTAL RESULTS	48
4.1	INTRODUCTION	48
4.2	ENERGY BALANCE	48
4.2.1	P_{SHAFT} , shaft power input	48
4.2.2	P_{PUMP} , pumping power output	52
4.2.3	P_{MECH} , power due to mechanical losses	55
4.2.4	Differential temperature	55
4.3	RESIDENCE TIME DISTRIBUTION	56
4.4	CONCLUSIONS	59
CHAPTER 5.	ANALYSIS AND DISCUSSION OF RESULTS	60
5.1	INTRODUCTION	60
5.1.1	Commissioning	60
5.2	ENERGY BALANCE	61
5.2.1	P_{SHAFT} and shaft torque	61
5.2.2	P_{PUMP} and differential pressure	64
5.2.3	P_{DISK} , disk friction	67
5.2.4	P_{MECH} , Mechanical losses	69
5.2.5	P_{LEAK} , leakage losses	70
5.2.6	$P_{\text{HYDRAULIC}}$, hydraulic losses	71
5.2.7	Differential temperature	71
5.3	OVERALL ENERGY BALANCE	73
5.4	RESIDENCE TIME DISTRIBUTION	76
5.4.1	General observations	76

5.4.2	Modelling of the residence time distribution curves	78
5.4.3	Modifications to improve the residence time distribution	79
5.5	MIXING IN THE ROTOR/STATOR MIXER	81
5.5.1	Macromixing	81
5.5.2	Micromixing	82
5.6	CONCLUSIONS	82
CHAPTER 6.	CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK.	84
6.1	CONCLUSIONS	84
6.2	RECOMMENDATIONS FOR FUTURE WORK	85
REFERENCES		87
APPENDIX A	EXPERIMENTAL DATA	
APPENDIX B	CALIBRATION DETAILS	
APPENDIX C	EXPERIMENTAL EQUIPMENT DETAILS	
APPENDIX D	DRAWINGS OF ROTOR AND STATOR	

FIGURES

	Page
1.1 An in-line rotor/stator mixer	1
2.1 Toothed ring rotor and stator	4
2.2 Rotor/stator mixer heads (courtesy Silverson)	5
2.3 Multi ringed rotor/stator mixer	6
2.4 Multi stage rotor/stator mixer	7
2.5 Inlet recirculation (Yedidiah, 1985)	14
2.6 Volute recirculation (Copley et al., 1962)	15
2.7 Residence time distribution curve for a rotor/stator mixer, taken from Studer (1990)	17
2.8 Residence time distribution curve for a centrifugal pump, taken from Bolzern and Bourne (1985)	18
2.9 Power balance for a double suction centrifugal pump (Stepanoff, 1957)	20
2.10 The Oakes mixer	22
3.1 Schematic diagram of test rig	32
3.2 Photograph of test unit	33
3.3 Rotor and stator used	34
3.4 Instrumentation	41
3.5 Schematic diagram of injector	44
3.6 Schematic diagram of conductivity probe	45
4.1 Shaft torque, Λ , against flow rate Q , at 1000 rpm	49
4.2 Λ against Q for a range of shaft speeds	50
4.3 P_{SHAFT} against flow rate	51
4.4 Δp against flow rate at 2000 rpm	53
4.5 Δp against flow rate at 500 rpm	53
4.6 Δp against flow rate	54
4.7 P_{PUMP} against flow rate	54
4.8 Differential temperature against flow rate	56
4.9 Normalised residence time distribution curve at $N = 500$ rpm. $Q = 1.5$ l/s	57
4.10 Background concentration against test number	58

	Page
5.1 Shaft torque against shaft speed	62
5.2 Shaft torque against shaft speed	62
5.3 Shaft power input against shaft speed	63
5.4 Shaft power input against shaft speed	63
5.5 Pumping power against shaft speed	66
5.6 Pumping power against shaft speed	66
5.7 Pumping efficiency against shaft speed	67
5.8 Overall energy balance for the mixer at 2000 rpm	73
5.9 Overall energy balance for the mixer at 2500 rpm	74
5.10 Residence time distribution curve (1500 rpm, 1 litre/ second) showing comparison with models	80
5.11 Residence time distribution curve (2000 rpm, 1.5 litres/second) showing comparison with models	81

TABLES

3.1	Range of shaft speeds	46
3.2	Range of flow rates	46
4.1	Power drawn due to mechanical (seal and bearing) losses	55
5.1	Disk friction calculated assuming thin disk and including thickness	68

NOMENCLATURE

Arabic symbols

c_i	concentration of substance i	mol m^{-3}
c_s	specific heat capacity	$\text{J kg}^{-1} \text{K}^{-1}$
C	normalised concentration	-
D	rotor diameter	m
	diffusivity	$\text{m}^2 \text{s}^{-1}$
E	engulfment rate	s^{-1}
	rotor thickness	m
K	constant	-
L_s	initial scale of segregation	m
N	rotational speed	s^{-1}
P_{SHAFT}	shaft power input	W
P_{PUMP}	pumping power output	W
P_{DISK}	power dissipated by disk friction	W
P_{MECH}	power dissipated by mechanical losses	W
P_{LEAK}	power dissipated by leakage flows	W
$P_{\text{HYDRAULIC}}$	power dissipated by hydraulic losses	W
Q	flow rate	$\text{m}^3 \text{s}^{-1}$
R	reaction rate	$\text{mol m}^{-3} \text{s}^{-1}$
t	time	s
t_m	mixing time	s
t_D	diffusive mixing time	s
t_E	engulfment time	s
t_{MACRO}	macromixing time	s
V_{ei}	volume of eddies containing substance i	m^3

Dimensionless groups

$Po = P_{\text{SHAFT}} / \rho N^3 D^5$	Power number	-
$Re = ND^2/\nu$	Reynolds number	-
$Sc = \nu/D$	Schmidt number	-

Greek symbols

α	element age	s
γ	an exponent	-
Δp	differential pressure	Pa
ΔT	differential temperature	K
ϵ	turbulent energy dissipation rate	W kg ⁻¹
η	pumping efficiency	-
λ_k	Kolmogoroff microscale	m
Λ	shaft torque	N m
ν	kinematic viscosity	m ² s ⁻¹
ρ	density	kg m ³

CHAPTER 1. INTRODUCTION

A rotor/stator mixer incorporates a rotor spinning near to a motionless stator. This thesis deals specifically with in-line rotor/stator mixers, of the type shown in Figure 1.1. These mixers look like centrifugal pumps, and are installed in the same way.

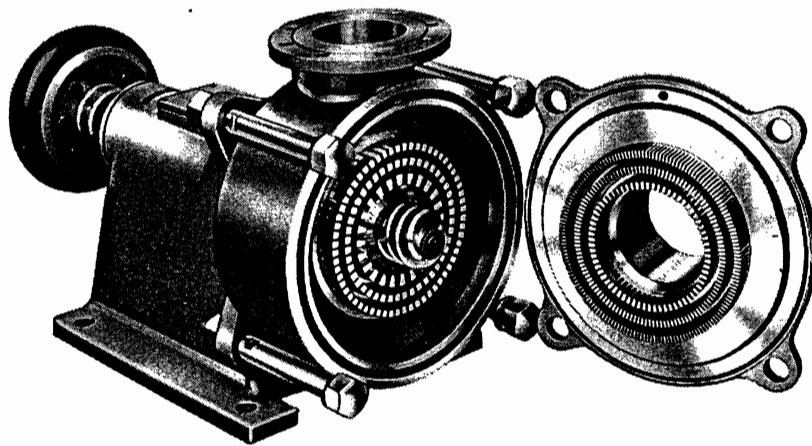


Figure 1.1 An in-line rotor/stator mixer

A number of companies produce this type of mixer; the majority are based in Europe, particularly in the UK, Germany and Switzerland. In-line rotor/stator mixers are perceived as brute force mixers. They are a successful product and are used in a wide variety of applications, for example mixing additives into bitumen, mixing batters or homogenising chocolate.

This thesis is an account of a study that has begun to look at the use of in-line

rotor/stator machines for the mixing of chemically reactive streams. Rotor stator mixers are compact, high intensity mixing devices; it is thought that their use could improve process efficiency.

The chemical and process industries are, more than ever, under commercial, environmental and legal pressure to increase efficiency. An efficient chemical process will use less energy and raw material, producing less waste.

The thesis includes a review of relevant literature, highlighting areas where more information is needed. The review is completed by a statement of the objectives for the work contained in this thesis.

The experimental equipment, methods and instrumentation used are described, and experimental data presented. Analysis and discussion of the results is provided, and an assessment of the performance of rotor/stator mixers as chemical reactors given.

The overall conclusions of this thesis and recommendations for future work in the field of rotor/stator mixing are given in the final chapter.

Wherever possible figures and tables are positioned close to where they are referred to in the text.

CHAPTER 2. LITERATURE REVIEW AND OBJECTIVES

2.1 INTRODUCTION

In this chapter relevant literature will be reviewed, and conclusions drawn about the performance of rotor/stator mixers. The purpose of this review is to determine what work has been conducted to date and where more information is required.

Very little work on rotor/stator mixing has been published. Literature searches using two separate on-line data-base search facilities and a CD-ROM located few relevant references. Because of the limited amount of specific literature, references relating to centrifugal pumps were also reviewed.

This chapter begins by describing rotor/stator mixers. Mixing processes are then described generally. Following this, characteristics of rotor/stator mixers (along with centrifugal pumps where relevant) are reviewed. This is to establish what the present level of knowledge can reveal about the suitability of rotor/stator mixers as chemical reactors, and to highlight what information is required to fill gaps in this knowledge.

A set of conclusions from the available literature follow, and the chapter is completed by a statement of the objectives for this study.

2.2 IN-LINE ROTOR/STATOR MIXERS

Many companies manufacture in-line rotor/stator mixers: a number of different designs are therefore available. All have the same essential components: a high speed rotor spinning close to a motionless stator. Probably the most common design is the toothed ring, shown in Figure 2.1. This design is typical of those offered by manufacturers in Germany and Switzerland. Many other designs are also available, for example see Figure 2.2.

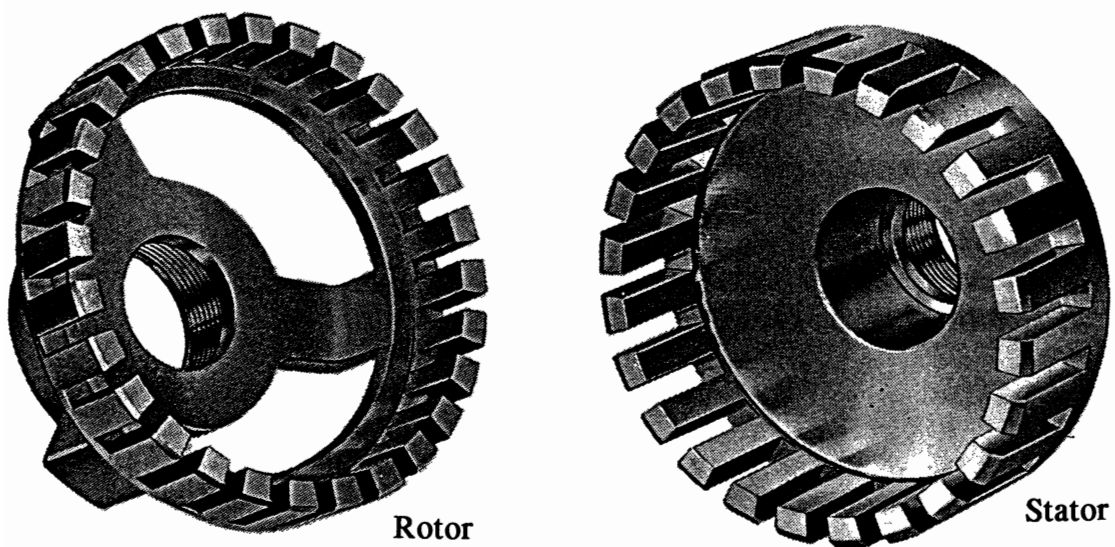


Figure 2.1 Toothed ring rotor and stator

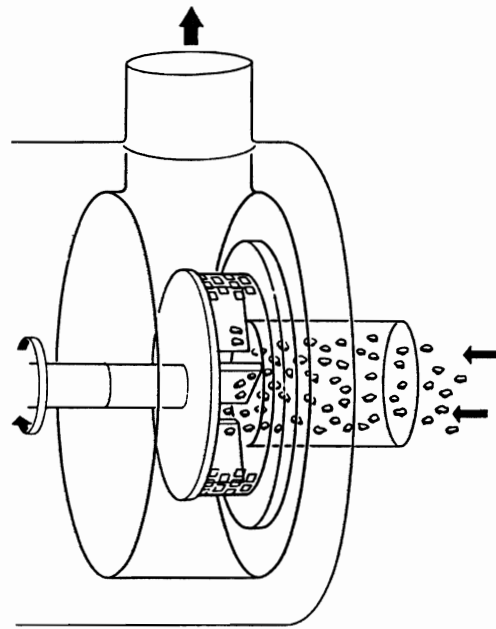


Figure 2.2 Rotor/stator mixer heads (courtesy Silverson)

Rotor/stator mixers are often used to mix thick, viscous liquids, or to disintegrate solid particles.

All designs have in common a narrow gap between rotor and stator, conventionally known as the "shear gap". This is believed to be a region of intense shear and most manufacturers' catalogues give it a great deal of attention. This is where any solid particles in the through flow are ground to a powder and is believed to be the location of the most intense mixing.

Multi-ringed rotor/stator mixers, see Figure 2.3, have more than one ring of teeth or blades. These force the fluid to pass through more than one shear gap. The

size of the teeth and gaps between teeth can be varied from ring to ring.

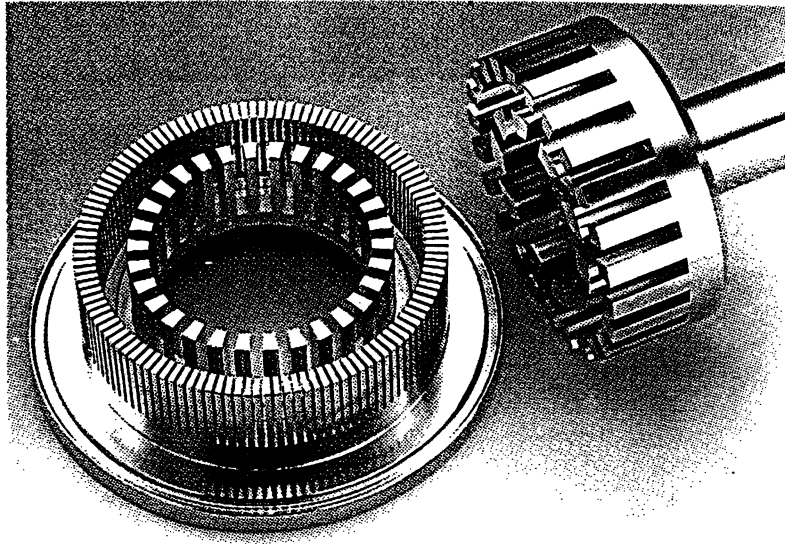


Figure 2.3 Multi ringed rotor/stator mixer

Multi-stage rotor/stator mixers, see Figure 2.4, have more than one rotor/stator pair on the mixer shaft, in a similar way to multi-stage pumps. This provides similar benefits to using multi-ringed mixers, and also improves pumping performance. It is possible to use different types of rotor/stator pairs for each stage, and it is possible to have a centrifugal pump impeller in a pump volute as one of the stages.

2.2.1 Industrial applications

One of the principal uses of rotor/stator mixers is in the food industry; any fluid that can be pumped can be mixed by an in-line rotor/stator mixer and examples

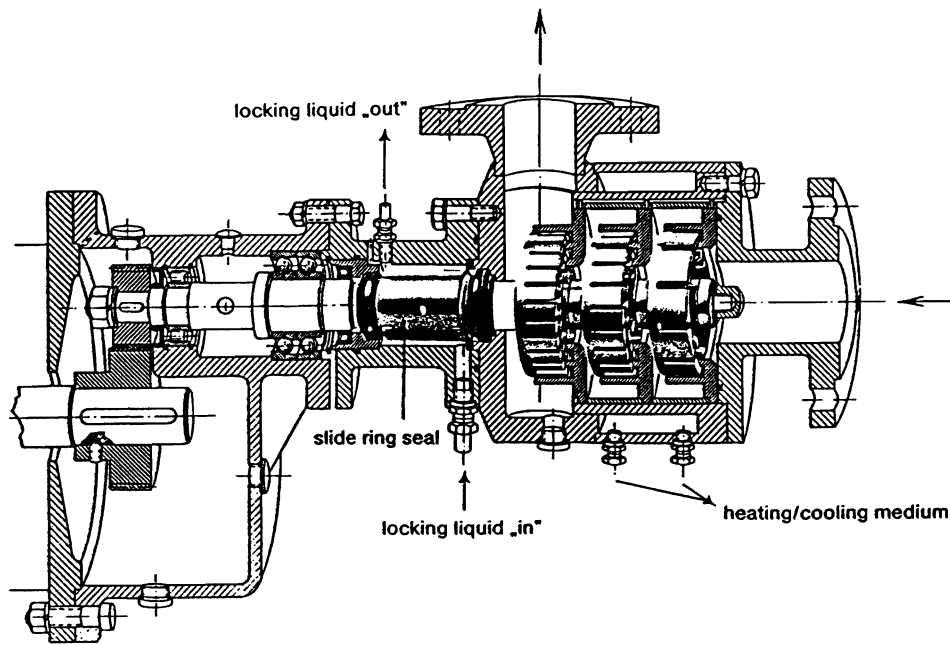


Figure 2.4 Multi stage rotor/stator mixer

include marshmallow, cake mixture and soft drink concentrates (Bouette, 1974 and Holley & Weisser, 1982). Secondary sewage sludge treatment is another application that has been proposed (Praus, 1991). Rotor/stator mixers can also be used to reduce the size of solid particles in a slurry, and have been used to crumb used car tyres.

The industrial application of rotor/stator mixers to chemical reactions with low viscosity reagents is comparatively rare (Rothman 1993).

2.3 MIXING PROCESSES

This section describes mixing processes generally, and will be used as a basis upon which to make an assessment of the mixing performance of rotor/stator mixers.

It is conventional to consider mixing as a whole in terms of two processes, termed "macromixing" and "micromixing".

2.3.1 Macromixing

The reduction in the variation of a contaminant concentration or temperature on a coarse scale is termed macromixing and can be thought of as resulting from bulk flow patterns and turbulent diffusion (Hannon, 1990). Provided flow is fully turbulent the macromixing rate is independent of viscosity. Only the scale of segregation (or un-mixedness) is reduced by macromixing; to enable molecules that are initially apart to meet (and to react) requires micromixing (Section 2.3.2).

A time constant for macromixing in an isotropic turbulent mixer was derived by Corrsin (1964):

$$t_{\text{macro}} = (L_s^2 / \epsilon)^{1/3} \quad (2.1)$$

where L_s is the initial scale of segregation and is of the same order as a characteristic dimension of the reactor. Equation (2.1) has been qualitatively

validated by several sets of experiments in batch reactors (e.g. Grenville, 1988). It is not clear how relevant this time constant will be in a rotor stator mixer, where the turbulence is inhomogeneous (i.e. depends upon position in the mixer), but an important point to note is the dependence on ε , the turbulent energy dissipation rate.

2.3.2 Micromixing

Micromixing is the term used to describe the final, small scale, stages of mixing. In turbulent flow this is believed to proceed through the formation of very fine laminated structures, brought about by fluid engulfment within the smallest turbulent energy dissipating eddies. This precedes molecular diffusion and reaction. Micromixing occurs at scales below the smallest turbulent eddy size, the Kolmogoroff micro-scale, λ_k defined as:

$$\lambda_k = \left(\frac{v^3}{\varepsilon} \right)^{1/4} \quad (2.2)$$

A characteristic half-life time for micromixing is the diffusive mixing time (Baldyga and Bourne, 1984) (t_d):

$$t_d \approx 2 \sqrt{\frac{v}{\varepsilon}} \operatorname{arcsinh}(0.05Sc) \quad (2.3)$$

Fluid engulfment is often rate-determining (Baldyga and Bourne, 1989) and the

appropriate characteristic half-life time (t_E) is then:

$$t_E \approx 17 \left(\frac{\nu}{\epsilon} \right)^{1/2} \quad (2.4)$$

A complete description of the flow pattern and turbulent energy levels for a reactor would be sufficient to predict the results of chemical reactions, if combined with good models for macro and micromixing. A description as complete as that cannot, yet, be provided for chemical reactors of any fluid mechanical complexity. Fortunately, if certain assumptions are made and estimates provided for the energy dissipation rate, a great deal can still be said about the relative merits of different reactors; a good quantitative prediction of the outcome of chemical reactions can also be made, see for example Bolzern & Bourne (1985).

Baldyga and Bourne (1984) proposed a model for micromixing, known as the engulfment-deformation-diffusion (EDD) model. Use of this model required the numerical solution of a set of partial differential equations, and therefore substantial computational effort was needed. Later Baldyga and Bourne (1989) proposed simplifications to this model, having identified the engulfment stage as rate determining under certain conditions; use of this model required the solution of a set of ordinary differential equations. The essence of this model is given here, so that again the importance of the turbulent energy dissipation rate can be shown.

The growth of eddies of reactant feed substance i is modelled:

$$\frac{dV_{ei}}{d\alpha} = E V_{ei} \quad (2.5)$$

where V_{ei} is the volume of eddies containing substance i , α is the element age (the time since it entered the reactor) and E is the engulfment rate (note that the engulfment rate is sensitive to ϵ):

$$E = 0.05776(\epsilon / \nu)^{1/2} \quad (2.6)$$

A mass balance on substance i inside and outside of the expanding reaction zone gave the E model:

$$\frac{dc_i}{d\alpha} = E(<c_i> - c_i) + R_i \quad (2.7)$$

where $<c_i>$ denotes the mean concentration of species i outside the reaction zone.

Micromixing models can also be used "in reverse" to evaluate the energy dissipation rates in a reaction zone, by examining the outcome of a reaction and comparing with model predictions.

2.3.3 Reactive mixing

The rate of a chemical reaction may depend upon both the reaction kinetics and the rate of mixing. Chemical reactions are classified as being slow, fast or instantaneous. If the mixing half-life time, t_M , (for example t_D or t_E) is comparable to the reaction half-life time, t_R , then the reaction is labelled "fast". Reactions where $t_M < t_R$ and $t_M > t_R$ are termed "slow" and "instantaneous" respectively.

Mixing will not severely affect the economics of simple (one-step) reactions, for example some neutralisations, when there is no great penalty if the reaction takes (say) 2 rather than 1½ seconds.

However, with competing (multiple) reactions the mixing rate can affect the final product distribution or yield. Examples of these reactions are azo-couplings, polymerisations and nitrations. Poor mixing can have undesirable economic and ecological consequences. This is illustrated well by the following reaction scheme (Levenspiel, 1972):



The distribution of products may display sensitivity to mixing conditions. If, for example, the first (desirable) reaction is fast and the second (undesirable) is slow then if B is fed slowly into A and then rapidly mixed it reacts to form R and R is

carried away.

However if mixing is insufficiently rapid, when **R** is formed it will not be carried away from the reaction zone and some will combine again with **B** to form **S**. The distribution of products is thus mixing-sensitive.

If **R** is a high value product and **S** is a worthless waste product then it becomes necessary to maximise the yield of **R**. This will improve the economics of the process, producing more **R** for given quantities of raw material and reducing the need for potentially expensive separation of **S** from the reaction products.

2.4 FLOW PATTERN

Because of the nature of rotor/stator machines - with rotors spinning inside stators at several thousands of revolutions per minute - it has been very difficult to observe flow patterns visually.

However, a large number of workers have investigated the flow patterns in centrifugal pumps, overcoming similar difficulties to those above. The results of these studies can be found in pump text books, for example Stepanoff (1957). These studies often focus on one particular region within the pumps, usually the inlet, impeller, impeller outlet or volute regions. The purpose of these studies is typically to improve the pumping efficiency, by understanding and then reducing

the losses. Almost the opposite approach may be required for rotor/stator mixers, where the energy dissipation rates (or losses) need to be maximised in order to improve mixing.

Recirculating flows exist, under certain operating conditions, throughout centrifugal pumps. Discharge and suction recirculation have been investigated by Karassik (1983). This and other references (e.g. Fraser, 1982) state that the amount of recirculation is dependent upon the throughput and rotational speed. At throughputs close to the best efficiency point (BEP), recirculation is minimal. The best efficiency point is defined as the pump capacity at which the pumping power efficiency is highest, the best efficiency point for a centrifugal pump is given in Figure 2.9. Inlet recirculating flow is shown in Figures 2.5 and volute recirculation in Figure 2.6.

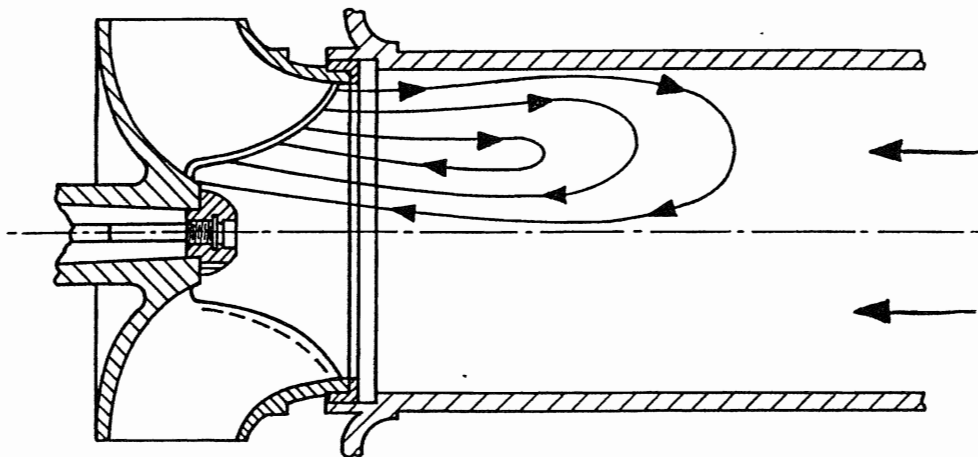


Figure 2.5 Inlet recirculation (Yedidiah, 1985)

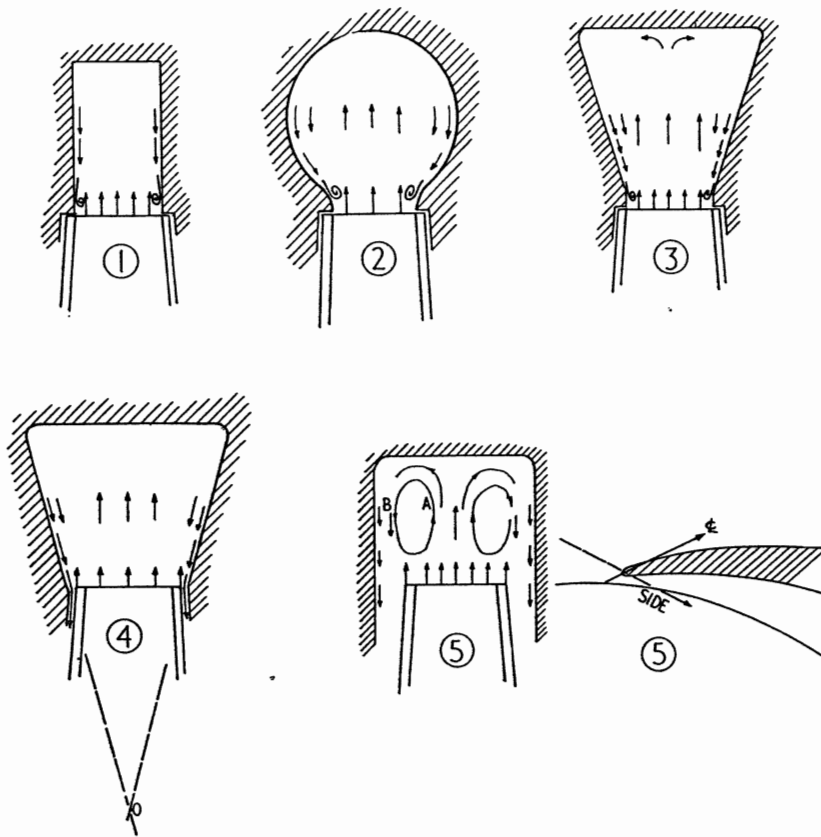


Figure 2.6 Volute recirculation (Copley et al., 1962)

Recirculating flows will contribute towards deviation from plug flow. There is no information in the literature on recirculating flows in rotor/stator mixers.

Flow separation and re-attachment at the leading edge of pump impeller blades can affect pumping performance (e.g. Copley & Worster, 1961). These phenomena will probably be more significant in rotor/stator mixers, as they are not generally designed to avoid them.

2.5 RESIDENCE TIME AND RESIDENCE TIME DISTRIBUTION

Residence time distributions (RTDs) provide information about the bulk flow through a mixer, and may be used to improve the predictions of chemical reactions by accounting for the residence times of reagents. They are obtained using stimulus-response experiments (Levenspiel, 1972).

Typical mean residence times ($\bar{t} = \text{volume}/\text{flowrate}$) for rotor/stator mixers are of the order of 1 second (Bourne & Garcias-Rosas, 1986, and Bouette, 1974). Short residence times are desirable for reactors containing fast, competing chemical reactions. These residence times are much lower than conventional reactors', for example stirred tanks. Therefore reaction products will spend less time in the mixer and this will reduce the chance of side reactions.

Studer (1990) presents some residence time distributions of in-line rotor/stator mixers, for both step (tracer turned on suddenly and kept on) and pulse (tracer injected instantaneously) stimuli. The rotor diameters were in the range 20-40 mm. Salt (NaCl) tracer was introduced into the centre of the rotor and the sampling probe was situated in the outlet pipe. Results indicated that the machines tested (manufactured by Ystral) exhibited back-mixed behaviour and that very little by-passing occurred, see Figure 2.7. Significant tailing was also observed, showing that tracer was trapped in the mixer for much longer than one residence time. These

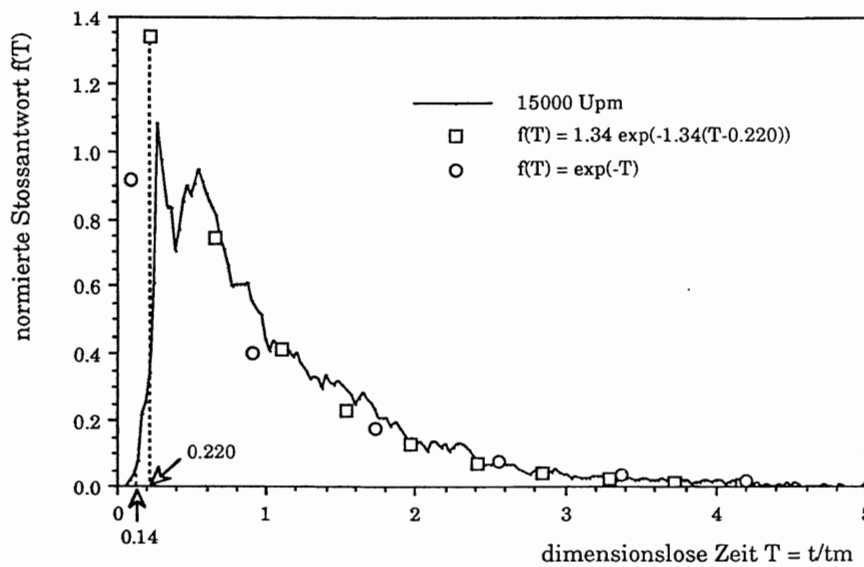


Figure 2.7 Residence time distribution curve for a rotor/stator mixer, taken from Studer (1990)

residence time distributions investigated the overall behaviour of the mixer, showing it to be backmixed. An ideal backmixed reactor is one in which a pulse of tracer would be instantly dispersed equally throughout the reactor and then flushed out over time. The normalised residence time distribution (see Section 3.4) would have a value 1 at $t = 0$, followed by an exponential decay. Studer also used a scheme comprising a plug flow reactor followed by a stirred tank reactor to model the RTDs. The experimental curves fell between the model curves for a stirred tank and for the combined plug flow and stirred tank reactor.

It is expected (Hannon, 1990) that plug flow might exist through the rotor and stator rings themselves, with backmixing occurring in the mixer volute.

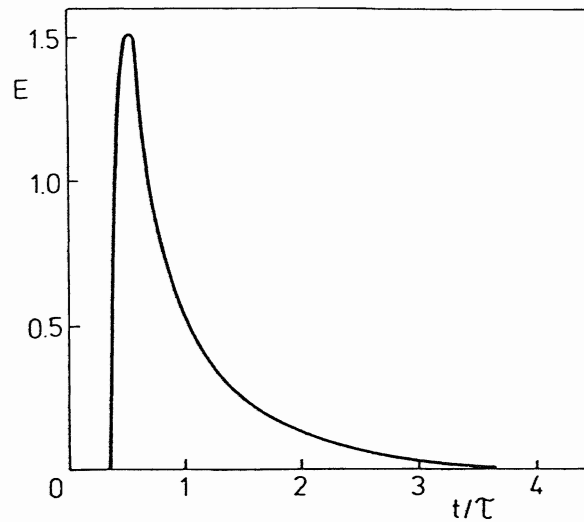


Figure 2.8 Residence time distribution curve for a centrifugal pump, taken from Bolzern & Bourne (1985)

Residence time distributions have been produced for centrifugal pumps by Bolzern & Bourne (1985) using concentrated dye and a photocell. Figure 2.8 shows a typical experimental result from that work. The authors characterised their residence time distributions using the "axial dispersion" and "stirred tanks in series models" (Levenspiel, 1972). Under all operating conditions they report substantial backmixing in the pump.

Studer (1991) also measured residence time distributions in a centrifugal pump. These exhibited tailing and the author also used the axial dispersion and stirred tanks in series models. The RTDs that were measured displayed pulsations after the peak concentration, indicating a rotary motion that was much slower than the

impeller tip speed. It was reported that tailing was due to "dead zones", regions in which tracer became trapped, within the pump.

Hearn (1992), measuring residence time distributions in a centrifugal pump with an open impeller found that the axial dispersion model was unsuitable. Instead it was recommended that a model representing two plug flow regions and a well mixed region be used (Levenspiel, 1972). This scheme fitted the experimental RTDs well.

2.6 ENERGY BALANCE IN A ROTOR/STATOR MIXER

It has been shown in Section 2.3 that knowledge of ϵ , the turbulent energy dissipation rate, is crucial for modelling both macro and micromixing in a reactor. Knowledge of how energy input into rotor/stator mixers is transferred will be required in order that the progress of chemical reactions can be predicted and controlled.

Not all energy input to a pump is transformed into pumping output. After many years of research, development and invention, modern centrifugal pumps are very efficient machines but energy losses are still significant. The study of these energy losses still occupies many researchers and all manufacturers of centrifugal pumps.

Shaft power input is transformed into pumping power output and dissipated through losses; these losses are defined and discussed in the following sections.

In pumps total losses are made up of disk friction, mechanical, leakage and hydraulic losses (e.g. Stepanoff, 1957; Anderson, 1980), so that:

$$P_{\text{SHAFT}} = P_{\text{PUMP}} + P_{\text{DISK}} + P_{\text{MECH}} + P_{\text{LEAK}} + P_{\text{HYDRAULIC}} \quad (2.10)$$

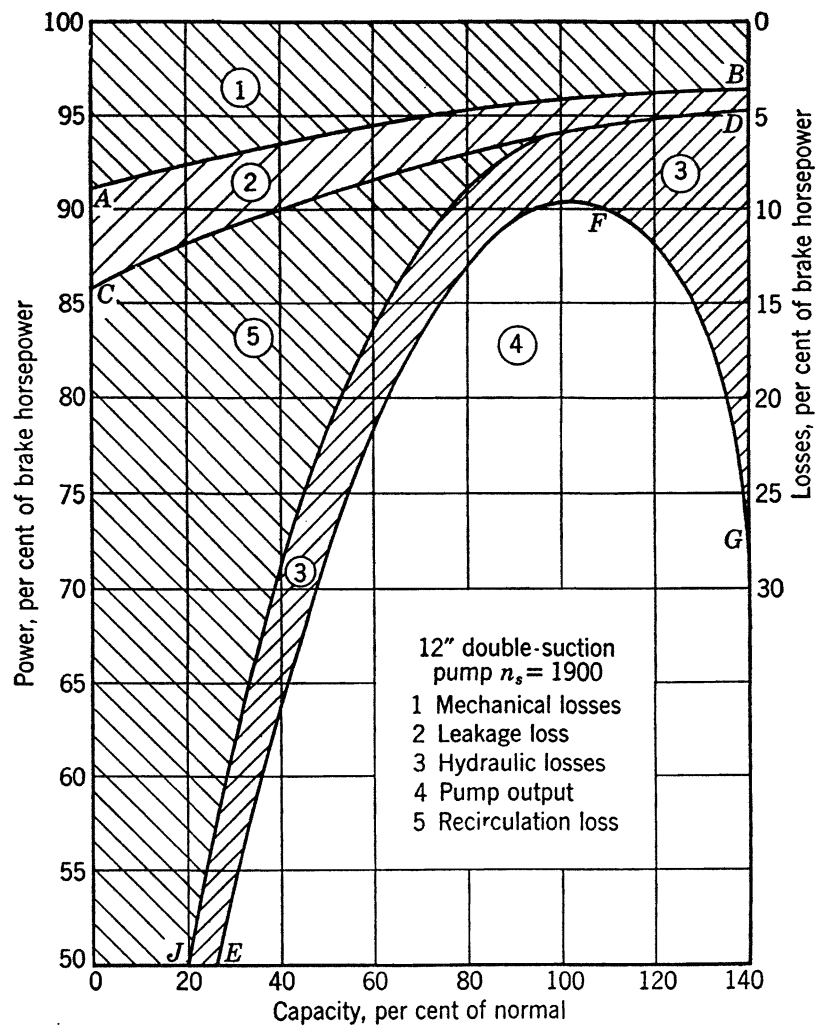


Figure 2.9 Power balance for a double suction centrifugal pump (Stepanoff, 1957)

Figure 2.9 shows a power balance for a double suction centrifugal pump (in this

figure disk friction is included in the mechanical loss and a recirculation loss is isolated from the leakage loss). Over a typical operating range, from 60 to 140 % of the normal capacity, the percentage of power losses attributed to hydraulic losses increases from around 4 % to more than 20%. Below the best efficiency point recirculation losses dominate, above the best efficiency point hydraulic losses dominate.

Rotor/stator mixers should suffer the same types of losses, although in very different proportions. The individual components of the power balance will now be considered.

2.6.1 P_{SHAFT} , shaft power input

Designers and operators of rotor/stator mixers need to know their power requirements, particularly for selecting motors or for scaling-up the performance from pilot to full scale. In fully baffled stirred tanks (Rushton et al., 1950) the shaft power is related to the rotational speed and diameter of the impeller by:

$$P_{\text{SHAFT}} = Po\rho N^3 D^5 \quad (2.11)$$

The coefficient Po is known as the power number and depends upon impeller type. At high impeller Reynolds numbers, fully turbulent conditions, the power number is constant for a given impeller.

In the laminar regime $Po \propto Re^{-1}$ where $Re = ND^2/\nu$

$$\therefore P_{\text{SHAFT}} \propto N^2 D^3 \quad (2.12)$$

Bouette (1974), working with a rotor/stator mixer, the Oakes model shown in Figure 2.10, reported that, at high Reynolds number, $P_{\text{SHAFT}} \propto N^3 D^5$ and that, at low Reynolds number, $P_{\text{SHAFT}} \propto N^2 D^3$. These expressions are identical to those established for an impeller rotating in a fully baffled stirred tank. Unfortunately, the definite values for high and low Reynolds number were not given.

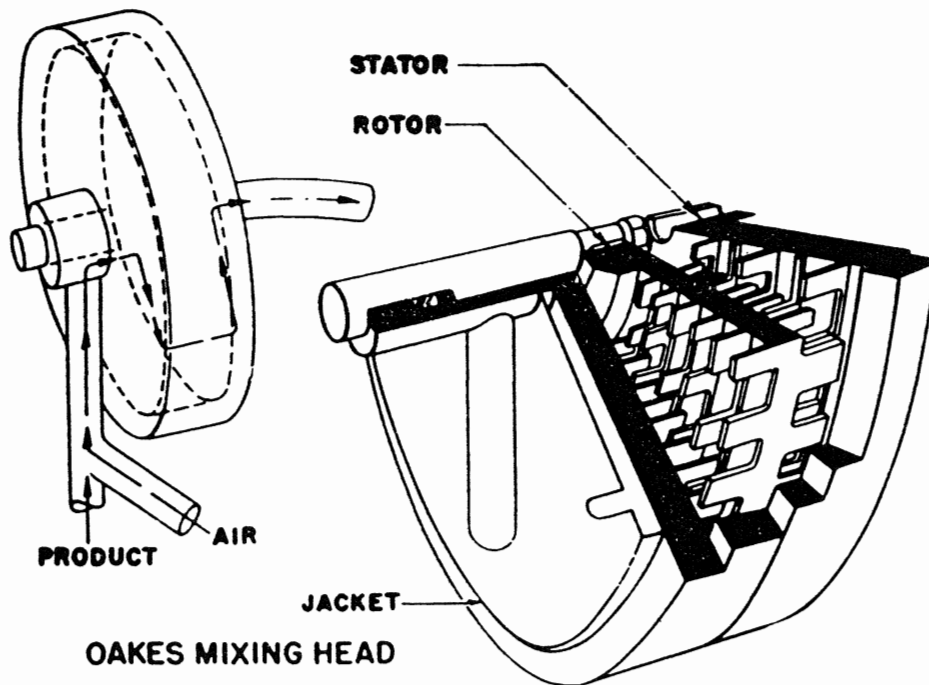


Figure 2.10 The Oakes mixer

Summets (1966) produced the following correlation, for the power requirement of a rotor/stator type of mixer, reporting an agreement with experiment to within 5 %.

$$P_{\text{SHAFT}} = 0.931 N^3 D^5 \rho \mu^{-0.086} Q^{0.045} H_t^{-0.0787} d_t^{0.785} Z^{0.068} S^{0.46} \quad (2.13)$$

where H_t is the distance between the centre lines of the teeth, d_t is the tooth size,

Z is the total number of mixing teeth and S is the number of stages. While this expression was formed for an unusual rotor/stator mixer (one that resembled the Oakes Mixer), it may be possible to produce something similar for other rotor/stator mixers. It is interesting that this expression shows a relatively weak dependence on flow rate and a fairly strong dependence on the number of stages. However, this expression was taken from a translation of a Russian paper and it is not stated over what range the expression is valid; nor are all of the units given. However, the expression includes the $N^3 D^5$ terms as before.

Studer (1990) reported an exponent of around 2.5 for N in an expression for the turbulent energy dissipation rate in a rotor/stator mixer. This suggests that the exponent on N in a power correlation may be below 3. Fractional exponents are sometimes found in expressions for power drawn in centrifugal pumps (Stepanoff, 1957).

2.6.2 P_{PUMP} , pumping power output

There is no independently published information on the pumping performance of rotor/stator mixers, although it is clear that it will be a much smaller component of the energy balance than in pumps.

Pumping performance will be required for design purposes in order to assess the additional pumping requirements of a process. It is possible that rotor/stator mixers can also provide some of the pumping requirements. Some manufacturers, for example Silverson and Greerco, supply rotors that more closely resemble pump impellers than toothed ring-type rotors, see Figure 2.2.

2.6.3 P_{DISK} , disk friction losses and P_{MECH} , mechanical losses

Disk friction losses are caused by the frictional drag on the surfaces of the impeller or rotor. Expressions of the form $P_{\text{DISK}} \propto N^3 D^5$ are given for this loss in pump text books, for example Stepanoff (1957) and Karassik et al (1984).

Mechanical losses are those due to the seals and bearings on pumps or mixers. The mechanical and disk friction losses will be comparable in rotor/stator mixers to centrifugal pumps if they are of similar size and operated at similar speeds, and if similar seals and bearings are used. However some rotor/stator mixers operate at very much higher rotational speeds than pumps, and therefore bearing and seal

losses could be very different.

Disk friction and mechanical losses do not contribute to mixing (Bolzern & Bourne, 1985)

2.6.4 P_{LEAK} , leakage losses

Leakage (or recirculation) losses in pumps are losses due to fluid not passing directly through in the required direction. This includes leakage through seals, a loss that will not be considered in rotor/stator mixers, and leakage from the high pressure side of the impeller to the low pressure (or inlet) side. The second of these is a mechanism for macromixing (Bolzern and Bourne, 1985) and will influence the residence time distribution.

2.6.5 $P_{\text{HYDRAULIC}}$, hydraulic losses

Knowledge of hydraulic losses is essential for a better understanding of rotor/stator mixing, as they are the only losses that contribute to micromixing.

A well designed pump would be a comparatively poor mixer, as hydraulic losses which could contribute to mixing would be low. Conversely in-line rotor/stator mixers are good at mixing but comparatively poor at pumping. Much work has been done on accounting for energy losses within pumps, most of it aimed at

reducing them. No similar work could be found for rotor/stator mixers.

Before micro-mixing models (e.g. those of Baldyga and Bourne, 1984 & 1989) can be applied, the magnitude and distribution of turbulent energy dissipation rate must be established.

The turbulent energy dissipation rate, ϵ , varies throughout any mixer/reactor. Locating regions of high ϵ is of vital importance in reactor design, as they disclose high local mixing rates. Unfortunately there is little information in the literature to assist in pinpointing these regions in rotor/stator mixers.

Bourne and Studer (1992) report that removal of the stator had no influence on the product distribution of a competitive chemical reaction. This implied that the reaction was complete before the shear gap was reached, or that the energy dissipated in the shear gap and stator regions was not useful for their mixing application. Pedrocchi and Widmer (1988), studying the formation of emulsions in rotor/stator mixers, found drop size distributions (stated in this paper as being strongly dependent upon ϵ) to be largely independent of shear gap size. These results appear to contradict the belief that the energy dissipation rate in the shear gap is the principal mixing mechanism.

Bourne and Garcias Rosas (1986), using the results of a competitive chemical reaction with the micromixing model (of Baldyga and Bourne) to investigate ϵ ,

report values of rates of turbulent energy dissipation in the reaction zone to be in the range 500-700 W/kg, for a rotor/stator mixer with a rotor diameter of approximately 20 mm. Bourne and Studer (1992) using the same technique report values of ϵ of more than 1000 W/ kg.

2.7 MIXING PROCESSES IN ROTOR/STATOR MIXERS

2.7.1 Macromixing

Macromixing may depend upon, for example, the operating condition, rotor/stator type and reactant feeding arrangement. No information on macromixing in rotor/stator mixers is given in the literature.

2.7.2 Micromixing

Micromixing depends strongly upon the rate of turbulent energy dissipation in a reactor. Values for ϵ of up to 1000 W/kg have been measured. For water - like fluids, this represents diffusive mixing times of around 0.5 ms; the time for fluid engulfment, t_E , is also of this order.

These energy dissipation rates are very high compared to those found with other reactors. In typical (say 2000 litre) stirred tanks, for example, the maximum ϵ is of the order of 5 W/ kg.

Provided that the reagents of fast chemical reactions are fed into appropriate regions within the mixer, and provided that the coarse scale distribution of reagents is satisfactory this will be an excellent environment for micromixing.

2.8 CONCLUSIONS

- A number of rotor/stator designs are available. The toothed ring type is probably the most common.
- Rotor/stator mixers are similar to centrifugal pumps and literature on centrifugal pumps contains useful information.
- Mixing can control the yield of chemical reactions and mixing rates depend upon the turbulent energy dissipation rate.
- In pumps and therefore rotor/stator mixers, energy is lost to:
 - i) Hydraulic losses
 - ii) Leakage losses
 - iii) Disk friction losses
 - iv) Mechanical losses
- Of these losses only hydraulic losses will contribute to micromixing.

- Shaft power is likely to be proportional to $N^3 D^5$, although the exponents in this expression may be fractional.
- One set of residence time distribution curves for rotor/stator mixers has been published. These show that they exhibit backmixing and that little bypassing occurs.
- Little work has been published on micromixing in rotor/stator mixers. What has been revealed that very high rates of turbulent energy dissipation are available, $\sim 1000 \text{ W/kg}$, which implies rapid micromixing.
- No information relating to micromixing or chemical reaction in rotor/stator mixers at industrial scales has been published.

2.9 OBJECTIVES FOR THIS STUDY

- To build a test rig that will incorporate an industrially sized in-line rotor/stator mixer.
- To devise methods for measuring key performance parameters and to use these to examine energy dissipation within the rotor/stator mixer and account for it.
- To examine, and account for, the residence time distribution (RTD) of an industrially sized in-line rotor/stator mixer.

CHAPTER 3. EXPERIMENTAL DETAILS

3.1 INTRODUCTION

In order to achieve the objectives in Section 2.10 a mixer was procured and fitted into a test loop, instrumentation was fitted and experimental data were collected using this rig.

This chapter describes the rig and instrumentation used for this work; equipment and software suppliers' names and addresses are given in Appendix C. The experimental methods used are also described in this chapter.

3.2 EXPERIMENTAL RIG

A custom built industrially sized rotor/stator mixer, based on a standard Silverson in-line 425 SL, was fitted into a test loop between two 2.5 m³ tanks. This is shown in Figure 3.1; Figure 3.2 is a photograph of the test unit. Water was used for all tests and flow rates through the test loop were generated by a Flygt pump, controlled by a manually operated valve and measured with an Altometer SC80 AS magnetic flow meter. The water tanks were connected by a pipe and water was recycled from the receiver tank to the delivery tank.

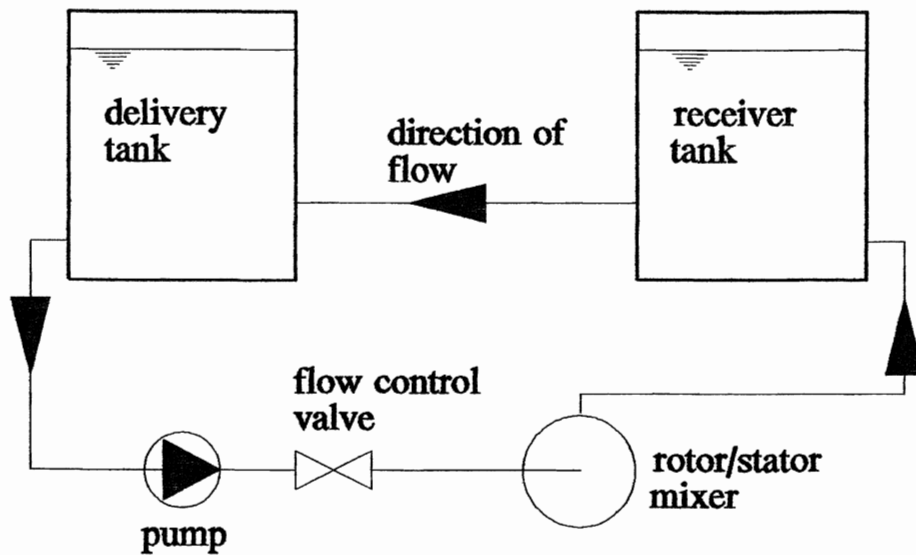


Figure 3.1 Schematic diagram of test rig

The rotor and stator were (approximately) square section toothed rings (18 teeth each) and are shown in Figure 3.3, drawings of the rotor and stator are given in Appendix D. The rotor was held on the shaft by a bullet-shaped nut. The rotor and stator were machined from solid stainless steel. The rotor external diameter was 120 mm and its internal diameter was 95 mm, the stator internal diameter was 124 mm and its external diameter was 147 mm. The rotor teeth were 16 mm long and the stator teeth were 19 mm long.

The mixer volume, with rotor and stator in place, was 1.59 litres ($1.59 \times 10^{-3} \text{ m}^3$) This was estimated from drawings and then measured by filling with a measured volume of water. The volute was a cylinder with diameter 172 mm and length 110

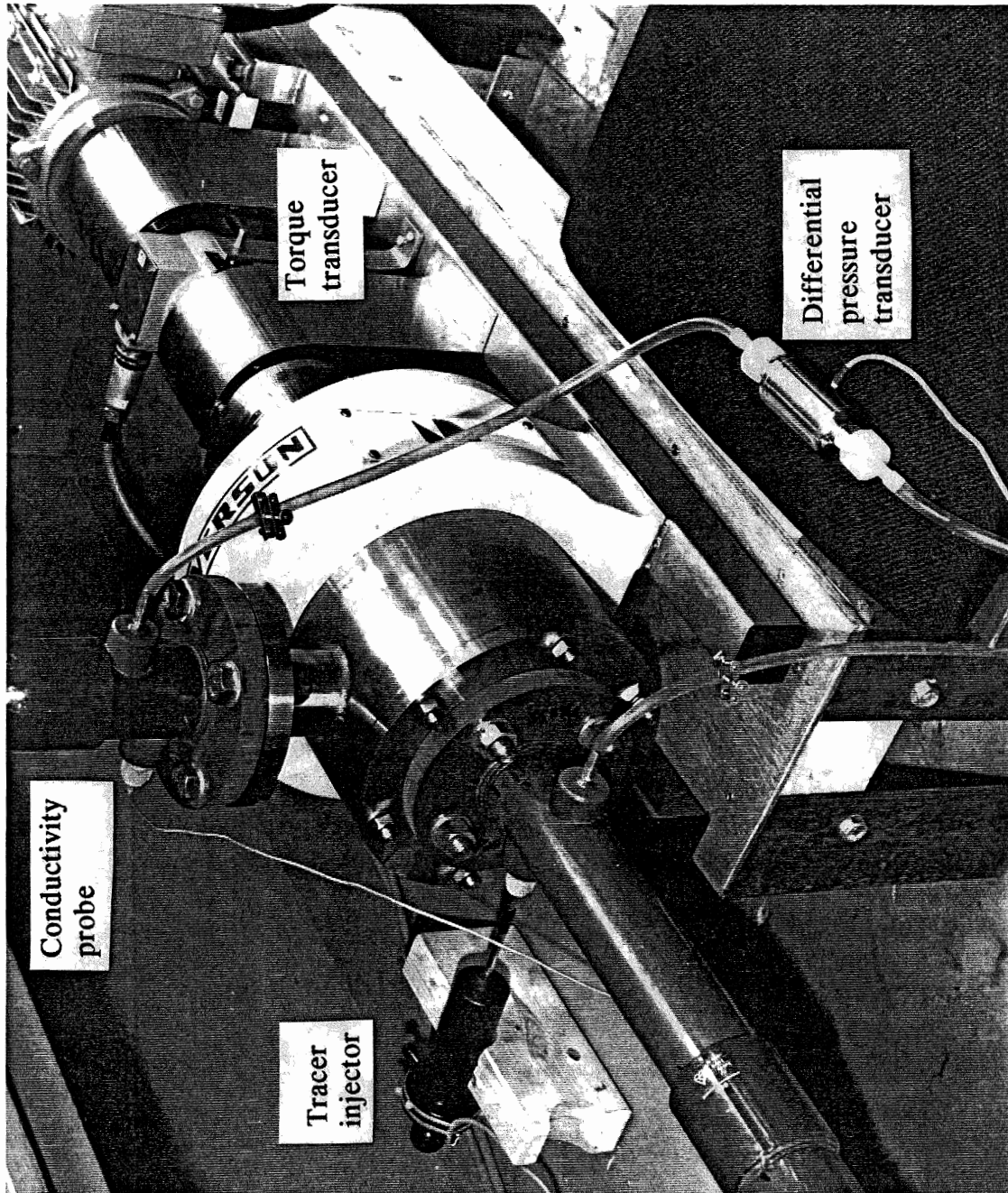


Figure 3.2 Photograph of test unit

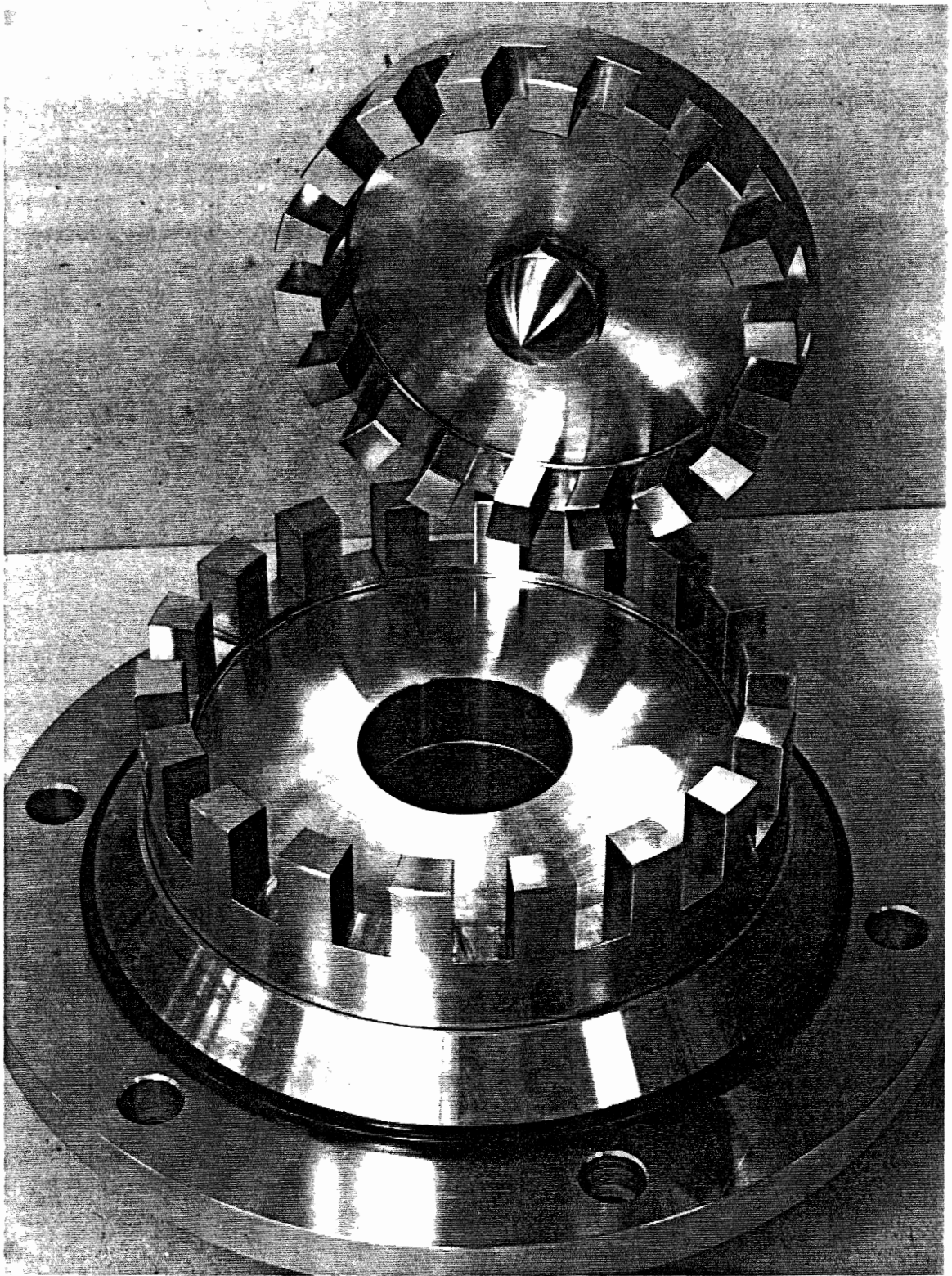


Figure 3.3 Rotor and stator

mm. The inlet and outlet pipe diameters were 50.8 mm (2").

The rotor and stator were designed by the author to represent a commercially available unit, and were made by Silverson, the suppliers of the machine. However it was not a standard Silverson design. The mixer size has been selected so that it will be possible to insert packing to reduce the size of the mixer volute and incorporate a smaller rotor/stator pair. This will be used in the future to investigate scale effects.

3.2.1 Band of Operation

Flow rates up-to approximately 3.5 litres per second were possible with the rig, the maximum shaft speed tested was 2500 revs per minute (42 revs per second), it was possible, however, to use the mixer at 3500 rpm (58 rps). The machine remained well balanced at most shaft speeds. At high speeds and low flow rates there was audible cavitation. This was partially alleviated by applying back pressure (half closing a downstream valve).

3.3 ENERGY BALANCE - BACKGROUND

The energy balance for this mixer is (Section 2.6):

$$P_{\text{SHAFT}} = P_{\text{PUMP}} + P_{\text{DISK}} + P_{\text{MECH}} + P_{\text{LEAK}} + P_{\text{HYDRAULIC}} \quad (3.1)$$

where P_{SHAFT} is the shaft power input, P_{PUMP} is the pumping power output, P_{DISK} is the power loss to disk friction, P_{MECH} is the power lost to seal and bearing losses and $P_{\text{HYDRAULIC}}$ and P_{LEAK} are the power due to hydraulic (useful to mixing) and leakage losses.

The energy balance required measurement of shaft torque, shaft speed and pressure rise. Temperature rise across the mixer was also measured to provide additional information on energy dissipation. The components of the balance were obtained in the following ways.

3.3.1 P_{SHAFT}

The rotational speed of the shaft, N , and the shaft torque, Λ , were measured and the shaft power calculated:

$$P_{\text{SHAFT}} = 2\pi N\Lambda \quad (3.2)$$

3.3.2 P_{PUMP}

The flow rate through the mixer, Q , and the pressure rise across the mixer, Δp , were measured and the pumping power output calculated:

$$P_{\text{PUMP}} = Q\Delta p \quad (3.3)$$

3.3.3 P_{DISK}

The rotational speed of the rotor, N , was measured and the theoretical power loss to disk friction calculated using two expressions. The first (Stepanoff, 1957) assumes that the rotor is a thin disk and the second (Jekat, 1986) includes the effect of rotor thickness:

$$P_{\text{DISK}} = KN^3D^5 \quad (3.4)$$

$$P_{\text{DISK}} = KN^3D^4(D+5E) \quad (3.5)$$

where K is a constant (the constant from Jekat was used in both equations) and E is the rotor thickness at the circumference. It would be possible to measure this directly if a disk with the same dimensions as the rotor were fitted to the shaft, this was not done in this work.

3.3.4 P_{MECH}

The rotor was removed and the shaft rotated in the mixer alone. It was assumed that power consumed due to drag on the shaft was small compared to the seal and bearing losses. The resulting torque and Equation 3.2 were used to calculate the mechanical power losses.

3.3.5 P_{LEAK} and $P_{\text{HYDRAULIC}}$

These losses were not directly measured, but it is assumed that together they account for the difference between the shaft power input and the sum of the losses above:

$$P_{\text{HYDRAULIC}} + P_{\text{LEAK}} = P_{\text{SHAFT}} - (P_{\text{PUMP}} + P_{\text{DISK}} + P_{\text{MECH}}) \quad (3.6)$$

Differential temperature across the mixer was measured to provide further information on the losses that dissipated to heat.

3.4 RESIDENCE TIME DISTRIBUTIONS - BACKGROUND

Residence time distribution curves were produced with a pulse stimulus technique using a salt tracer. The pulse was injected at 1½ pipe diameters upstream of the mixer and sampled 3 diameters downstream of the outlet.

The absolute quantity and concentration of salt was not measured. So, in order that the residence time distributions could be compared with theoretical models and with each other on equal terms, they were normalised. The time axis was made dimensionless by dividing through by the mean residence time, $t' = t/\bar{t}$.

The RTD curve was normalised by making the area under the curve equal to one:

$$\int_0^{\infty} C \, dt' = 1 \quad (3.7)$$

All of this was done using spreadsheet formulae.

3.5 DATA ACQUISITION AND ANALYSIS

Shaft torque, differential pressure and residence time distribution measurements were collected with a standard data acquisition package running on a personal computer. The package used was Labtech Notebook Version 5. The transducers were plugged in to a sixteen channel Metrabyte Dash-16 digital-to-analogue converter board, and the acquisition package was used to convert this signal into either a screen output or (for the residence time distributions) an ASCII text file.

The shaft torque and pressure transducer outputs were constantly averaged by the acquisition package, so that the figure on the screen was the average of the last 50 values. Values were sampled 100 times per second and the screen display was refreshed two times per second.

For the residence time distributions, conductivity probe responses were sampled for 10 seconds during each test. 5000 values were taken per second and these were stored in a temporary ASCII file. This file was then block averaged (in blocks of

50) and the resulting file imported to a spreadsheet package for analysis.

The spreadsheet package used for the analysis, regression and presentation of all data was Quattro Pro Version 4.0.

3.6 INSTRUMENTATION

Instrumentation was used to measure shaft speed, shaft torque, differential pressure across the mixer, differential temperature and residence time distribution data. The instrumentation used, except the torque transducer, is shown in Figure 3.4. Calibration details are given in Appendix C.

3.6.1 Shaft speed

A hand held optical TM -2011 tachometer was used. The tachometer was pointed at the rotating shaft and a small lamp illuminated by pressing the button. Light was reflected periodically by a small piece of foil attached to the shaft and the speed of the shaft, in revolutions per minute, was shown on the tachometer display.

At low shaft speeds (< 200 rpm) it was possible to calibrate by eye and a stopwatch. By pointing the tachometer at a fluorescent tube it was possible to check the accuracy against the mains cycle (6000 maxima and minima per minute). These checks showed the tachometer to be accurate.

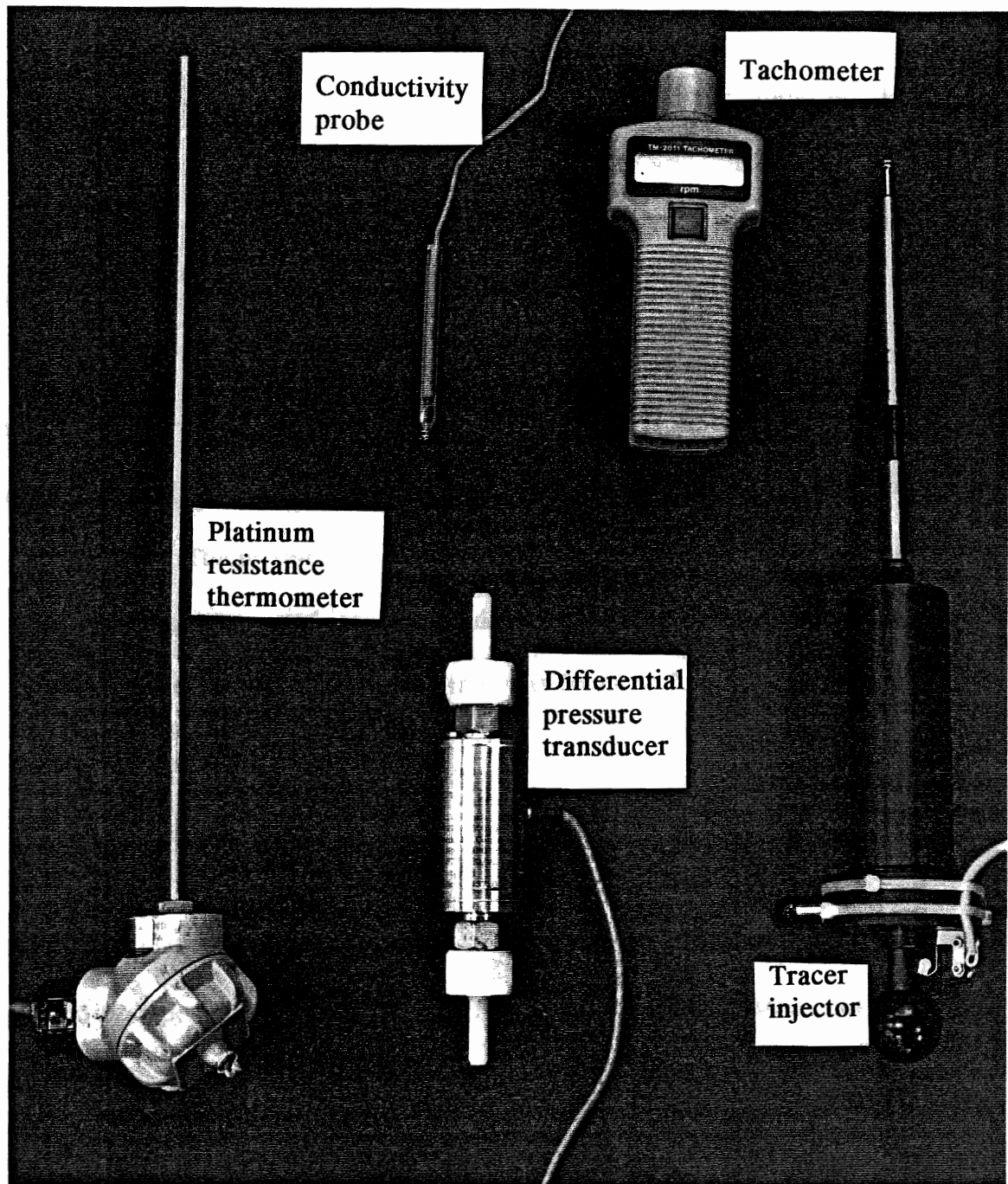


Figure 3.4 **Instrumentation used**

3.6.2 Shaft Torque

A T 30 FD torque transducer with a KMN 902.D signal amplifier, both by Hottinger Baldwin Messtechnik, were used. The shaft was split and the transducer fitted between flexible couplings. This work was done by Silverson, the manufacturers of the mixer.

The transducer was calibrated, over the range 0 to 4 N m, by locking of the shaft at one end and adding water to a beaker suspended from a bar fixed horizontally onto the shaft. The incremental volumes of water, the moment arm length and the meter readings were typed into a spreadsheet. A regression was then performed, see Appendix C; the calibration constant was found to be 0.001806 N m/unit.

3.6.3 Differential Pressure

A 1 bar Druck differential pressure transducer was connected to wall tapings located $1\frac{1}{2}$ pipe diameters upstream and 3 pipe diameters downstream of the mixer. The transducer was calibrated using a Druck DPI 600 digital pressure indicator. The calibration chart and calculation are given in Appendix C.

3.6.4 Differential Temperature

Differential temperature was measured using two platinum resistance thermometers, one of which is shown in Figure 3.4. Differential temperature was read off from the front of a signal conditioning box, the display read to a resolution of 0.001 K. The signal conditioning box contained an auto-balancing current bridge.

The offset between the two transducers was measured by immersing them both in a thermally insulated, well mixed, water bath. This provided a way of checking the transducers' response time. It took approximately one minute for the reading to settle after one of the probes was removed from and returned to the water bath. During tests the temperature readings were taken only after the reading had settled down, over this time scale.

Whenever the temperature probes were immersed in the water bath, that was assumed to contain water at a constant temperature throughout, an offset reading within ± 0.005 K of 0.037 K was obtained. Therefore the accuracy of the temperature probes was taken as ± 0.005 K

3.6.5 Residence time distribution

The tracer injector was made in the workshops at the Eidgenössische Technische

Hochschule, Zürich. The injector is photographed in Figure 3.4 and shown schematically in Figure 3.5. The small chamber in the end of the injector was filled with a salt (NaCl) solution. This was released by pulling a small trigger, causing a spring to push the chamber out of the tube. The injector was fitted with a micro-switch that triggered the data acquisition package to collect data.

The absolute concentration of the salt solution was not important, as the residence

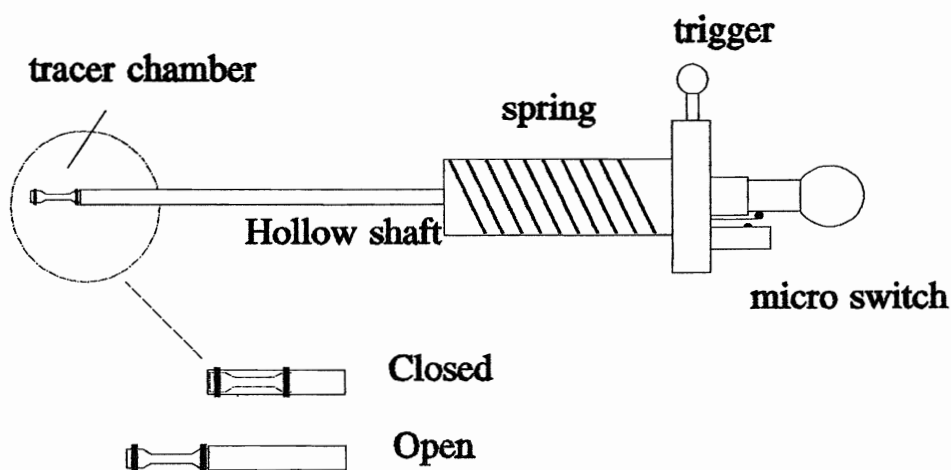


Figure 3.5 Schematic diagram of injector

time distribution curves were normalised. A saturated salt solution was prepared and then diluted to around half strength.

The conductivity probe is shown in Figure 3.6 (and Figure 3.4) and was made of two platinum wires coated in black, amorphous platinum which were set into a resin shaft (Ruszkowski, 1985). These probes give a linear response to concentration change (see Figure 4.10 & Studer, 1991). The probe output was amplified by a standard signal amplifier and recorded using Labtech Notebook.

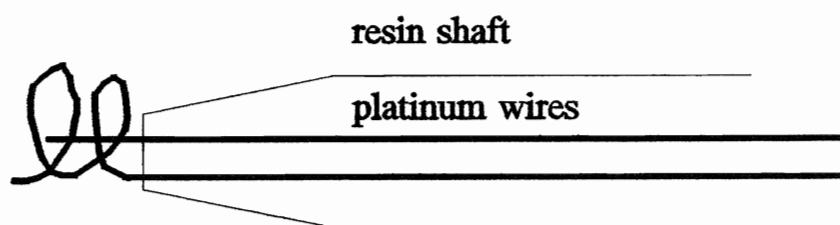


Figure 3.6 Schematic diagram of conductivity probe

3.7 EXPERIMENTAL METHOD

3.7.1 Energy balance

The following operating conditions were investigated:

Table 5.1

Shaft speed (rpm)
0
500
1000
1500
2000
2500

Table 5.2

Flow rate (litres/sec)
0.0
0.5
1.0
1.5
2.0
2.5
3.0
3.5

For all energy balance tests the shaft speed was set and recorded and the flow rate varied by adjusting the flow control valve (Figure 3.1). Measurements of shaft torque and differential pressure were taken at each flow rate.

Mechanical losses due to the bearings and seals were measured by removing the rotor and running the mixer. As the rotor was used to retain the seal setting spring, a small plastic cap was made and fitted to the end of the shaft for these tests.

3.7.2 Residence time distribution

The procedure used for obtaining residence time distribution curves was as follows:

- i) The rig was drained and the injector removed.
 - ii) The injector was filled with tracer and replaced in the rig.
 - iii) The pump and mixer were started and the flow rate and shaft speed set.
 - iv) The data acquisition package was started.
 - v) Tracer was released.
 - vi) Data sampled for 10 seconds
 - vii) Data block averaged.
- back to i)

CHAPTER 4. EXPERIMENTAL RESULTS

4.1 INTRODUCTION

This chapter is used to present the results of the tests conducted using the experimental equipment and methods described in Chapter 3. Raw data are presented in Appendix A. Three complete sets of data were collected for the energy balance (Λ , Δp , N), and a number of residence time distribution curves were obtained.

Comments are given on the reproducibility and quality of the results, but no analysis or discussion is included in this chapter.

4.2 ENERGY BALANCE

Those measurements relevant to the study of energy dissipation are presented in this section. Three complete sets of tests were done, these are numbered Test 1, Test 2 and Test 3.

4.2.1 P_{SHAFT} , shaft power input

So that shaft power input could be calculated, shaft torque and shaft speed were measured. Three typical set of results, showing shaft torque against flow rate at

1000 rpm, are shown in Figure 4.1.

This graph shows that the results are reproduced but off-set from each other. This

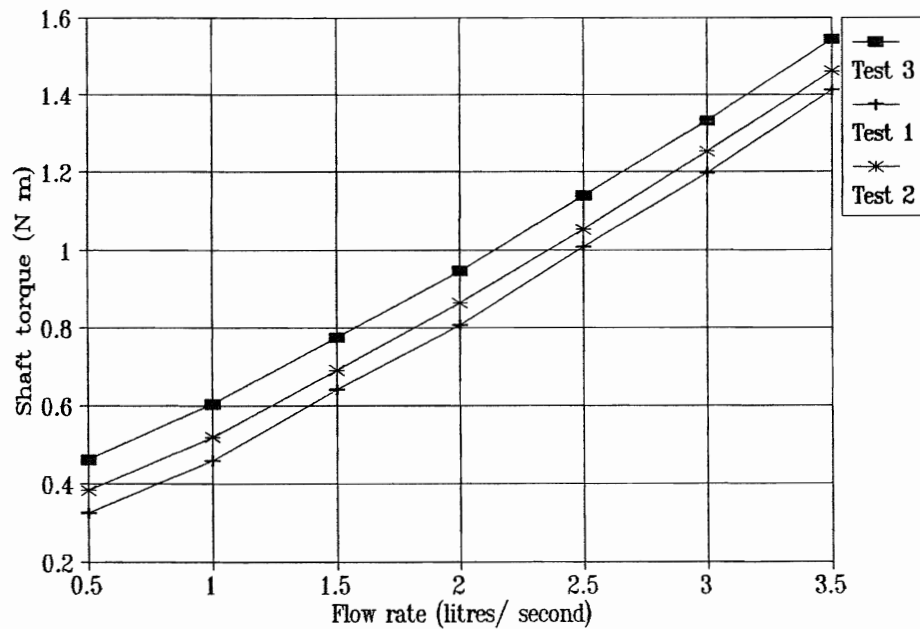


Figure 4.1 Shaft torque, Λ , against flow rate, Q , at 1000 rpm

became apparent after the second set of tests had been completed. It was discovered that, when at rest, the transducer (or the shaft) was retaining a torque of up to approximately 0.2 Nm. When the torque transducer was zeroed for the tests therefore there could still be a 'stored' torque.

The 'stored' torque was released for the final set of tests (Test 3 in Figure 4.1) by gently tapping and slowly rotating the shaft before each run. This means that

greatest confidence is placed upon the final set of tests. However an experimental error of up to 0.1 Nm may still be included in the results for shaft torque.

Corrected shaft torque (Test set 3) is shown against flow rate for a range of shaft speeds in Figure 4.2, and shaft power is shown against flow rate in Figure 4.3.

The torques measured for this work, up to approximately 4 N m, cover only a small part of the transducer rating (up to 200 N m). While the readings taken were very

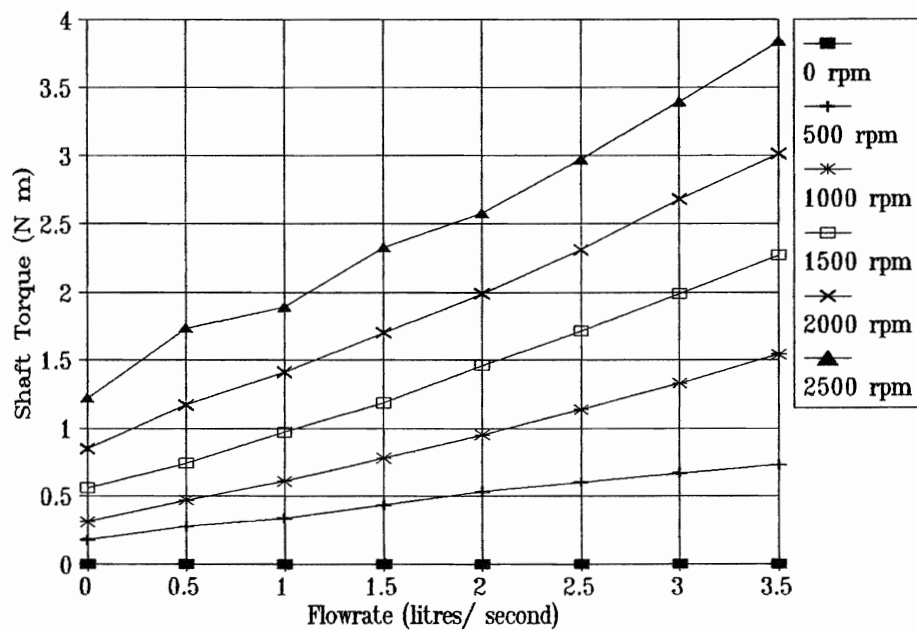


Figure 4.2 Λ against Q for a range of shaft speeds

reproducible (and steady) it is thought that the transducer or bearings may be responsible for holding the stored torque.

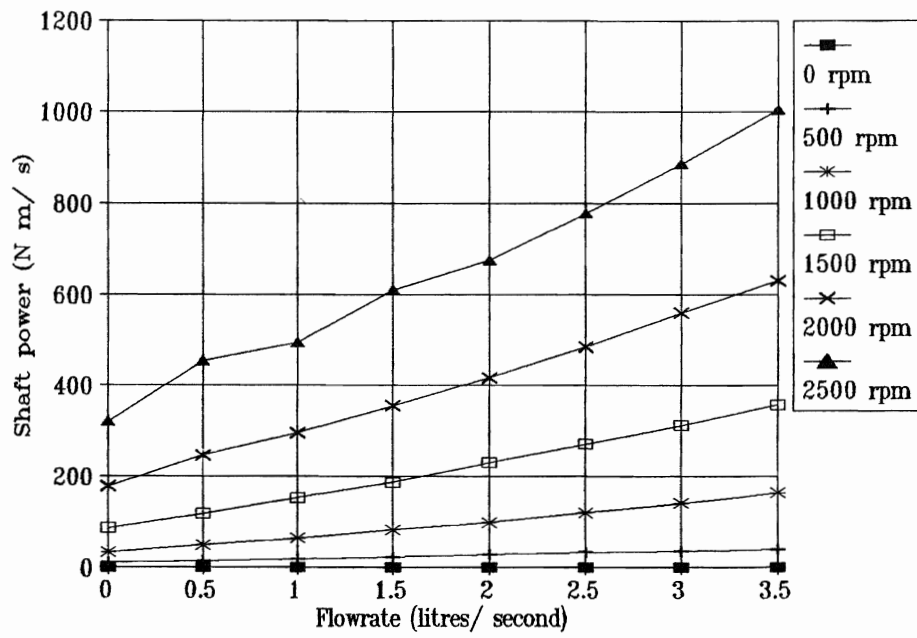


Figure 4.3 P_{SHAFT} against flow rate

4.2.2 P_{PUMP} , pumping power output

Figure 4.4, showing three sets of readings taken at 2000 rpm, demonstrates that the differential pressure measurements are reproducible. At lower shaft speeds, at 500 rpm in Figure 4.5, the reproducibility is not as good. At low shaft speeds the differential pressure is very low and accuracy is beginning to be limited by the transducer and the resolution of the analogue-to-digital conversion by the computer.

Differential pressure is shown against flow rate for a range of shaft speeds in Figure 4.6. The pressure rise decreases as the flow rate is increased (at fixed speed). At very high flow rates, or low speeds, it tends to zero. Overall pressure drops are recorded at zero speed.

Pumping power output is shown against flow rate for a range of shaft speeds in Figure 4.7. This increases with speed, but for each speed (except 2500 rpm) there is a distinct maximum pumping power. The flow rate corresponding to the maximum pumping power increases with shaft speed.

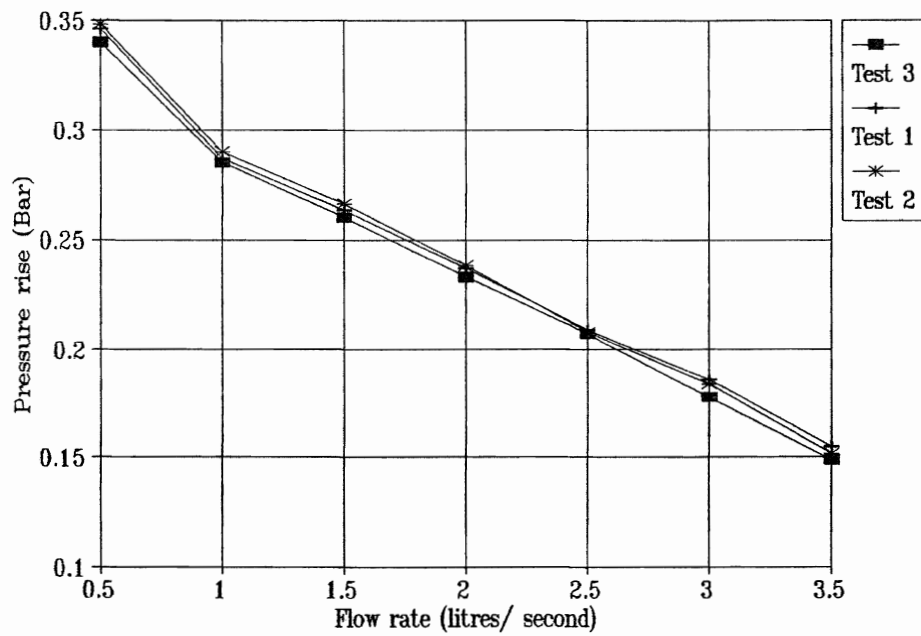


Figure 4.4 Δp against flow rate at 2000 rpm

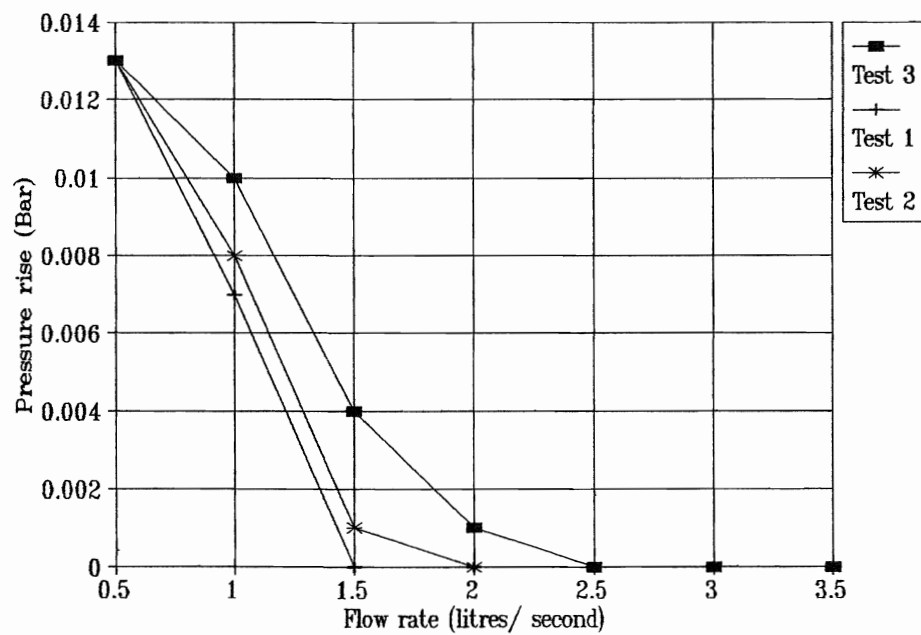


Figure 4.5 Δp against flow rate at 500 rpm

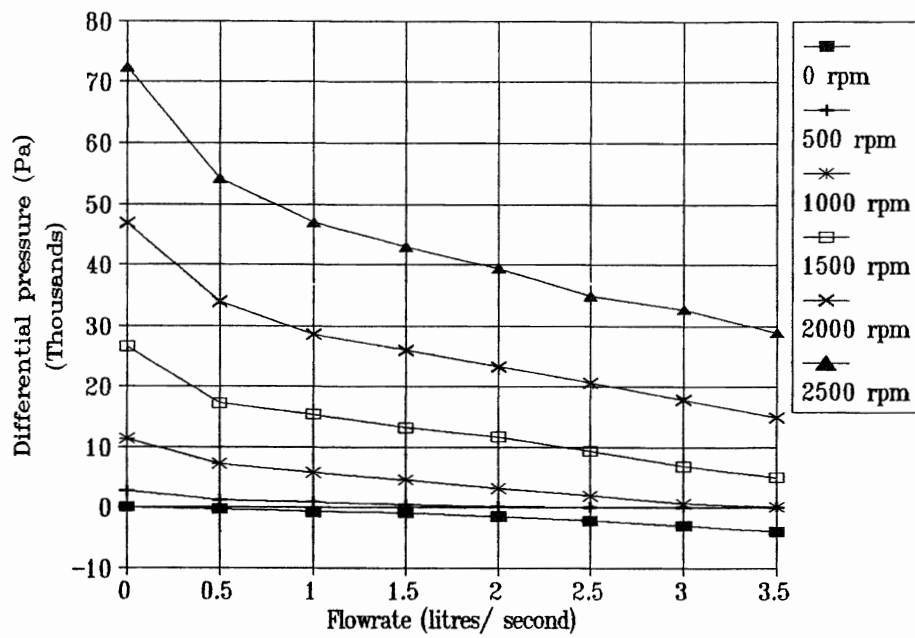


Figure 4.6 Δp against flow rate

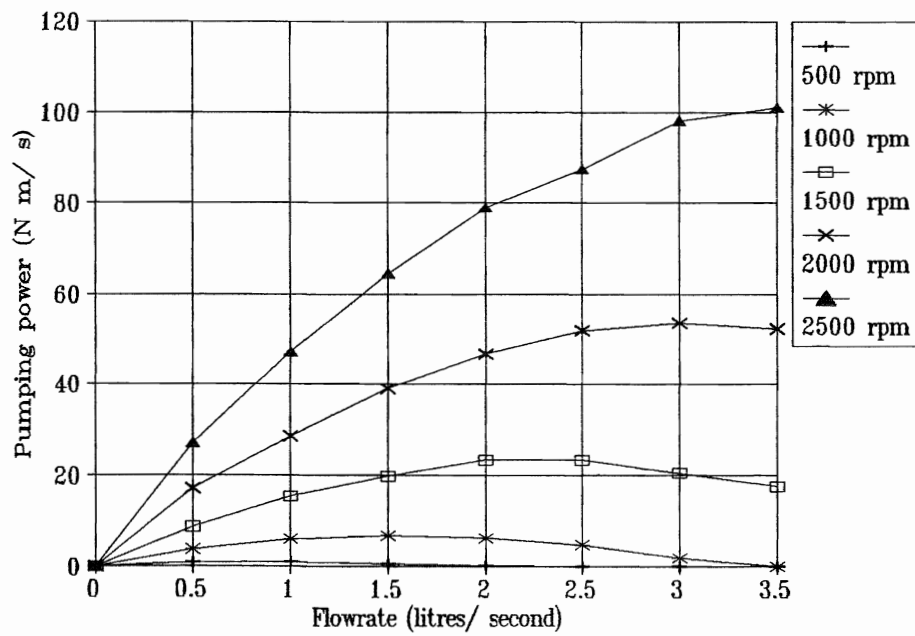


Figure 4.7 P_{PUMP} against flow rate

4.2.3 P_{MECH} , power due to mechanical losses

Shaft power with the rotor removed (P_{MECH}) is given in Table 4.1. The shaft torque was found to be independent of water throughput under these conditions. The results were not reproducible.

Table 4.1 Power drawn due to mechanical (seal and bearing losses)

N (revs/ min)	P_{MECH} (Test 1)	P_{MECH} (Test 2)	P_{MECH} (Test) 3
500	7.4	8.5	10.1
1000	15.1	17.6	21.4
1500	21.3	26.7	33.5
2000	31.4	37.4	46.1
2500	39.7	49.2	61.5

4.2.4 Differential temperature

One set of differential temperature results is shown against flow rate for a range of shaft speeds in Figure 4.8. The differential temperature measurements were not reproducible, and displayed sensitivity to how far the inlet transducer projected into the pipe. Investigative tests were carried out and it was found that readings were time dependent (over time scales of several minutes) and that the rotor shaft became warm (when stationary) to touch after the mixer had been run. The tests in this graph were conducted in the order 500 rpm, 0 rpm, 1000 rpm, 1500 rpm,

2000 rpm. The time difference between each test was not recorded. No full repeats were done.

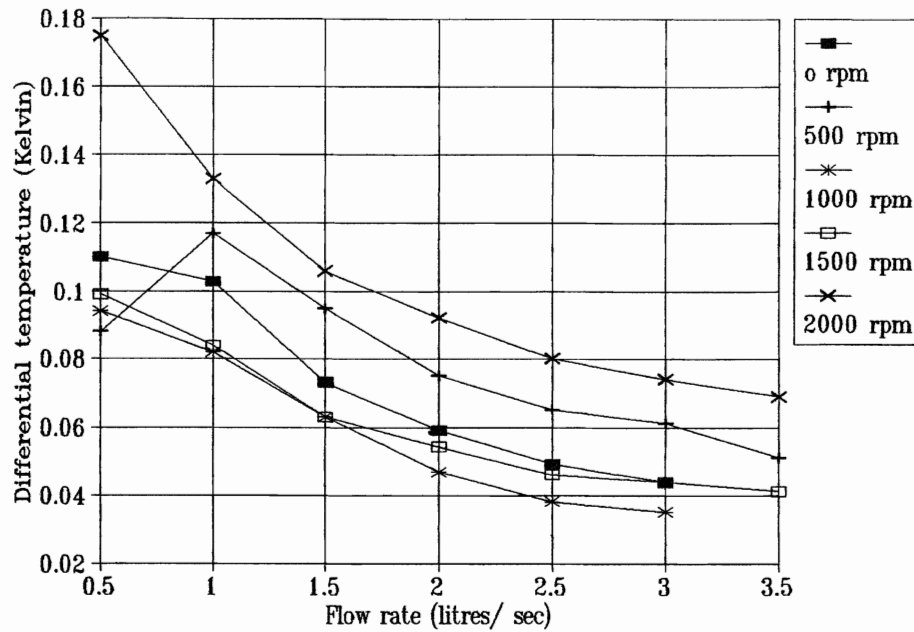


Figure 4.8 Differential temperature against flow rate

4.3 RESIDENCE TIME DISTRIBUTION

A typical normalised residence time distribution is shown in Figure 4.9; the remaining curves are shown in Appendix A. Figure 4.10 is a plot of background concentration (the output from the amplifier before each test) against test number. Water was recycled between two well mixed (based upon visual assessment) 2.5 m³ tanks and the volume of salt tracer introduced during each test was approximately 0.2 ml. That the tiny volume of salt changed the measured background

concentration highlights the sensitivity of the probes.

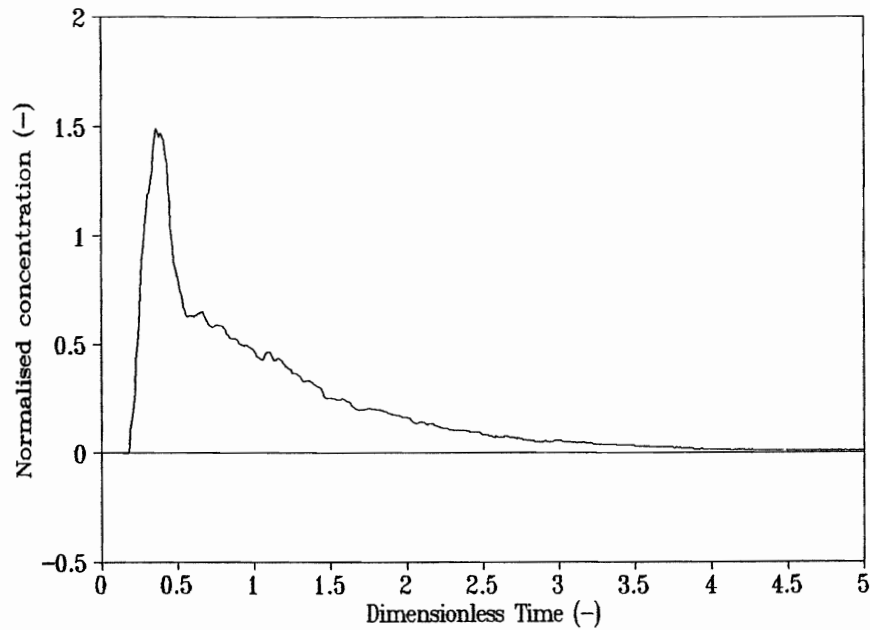


Figure 4.9 Normalised residence time distribution curve at $N = 500$ rpm, $Q = 1.5$ l/ s

Two of the RTDs given in Appendix A are faulty. These are A.14 and A.17. It is thought that A.14 shows a run in which no tracer was added to the injector, so that small fluctuations in the background concentration are amplified by normalisation. A.17 shows the tail of an RTD, probably caused by late triggering.

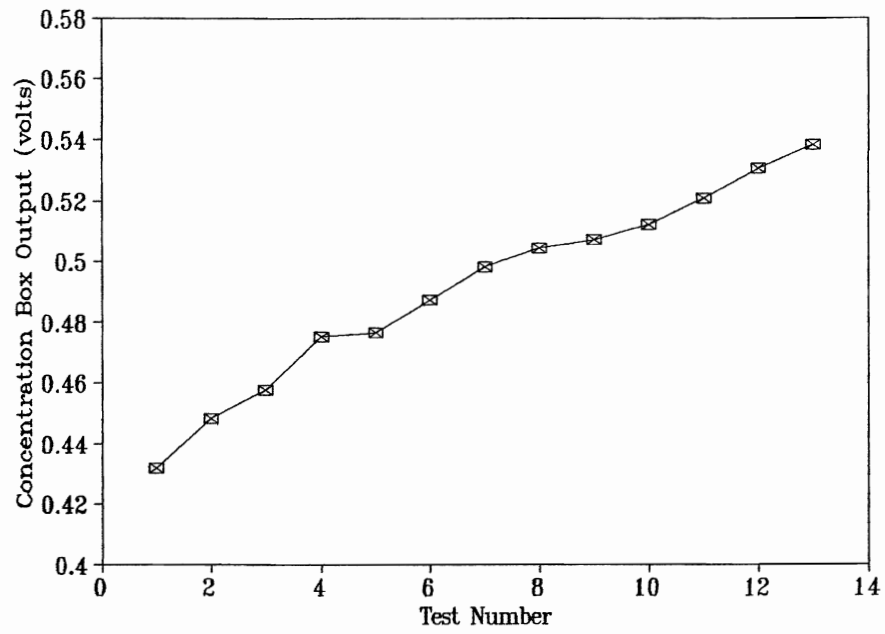


Figure 4.10 Background concentration against test number

4.4 CONCLUSIONS

- Torque measurements were found to be off-set from test to test; this problem can be alleviated by releasing the 'stored' torque. A transducer that measured a smaller torque range would be better suited for future work.
- Differential pressure measurements are reproducible and accurate.
- Shaft power input with the rotor removed was used to investigate the losses due to the seal and bearings. These losses were not found to be reproducible.
- Differential temperature measurements are not reproducible with the set-up used. However they indicate that the amount of energy dissipated to heat by the machine is large and measurable.
- Residence time distributions have been measured, and the sensitivity of the conductivity probes has been demonstrated.

CHAPTER 5. ANALYSIS AND DISCUSSION OF RESULTS

5.1 INTRODUCTION

The data presented in Chapter 4 will be used in this chapter to examine and account for the energy balance and residence time distribution of the rotor/stator mixer.

The energy balance and residence time distributions will be used to provide information about the mixing performance of rotor/stator mixers.

5.1.1 Commissioning

The mixer was fitted into a test loop between two large tanks (see Figure 3.1), so that water could be re-used. Care was taken so that conditions close to plug flow could be assumed at the mixer inlet: a large radius elbow was fitted to the upstream pipe and there were 6 diameters of straight pipe before the mixer.

Few problems were experienced during commissioning, but time was lost while these problems were sorted. The first major problem was a shaft seal failure. This was because the set spring was too short, caused by a spacer that was too long. The spring was over compressed (in fact coil-bound) and the pressure on the seal faces was therefore too high. The carbon and ceramic faces of the seal were

cracked by the heat generated by friction. Time was lost while this problem was investigated and, once the machine supplier had been informed of the problem, a new seal and spacer were sent and fitted.

The other major setback was a cracked inlet pipe flange, caused by over-tightening a nut. Time passed while a new pipe section was made.

5.2 ENERGY BALANCE

The energy balance for the mixer is:

$$P_{\text{SHAFT}} = P_{\text{PUMP}} + P_{\text{DISK}} + P_{\text{MECH}} + P_{\text{LEAK}} + P_{\text{HYDRAULIC}} \quad (5.1)$$

where:

P_{SHAFT} ($= 2\pi N\Lambda$) is the shaft power input, P_{PUMP} ($= Q\Delta p$) is the power output, P_{DISK} ($= KN^3D^5$) is the disk friction loss and $P_{\text{HYDRAULIC}}$ and P_{MECH} are the hydraulic (useful to mixing) and mechanical losses.

This section shall examine each of the components of the energy balance.

5.2.1 P_{SHAFT} and shaft torque

Shaft torque plotted against shaft speed is shown in Figures 5.1 and 5.2. The torque is approximately proportional to shaft speed.

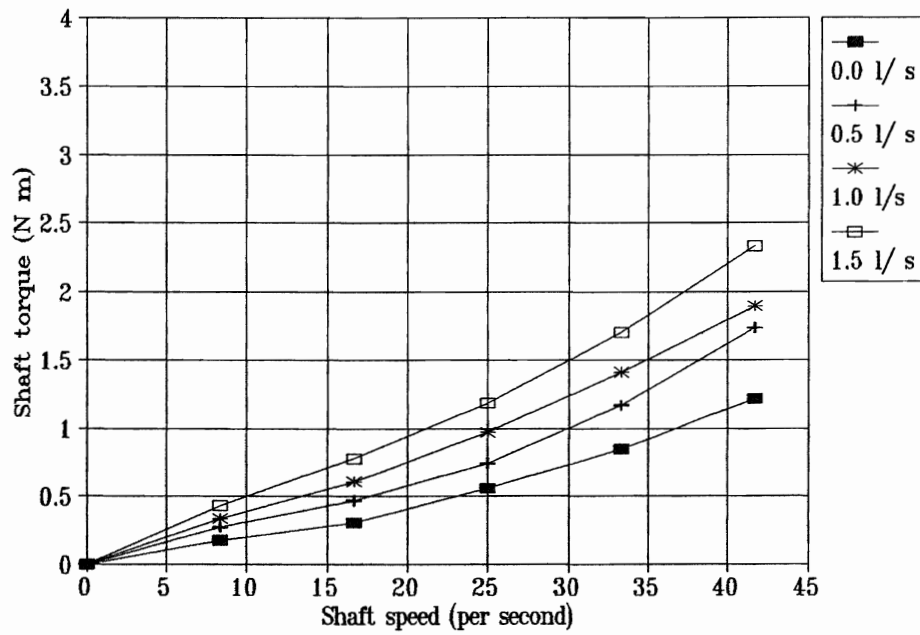


Figure 5.1 Shaft torque against shaft speed

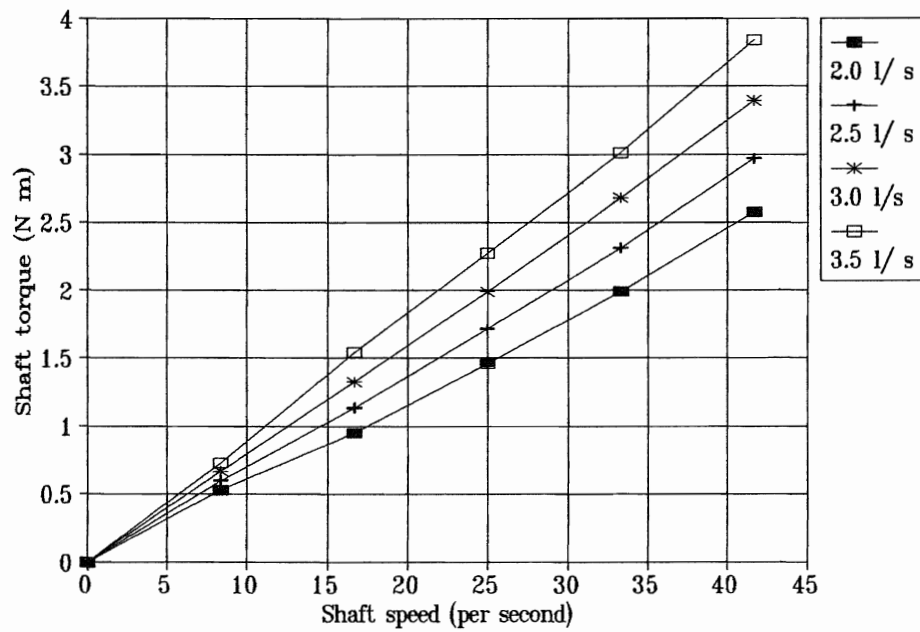


Figure 5.2 Shaft torque against shaft speed

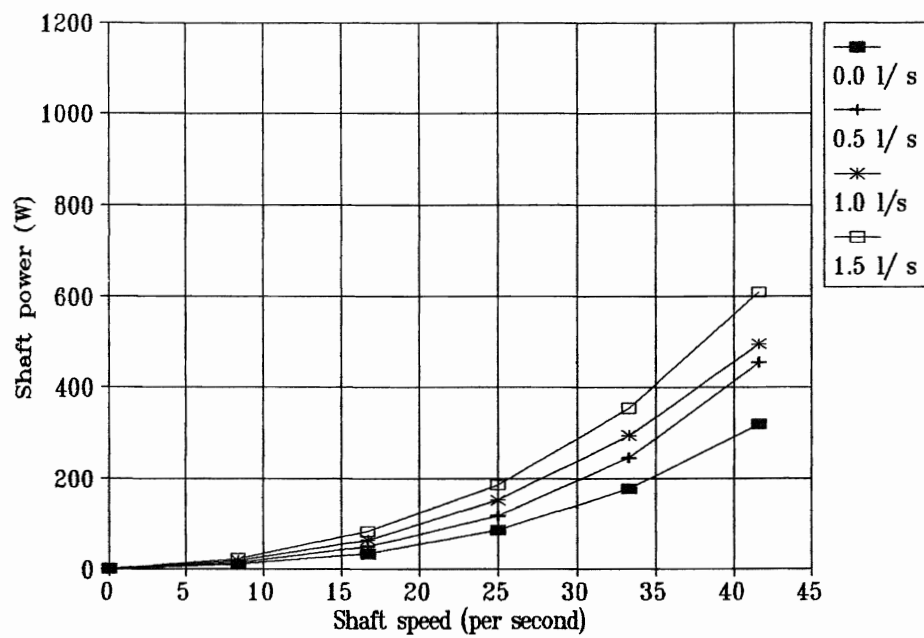


Figure 5.3 Shaft power input against shaft speed

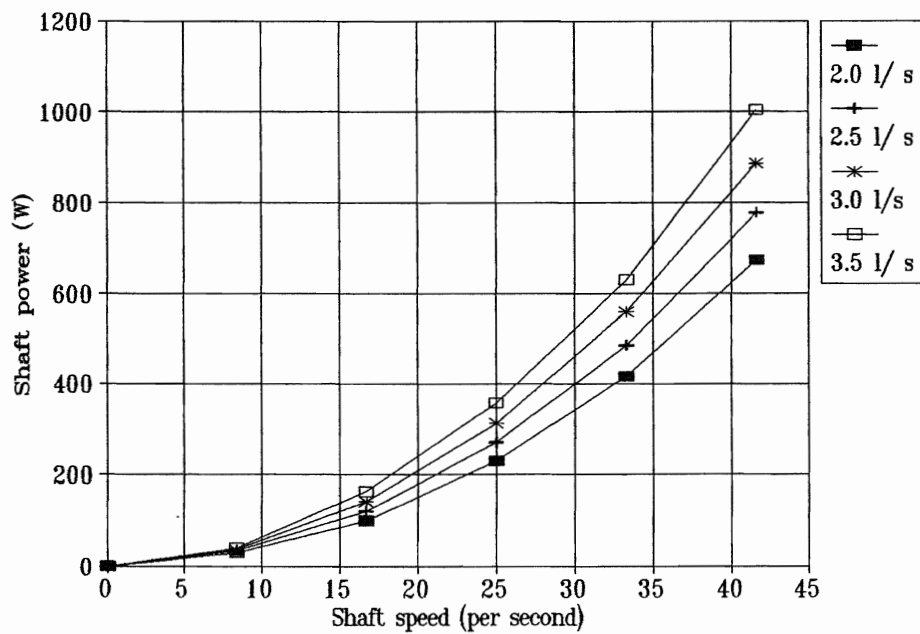


Figure 5.4 Shaft power against shaft speed

It was assumed that an expression of the form $P_{\text{SHAFT}} \propto N^\gamma$ existed and a regression was performed on these data, to determine the exponent γ . It was found that this exponent was between 1.99 and 2.2 for the data collected. The limited amount of stirred tank literature suggests that this value for γ is low, and that it should be closer to 3. Studer (1992) reports an exponent 2.5 in a similar expression for turbulent energy dissipation rate in a rotor/stator mixer. Rotor/stator mixers are very different to stirred tanks and further investigation will be required to investigate this result.

5.2.2 P_{PUMP} and differential pressure

Differential pressure is made up of pressure losses due to the sudden expansion (as the inlet pipe enters the mixer), sudden contraction (at the outlet pipe), other similar losses (e.g. wall friction) and a pressure rise due to the pumping effect of the rotor.

The sudden expansion and sudden contraction losses are shown in Figure 4.6, which includes the curve for pressure loss against capacity with the rotor stationary. When the data were analysed it was found that these losses were proportional to the pipe velocity to the power 1.85. Given the relatively large error associated with these small pressure measurements (Section 4.2.2) the exponent is close enough to 2 and the resulting loss coefficient was found to be approximately 2.8. This is higher than would be expected if the loss at zero shaft speed was due to the sudden

expansion and contraction alone (Miller 1990) in which case the loss coefficient would be approximately 1.5. This result implies that other losses are contributing to the overall loss. When the tests were conducted at zero shaft speed, the shaft was rotated to check for dependence upon the relative positions of gaps and teeth between the rotor and the stator. The pressure loss was found to be independent of this.

The pressure loss due to expansion and contraction losses has not been subtracted from the pressure rise plots as the losses through the inlet and outlet would be altered by any recirculation that may occur in these regions. However, it is included in plots of the overall energy balance (Section 5.3) to show their relative importance.

The pumping power was calculated by multiplying throughput by the pressure rise. The pumping power is shown against shaft speed in Figures 5.5 and 5.6.

Using these data it was possible to plot pumping efficiency curves for the shaft speeds used. The pumping efficiency is defined as $\eta = P_{\text{PUMP}} / P_{\text{SHAFT}}$. These curves are given in Figure 5.7. The maximum pumping efficiency obtained was of the order of 12%. This is much lower than the pumping efficiencies of centrifugal pumps, which can be above 90%. The flow rate at the best efficiency point increases with shaft speed.

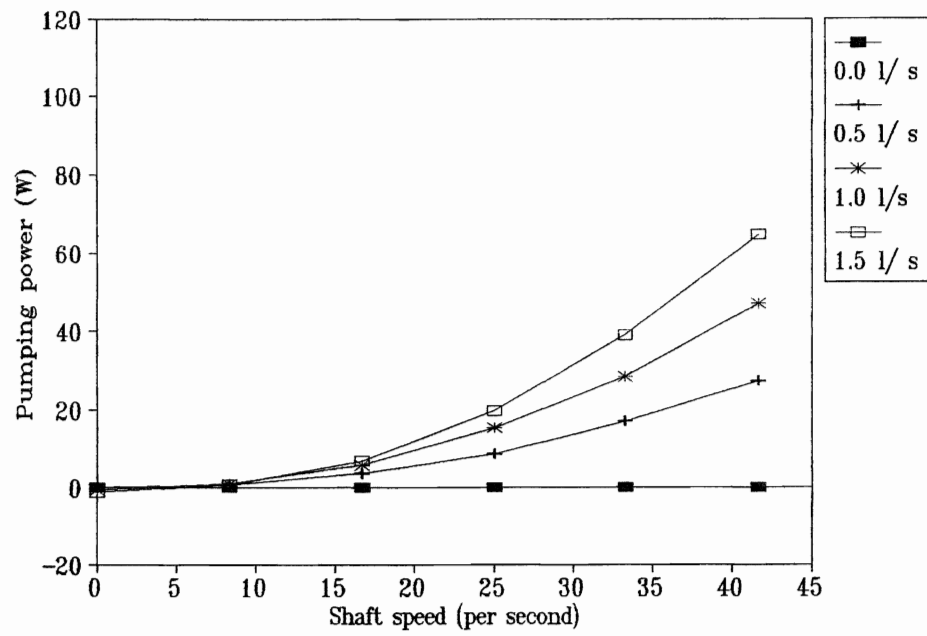


Figure 5.5 Pumping power against shaft speed

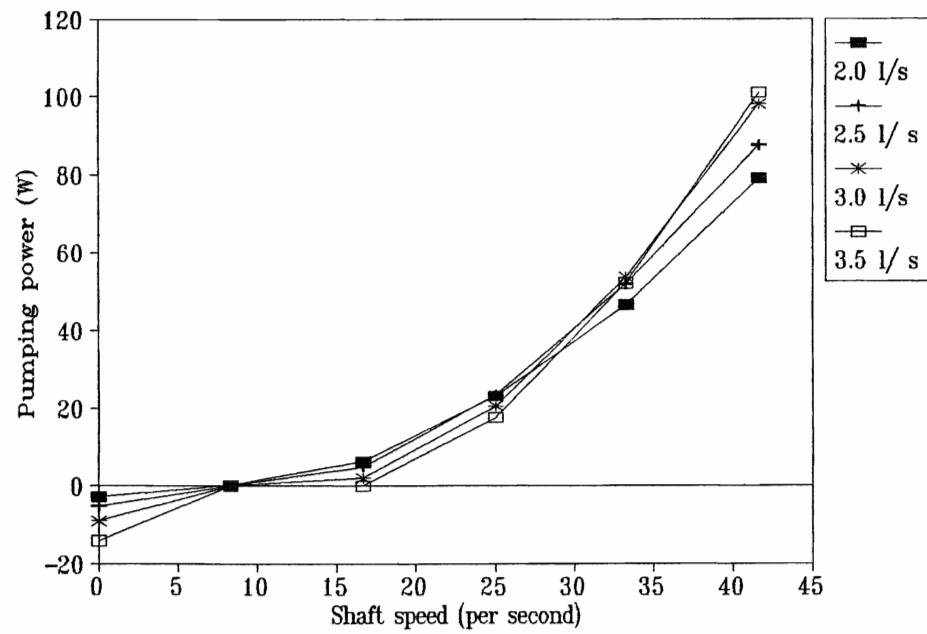


Figure 5.6 Pumping power against shaft speed

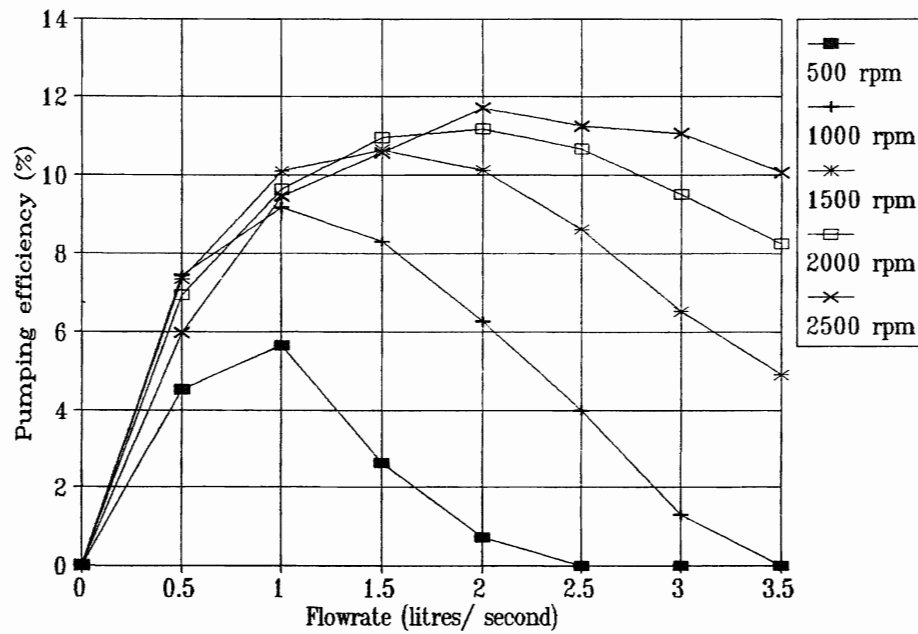


Figure 5.7 Pumping efficiency curves against flow rate

5.2.3 P_{DISK} , disk friction

Disk friction was not directly measured for this study. However a theoretical expression for the effects of disk friction is available and power calculated using this expression will be compared with the overall shaft power to give an assessment of this loss. As stated in Section 2.6.3 disk friction losses do not contribute to mixing, so their importance must be established in order that the proportion of shaft torque that is 'usefully' dissipated can be found.

Stepanoff (1957) reports that P_{DISK} , the power absorbed by disk friction, for a thin disk, will be:

$$P_{\text{DISK}} = K N^3 D^5 \quad (5.1)$$

where K is an empirical constant.

An expression, of the same basic form, that takes into account impeller thickness is given by Jekat (1986):

$$P_{\text{DISK}} = K N^3 D^4 (D + 5E) \quad (5.2)$$

where E is the impeller thickness. Stepanoff (1957) states that these expressions give values that are too high. Using Equations (5.1) and (5.2), together with the constant supplied by Jekat, the following values are obtained.

Table 5.1 Disk friction calculated assuming thin disk and including thickness

N (RPM)	N (RPS)	P_{DISK} Thick (W)	P_{DISK} Thin (W)
0	0.0	0.0	0.0
500	8.3	0.8	0.4
1000	16.7	6.1	2.9
1500	25.0	20.5	9.6
2000	33.3	48.6	22.9
2500	41.7	94.9	44.7

5.2.4 P_{MECH} , Mechanical losses

Mechanical losses due to the seals and bearings were measured by removing the rotor and spinning the shaft alone in water. It was assumed that the drag on the spinning shaft due to the water is small. The results obtained (Section 4.2.3) show that the mechanical losses are a small part of the overall picture and that they are unpredictable. The measurements taken are tiny in comparison to the transducer rating, this means that accuracy will be lower than for other shaft torque measurements with the rotor in place. Seal mechanical losses are not 'well behaved' (Nau, 1993), but these results do provide an order of magnitude value for the seal losses that can be incorporated into the overall energy balance.

Seal and bearing losses were found to be independent of throughput. This partially confirms the assumption that drag on the shaft is small in comparison to the seal and bearing losses.

A variation in measurements between tests was found, this is thought due to the unpredictability of friction. It is possible that changes in the fluid film that exists between the seal faces is responsible for this variation.

Seal losses are not subtracted from the shaft torque values in this thesis. However these losses account for the small kink in the curves for shaft torque against rotational speed (Figure 5.1).

5.2.5 P_{LEAK} , leakage losses

No flow visualisation was carried out in this study and therefore no information is directly available on internal recirculation or short circuiting through the mixer.

During measurement of the mixer volume, when the mixer was filled with water and the outlet flange was open to the atmosphere, the rotor was spun at low speed. A strong swirl was noticed at the outlet flange. This is evidence that secondary flow exists in the outlet pipe, but the strength of this flow cannot be judged from these observations.

The long tailing of the residence time distribution curves is evidence that a significant amount of recirculation does occur in the mixer. However, recirculation from the mixer volute back through the stator and into the rotor region is not thought possible. Any recirculation on the delivery side of the rotor will occur in the mixer volute. Similarly it is possible that inlet recirculation occurs. The sensitivity of the temperature probe measurements to the inlet pipe transducer position may confirm this. Flow visualisation will be required to further examine leakage flows.

5.2.6 $P_{\text{HYDRAULIC}}$, hydraulic losses

The hydraulic and leakage losses make up the remainder of the shaft power input not accounted for by any of the loss components above. It is not, therefore, possible to provide an accurate figure for the hydraulic losses alone. In Section 5.3 the overall energy balance will be examined and conclusions will be drawn on the magnitude of the hydraulic losses.

5.2.7 Differential temperature

It was hoped that differential temperature measurements could be used to provide additional information on the efficiency of the mixer. This has not proved to be true. Heat from sources other than the hydraulic losses in the mixer appear to be masking the results.

A typical differential temperature measurement was 0.069 K, at 3.5 litres/ second and 2000 rpm. The power input required to cause this rise is given by:

$$P = \rho Q c_s \Delta T \quad (5.3)$$

where c_s is the specific heat capacity of water.

For the given temperature rise and flow rate, this gives a power consumption of

approximately 1000 W. This value is more than the shaft input power at this condition (630 W).

The motor on the mixer was rated at 4.0 kW, and the highest shaft power was around 1 kW. The motor was therefore operating at a point well away from its best electrical efficiency and this could explain the warmth of the shaft. The pipe (PVC) may also allow heat to pass from the water to the surrounding air and vice versa.

If this technique is to be used successfully in the future it will be necessary to thermally isolate the mixing unit from its surroundings. This would involve using a heat insulator to isolate the motor heat source from the fluid. This may be possible at the flexible couplings on the current mixer, although heat generated in the bearings and seal will not be isolated. The pipe work would need to be insulated well to prevent heat transfer through the pipe wall.

This method would also require a number of downstream temperature probes, to ensure that all of the kinetic energy has been passed from the large turbulent eddies to the small, energy dissipating, eddies where it is dissipated as heat.

The extremely accurate and precise platinum resistance thermometers used were mostly useful in demonstrating the problems with this technique.

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The extremely accurate and precise platinum resistance thermometers used were mostly useful in demonstrating the problems with this technique.

5.3 OVERALL ENERGY BALANCE

The overall energy balance for the rotor/stator mixer has been examined at 2000 and 2500 rpm, as shown in Figures 5.8 and 5.9. These figures are of the same form, but very different in appearance, as the one presented by Stepanoff (Figure 2.9).

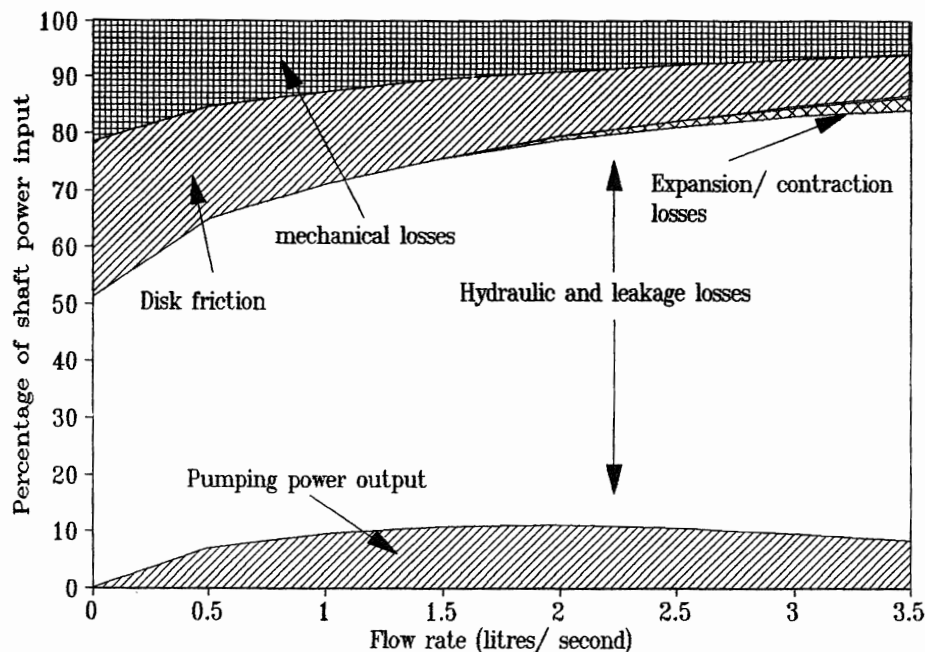


Figure 5.8 Overall energy balance for mixer at 2000 rpm

This shows how much of the total shaft power input is transformed into the various components of the energy balance. The disk friction and the mechanical losses are the highest of the estimates that were calculated or measured, therefore this gives the lowest hydraulic and leakage losses from the data collected. The pressure loss

measured when the rotor was stationary is included and is labelled as the expansion/ contraction loss.

This plot shows that, at 3.5 litres/ second, approximately 70 % of the shaft power input is given over to a combination of hydraulic and leakage losses. This implies that approximately 450 W is accounted for by these losses.

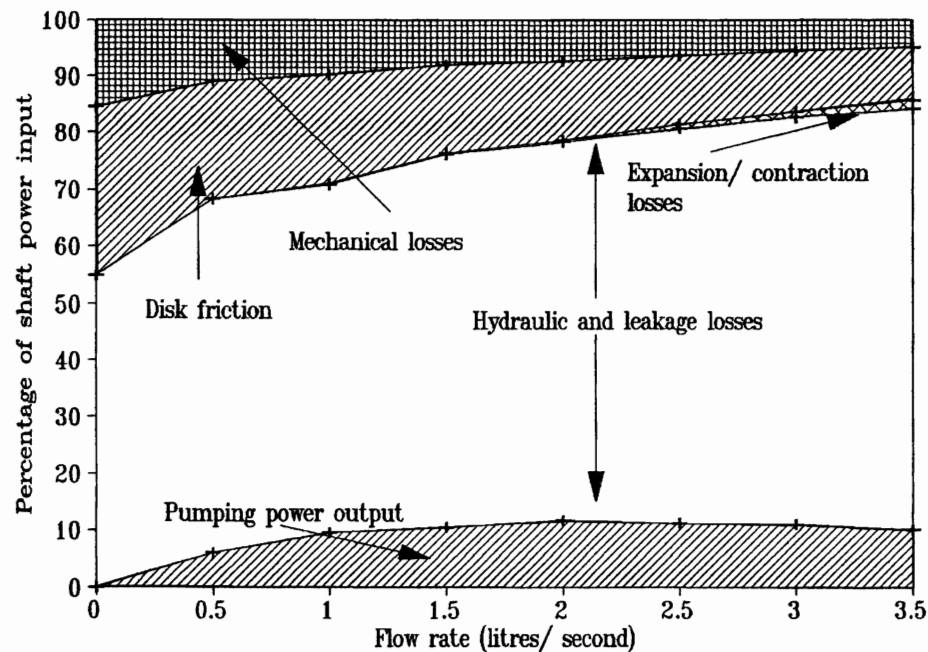


Figure 5.9 Overall energy balance for mixer at 2500 rpm

Figure 5.9 shows the same plot for 2500 rpm. The two plots are very similar. Once again approximately 70 % of the shaft power is accounted for by hydraulic and leakage losses, in this case corresponding to approximately 700 W. The volume of the mixer is $1.59 \times 10^{-3} \text{ m}^3$. Therefore these losses amount to $\sim 450 \text{ W/ kg}$, averaged

over the whole mixer volume.

It was shown in Section 5.2.1 that $P_{\text{SHAFT}} \approx \text{const} \times N^2$. Therefore, if the percentage of shaft input given over to hydraulic and leakage losses remains at approximately 70 % of the shaft power, at 3500 rpm this would amount to ~900 W/kg, averaged over the whole mixer. There is further likely to be strong inhomogeneity in the values of ϵ throughout the mixer, with the highest rates in the region of the rotor and stator. The inhomogeneity of ϵ in stirred tanks has been measured by Wu and Patterson (1989), local values of ϵ that were twenty times greater than the tank average were reported.

The proportion of this loss that can be attributed to turbulent energy dissipation, ϵ , cannot be estimated from this work. However, Bolzern and Bourne (1985) and Hearn (1992), both using reactive mixing experiments, report that the proportion of the total losses given over to turbulent energy dissipation in centrifugal pumps is between 1 and 4%. This would correspond to up to approximately 40 W/kg in a rotor/stator mixer. However it has been shown by Studer (1992) that turbulent energy dissipation rates in the reaction zone of a small scale rotor/stator mixer can exceed 1000 W/kg.

Rotor/stator mixers are not designed to be hydraulically efficient, as are centrifugal pumps. Therefore a greater proportion of the total losses will be due to turbulent energy dissipation. Also the inhomogeneity throughout the mixer of the energy

dissipation implies that the turbulent energy dissipation rate will be higher still in the rotor/stator region.

In conclusion it is therefore estimated that, at 3500 rpm, the turbulent energy dissipation rate in the rotor/stator region of the mixer will exceed 1000 W/ kg.

5.4 RESIDENCE TIME DISTRIBUTION

The residence time distribution curves obtained were normalised, making the time axis dimensionless by dividing by the theoretical residence time and making the area under each curve equal to 1, see Section 3.4. This section will discuss the features of these curves, providing possible physical explanations. Models for characterising the overall behaviour will be tested.

5.4.1 General observations

The residence time distribution curves are given in Appendix A. The following general observations can be made about these curves:

- i) The curves exhibit substantial tailing. This shows that tracer is trapped in the mixer for up to 5 dimensionless times. If the mixer were used for multiple chemical reactions see Section 2.3.3, then reaction products will remain in the mixer to compete for fresh reagents. Product distribution

therefore will be affected.

- ii) Some of the curves (particularly at low speed, high flow) resemble those obtained in continuous stirred tank reactors (Levenspiel, 1972).
- iii) As the flow rate is increased the curves begin to more closely resemble dispersed plug flow traces. The peak shifts towards a dimensionless time of one.
- iv) At low shaft speeds fluid is short circuiting through the mixer. This is shown by early peaks, the curves for 500 rpm at 1 and 1.5 litre/ second show this best (Figures A.1 and A.2). Short circuiting means that the whole of the mixer volume is not being used effectively.
- v) As the shaft speed and flow rate are increased the curves become appreciably smoother. This gives a possible measure of the macromixing in the mixer. Smoother curves show that the coarse scale variation is reduced. Because of the size of the probe (~ 4 mm) it is not thought that a quantitative measure of the concentration variation from the curves would be valid. This is because the probe size is of the same order as the expected scale of the segregation.

However it is possible to say, qualitatively, that the macromixing is

improved as flow rate and shaft speed are increased.

vi) As the shaft speed is increased the peak value of concentration is lowered.

This is a sign that the short circuiting is being lessened.

vii) As the flow rate increases the dimensionless time at which the first sign of tracer arrives also increases.

5.4.2 Modelling of the residence time distribution curves

In order to model the behaviour of a reactor, knowledge of the turbulent energy dissipation rate is required. It is also necessary to have knowledge of the residence time distribution. For rotor/stator mixers, models of the residence time behaviour may (in the future) be combined with models for the mixing behaviour to give an overall reactor model.

Figure 5.10 shows an experimental residence time distribution curve. Superimposed on this curve are two model residence time distribution curves. The first of these is the stirred tank model:

$$C = e^{-t/\bar{t}} \quad (5.4)$$

The second is a model that represents a plug flow region in series with a stirred tank (Levenspiel, 1972):

$$C = \left(\frac{V}{V_m} \right) e^{-\left(\frac{V}{V_m} \right) \left(t' - \frac{V_p}{V_m} \right)} \quad (5.5)$$

where V_m and V_p are the volume of the stirred tank and plug flow regions.

From the RTD curves in Appendix A, there is a dependence on N and Q . In order that the model given in Equation (5.5) can be used, the values of V_m and V_p must be varied.

It is possible that a relationship can be found so that the model parameters can be calculated given the operating condition. Flow visualisation would assist in the physical validation of the models given.

In Figure 5.10 the value of V_m is $1.32 \times 10^3 \text{ m}^3$ and V_p is $0.27 \times 10^3 \text{ m}^3$. In Figure 5.11 these values are 1.19×10^3 and $0.4 \times 10^3 \text{ m}^3$ respectively.

5.4.3 Modifications to improve the residence time distribution

It is probable that recirculating flows in the mixer volute are responsible for the backmixed behaviour, as recirculation through the rotor/stator region itself is unlikely. Modification to the design of the mixer to improve the residence time

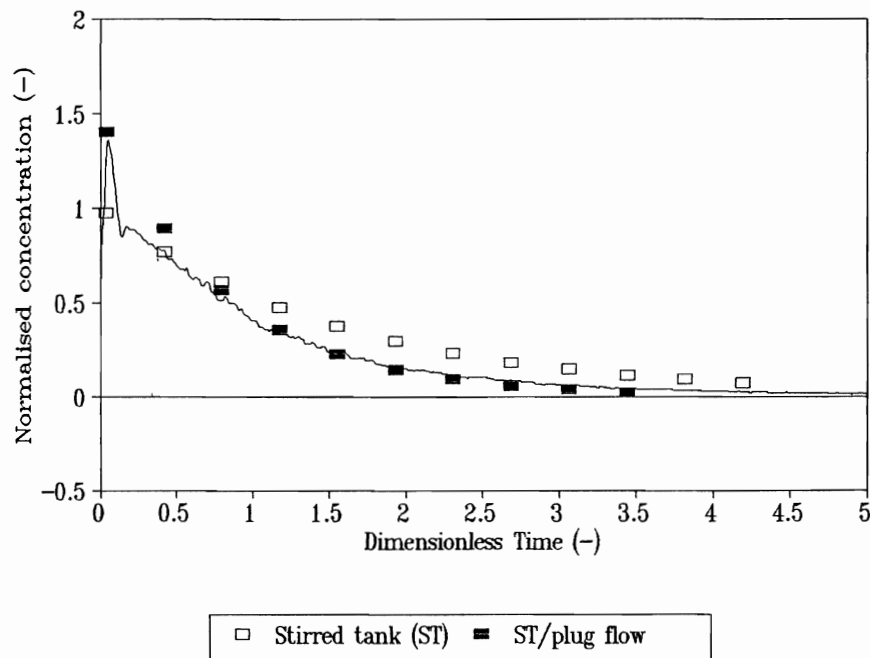


Figure 5.10 Residence time distribution curve (1500 rpm, 1 l/sec), showing comparison with models

distribution might need to reduce the size of the volute to cause the fluid to pass through the volute more quickly.

It is suggested that if the volute is reduced in size (by packing) and if the number of outlets is increased (reducing the average path length to exit) then the residence time distribution would better approximate plug flow. This modification would also reduce leakage losses.

All of the residence time distribution curves presented in this thesis have been adjusted by subtracting the effects of (assumed) plug flow regions upstream and downstream of the mixer.

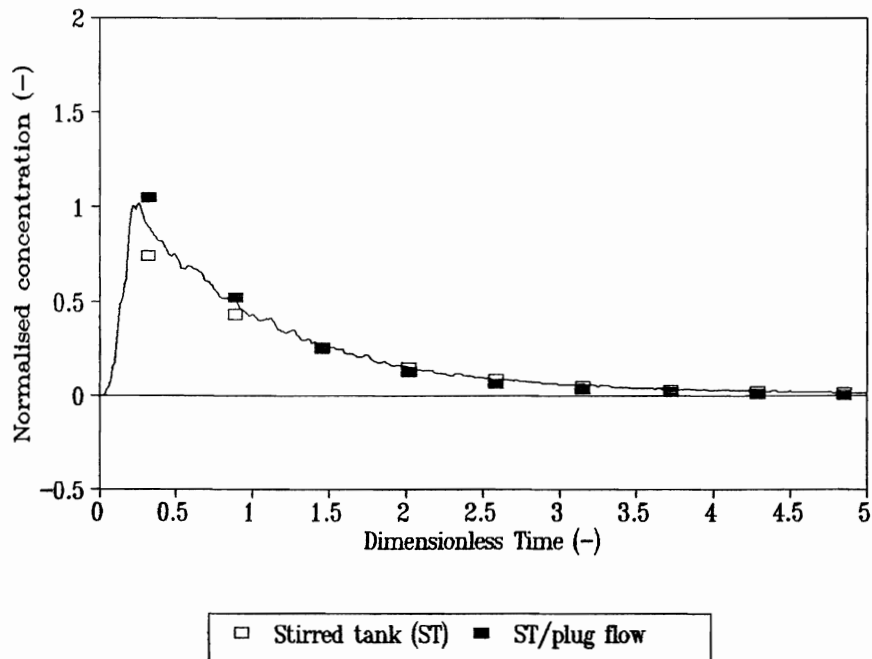


Figure 5.11 Residence time distribution curve (2000 rpm, 1.5 l/s) showing comparison with models

5.5 MIXING IN THE ROTOR/STATOR MIXER

5.5.1 Macromixing

The rotor and stator may be responsible for a number of macromixing mechanisms.

A toothed rotor, spinning at speed, would slice up an element of fluid as it passed through. This "mechanical" mixing has relatively little to do with fluid mechanics and would have a significant effect regardless of flow conditions elsewhere in the mixing chamber. An analogous mechanism may occur as a tangentially moving element of fluid passes a set of stator teeth.

Rotors can impart very high shear stresses, especially when held close to a motionless stator. This will have the effect of thinning out segregated bands of reactant.

The high rates of turbulent energy dissipation estimated (Section 5.3) also imply that macromixing through turbulent diffusion will be rapid (Section 2.3.1).

5.5.2 Micromixing

It is estimated, in Section 5.3, that ϵ exceeds 1000 W/ kg in the rotor/stator region of the mixer. This figure cannot be confirmed until it has been verified (e.g. by using a competitive chemical reaction experiment, see Section 2.3.2). However $\epsilon > 1000$ W/ kg implies that micromixing will be rapid (engulfment times t_E will be of the order of $\frac{1}{2}$ msec).

5.6 CONCLUSIONS

- The data indicate that $P_{\text{SHAFT}} \propto N^2$. The value of the exponent was expected to be ~ 3 . Further work is required to investigate this finding.
- The maximum pumping efficiency obtained is $\sim 12\%$. This rotor/stator mixer is an inefficient pump.

- Energy balances show that up to 70% of power input is dissipated by hydraulic and leakage losses.
- It is estimated that ϵ exceeds 1000 W/kg in certain regions of the mixer.
- RTDs show that the mixer exhibits backmixing; modifications that should make the RTD better approximate plug flow have been proposed.
- Both macro mixing and micromixing are rapid in the mixer.
- The differential temperature technique for monitoring the energy dissipated to heat by losses did not give directly useful results, it is thought that heat from other sources is masking results. Further development will be required.

CHAPTER 6. CONCLUSIONS AND RECOMMENDATIONS FOR FUTURE WORK.

6.1 CONCLUSIONS

- Very little previous work has been published on rotor/stator mixing.
- Rotor stator mixers are rarely reported as being used as chemical reactors. Available literature suggests that they provide promising environments for conducting fast chemical reactions.
- Energy balances in this work have shown that the rotor/stator mixer is an inefficient pump, up to 70 % of power input is dissipated by hydraulic and leakage losses.
- Residence time distributions show that the mixer exhibits backmixed behaviour. Modifications have been suggested to make the RTD better approach plug flow.
- Both macro and micromixing have been shown to be rapid.

6.2 RECOMMENDATIONS FOR FUTURE WORK

To confirm the conclusions of this study, and to better understand rotor/stator mixing the following are recommended:

- The range of test data should be extended. To date this mixer has not been used over its full operating range: higher shaft speeds require examination. Both the residence time distribution and power balance studies are in need of more data.
- The effect of rotor/stator head design needs to be investigated. A number of different designs are commercially available. It is necessary to find out what features of design can be changed to improve mixing performance.
- Flow visualisation. Although there are intrinsic difficulties in doing this, it will produce information that will lead to improved mixing performance; flow visualisation in pumps has led to improved pumping performance.
- Competitive chemical reactions can be used to locate the best location to feed in reagents for fast chemical reactions. Work has already been done in smaller equipment with competing chemical reactions to investigate micromixing (see Studer, 1990, Bourne and Garcias-Rosas, 1985). Work is needed at a large scale, and new reagent feed positions should be examined.

- Numerical modelling of the flows in rotor/stator mixers may lead to a better understanding of their performance. It will be difficult to model the whole flow and many simplifying assumptions will be required.

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APPENDIX A Experimental results

This Appendix contains experimental results. Differential pressure and shaft torque measurements are given 'as read'. The residence time distribution curves have been normalised and adjusted to account for assumed plug flow regions upstream and downstream of the mixer.

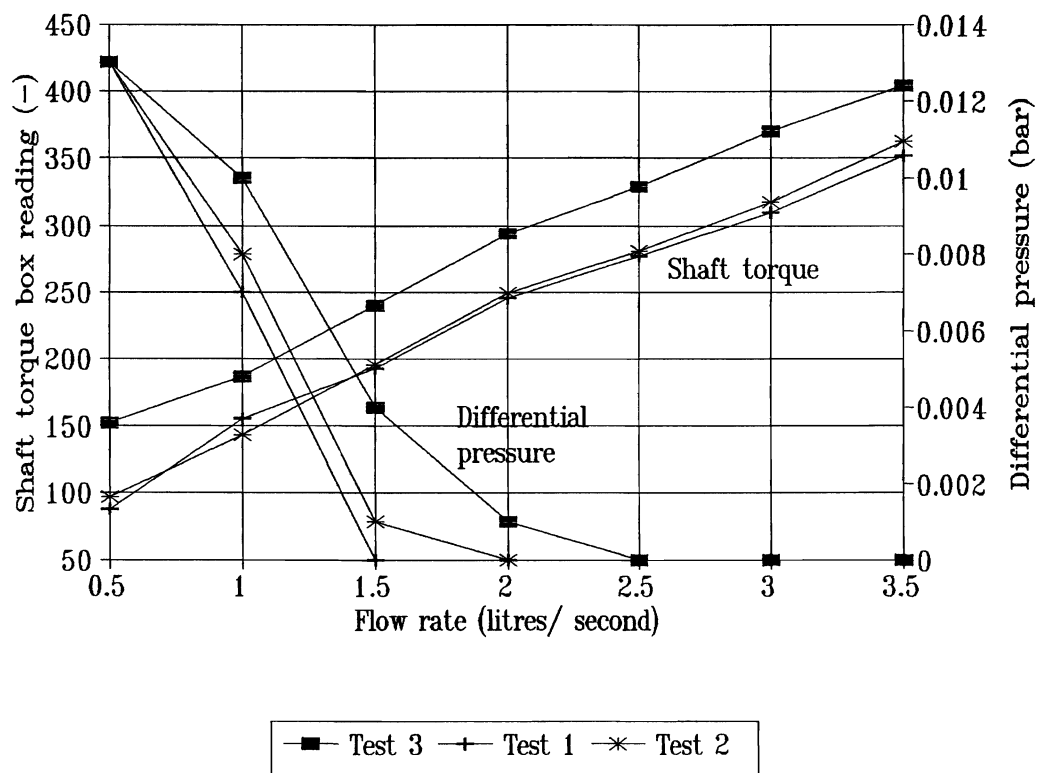


Figure A.1 Shaft torque and differential pressure at 500 rpm

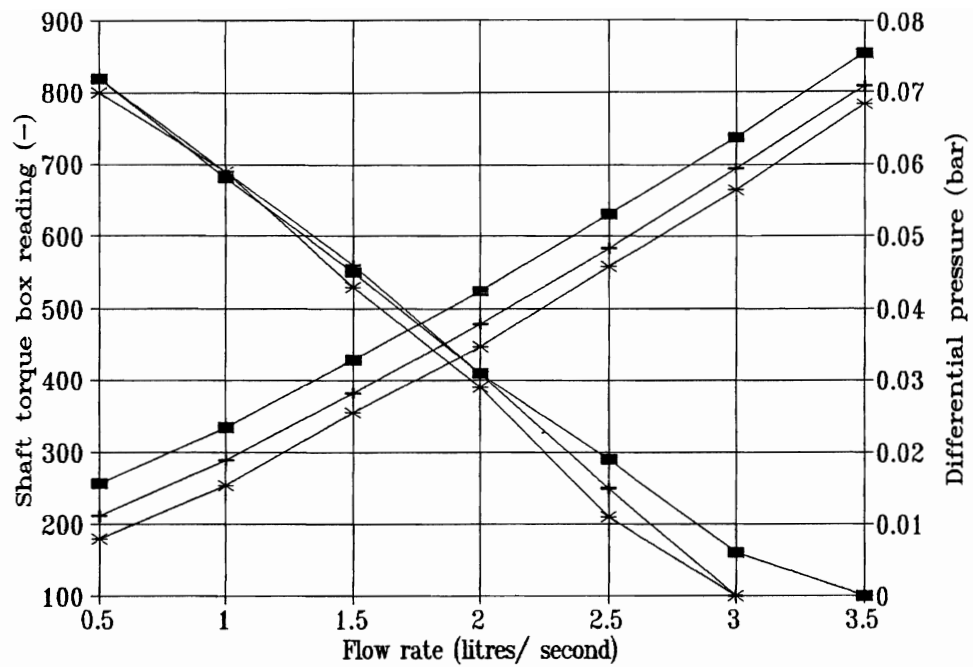


Figure A.2 Shaft torque and differential pressure at 1000 rpm

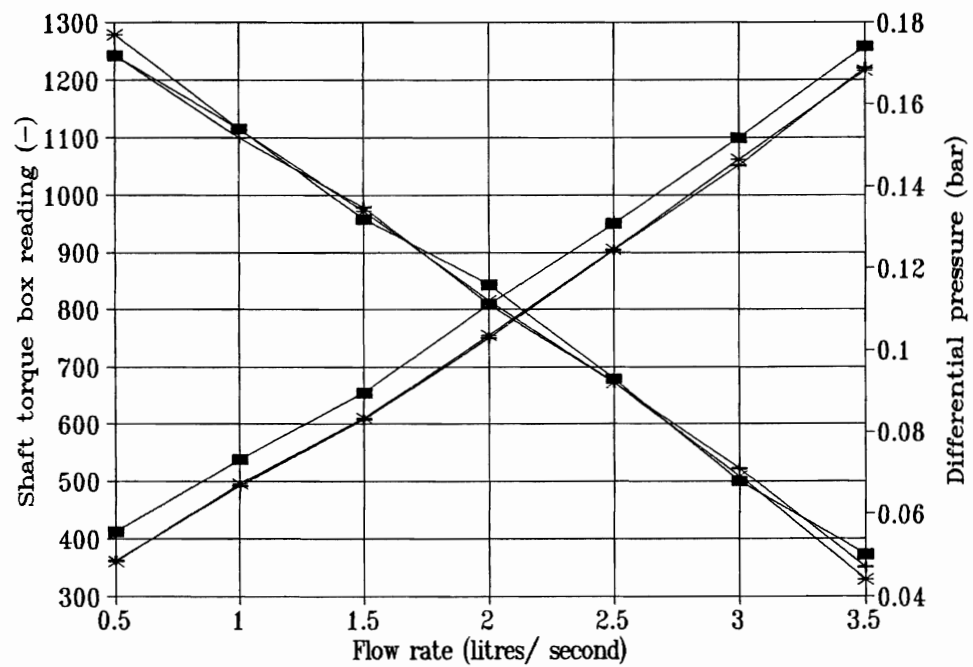


Figure A.3 Shaft torque and differential pressure at 1500 rpm

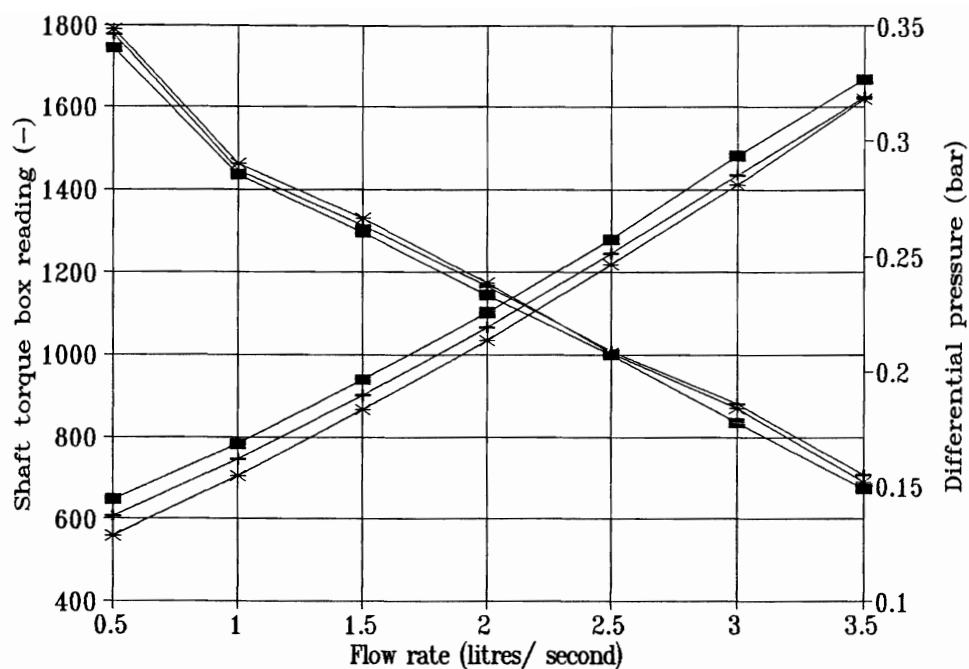


Figure A.4 Shaft torque and differential pressure at 2000 rpm

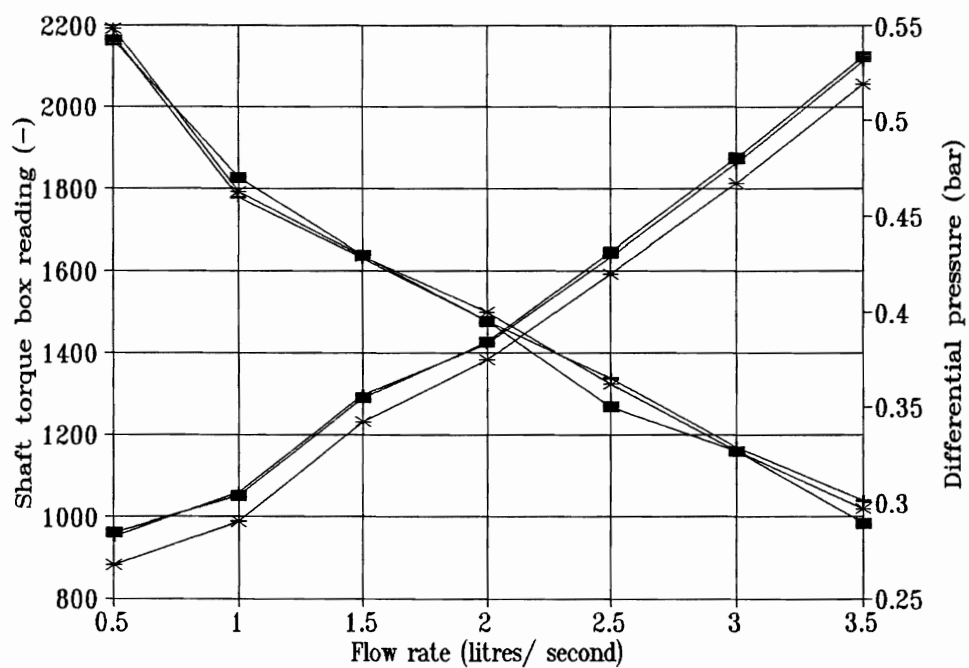


Figure A.5 Shaft torque and differential pressure at 2500 rpm

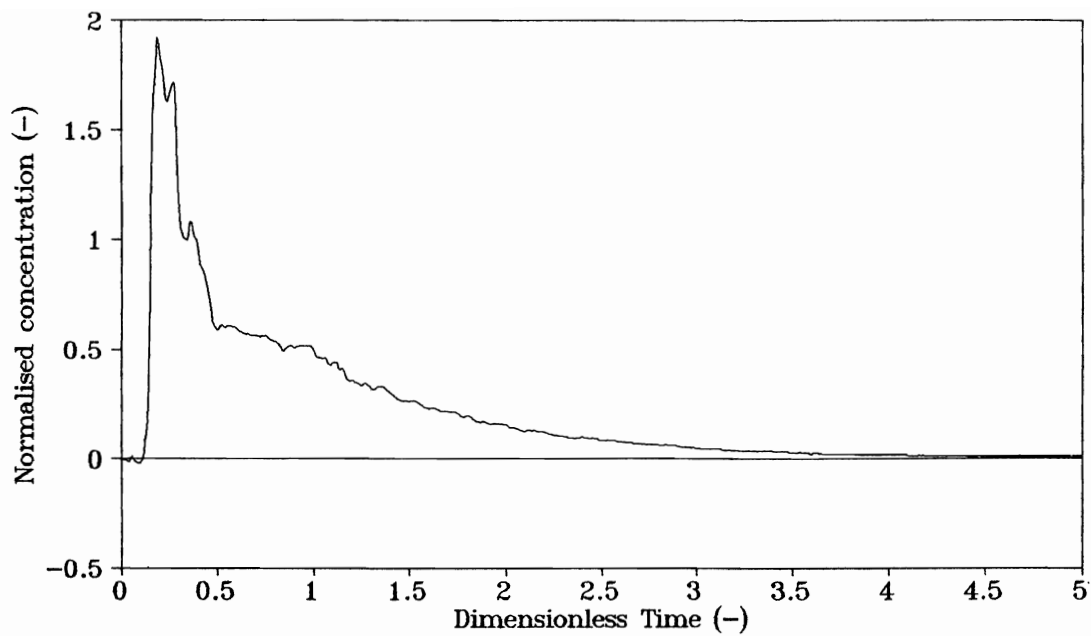


Figure A.6 **500 rpm, 1 litre/ second**

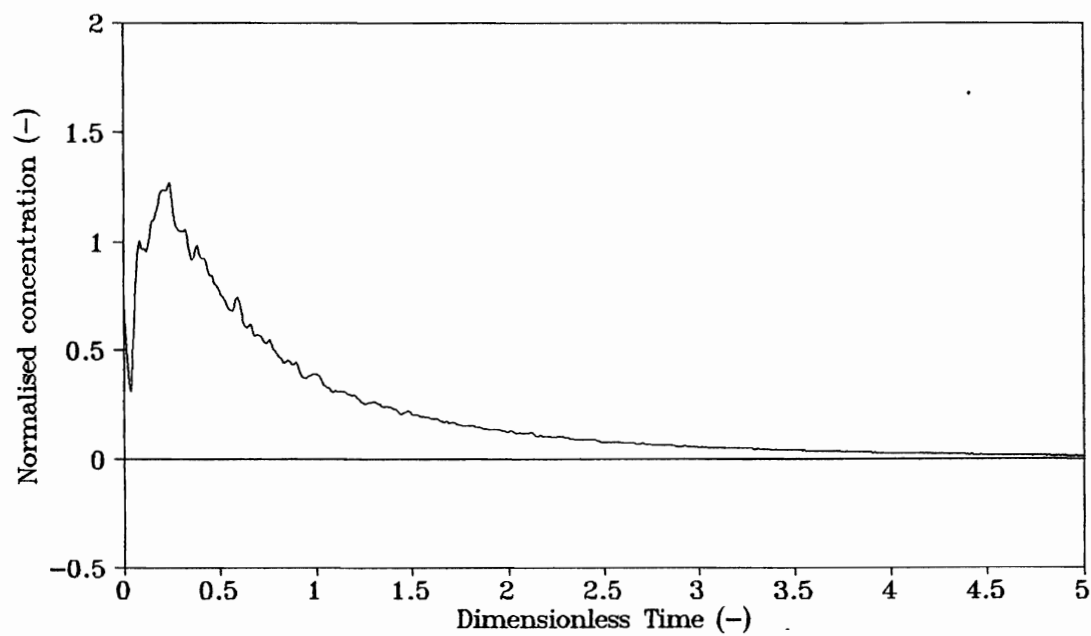


Figure A.7 **1000 rpm, 1 litre/ second**

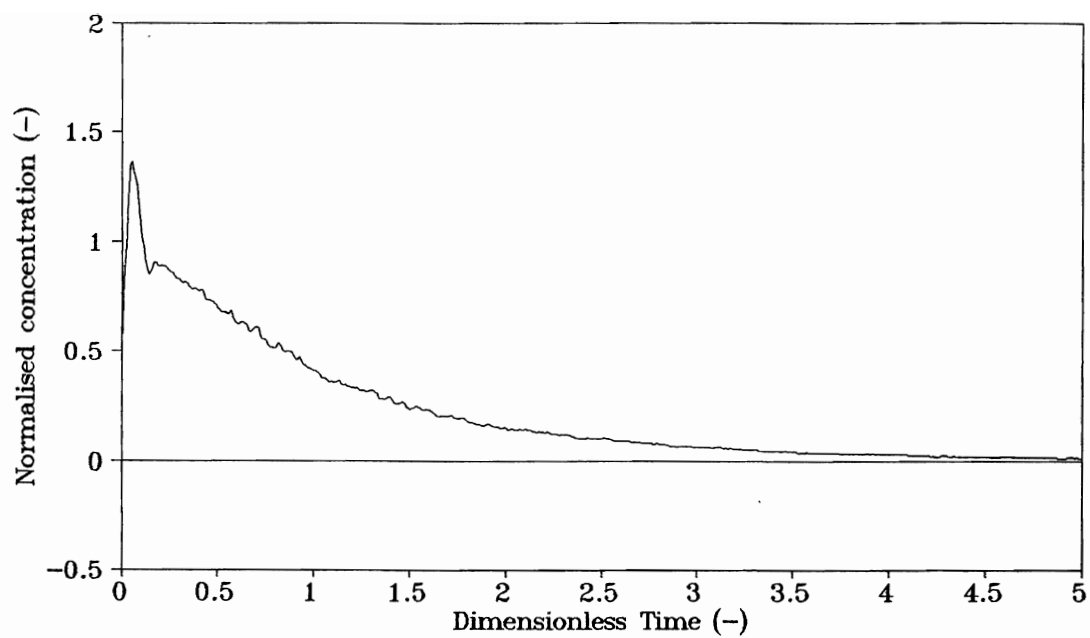


Figure A.8 1500 rpm, 1 litre/ second

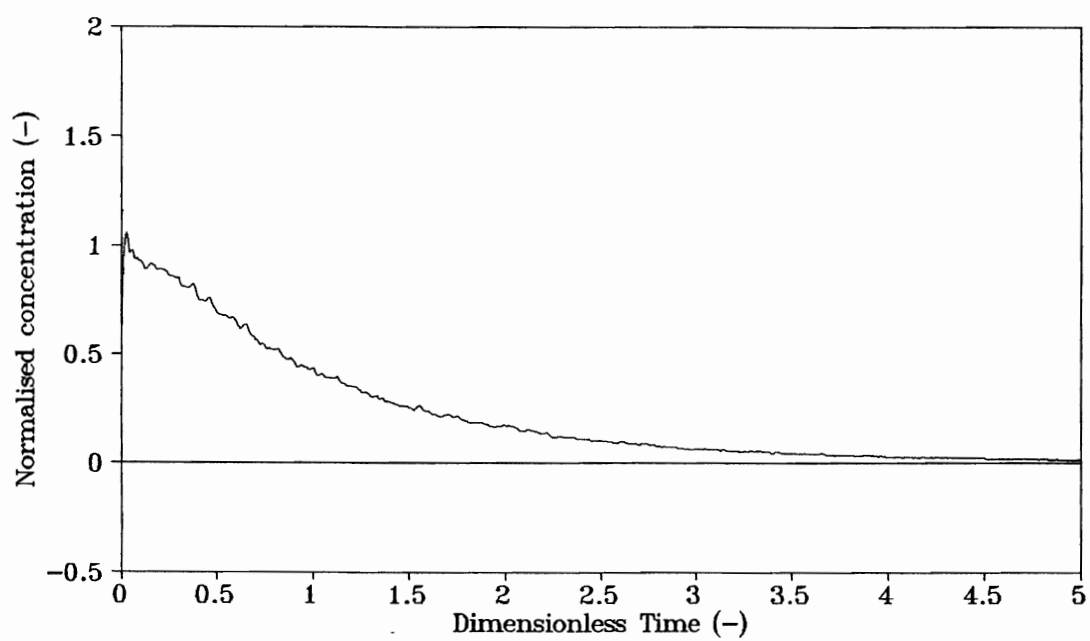


Figure A.9 2000 rpm, 1 litre/ second

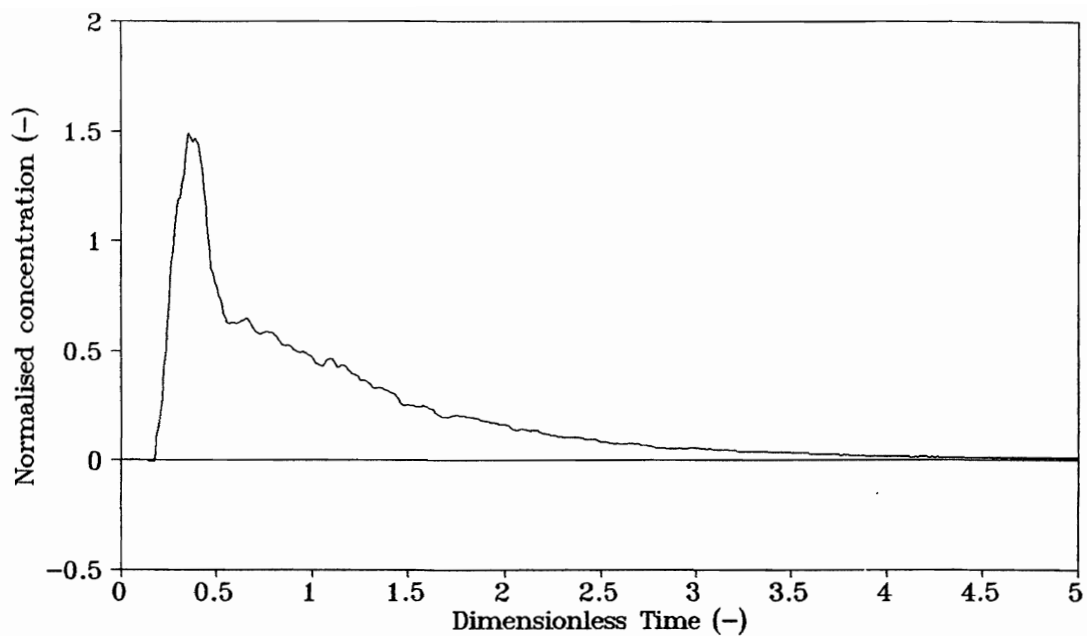


Figure A.10 **500 rpm, 1.5 litres/ second**

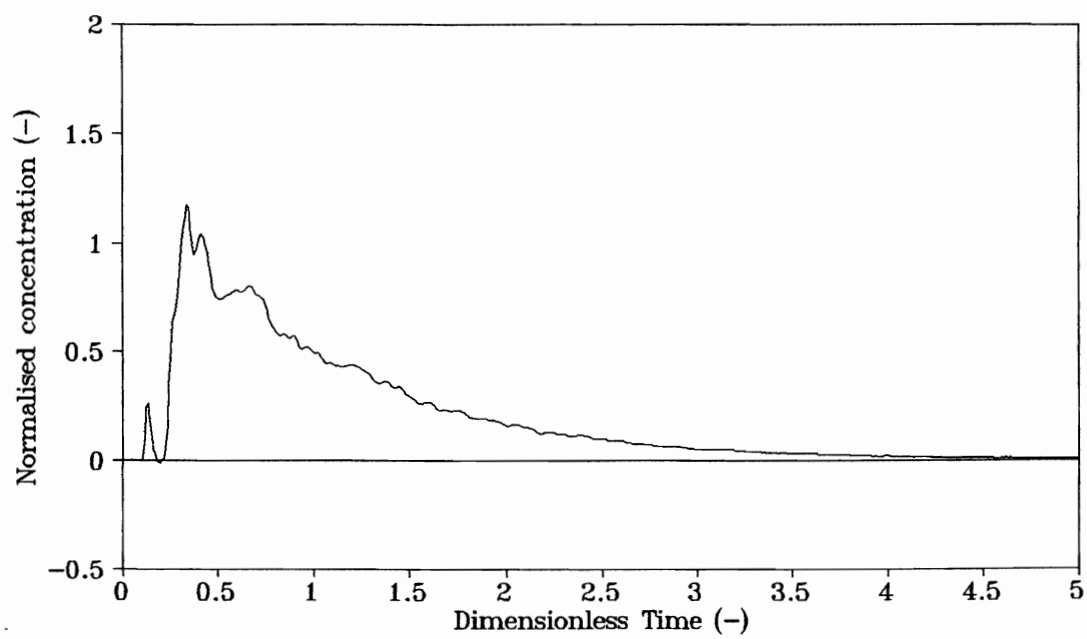


Figure A.11 **1000 rpm, 1.5 litres/ second**

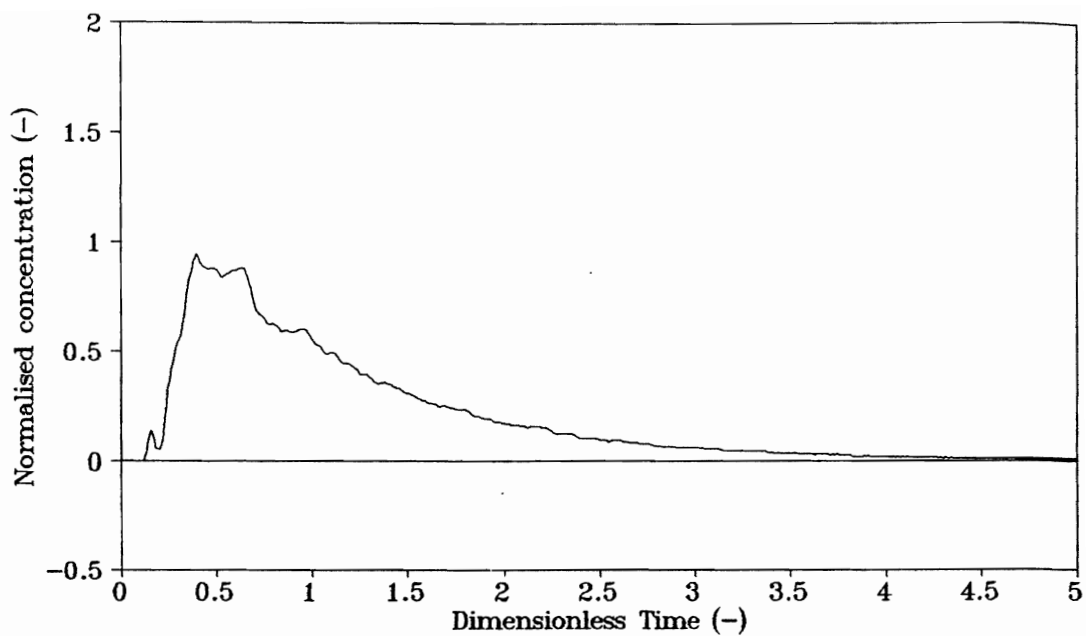


Figure A.12 1500 rpm, 1.5 litres/ second

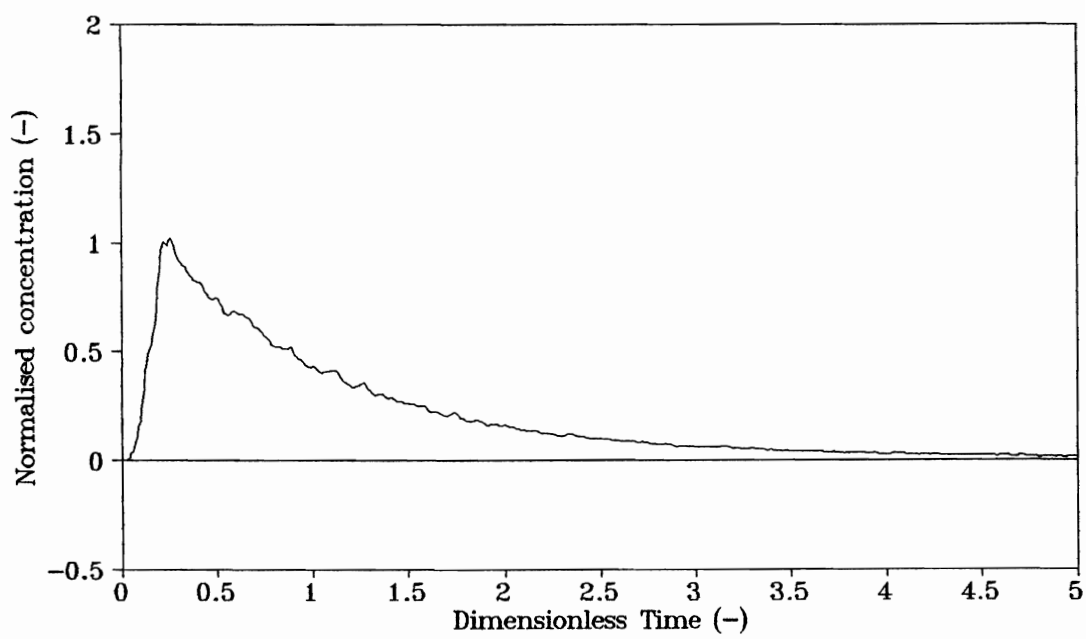


Figure A.13 2000 rpm, 1.5 litres/ second

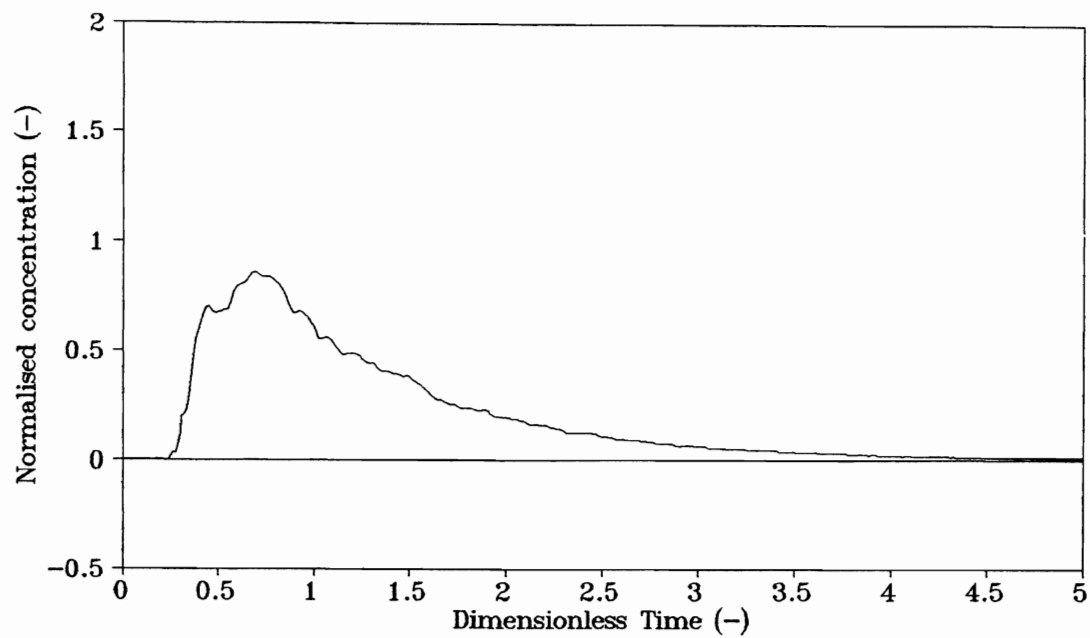


Figure A.16 **1500 rpm, 2.0 litres/ second**

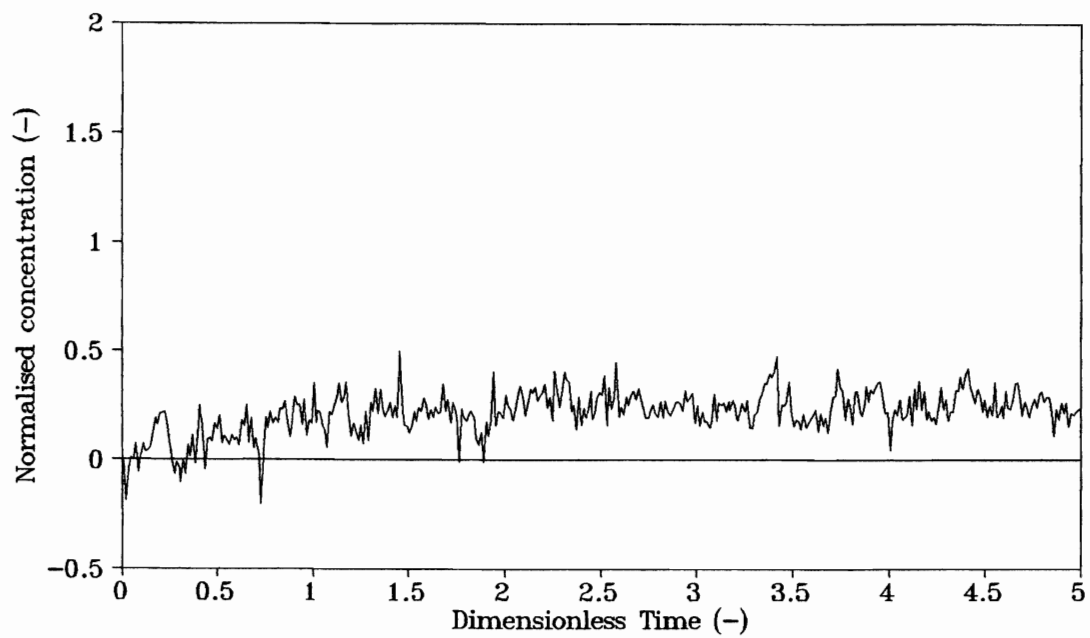


Figure A.17 **2000 rpm, 2.0 litres/ second**

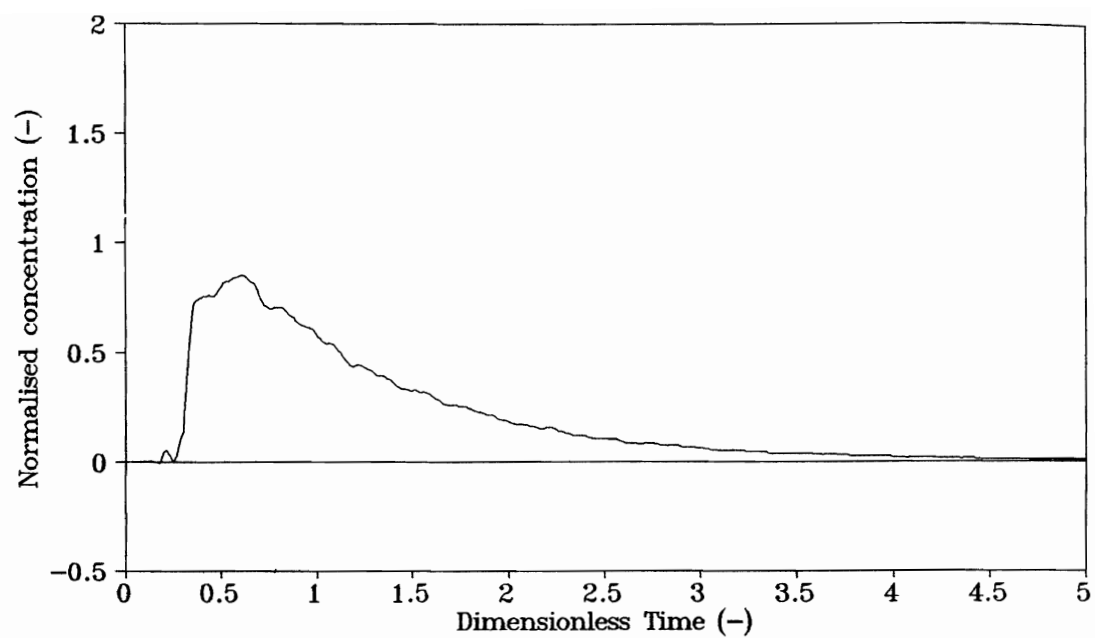


Figure A.18

2500 rpm, 2.0 litres/ second

APPENDIX B Calibration details

Table B.1 Differential pressure transducer calibration
(C:\mphil\q\pres.wk1)

input units bar	output units volts	Regression	
0	-0.001		
Output:			
0.2	0.999	Constant	0.000164
0.4	2	Std Err of Y Est	0.000114
0.6	3.001	R Squared	1
0.8	4.002	No. of Observations	11
1	5.002	Degrees of Freedom	9
0.8	4.002		
0.6	3.002	X Coefficient(s)	0.1999
0.4	2	Std Err of Coef.	0.000021
0.2	1		
0	0		

Table B.2 Torque calibration for rotor/ stator mixer (C:\mphil\q\torque)

lever arm = (m)		0.35			
Vol water	Mass kg	Force N	Torque N m	Meter -	
0	0	0.000	0.000	0	
94	0.094	0.922	0.323	177	
167	0.167	1.638	0.573	312	
204	0.204	2.001	0.700	383	
264	0.264	2.590	0.906	501	
328	0.328	3.218	1.126	616	
395	0.395	3.875	1.356	741	
454	0.454	4.454	1.559	855	
496	0.496	4.866	1.703	929	
556	0.556	5.454	1.909	1047	
636	0.636	6.239	2.184	1197	
700	0.7	6.867	2.403	1322	
765	0.765	7.505	2.627	1447	
819	0.819	8.034	2.812	1550	
892	0.892	8.751	3.063	1688.5	
947	0.947	9.290	3.252	1795	
990	0.99	9.712	3.399	1880	
1046	1.046	10.261	3.591	1985	
1130	1.13	11.085	3.880	2145	
1156	1.156	11.340	3.969	2195	

Regression Output:

Constant	0.010719	
Std Err of Y Est		0.007036
R Squared		0.999969
No. of Observations		20
Degrees of Freedom		18
X Coefficient(s)	0.001806 (Nm/ unit)	
Std Err of Coef.	0.000002	

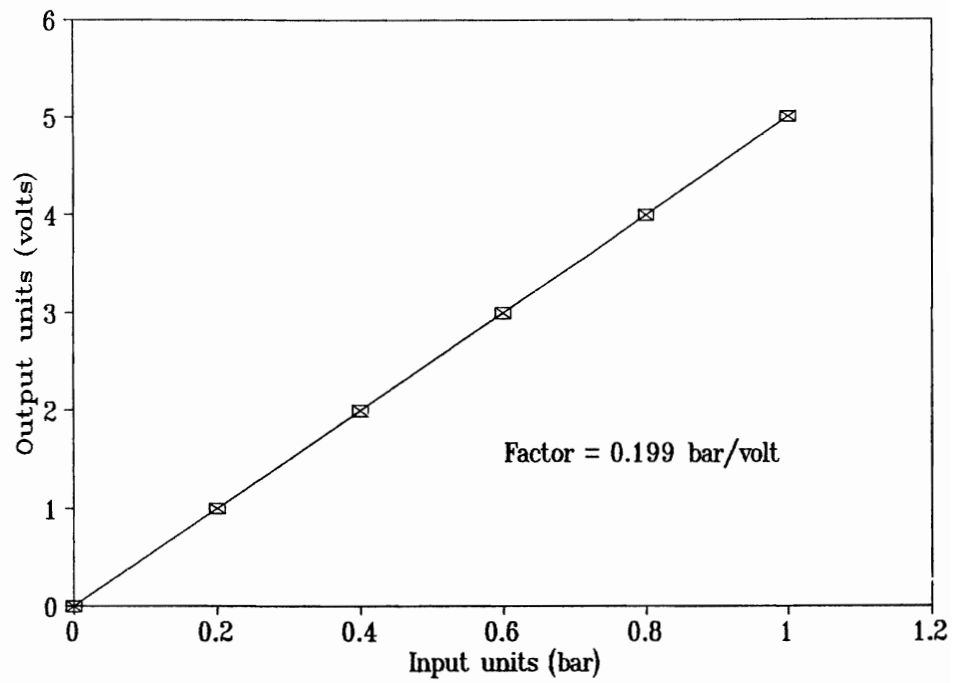


Figure B.1 Calibration curve for differential pressure transducer.

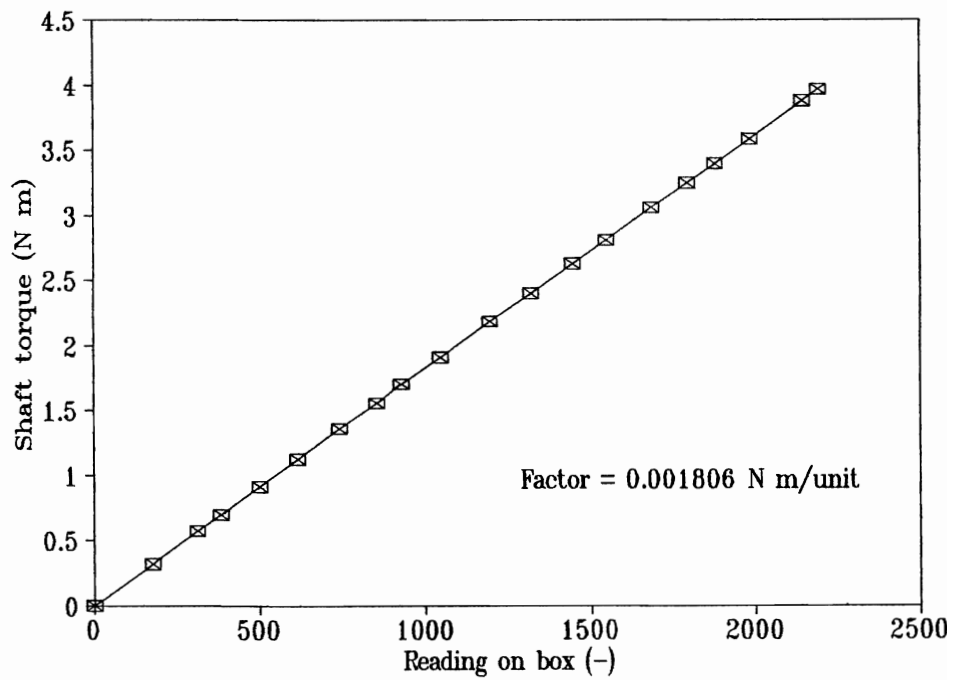


Figure B.2 Calibration of torque transducer

APPENDIX C EXPERIMENTAL EQUIPMENT DETAILS

Silverson in-line 425 LSM rotor/stator mixer, machine No 425LSM276

**Silverson Machines Limited,
Waterside,
Chesham,
Buckinghamshire,
England HP5 1PQ.**

Flygt BS 20066.171 submersible pump

**ITT Flygt Limited,
Colwick,
Nottingham,
England NG4 2AN**

LABTECH NOTEBOOK version 5

**Laboratories Technologies Corporation,
400 Research Drive,
Wilmington MA,
USA 01887**

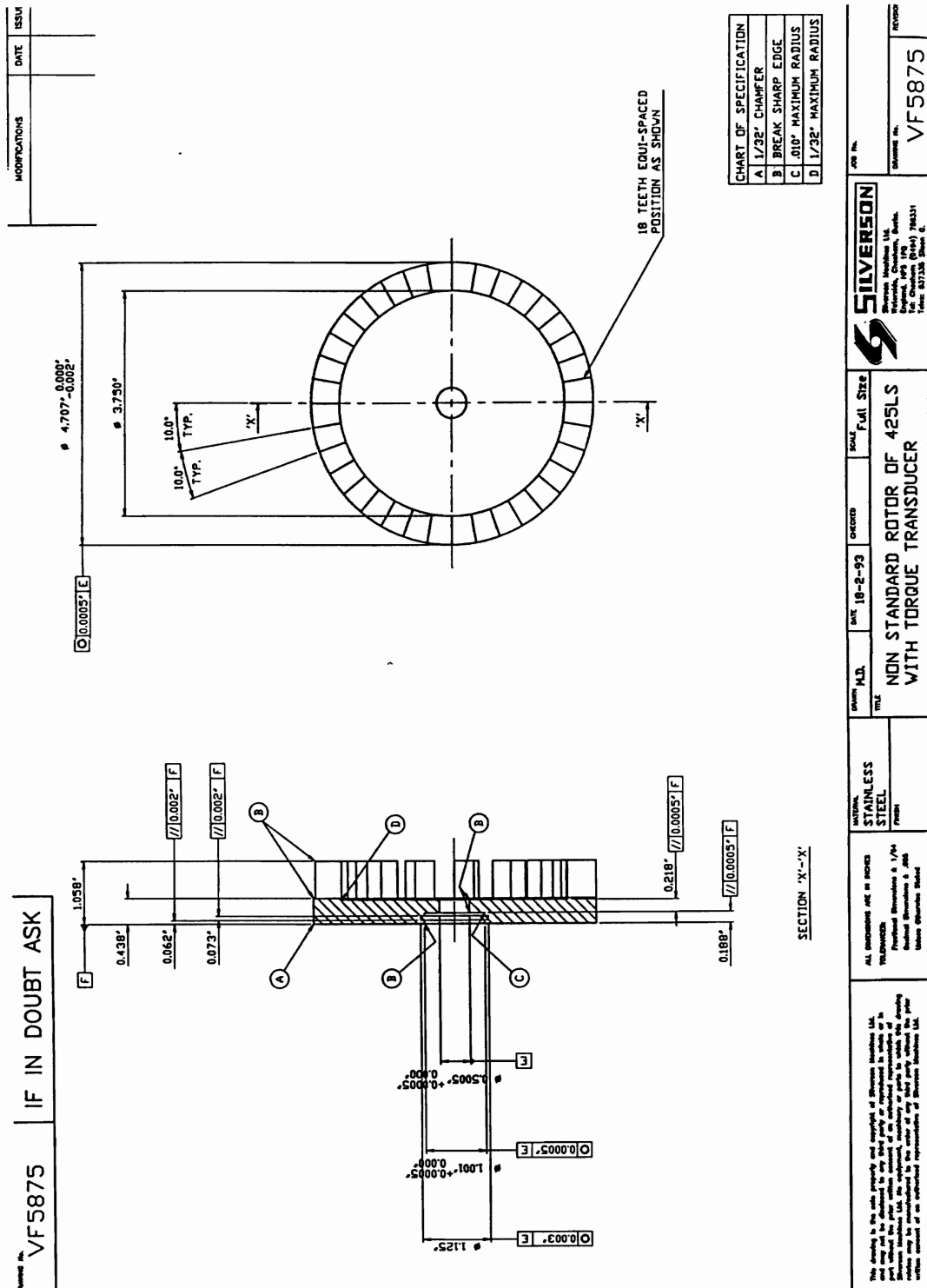
T 30 FD torque transducer and KMN 902.D signal amplifier

**Hottinger Baldwin Messtechnik GMBH
Postfach 4235,
Im Tiefen See 45,
D-6100 Darmstadt
Deutschland**

Druck 1 bar differential pressure transducer

**Druck Limited,
Fir Tree Lane,
Groby,
Leicester,
England LE6 0FH**

APPENDIX D Drawings of rotor and stator



VF5876 IF IN DOUBT ASK

MODIFICATIONS	DATE	ISSUE

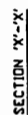


CHART OF SPECIFICATION	
A	1/32" CHAMFER
B	BREAK SHARP EDGE
C	.010" MAXIMUM RADIUS
D	1/32" MAXIMUM RADIUS
E	TO BE CONCENTRIC

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