The coil pumps

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The Coil Pumps

by

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A Doctoral Thesis submitted in partial fulfilment of the requirements for the award of

Doctor of Philosophy of the Loughborough University of Technology

March 1988

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A family of Coil Pumps has been developed at Loughborough University over the last 10 years by the author. The coil pump, sometimes known as the 'hydrostatic pump', was known and used in the 18th and 19th centuries, since then it appears to have been forgotten. Laboratory investigations lead to a theory for predicting the behaviour of this little known pump. The theory is based on an assessment of the boundary levels of the liquid plugs within the loops. From this work, two types of suction pump were derived and the lift pump theory was developed and adapted to predict their behaviour.

One suction pump was based on one helical coil and it required a regulated air supply. The second pump used two helical coils 'back to back' one taking water into the pump and the other (of larger capacity) withdrawing it.

Laboratory tests were carried out on a number of versions of both suction pumps and the experimental results agreed well with those produced by the theory.

Practical applications of this family of pumps included a low cost stream powered lift pump, a dosing pump, a sewage suction pump and a sewage treatment process.
STATEMENT OF RESPONSIBILITY

All the theoretical work presented in this thesis was originated and developed by the Author. The laboratory results on the lift coil pump have been taken from investigations carried out by research and project students where the author acted as supervisor. Independent checks on the accuracy of their work were made by the author.

The laboratory work and the development of the theory for the two suction pumps has been the sole responsibility of the Author.
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a
An
Cd
Con(n)
D
Dimensioner
Dimer%
Din
Exp(n)
Fout
he(n)
H(n)
Hai
Hat
Hlift
Hout
Hsuc
fili(n)
k
Lai

Cross sectional area of helical pipe
Identification tag for an air plug in loop n.
Coefficient of discharge
the relative movement of the trailing edge (or leading) of liquid plug Wn towards the inlet as the liquid plug moves through its last revolution.
Effective diameter of the drum. The centre of drum to centre of helical pipe.
Distance from the water surface to the lowest point of the centre line of the helical pipe.
Percentage depth of immersion. The ratio of Dimer to the effective drum diameter.
Internal diameter of helical pipe.
Internal diameter of the inlet tube
the relative movement of the trailing edge of liquid plug Wn away from the inlet as the liquid plug moves through its last revolution.
Force on outlet bend.

Acceleration due to gravity - 9.81 m/sec².

Pressure head difference across liquid plug in loop n.
Effective head across a liquid plug allowing for dynamic losses.
Absolute pressure head in air plug n.
Ambient pressure at the inlet to the pump.
Atmospheric pressure head.
Height of liquid surface in header tank above pump centre line.
Pressure head at the outlet to the pump.
Suction lift (pump centre line to sump water surface).
Change in pressure head from ambient to the pressure in loop n.
Coefficient of compression or expansion of the air in the coil.
Length of the incoming air plug at inlet pressure.
Lao  Length of the outgoing lair plug at outlet.
L(a(n)) Length of the air plug in loop n.
Lit  Effective length of the inlet tube.
Lrel  Relative movement of the inlet liquid plug causing 'swing-in'.
Lwi  Length of the incoming liquid plug at inlet.
Lwo  Length of the outgoing liquid plug at outlet.
Lw(n) Length of the liquid plug in loop n.
Lx  Effective length of the inlet arm on the suction pump.

N  Total number of loops on the coil.
Nb  Number of bubbling loops.
Ni  Number of loops on the inlet coil (Double coil suction pump).
No  Number of loops on the outlet coil (Double coil suction pump).
Ns  Number of spilling loops.
Nns Number of non-spilling loops.
Nsb  Number of spilling-bubbling loops.
n  Loop number from the inlet-variable in a coil.
ns  Loop number of the last non-spilling plug from the inlet.
nb  Loop number of the last non-bubbling plug from the inlet.
P  Power consumption of the pump.
p  Pressure.
ρ  Density of the fluid.

Qa  Measured flow rate.
Qd  Dosing flow rate for the double coil suction pump.
Qw  Measured flow rate for the suction pump.
Qt  Theoretical flow rate.

R  Effective radius of drum.
r  Radius of helical pipe.
RotW(1) Rotational swing occurring whilst liquid plug enters through the inlet.
RotA(1) Rotational swing occurring whilst air plug enters through the inlet.

S  Speed of rotation of the drum - rpm

t  Time (variable).
T  Temperature.
Tain Time for air plug to enter the inlet arm.
Td  Time for one revolution of the drum.
Tol Tolerance used in the convergence methods.
Identification tag for a liquid plug in loop n.

\( \beta(n) \) Rotation of lower edge of the liquid plug from the vertical.

\( \Theta(n) \) Rotation of upper edge of the liquid plug from the vertical.

\( \Omega_{sp} \) Angle of liquid plug from vertical required to cause spilling.

\( \Omega_{bub} \) Angle of liquid plug from vertical required to cause bubbling.

\( \%sw \) Amount of 'swing-in' of the liquid plug expressed as a percentage of the maximum possible.

Subscript "i" refers to the variable associated with the inlet coil.
Subscript "o" refers to the variable associated with the outlet coil.
Superscript "'" refers to a modified value of a variable.
CHAPTER 1

INTRODUCTION

This thesis is concerned with predicting the behaviour of a family of simple pumps which we have termed the "coil pumps". We did not invent the coil pump since the idea stretches back over hundreds of years. What we have achieved is to propose a theory to predict the behaviour of the original pump which, to distinguish it from the later types, we have called the "lift coil pump". From the work on this pump we developed a new version which lifts a liquid by suction. This new version lead to two variations, one we called the "single coil suction pump" and the other is called the "double coil suction pump".

The thesis is basically divided into three parts, Chapters 3 to 6 are concerned with the behaviour of the lift coil pump. When we started this investigation in 1974, there was very little published information on the behaviour of this pump, so we started with a minimum of work to build on.

Chapters 7 to 9 deal with a new version of the coil pump which works by suction and has a single coil.

Finally, Chapters 10 to 12 describe the workings of the other type of suction pump which has a double coil.

In the above description I have used the plural pronoun "we". This is intended to reflect the contribution of two research students, five undergraduate project students and two excellent technicians.

The rest of this chapter provides an introductory description of the three types
Figure 1.1 Diagrammatic Layout of the Lift Pump
of coil pump which are the concern of this thesis.

1.1 A DESCRIPTION OF THE LIFT COIL PUMP

In its simplest form, the pump consists of a length of flexible tube wound around the outside of a cylindrical drum which has its long axis horizontal. This drum is then submerged to about half its depth in a liquid. A flexible pipe wound around the drum forms a helix and one end of the pipe is secured to the drum and left open. This forms the inlet to the pump. See Figure 1.1. The other end of the pipe is connected via a sealed rotary joint to the delivery pipe. The rotary joint lies on the drum's axis and connects the pump outlet to the stationary delivery pipe.

Rotation of the drum at a slow speed, for example 3 rpm, causes the inlet to take in alternate plugs of liquid and air which then move through the helical pipe, into the rotary joint, and up the delivery pipe. The pressure required to force the liquid and air up the delivery pipe is developed by each liquid plug acting as a manometer and sustaining a pressure difference across it. The flow rate is governed by the amount of liquid taken in by the inlet during one rotation of the drum and, of course, the drum speed.

In Figure 1.1, the pipe is wound around the drum in the form of a helix, other investigators have used a spiral winding. The reasons for choosing the helical winding are explained in Section 4.1.

1.2 A DESCRIPTION OF THE SINGLE COIL SUCTION PUMP

The suction pump is similar in construction to a lift coil pump, but here the delivery pipe is lowered into a sump whose liquid level is well below the pump, see Figure 1.2. In order to operate, the pump needs an air supply since it cannot take in air through the suction pipe. This air comes from an air inlet
Figure 1.2 Diagrammatic Layout of the Single Coil Pump
Figure 1.3 Diagrammatic Layout of the Double Coil Pump
on the stationary part of the inlet tube directly behind the rotary joint. The air inlet is regulated by a needle valve which controls the air drawn in from the atmosphere.

If the pump is rotated in the opposite direction to the lift pump, it will pull up liquid from the sump, provided it is primed beforehand. The hydraulic action of this pump is similar to the lift pump, as it takes in air and liquid to form plugs which act as manometers as they move through the helix.

### 1.3 A DESCRIPTION OF THE DOUBLE COIL SUCTION PUMP

This pump consists of two helical coils wrapped around a drum. One coil is wound clockwise and the other is wound anti-clockwise. The two coils are connected in series and a suction pipe is connected at their junction. This suction pipe passes through the rotary joint and down into the sump. See Figure 1.3. The drum is submerged to about half its depth in a liquid.

As the drum is turned, the inlet coil takes in plugs of liquid and air, and passes them to the outlet coil. The outlet coil then pulls the plugs through its own helix and back out into the holding tank. If the capacity of the outlet coil is greater than the inlet coil, the liquid from the sump will be pulled up the suction pipe into the outlet coil to make up the volume difference. The hydraulic action of this pump is again based on the manometric effect.

The capacity of the outlet coil can be made greater than the inlet coil either, by using a larger diameter for the helical pipe or, by increasing the effective diameter of the drum. This pump is self priming and self regulating.
CHAPTER 2

AIMS OF THE THESIS

The main aim of this thesis is to describe the work carried out at Loughborough on a family of coil pumps.

The investigation of these pumps covers a span of fourteen years and is made up of a sequence of projects which I have undertaken or supervised. There was no specific single aim for all the work carried out at Loughborough since this work was made up of a number of projects, each having its own aim. The projects and individual aims can be divided as follows.

(a) The first main project was carried out after a series of final year student projects. This 18 month project was financed by SERC and was undertaken by R. Annable, a research student. The aim of his work was to carry out a laboratory study of the lift coil pump and propose a theory which would predict the basic hydraulic behaviour of the pump. Annable was also asked to look at possible practical applications for the pump. From the laboratory work, he developed a low-cost, stream-powered pump which fulfilled his latter aim.

From Annable's work, two new projects developed, these were the Coil Treatment Unit described in section (b) and the suction coil pumps described in (c).

(b) It became apparent that a modified version of the lift coil pump could be used to treat sewage since the coil pressurizes both liquid and air, and these could be used to form the basis of a treatment process. S. Galvin, a research student, carried out an investigation into the application of the
coil pump theory to this treatment process. His work is not covered by this thesis, but his laboratory experiments did assist in furthering the theory of the lift pump.

(c) The two versions of the suction coil pump were conceived at approximately the same time and I carried out the two projects simultaneously. The aim of these two projects was the same, that of discovering the capabilities of the pump and providing a workable theory which would predict the important characteristics of the pumps. A secondary aim, in both cases, was to look at possible applications for the two pumps. In the case of the double coil suction pump, this lead to the development of the Channel Doser.

The pumps described in this thesis all have helical coils and they share the same general hydraulic characteristics, hence, one basic theory can be used to describe the behaviour of each. It is this common theory that provides the backbone to the thesis.

The thesis describes the work on the coil pumps spread over a period of fourteen years. Some aspects of this work are only briefly mentioned, others have not been included at all. All the important factors that have emerged from these investigations over this period, however, are included here.

To summarise, this thesis covers the development and testing of the lift pump theory and its adaption for use with other pumps in the same family.
CHAPTER 3

ORIGINS AND PAST WORK

3.1 ORIGINS

3.1.1 Introduction

The two common forms of the coil pump, the spiral and helix versions have been known for a long time. The first documented evidence I have found is a description of both types of pump in a treatise of mechanics dated 1806 (Gregory 1806). The invention of this pump is attributed to Andreas Wirtz in the mid-18th century. I suspect that the origins of this pump may stem back many centuries before this date, but I have no proof.

If the pump has ancient origins, then why are so few engineers familiar with this type of pump? Two factors would have made this pump difficult to construct in the past. The helical or spiral coil would have had to be made from natural materials such as bamboo or leather. Alternatively, it would have had to be cast in a metal such as lead which would have been heavy and cumbersome. Both these types materials would have needed great care and skill to form the coil. The second factor is the rotary joint. Though the pump constructors in the past were capable of making an effective rotary joint from leather and wood, they would have experienced difficulty combining a water tight seal with a low rotational resistance, as we do today.

Whatever the reasons, the coil pump has remained in obscurity, except for the period around the mid 18th century when Wirtz constructed his version of the spiral pump.
3.1.2 Andreas Wirtz

The information used here is taken from the 'Treatise on Mechanics' by Olinthus Gregory (1806). It would very interesting to trace the work of Wirtz back to Zurich where he operated. This would not have aided our investigation, only satisfied an historical curiosity.

H. Andreas Wirtz was a tin-plate worker and he 'invented', designed and constructed a spiral coil pump in a dye-house in Limmat in 1759. It is unclear as to the size of the outside diameter of this spiral or, how many turns were on the spiral. The inlet part of the spiral appears to be a horn shape which acted as a form of scoop to increase the intake of water.

Gregory credits Wirtz with the invention of the spiral pump, but mentions little more about the pump used at Limmat. He does describe well the basic manometric action of the pump and suggests a number of design rules. He also discusses briefly the helical coil version of the pump and a hybrid arrangement which is effectively a tapering helix.

3.1.3 Other Early Work

Gregory also describes what I assume to be a helical pump, though he refers to it as a spiral pump. This pump was constructed at Florence and had an outside diameter of 10ft with a pipe diameter of 6 inches. The pump rotated at 6 rpm. and raised 22 cu. ft per min. to a surprisingly low height of 10ft. One advantage of a coil pump is that it can lift to a height greater than it own dimensions, which is not the case here.

Young (1807) describes the coil pump in his book 'A Course of Lectures on Natural Philosophy and the Mechanical Arts'. He also attributes its invention to Wirtz, and comments briefly on a pump used in Florence. He also mentions that the pump was used in Russia, but he gives no further details.
More interestingly, he states that Lord Stanhope used a spiral pump in England and he, Young, carried out some trials himself. He gives few details of these experiments except that he raised water to a height of 40ft; to do this he used a 100ft length of 0.75 inch diameter lead pipe to make up the spiral. As might be expected, the pump was heavy and cumbersome and Young abandoned it in favour of a 'force pump'. Young also discusses the problems of the centrifugal forces and high friction losses in the pump and delivery pipe which suggests that the speed of drum rotation of his drum was high. The tone of Young's report on his experimentation with the spiral pump is one of disappointment.

Another 19th Century author, Ewbank (1842), also reports on the coil pump. He quotes extensively from Gregory's book, but he does provide additional information about the Russian pump. He states that in 1784, a pump was constructed in Archangelsky and that it raised a hogshead of water per minute (50 gals/min) to a height of 74 ft through a pipe 760ft long: no more information is given. This must have been a substantial pump to achieve the quoted flow rate and, because of its high power demand, it would have probably have been driven by a water wheel. It is a pity more details were not available on this pump.

The work carried out on the coil pump in the 18th and 19th Centuries, produce a number of pumps which would lift water but there seems to be an underlying feeling amongst these authors that this simple pump could be used more effectively.

3.2 RECENT INVESTIGATIONS

Interest in the coil pump was rekindled in the early 1970's when A. E. Belcher (undated) wrote a short report claiming he had invented the pump. He gave a
description of the pump but he provided few details. R Ohlemutz was aware of Belcher's report and he undertook an investigation into the spiral pump. Ohlemutz(1976) carried out tests with a small perspex model and produced a theory based on the assumption that the air in the pump is incompressible. This assumption was valid for his small laboratory models, but it unsuitable for the situation where the pump is lifting a few metres or more. Ohlemutz published a report on his work in 1974 followed by a Ph.d Thesis in 1976.

In the mid 70's, work on the coil pump started at Loughborough, Salford and Dar es Salaam Universities, and other work was carried out in Denmark and Zimbabwe.

Weir (1979) in Dar es Salaam published a report containing a theory for the pump and a turbine design to power it. His theory took into account the compressibility of the air in the pump but did not consider such phenomena as bubbling and spilling. Also in 1979 Morgan(1979) published a short article describing a stream powered spiral pump he built for an irrigation scheme in Zimbabwe. The article concentrates on the practical design and operation of the pump and not on the hydraulic behaviour.

In 1980 Stuckey and Wilson (1980) at Salford published a paper describing the results of the tests on a laboratory stream-powered pump. They also offered some guidance on the hydraulic design of coil pumps.

The Danes produced three reports, Sydfynsgruppen,1980 (1),(2), & (3) on their work on constructing coil pumps. The emphasis of their effort was on providing low cost pumps for Developing Countries but they did produce a simple theory based on the manometric effect. In 1980, Mortimer and Pickford published their first paper on the coil pump.
3.3 WORK AT LOUGHBOROUGH

Work on the coil pump started at Loughborough in 1974 with a simple laboratory model, the idea for building the pump came from Belcher's report (undated). Final year undergraduate project students, Bamforth (1977), Winstanley (1977), Robinson (1978), Annable (1979) and Forrester (1985) carried out laboratory investigations on the helical pump under my supervision. In 1979 SERC awarded Mortimer and Pickford a research contract valued at £10,000 to investigate the behaviour of the pump. Annable, who had investigated the pump in his undergraduate project accepted a research studentship for 18 months to work on this project.


In 1984 Severn Trent Water Authority and The British Technology Group provided a grant of £12,000 and extensive site facilities to carry out research on the Coil Treatment Unit which is a variation on the coil pump. S. Galvin worked as a research student on this project for 12 months, this lead to the publication of a further paper Mortimer & Galvin (1986).

The work described in these papers forms part of the work described in this thesis.
CHAPTER 4

THE THEORY OF THE LIFT PUMP

4.1 INTRODUCTION

The lift coil pump can take many forms, for example, a coil can be wound in a spiral, helix, or a tapering helix, but it is not practical to consider them all. The work described in this thesis has concentrated on the pump with a helical pipe wound around a drum. This configuration is simple and it provides a good starting point for more complicated arrangements, see Figure 4.1. In fact, the helical pipe and drum concept is used in this figure for the sake of simplicity, as in practice, it may look superficially quite different. See Chapter 6 for more details.

Why develop a theory to predict the behaviour of a simple pump such as this? Firstly, it allows the existing designs of the coil pump to be fully exploited and secondly, it can lead to new variations, as in this case.

If a theory is to be useful, it must be capable of predicting the hydraulic characteristics of the pump. With many types of pumps, the characteristics consist of a relationship between the flow rate through the pump and the following:

the head difference produced across the pump,
the efficiency of the pump,
and the suction lift.

With any pump, the head/flow rate relationship must be of great importance and this is so for the coil pump. In most cases, the efficiency can only be
Figure 4.1 General arrangement of Lift Coil Pump
determined by experimental methods, so a theoretical prediction is of little use. The third relationship, that for predicting the suction capabilities, is not applicable in this case, since this version of the pump cannot generate a suction pressure.

The major part of this chapter, Sections 4.3 to 4.6, is concerned with defining the head/flow rate curve by using the liquid level differences in the loops. Section 4.3 describes the hydraulic mechanisms that could be present in the pump. In Section 4.4 the basic equations for these mechanisms are derived and Sections 4.5 and 4.6 shows how these equations are used to calculated the pressure differences across the pump. Section 4.7 deals with the estimation of the flow rate and Section 4.8 gives some consideration to the power required by the pump. Finally, Sections 4.9 to 4.11 are concerned with secondary problems.

For reasons which will be be explained later, the construction of the head/flow rate characteristic curve for a particular pump is left until Chapter 5.

4.2 GENERAL APPROACH TO THE THEORY

4.2.1 Introduction

After a short period of observation of a working coil pump where the helical coil is transparent, it becomes obvious that the pump is pressurizing the liquid and the air by a manometric effect. It can also be seen that the fluid is being moved through the pump by a positive displacement action similar to an archimedean screw.

When the pump is stopped, the liquid levels in the loops remain virtually in the same position as they were when the pump was working. When the investigation was started, it seemed very appropriate to develop a theory
which would predict the liquid levels in the pump since they appeared to be the key to its performance.

Stuckey and Wilson (1980) used dimensionless groups of pump parameters based on external measurable parameters to define the pump's performance whilst Weir (1979), Galvin (1987) and Ohlemutz (1974) focussed their efforts on the liquid levels.

Dimensionless groupings do provide a neat method of presenting the results, but it was felt that they would not give the same insight into the workings of the pump as an analysis of liquid level differences. In fact, Annable (1982) used his computer program based on level difference assessment to construct dimensionless design curves and so one can be used to find the other.

An alternative method of developing a theory for the Coil Pump could be based on the large amount of work that has been carried out on multiphase flow. Many papers are published on this subject each year and the International Journal of Multiphase Flow is one journal dedicated to the subject. The main applications for this work are in the field of nuclear energy and the chemical engineering industry.

The theories of multiphase flow behaviour cover a range of conditions from the flow of water with well dispersed, small diameter air bubbles in it, to slugs of air moving through the water flow, see Wallis (1982), Nakkoryakov et al (1986) and Delhaye (1983). However, none of these theories are directly applicable to the coil pump where essentially neither the two phases are inter-mixing, nor, are the viscous forces playing an important part in the relationship between the two phases. The flow in a coil pump is dominated by gravitational forces, with most other systems described by multiphase flow, viscous forces control flow conditions.
4.2.2 A Static or Dynamic Analysis of the System?

The coil pump is basically an unsteady system when viewed over a time period of one cycle or drum revolution, that is, if it is pumping liquid up to a header tank. The outlet head is continually changing and the composition of the plugs nearest to the inlet and outlet is varying.

A theoretical model for the coil pump could be based on one of the assumptions listed below.

(a) A static analysis of the system can be carried out which ignores the dynamic effects of viscosity. This analysis would also assume that the plug movements and pressure variations within one cycle can be described by one analysis; though this sounds unlikely, it can be achieved with some success as will be shown later. If the outlet head variations are known it is then possible to repeat the static analysis for a number of the values of outlet head.

(b) Method (a) can be improved by carrying out a series of static analyses at different times throughout one cycle. These analyses could include some of the steady state dynamic effects such as an approximation for the liquid plug friction effects and inlet losses.

(c) The third method is to carry out an analysis including time as a continuously increasing variable. This method has been adopted by Boyce et al. (1969), Rippel et al. (1966), for analysing two phase flow.

The majority of uses envisaged for the Coil Pump involve speeds of rotation of less than 10 r.p.m. At these speeds the fluid velocity along the pipe axis is unlikely to exceed 0.75 m/s, and this yields a velocity head of about 30mm of
Figure 4.2(a) Cascading Manometer Concept

Figure 4.2(b) 'Pulled Pipe' Concept
water. This velocity head is based on a velocity relative to the pipe and not the external liquid surface, so it is not a true hydraulic loss. The energy dissipated through minor losses in the helical coil is difficult to estimate, but it is probably of the order of 1% of the total head gain. Friction losses in a typical coil pump (using the D'Arcy equation) amount to less than 0.5% and the head associated with the whirl velocity is also less than about 0.5% of the total head gain across the pump.

Though these approximate percentage dynamic losses are small when compared to the total head gain, their effect on the hydraulic behaviour of the pump may be more significant since these losses may be concentrated locally, as in the case of minor losses. However, it was felt that any concentration of these effects would not affect the general behaviour of the pump significantly except, possibly, at the inlet.

Since the effects of the dynamic losses appear relatively small and also since the liquid levels in the coil change little when the drum is stopped, encouraged the view that a static analysis was valid. It was also strongly felt that a relevant dynamic theory, as suggested in option (c) above, may be extremely difficult to construct and calibrate. Boyce, and Rippl all had much simpler hydraulic systems than the one developed in the coil pump since the two fluid phases were better mixed than this case. For all these reasons it was decided to adopt option (a) for the major part of the work but also carry out checks on the quasi-dynamic approach where a static situation would be determined at regular times throughout the cycle of the pump i.e. in one revolution.

4.2.3 Means of Modelling the Pump

It is very difficult to draw the helical coil pump and to define the geometry mathematically is cumbersome. Two ways of visualizing the pump are used
to ease the process of analysis; both are shown in Figure 4.2. In Figure 4.2(a) the coil is represented as an 'unwound' helix which forms a cascading manometer. This concept is used to investigate the pressure build up in the coils. The term 'cascading' used here refers to a number of manometers connected in series where the pressure can 'cascade' from one loop to the next.

The second form of visualisation is shown in Figure 4.2(b) the coil pump is shown as a straightened pipe which is being 'pulled' over the liquid and air plugs. This concept of 'pulling' the pipe, and hence, moving the higher pressure zone at the outlet towards and over the plugs, can be used because of the way in which the liquid plugs are held in the lower portion of each loop by gravity. This concept is used to study the plug volume changes and fluid movements and is developed further in Section 4.4.5.

Both Weir (1979), Sydfynsgruppen (1980(1)) and Ohlemutz (1974) used the cascading manometer concept but not the 'pulled pipe' idea. The cascading manometer can be used to study the pressure build up, but to study the other important phenomena, the 'pulled pipe' idea is necessary.

In the following sections, a static situation will be assumed unless otherwise stated.

4.2.4 Initial Plug Configuration used for Analysis

The position of the air and liquid plugs, (along the axis of the helical pipe) in relation to the inlet and outlet of the helical pipe, varies as the pump rotates through one revolution. At some points in the cycle only part of an air, or liquid, plug will have entered the inlet and, similarly, there are times when part of a plug will be discharging through the outlet. The outlet is defined here as the point on the helical pipe where it turns through 90 degrees before leading down to the rotary joint. See Figure 1.1.
Figure 4.3 The Cascading Manometer
In the main part of the analysis, it is proposed to consider the configuration of air and liquid plugs where there is a whole number of pairs of plugs in the coil. It can be assumed that an air plug is just about to leave through the outlet. The coil is also assumed to have an exact number of turns, or loops. The inlet and outlet are taken to be on the same circumferential position on the drum, though both are not in the same vertical plane. If the plugs in the coil were all at atmospheric pressure then a number of pairs of air and liquid plugs, equal to the number of loops, would fit precisely in the helical pipe.

Under pressure, the air plugs will be compressed, and with an air plug about to leave the outlet, there will be space at the inlet to be occupied by air at atmospheric pressure. In this initial analysis the problem of partial plugs will be ignored but in Section 4.10 the effects of partial plugs at the inlet and outlet will be briefly considered.

4.3 PRESSURIZATION WITHIN THE LOOPS

4.3.1 Introduction

Each time the inlet of the helical coil moves through one revolution, it takes in a plug of liquid and a plug of air. In the first turn of the coil, the position of the air/liquid interfaces are, in most cases, close to the external liquid levels.

As each pair of plugs moves through the helical coil, it acts as a manometer sustaining a pressure difference across the pair. The action of a number of pairs of liquid and air plugs constitute a cascading manometer as shown in Figure 4.3. The plan view of the manometers is shown in Figure 4.4(a).

As the head difference across the pump increases, so the individual liquid level differences have to increase to balance the total pressure difference
FIGURE 4.4a (at time t=t)

FIGURE 4.4b (at time t=t+60/S secs)

Air plug identification tag

Water plug identification tag

Plug lengths at fixed pipe positions

**Figure 4.4 Plug layout in Unwound Pipe**
FIGURE 4.5 Spilling in Liquid Plug Wn

Figure 4.6 Liquid plug Wn with Bubbling
across the pump, see Figure 4.3.

4.3.2 Liquid Level Differences

The liquid level differences in the coils are governed by the rotation, or swing, of the liquid plugs relative to the external liquid level. This, in turn, is controlled by the compression of the air plugs.

As the level differences across a liquid plug increases, it is possible that other phenomena may occur. The two most important are termed 'spilling' and 'bubbling' regions. The situation shown in Figure 4.3 is termed the 'non-spilling region'.

4.3.3 Spilling Mechanism

As a liquid plug progresses through a coil, it rotates relative to the external liquid surface as already mentioned. If this rotation is sufficiently great so that the trailing edge of the liquid plug reaches the crown of the pipe, then spilling will occur as the liquid spills over the crown of the loop.

Spilling is assumed to take place when the horizontal liquid surface of the trailing edge of the liquid plug just reaches the lower wall of the crown of the pipe as shown on Figure 4.5. (See Section 4.4.7 for further details). When spilling occurs, liquid flows from one plug back to the following plug.

The spilling region which consists of a number of spilling loops in a coil is bounded by the outlet on one side and the non-spilling region on the other. In cases where the pump is subjected to a high lift with a low depth of immersion, a spilling region will be followed by a bubbling region on the inlet side. This second type of spilling is termed "high spilling".
4.3.4 Bubbling Mechanism

If the leading edge of the liquid plug dips to the invert of the loop before the trailing edge reaches the crown, then bubbling will occur. The word 'invert" here is used to signify the lowest part of the circular coiled pipe. See Figure 4.6.

With bubbling, air passes from an air plug back through the liquid plug and into the following air plug. Whilst an air plug is losing air, it will also be gaining air from the preceding plug. The bubbling region is bounded by the outlet on one side and a non-spilling region on the inlet side. In some cases, there may be a spilling region between the outlet and the bubbling region.

The occurrence of spilling or bubbling depends largely on the depth of immersion of the drum supporting the coil which, in turn, determines the length of the liquid plug. Low depths of immersion tend to cause bubbling, whereas, a greater depth of immersion produces spilling. The diameter of the helical pipe also plays an important part in bubbling. With diameters of less than 10mm, it is unusual to observe bubbling phenomenon, but with diameters over 100mm bubbling occurs freely when the depth of immersion is less then 50%.

4.4 EQUATIONS REQUIRED TO DETERMINE THE LEVEL DIFFERENCES

4.4.1 Introduction

The calculation of the level differences is carried out using a string of calculations termed a 'routine', to do this a number of equations are required and these are derived in this section.

The first part of Section 4.4 (4.4.2 and 4.4.3) is concerned with behaviour of the air plugs under compression. The effects that this compression has on the
plugs, generally, are dealt with in Sections 4.4.4 to 4.4.8. Finally, Section 4.4.9 develops three further equations that are required for the level difference calculations which come later in this chapter.

4.4.2 Cascading Manometer

A brief inspection of the coil pump lifting liquid will indicate that as the pressure builds up at the outlet, the plugs of liquid rotate in a manner that suggests they are acting as a form of multiple manometer which is resisting the head difference across the pump. In Section 4.2.3, it was suggested that the action of the pump may be represented by a cascading manometer and this is shown in Figure 4.3. If \( h(n) \) is the head difference across loop \( n \), \( H_{ai} \) is the absolute pressure head at the inlet and \( H(N) \) is the absolute head in the last air plug then,

\[
H(N)-H_{ai}=h(1)+h(2)+h(3)+\ldots+h(N). \quad \text{Equation 4.1}
\]

\( H_{ai} \) is the ambient pressure, normally assumed to be atmospheric. \( N \) is the number of loops.

If the air were incompressible then, \( h(1)=h(2)=h(3)=\ldots=h(n) \). In reality \( h(n+1)>h(n)>h(n-1) \) because of the compressibility of the air.

With a static analysis of the system, \( h(n) \) is the measured head across the liquid plug, \( n \). If dynamic losses are taken into account then \( h(n) \) is the effective head which is the measured head minus the dynamic losses. This is discussed further in Section 4.9.

4.4.3 Compression of the Air Plugs

For a perfect gas

\[
p.V=R.T
\]

where \( p=\) absolute pressure of the gas, \( V=\) specific volume, \( T=\) temperature of
the gas, \( R \) = the gas constant.

For a polytropic process, one frequently used equation is,

\[ pV^k = \text{Constant}. \tag{4.2} \]

where \( k \) is the coefficient of compression or expansion.

Many relationships define the behaviour of real gases, see Reid et al (1966), but most involve a number of parameters that have to be assumed or estimated. This makes their use in any analysis difficult if tables or graphs have to be incorporated, particularly if a computer is involved. Equation 4.2 is widely used and accepted and it only requires one parameter to be estimated; for these two reasons it is proposed to adopt this equation. Taking the case of a plug of air situated in a pipe of constant cross-sectional area and where the pressures are measured in terms of pressure head, then Equation 4.2 will become,

\[ H_{ai}L_{ai}^k = H(n)L_{ai}^k. \]

Rearranging this equation,

\[ L_{n} = L_{ai}(H_{ai}/H(n))^{1/k}. \tag{4.3} \]

\( L_{ai} \) is the initial length of the air plug at a pressure head \( H_{ai} \) which, in this case, is normally assumed to be atmospheric pressure. \( L_{n} \) is the length of the air plug at a pressure head of \( H(n) \). If \( \Delta H(n) \) is the change in the length of air plug \( n \) as the pressure changes from \( H_{ai} \) to \( H(n) \) then,

\[ \Delta H(n) = L_{ai}[1-(H_{ai}/H(n))^{1/k}]. \tag{4.4} \]

For any analysis, an estimate of \( k \) is required; for the work in this thesis, a value of 1.15 is used. See Appendix 1 for details.

4.4.4 'Swing' of the Plugs

Consider the situation shown in Figure 4.3 where the cascading manometer has no pressure difference across it. The liquid levels bounding the plugs would all be at the same height, assuming that the liquid plugs were all the same length. If a pressure builds up at the outlet, all the air plugs will
compress due to the increase in pressure and they will contract in length. This contraction must cause the liquid plugs to move or swing, and this movement can only occur in a direction towards the inlet if the liquid plugs are to manometrically balance the pressure difference across the pump. Logically, it is possible to argue that other combinations of movements can occur, but none will satisfy the equations governing the manometer principle and conform to the observed levels.

4.4.5 Relative Movements of Plugs

The analysis of this aspect is best carried out by considering the pulled pipe concept introduced in Section 4.2.3.

Figure 4.4(a) shows a simplified arrangement of air and liquid plugs at an arbitrary time t=0. Under the quasi-steady state conditions in the pump, it is assumed that at any point on the pipe, all air plugs passing the point will have the same length and all liquid plugs will also be of equal length. Two points at different positions on the pipe may, or may not, have liquid or air plugs of the same length passing those points. The plug lengths shown on Figure 4.4 are related to the point in the pipe when the trailing edge (or, for convenience, the leading edge in some later cases) of the liquid plug reaches that point. An identity tag is also given to each plug in Figure 4.4. This enables the progress of particular plugs to be followed.

Figure 4.4(b) shows the same set of plugs, but at a time t+60/S later. If S is the speed of rotation in r.p.m and, if t is measured in seconds, then the situation shown in Figure 4.4(b) occurs exactly one revolution of the drum after the situation shown in Figure 4.4(a).

All the plugs shown in Figure 4.4(a) will move forward by the action of the rotating helical coil to the positions shown on Figure 4.4(b); this allows two
more plugs to enter the pipe, liquid plug W0 and air plug A0. If we consider the pair of plugs Wn and An, the inlet end of Wn, shown by line XX on Figure 4.4(a), will have moved in one revolution to the position shown by the line YY on Figure 4.4(b). If plug Wn was subjected to no other forces than those produced by the helical coil moving it, then

Distance YY to XX = πD.

D is the effective drum diameter.

To view it a different way, the above equation is valid if no further compression of the air plugs occurs during the considered revolution. In fact, air compression does occur and so the equation becomes,

Distance YY to XX = πD − Con(n). Equation 4.5.

Con(n) is the relative movement of the trailing edge of liquid plug Wn towards the inlet as the liquid plug moves from its position shown on Figure 4.4(a) where its length is Lw(n) to the position on Figure 4.4(b) where its length is now Lw(n+1). The relative movement here must be caused by the compression of the preceding air plugs.

From Figure 4.4, Distance YY to XX = Lw(n) + La(n), therefore, Equation 4.5 becomes

Lw(n) + La(n) = πD − Con(n). Equation 4.6.

At lower pumping heads, πD = Lai + Lwi, (see Section 4.4.10) and so,

Lw(n) + La(n) = Lai + Lwi − Con(n). Equation 4.6(a).

At higher heads where there is a significant swing of the first plug, it is more correct to leave Lai + Lwi as πD.

Equation 4.6 assumes, as does Figure 4.4, that at the same point in each loop, the rotational positions of the air and liquid plugs relative to the inlet levels will always be the same. Experimental observation using a video camera suggests that this assumption is valid.

It could be argued that in Equation 4.6(a), Lw(1) should be used instead of Lwi and La(1) should replace Lai, but there some situations where unusual inlet
conditions can alter the plug lengths as they enter the coil. At this stage, therefore, it is proposed to keep the equations general.

Annable (1982) and Weir (1979) considered this relative movement as a superimposed velocity on the forward plug velocity which, in fact, is what it is. Since the combined velocity is then continuously changing, it adds further complications to the understanding of the plug behaviour. Viewing the plugs at time intervals of one drum revolution simplifies this problem considerably.

4.4.6 Movement of Non-Spilling Plugs

"Non-spilling plugs" is a slight misnomer here, since the term should be "non-spilling, non-bubbling plugs" but it was felt that the use of this term was too cumbersome.

Figure 4.3 suggests that all boundaries between the air and liquid plugs in the coil are different lengths. As the air plugs move from the inlet to regions of higher pressure, they will reduce in length, that is, provided no air passes from one loop to the next. Throughout a coil under pressure therefore, the air plug lengths will vary.

Assuming the liquid plugs to be incompressible, a change in plug length can only occur if liquid moves (or spills) from one plug to the next. The region of non-spilling loops requires, therefore, that all the liquid plugs in that region are the same length. The non-spilling region occurs close to the inlet of the pump.

If all the liquid plugs in the non-spilling region are the same length then, in general, \( L_{wi}=L_{w(n)} \). Equation 4.6(a) will then become,

\[
C_{on}(n)=L_{ai}-L_{a(n)}. \tag{Equation 4.7}
\]
or more correctly, $\text{Con}(n) = \pi \cdot D - (Lw(n) + La(n))$ from Equation 4.6.

Equation 4.7 shows that, as a liquid plug moves through one revolution within the coil pipe, its relative movement towards the inlet is equal to the compression of an air plug as it moves from the inlet to a position immediately following the liquid plug at the end of the considered revolution. This relative movement manifests itself as a plug rotation.

The relative movement $\text{Con}(n)$ given in Equation 4.7 equates to $\Delta H(n)$ as defined in Equation 4.4.

Equation 4.7 relates a liquid plug swing ($\text{Con}(n)$) to the air plug lengths and so it can be used to establish the pressure build up in the loops. In Loop $n$ on Figure 4.7(a), the liquid plug $W_n$ has rotated by an amount $[\beta(n) - \beta(n-1)]$ during its subsequent revolution and this has been caused by the air contraction $\text{Con}(n)$, therefore,

$$[\beta(n+1) - \beta(n)] = \text{Con}(n)/R.$$

$\beta(n)$ is the angle between the trailing edge of the liquid plug and the vertical line below the centre of the pump. In Section 4.4.9, $\beta(n)$ is, in turn related to the liquid level difference $h(n)$ across Loop $n$ which governs the pressures in the air plugs.

The explanation of the relationship between $\text{Con}(n)$ and $\beta(n)$ above has been given to illustrate the plug movement mechanism. In the calculations used to determine the level differences across the loops, a much simpler method is adopted which dispenses with the need to calculate $\text{Con}(n)$ directly: this is explained in Section 4.5.2.

4.4.7 Movement of the Spilling Plugs

If the rotation of the liquid plug is sufficiently large, the trailing edge of the liquid plug will reach the crown of the pipe and liquid will spill through the following air plug into the following liquid plug. This phenomenon is
known as 'spilling'. Figure 4.5 shows liquid plug Wn in the process of spilling. In fact, the profile of the liquid surface of the trailing edge is not horizontal as shown by the boundary of the shaded area, but it would follow the solid line shown on the figure. A depth of liquid will be required at the crown to allow the water to spill (as shown by the solid line) and viscous forces caused by the pipe friction on the 'inner' wall will tend to incline the liquid surface upwards towards the crown. Initially, for this analysis, the assumption of a horizontal liquid surface is felt to be reasonable. This is discussed further in Section 5.4.3 and in Appendix 3.

From Figure 4.5, if $\Omega_{sp}$ is the angle of rotation of the liquid plug from the vertical (above the centre line) that will just cause spilling,

$$\Omega_{sp} = \arccos \left( \frac{R-r}{R} \right).$$  \hspace{1cm} \text{Equation 4.8.}

R is the effective radius of the drum from the drum centre to the helical pipe centre line, r is the radius of the helical pipe.

If a liquid plug has reached the position shown in Figure 4.5, then any further spilling must be caused by the liquid plug moving, relatively, back towards the inlet under the influence of the compression of the following air plugs.

For spillage to occur in loop n, therefore,

$$\beta(n) > \pi - \Omega_{sp}.$$ \hspace{1cm} \text{Equation 4.9.}

$\beta(n)$ is the angle of the trailing edge of the liquid plug from the loop soffit (6 o'clock on coil elevation).

In practice, $\beta(n) = \pi - \Omega_{sp}$ when a coil is spilling, but when a check is being made in the non-spilling calculations, $\beta(n)$ can be greater than $\pi - \Omega_{sp}$.

If $\beta(n) = \pi - \Omega_{sp}$ then, from Figure 4.5, the angle of the leading edge can be found by,

$$\phi(n) = 2\pi - \Omega_{sp} - Lw(n)/R.$$ \hspace{1cm} \text{Equation 4.10.}
When spilling occurs, there can be no relative movement of the trailing edge of the liquid plug back towards the inlet since the plug surface remains in the same relative rotational position during a revolution.

If no relative movement occurs, then \( \text{Con}(n) \) in Equation 4.6(a) must be zero, therefore,

\[
L_w(n) = (L_{ai} + L_{wi}) - L_a(n).
\]

or, more correctly, \( L_w(n) + L_a(n) = \pi D \) from Equation 4.6.

It may seem a little strange that if \( W_n \) is spilling in Figure 4.4(a), then its length will be dependent on the length of \( A_n \) which is preceding it. In fact, it is the compression of the preceding air plug \( A_n \) which governs the spilling into plug \( W_n \). The compression of plug \( A_{n-1} \) governs the liquid spilling out of \( W_n \). In two adjacent spilling liquid plugs, the trailing edges will be at the same rotational angle to the vertical, \( \omega_{sp} \). One complete liquid plug and the whole of the preceding air plug, therefore, must fill one loop (360 degrees of pipe rotation). This condition is fulfilled by the version of Equation 4.11 derived from Equation 4.6, and so the length of \( W_n \) is governed by the length of \( A_n \).

It will be useful to determine the spillage to and from any loop at this stage. Consider \( W_n \) on Figure 4.4(a) and assume it is spilling and also it is receiving spillage from \( W_{n+1} \). Now by continuity,

\[
\text{Spill}_{in}(n) - \text{Spill}_{out}(n) = L_w(n+1) - L_w(n).
\]

Spill\(_{in}(n)\) and Spill\(_{out}(n)\) are the volumes, expressed as a length of spillage into and out of \( W_n \) as it moves from its position on Figure 4.4(a) to its position on Figure 4.4(b).

If there is a spilling region in the coil, then at its boundary on the inlet side, there will be a liquid plug which is receiving spillage but is not spilling itself; this is termed the 'last non-spilling loop' and is given the loop number 'ns'. If in Figure 4.4(a) \( n=ns \) and it is assumed that the liquid plug is just starting to
receive its first spillage then, Spillout will be zero and $L_w(n) = L_{wi}$. For this plug, Equation 4.12 will become,

$$\text{Spillin}(ns) = L_w(ns+1) - L_{wi}.$$  \hspace{1cm} \text{Equation 4.13.}$$

With the plug in loop number $ns+1$, the equation becomes

$$\text{Spillin}(ns+1) - \text{Spillout}(ns+1) = L_w(ns+2) - L_w(ns+1)$$

Spillout$(ns+1)$=Spillin$(ns)$ and substituting this, and Equation 4.13, into the above equation will yield,

$$\text{Spill}(ns+1) = L_w(ns+2) - L_{wi}.$$ Substituting in for $L_w(ns+2)$ derived from Equation 4.11, the above equation becomes,

$$\text{Spillin}(ns+1) = L_{ai} - L_a(ns+2) = \text{Spillout}(ns+2).$$

In a general form,

$$\text{Spillout}(n) = L_{ai} - L_a(n) = \text{Spillin}(n-1) \hspace{1cm} \text{Equation 4.14.}$$

This laborious proof shows that the amount of liquid spilling out of a plug during one drum revolution is dependent on the compression of the preceding air plug. This same air compression causes the plug rotation in the non-spilling region.

Neither Weir (1979) nor Ohlemutz (1974) carried out an analysis of the spilling loops. Weir concentrated his efforts on coil pumps which did not spill. He appears to assume that a spilling pump is unacceptable since any spillage reduces the flow rate; this is not true, except at very high lifts.

4.4.8 Movement of the Bubbling Plugs

With spilling loops, the trailing edge of a liquid plug reaches the crown of the pipe and any further relative movement causes spilling. If the leading edge of the liquid plug reaches the soffit of the loop before spilling occurs then air bubbles back with any further plug movement along the coil. The plug position when bubbling is shown in Figure 4.6.

As with spilling, the position of the leading edge of the liquid plug that is
required to cause bubbling has to be defined. Observations of the pump suggest that the mechanism is complex in that the leading edge is depressed by the pressure until a bubble is released to rise up through the liquid plug. The leading edge of the liquid plug then swings back a little in a clockwise direction on the layout shown on Figure 4.6 and the cycle repeats itself.

Other factors also influence the problem, larger diameter coil pipes bubble more easily than small diameter pipes. A higher speed of rotation of the drum also assists the bubbling action.

The criterion, therefore, for bubbling just to occur is that, if $\varnothing(n)$ is the angle from the vertical, of the leading edge of the liquid plug $W_n$ then,

$$\varnothing(n) = \pi - \Omega_{bub}$$

Equation 4.15.

$\Omega_{bub}$ is the angle from vertical line, below the drum centre line, of the leading edge of the liquid plug that just causes bubbling. Since a value of $\Omega_{bub}$ has not been accurately assessed in practice it is assumed to be equal to zero in the calculations. This is discussed further in Section 5.4.3.

If bubbling is occurring during one revolution of the coil, then the leading edge of the bubbling plug will stay in the same relative position. The bulking effects of the air bubbles in the liquid are ignored.

If the liquid plug length remains the same during a revolution, then the relative movement of the plug towards in the inlet will be zero. With bubbling, the plug movements under study are now related to the leading edge of the liquid plugs. Consequently, lines XX and YY on Figure 4.4 should be transferred to the outlet side of $W_n$. Equation 4.6 will then become,

$$L_w(n+1) + L_a(n) = \pi . D . Con(n).$$

Con(n) has to be redefined to refer to the leading edge of the plug and not the trailing edge used for the spilling case.
In above equation, therefore, $\text{Con}(n)$ is zero and so, 
$$L_a(n) = \pi.D - L_w(n).$$
If spilling is not occurring, $L_w = L_w(n)$ and then, 
$$L_a(n) = \pi.D - L_w$$
Equation 4.16.
If, $L_a + L_w = \pi.D$ then, $L_a(n) = L_{ai}$.

At first sight this equation seems odd since, if An in Figure 4.4(a) is receiving and losing air by bubbling, it has exactly the same length as when it entered the pump.

In two adjacent bubbling plugs, the leading edges of the liquid plugs will be at the same rotational angle to the vertical, $\Omega_{bub}$. One complete liquid plug and the whole of the following air plug, therefore, must fill one loop (360 degrees of pipe rotation). This condition is fulfilled by the equation, $L_a(n) = \pi.D - L_w(n)$, if the liquid plug plug length has not changed in length then the air plug length cannot have changed. Annable(1982), Weir(1979) and Ohlemutz(1974) did not consider the bubbling mechanism in any detail.

4.4.9 Liquid Level Differences and Rotational Angles in a Loop

A number of equations are developed in this section but their use and inter-relationships are left to Section 4.5. The liquid level differences are caused by the relative movement of the plugs in the coil and so it is appropriate to derive a number of equations governing these differences before proceeding to the calculation routines.

$\Omega(n)$ and $L_w(n)$ are defined on Figure 4.7 and are initially controlled by the inlet conditions and the plug movements. If they are known for liquid plug $W_n$, then it is possible to calculate the liquid level difference across the plug using the equation,
$$h(n) = R \left[ \cos(2\pi - \Omega(n) - L_w(n)/R) + \cos(\pi - \Omega(n)) \right].$$
Equation 4.17.
See Figure 4.7(a). This equation is valid for the spilling and bubbling regions as well as the non-spilling situation shown on Figure 4.7.
Figure 4.7(a) Level differences across Liquid Plug \( n \)
If $\varnothing(n)$ and $L_w(n)$ are known then, Equation 4.17 can be used to calculate $h(n)$. If $H(n)$, the pressure in air plug $n$, is also known, the air pressure in $A_{n+1}$, can be found by using,

$$H(n+1)=H(n)+h(n),$$
or, if the calculations work towards the inlet,

$$H(n-1)=H(n)-h(n)$$

Equation 4.18.

Three further equations are required to maintain the progress of the calculations from one loop to the next. The first relates $\varnothing(n+1)$ to $\varnothing(n)$ and is given by,

$$\varnothing(n+1)=\frac{L_a(n)+L_w(n)}{R}+\varnothing(n)-2\pi$$

Equation 4.19

This assumes that $L_w(n)$, the liquid plug length, and $L_a(n)$, the air plug length are already known.

The second equation required is the relationship between $\varnothing(n)$ and $\beta(n)$; this is,

$$\beta(n)=\varnothing(n)+(L_w(n)/R)-\pi.$$  

Equation 4.20.

This can be derived from consideration of the geometry of Figure 4.7. The final equation relates $\varnothing(n)$ and $\beta(n+1),$

$$\varnothing(n)=(L_a(n)/R)-(\pi-\beta(n+1))$$

Equation 4.21.

4.4.10 Inlet Plug Lengths

It was mentioned in the previous section that the inlet conditions provide a constraint on the calculation of the level differences, in fact, they govern the relative lengths of the liquid and air plugs.

If Dimer is the depth of immersion of the drum in Figure 4.7(b), then the liquid plug length can be found by geometry using the equation,

$$L_w=2R\cos\left[\frac{Dimer}{R}\right]$$

Equation 4.22

The air plug length will then be,
Equation 4.22 assumes that the liquid plug can enter the inlet freely and take up a position where its two surfaces are on the same level as the external liquid surface. When the lift is well below the maximum limit of the pump, this assumption is valid. However, if the pump is approaching its maximum lift, the level difference across the first coil becomes significant. This has the effect of depressing the leading edge of the plug as it moves into the first loop. The trailing edge of the plug must be at the level of the external liquid surface at the time the inlet of the coil makes contact with the liquid surface. See Figure 4.7(b). As the inlet rises out of the liquid, so the trailing edge will react to a further depression of the leading edge by rising above the level of the external liquid surface. At higher pumping heads, a first estimate of $L_{wi}$ is made using Equation 4.22. From the calculation routines, a value of $\theta(1)$ is found. $\theta(1)$ as defined on Figure 4.7(b) is given by,

$$\theta(1) = \cos^{-1}\left(\frac{\text{Dimer}}{R}\right)$$

Equation 4.24

From this and the calculated value of $\theta(1)$, the liquid plug length can be found by using the equation,

$$L_{wi} = R(\theta(1) + (\pi - \theta(1)))$$

Equation 4.25.

The amount by which the leading edge of the liquid plug rotates whilst it is being taken through the inlet is given by,

$$\text{RotW}(1) = \theta(1) - \left[\pi - \cos^{-1}\left(\frac{\text{Dimer}}{R}\right)\right].$$

If the rate of rotation of the liquid and air plugs is constant throughout the drum cycle, then the trailing edge of the air plug will rotate by $\text{RotA}(1)$.

$$\text{RotA}(1) = \frac{\text{Lai.RotW}(1)}{L_{wi}}.$$ 

This rotation occurs as the inlet moves around to take in the next liquid plug. Lai is the value calculated in Equation 4.23.

The equation above is only approximate since it is based on assumptions that are difficult to prove practically, however, it does seem to produce satisfactory results.

The modified air plug length Lai' will be given by the equation
The description of the modified plug lengths above are given as an introduction, for more detail see Section 4.6.7.

4.5 CALCULATION ROUTINES FOR THE LEVEL DIFFERENCES

4.5.1 Introduction

Two levels of calculations are used here, these are a procedure and a routine. The procedure is the overall calculation that provides the solution to the problem and within the procedure are calculation routines which are a well defined set of operations which deal with a particular hydraulic behaviour e.g. spilling or bubbling regions. The routine here is similar to the subroutine used in computer programming.

The calculation routines within a procedure cover the non-spilling calculations and, either, the spilling or bubbling calculations. The non-spilling calculations can start at either end of the non-spilling region in the coil and then proceed towards the inlet or outlet. The bubbling or spilling calculations start at the pump outlet and proceed loop by loop towards the inlet. In each case, the routine calculates the level differences across its particular region and individual loops.

At the interface between the nonspilling and the spilling or bubbling loops, a calculation routine has to be carried out in the loop which is referred to as the last non-spilling loop or the last non-bubbling loop.

It is proposed to describe the routines first, so Sections 4.5.2 to 4.5.4 deal with non-spilling, spilling and bubbling calculation routines separately. Section 4.5.5 is concerned with the last non-spilling loop, whilst Section 4.5.6 deals with the last non-bubbling loop. Finally Section 4.5.7 describes an unusual
START

Is there a spilling region?

Yes

\( n = n_{ns} \)

Known \( B(n), H(n-1) \) (see Figure 4.12)

\( La(n-1) \) from Equation 4.3

\( n = n - 1 \)

\( \Omega(n) \) from Equation 4.21

Lw(n) = Lw1

If \( \Omega(n) < 0 \) abandon calculations

h(n) from Equation 4.17

H(n-1) = H(n) - h(n)

\( La(n-1) \) from Equation 4.3

Is \( n = 1? \)

No

\( n = N \)

\( \Omega(N) \) Known

Value calculated from given equation

Yes

END

Figure 4.8 Calculation Routine for Non-Spilling Region
form of spilling called "high-spilling ".

4.5.2 The Calculation Routine for Non-Spilling Region

The non-spilling calculations start at the inlet side of the last non-spilling loop (from the inlet) and move loop by loop towards the inlet. The calculations could progress towards, or away from, the inlet; the reasons why the former was chosen is explained in Section 4.6.2.

In this section the following parameters will be assumed to have known values in addition to the information on the pump geometry.
(a) The loop number from the inlet in which the last non-spilling liquid plug is situated. The loop number of the last non-spilling loop is ns. This plug is receiving liquid spillage but is not spilling itself and so it defines the outlet end of the non-spilling region.
(b) B(ns), this is the inlet rotation of the last nonspilling plug.
(c) The absolute pressure in the air plug on the outlet side of the last non-spilling liquid plug.
(d) The length of the air and liquid plugs at the inlet. See Section 4.4.10.

The determination of values for (a) to (c) is carried out in the overall calculations procedure described in Section 4.6.

The calculation routine for the non-spilling region of the coil is explained on the flow chart in Figure 4.8.

The calculations start with the loop next to the last non-spilling loop (loop number ns-1) and proceed until loop number 1 is reached. If there is no spilling region, then the calculations start at the outlet and the value of $\Theta(N)$ is found by an iteration process in the overall procedure. When the non-spilling routine is carried out on a single run, which is part of the iteration
Known Hout, Lwi, ns and that spilling is occurring

\[ H(N) = H_{\text{out}} \]

\[ B(n) = \frac{\omega}{\omega_s} \]

\[ n = n - 1 \]

\[ \text{START} \]

\[ n = N \]

\[ H(N) = H_{\text{out}} \]

\[ \text{La}(n) \text{ from Equation 4.3} \]

\[ \text{Lw}(n) \text{ from Equation 4.11} \]

\[ \beta(n) = \pi \cdot \omega_s \]

\[ \omega_s \text{ known} \]

\[ \varnothing(n) \text{ from Equation 4.10} \]

If \( \varnothing(n) < 0 \) abandon calculations

\[ h(n) \text{ from Equation 4.17} \]

\[ H(n-1) = H(n) - h(n) \]

Is \( n = n_s + 1? \)

\[ \text{END} \]

Figure 4.9 Calculation Routine for Spilling Region
process, the calculated pressure at the outlet, $H(0)$, may greater or less than atmospheric pressure. This cannot occur in a stable pumping situation and the calculations required to satisfy this boundary condition are explained in Section 4.6.7.

4.5.3 The Calculation Routine for the Spilling Region

Apart from the pump geometry, the outlet lift pressure and the inlet fluid plug lengths need to be known to carry out the calculations in this routine. The number of spilling loops must also be known. Obviously, it must also be assumed that spilling is occurring. The determination of all these parameters is carried out in the overall calculations procedure which is explained in Section 4.6.

The method of finding the head differences and air plug pressures is explained in Figure 4.9. The calculations start at the outlet and continue from loop to loop until $n = ns+1$, where $ns+1$ is the number of the spilling loop nearest the inlet and $n$ is the loop number under analysis. This means that the calculations have been carried out for the required number of loops.

There is another situation where the spilling mechanism can occur at the pump outlet. This type of spilling is produced where the depths of immersion are low and pumping heads are high. $Lwi$ will be less than half the circumference, but the large air compression effects will eventually cause spilling near the outlet. The calculation routine for the high spilling region which is sandwiched between the outlet and the bubbling region follows the same method of calculation as shown in Figure 4.9. The calculations start at the outlet and progress towards the inlet. With each successive loop, the liquid length decreases which causes the leading edge of the plug to swing down towards the soffit of the loop. Eventually $\varnothing(n) < \pi - \Omega_{bub}$ and this causes a sudden change from spilling to bubbling in the calculations. How does one
If High spilling Region, \( n = N - N_{hs} \)

START

\( n = N \)

La(n) from Equation 4.16

\( \Theta(n) = \pi, \quad L_{wi} = Lw(n) \)

h(n) from Equation 4.17

\( H(n-1) = H(n) - h(n) \)

Is \( n = n_{b} + 1 \) ?

Yes

END

Figure 4.10 Calculation Routine for Bubbling Region
Loops adjacent Last Non-Spilling plug

Air pressure \( H(\text{ns}) \)

\( \text{Ans-1} \) No spillage \( \text{Spillage} \)

\( \text{Wns-1} \) \( \text{Wns} \) \( \text{Wns+1} \)

\( h(\text{ns}) \)

\( \Omega_{sp} \)

\( \beta(\text{ns}) \)

\( Lw(\text{ns})/R \)

Last Non-spilling Loop

Figure 4.11 Layout of the Last Non-Spilling Loop
know if this phenomenon is occurring? This is explained in the Section 4.5.7.

4.5.4 The Calculation Routine for the Bubbling Region

The known parameters to start this routine are the same as for the spilling region routine.

As with the spilling calculations, the calculations to accommodate the affects of bubbling start at the outlet end where n=N (unless high-spilling region exists) and continue to the outlet side of the last non-bubbling loop, nb+1. The method used is as shown on Figure 4.10 and, because the head differences across the loops experiencing bubbling are all the same, the calculations are relatively simple.

4.5.5 Calculations for the Last Non-Spilling Loop

The last non-spilling loop is the boundary between the non-spilling region and the spilling region.

Consider the case in liquid plug Wns which is receiving spillage from Wns+1 but is not spilling itself. See Figure 4.11. In the revolution of the last non-spilling plug, Wns starts to receive spillage at the beginning of the revolution and at the end starts to spill itself.

$\Phi(ns)$ is governed by the length of $La(ns)$ which it turn is determined by $H(ns)$; $\beta(ns+1)$ is fixed by the spilling conditions. $H(ns)$ will be known from the spilling region calculations. The unknown parameter is $Lw(ns)$ and this will lie within the limits of $Lwi$ and $Lw(ns+1)$ depending on how far the plug has progressed through last non-spilling plug cycle. The calculations to find the plug length from a known value of $\beta(ns)$ are incorporated directly into the calculation procedure shown in Figures 4.13. In Figure 4.13, $\beta(ns)$ is reduced in steps from $\pi-\Omega sp$ (just spilling) down to the value it would have
had if it was in the non-spilling region; this continues until a solution is found. In effect, this method progressively reduces the liquid plug length from the upper limit stated above until the boundary conditions are satisfied. The reason of doing this is explained in Section 4.6.5.

4.5.6 Calculation Routine for the Last Non-Bubbling Loop

If nb is the loop number of the last non-bubbling loop, then air plug Anb will be receiving air from Anb+1 but not losing any itself. The maximum length of this plug will be Lai when it is about to bubble itself. The minimum length will be Lai compressed by the pressure of H(nb), in this case, this will occur when Anb will just be starting to receive air.

The method, therefore, is to assume the length of Anb to equal Lai initially, then the length is progressively reduced by adjusting Ø(nb) in the calculation procedure until the boundary conditions are satisfied. This is very similar to the spilling case and it is explained more fully in Section 4.6.6.

4.5.7 Calculations of High-Spilling Region

The high-spilling region occurs between the outlet and the bubbling region. It only happens at high pumping heads and the plugs act in a very similar way to the spilling loops.

What causes a region of spilling loops to appear in a coil that should be dominated by bubbling? As already mentioned, this behaviour occurs with a low depth of immersion and a high lift. Under these conditions, the air plugs leaving the coil have been subjected to a significant amount of compression which has caused the liquid plugs to swing towards the inlet. With the short length of liquid plug, this swing causes bubbling to occur.
Figure 4.12(a) Outlet approaching a Bubbling Plug

Figure 4.12(b) Outlet Stopped Bubbling

Figure 4.12 Starting Mechanism for High-Spilling
Consider a bubbling plug leaving the coil as shown on Figure 4.12. As the outlet arm approaches the leading edge of the plug, we will assume that bubbling is occurring; see Figure 4.12(a). Once the outlet arm starts to take in the liquid plug, bubbling cannot occur and so the liquid plug will swing towards the inlet. If this swing is sufficient, spilling will occur which will lengthen plug WN-1 preventing spilling occurring there. Once this mechanism is set up it is self sustaining.

The swing of liquid plug WN towards the inlet during the last revolution is equal to that given by Equation 4.7. The swing per radian of drum movement will be,

\[ \text{Swing/rad} = \frac{(\text{La}-\text{La}(N))}{2\pi}. \]

Equation 4.28.

In Figure 4.12(a) the amount that plug WN can swing is dependent on the time it takes of the inlet arm to sweep in the liquid plug. The angle subtended by WN is \( \frac{Lw(N)}{R} \).

If spilling is to occur, then the trailing edge of WN will be at \( \Omega_{sp} \) to the vertical. For a bubbling plug to spill and start the high-spilling mechanism, the swing of the liquid plug must lift it up to the spilling position. Therefore, from Figure 4.12(b) if,

\[ 2\pi \cdot \Omega_{sp} \left[ \frac{Lw(N)}{R} - \Omega_{bub} \right] < \left[ \frac{(\text{La}-\text{La}(N))}{2\pi} \cdot \frac{Lw(N)}{R} \right] \]

Equation 4.29

then high-spilling will occur. The right hand term of this equation is derived from the swing in Equation 4.28 multiplied by the angle subtended by WN.

Once it has been established that high-spilling is occurring, then the spilling procedure is used with a check on each loop to see if bubbling has occurred (See Section 4.4.7).

Annable (1982) measured the level differences in loops containing a high-spilling region, but he did not identify it as having different hydraulic
behaviour.

4.6 THE OVERALL CALCULATION PROCEDURE

4.6.1 Introduction

Two boundary conditions have to be satisfied. Firstly, the pressure at the inlet to the pump has to be at atmospheric pressure and, secondly, the pressure at the outlet has to equal the applied lift pressure, Hout. Two further constraints apply; the lengths of the inlet plugs are governed by the pump geometry and the spilling and bubbling conditions have to be catered for. Because of these restrictions, an iterative process has to be used to obtain a solution.

The calculations can be divided into two procedures. The first varies the number of spilling or bubbling loops to obtain the closest value of the calculated outlet pressure (Ha(0)) to that of Hai, the atmospheric pressure. This provides the coarse adjustment in finding the solution. The fine adjustment is carried out by varying the plug lengths of the last non-spilling or the last non-bubbling loop until Ha(1)=Hat within the required tolerances.

The calculation routines and the iteration processes cannot be undertaken by hand and a series of computer programs were written to undertake these tasks. Firstly, the programs were written in Basic for ease of development and later they were rewritten in Fortran for speed of processing.

4.6.2 Methods of Analysis

Two alternative methods of convergence on a solution are possible. The first uses two sets of calculations, one starting from the outlet and the other starting from the inlet. A balance is then determined by an iteration process
at the boundary between non-spilling and the spilling/bubbling regions.

Galvin (1987) adopted this first method and he devoted a lot of effort to ensure that the process always converged. His computer program worked well, but it consumed a large amount of computer processing time. Weir (1979) also adopted this approach but his program was relatively unsophisticated.

The method adopted here also requires a substantial amount of computational time, but it does avoid the problems of homing in on 'false' solutions that occur with the method described above. In this method, each calculation run starts at the outlet and moves to the inlet. After each run, the number of spilling or bubbling loops is increased by 1 and this process continues until the outlet pressure drops below atmospheric. The rotation of the last non-spilling loop, (or non-bubbling loop) \( \theta_{(ns)} \) is then reduced from its maximum value to give a fine adjustment to satisfy the boundary condition of atmospheric pressure at the inlet.

4.6.3 Initial Data and Assumptions

In order to carry out these overall calculations, a number of parameters must be known or have to be assumed. These are as follows;

(a) drum diameter,
(b) pipe diameter,
(c) depth of drum immersion,
(d) pressure at pump outlet (this is discussed in Section 4.11),
(e) speed of drum rotation,
(f) total number of loops (a whole number),
(g) inlet pressure, normally assumed to be atmospheric.

The assumptions used in these calculations are as follows:

(a) dynamic effects can be neglected and a static analysis at one set of plug positions can be used to provide a good indication of the pump's
behaviour;
(b) the length of the helical coil will effectively hold a number of complete liquid plugs;
(c) a liquid plug is about to leave the coil through the outlet arm;
(d) liquid plugs are all the same length in the non-spilling region,
(e) the pipe diameter of the helical coil is sufficiently large to allow spilling or bubbling to occur freely.

Atmospheric pressure $H_{a}$ is taken to be 10.13m in all the calculations. Forrester (1985) investigated the effects of pressure changes on the pump behaviour and found them not to be significant.

4.6.4 Spilling or Bubbling Loops?

It is assumed that only one of these two phenomena will predominate in the coil, though, it is possible for high-spilling and bubbling regions to occur together. Referring to Figure 4.7(b) and also Figures 4.5 and 4.6, if,

$$\pi \cdot \frac{L_{w}}{2R} - \Omega_{sp} < \frac{L_{w}}{2R} - \Omega_{bub},$$

then the leading edge will dip to the bubbling position before the trailing edge rises to the spilling position.

It is possible that neither spilling nor bubbling will occur at low lifts and this is catered for in the main calculation procedure by initially setting the number of spilling/bubbling loops to zero.

4.6.5 Determination of the Number of Spilling Loops

Assuming that it has been established that there is a spilling region in the coil, then for the initial calculation runs, it is also assumed that the last non-spilling loop is just about to spill, that is,
Figure 4.13 Calculation Procedure with varying No of Spilling Loops
If $Ns$ is the number of spilling loops then $\beta(ns)$ will equal $\beta(N-Ns)$.

This condition will produce the greatest cumulative head that can be achieved across the non-spilling region and hence, the lowest inlet pressure for a fixed outlet pressure.

In the calculations, the number of spilling loops, $Ns$, is varied from zero, i.e. no spilling occurring at all, to $Ns=N$ where all the loops are spilling. $Ns$ is the number of spilling loops.

For each value of $Ns$, the spilling calculation routine is used for the spilling region. The head difference is calculated across the last non-spilling loop and the non-spilling calculation routine is carried out on the rest of the loops. The procedure is best explained on the flow chart on Figure 4.13.

As the number of spilling loops increases so the total head across the coil increases. The total head that can be generated across the non-spilling region is limited to a maximum value and so additional head capacity can only be gained by adding in more spilling loops. This maximum value in the non-spilling region is caused by the level difference decrease towards the inlet in an exponential manner and the maximum value is controlled by the pump geometry.

Since the head at the pump outlet is fixed, increasing the number of spilling loops decreases the head at the outlet. The process of increasing $Ns$ continues until $H(0)<H_{at}$.

4.6.6 Determination of the Number of Bubbling Loops

$Nb$ is the number of bubbling loops and $nb$ is loop number which contains
the last non-bubbling liquid plug. $\Omega(\text{nb})$ is assumed to be equal to $(\pi - \Omega(\text{bub}))$ which is equivalent to $A_\text{nb}$ having a length of $L_\text{ai}$. The logic and method is then the same as the spilling case shown on Figure 4.13.

If high-spilling is occurring, then the method described in Section 4.5.7 is used to determine the number of loops and the air plug pressure head on the inlet side of this region. The bubbling region routine is then applied to the reduced 'outlet head' and the fewer number of loops.

4.6.7 Determination of $\beta(\text{ns})$ to satisfy Boundary conditions

For this part of the calculation it is assumed that the number of spilling or bubbling loops have already been determined. $\beta(\text{ns})$ is then used as a fine adjustment to the calculations.

Taking the spilling case, in Section 4.6.5, $\beta(\text{ns})$ was assumed to be $\pi - \Omega(\text{sp})$, that is, just spilling. In fact, it is unlikely that this is the case. This value for $\beta(\text{ns})$ represents the upper limit and so $\beta(\text{ns})$ is reduced by a small decrement $\Delta\beta(\text{ns})$ each time and the calculations for the last non-spilling loop and the non-spilling region are repeated. This process continues until $(H_a(0)-H_a(\text{i}))$ is within the required tolerance. The action of reducing $\beta(\text{ns})$ has the effect of rotating the non-spilling region backwards in relation to the other coils. At the beginning of the duration of the last non-spilling loop it starts to receive spillage, but does not spill, the end of the duration, it is about to spill itself.

With this analysis, the number of spilling loops and the last non-spilling loop is being used to find a stable arrangement of level differences that match the applied head. In fact, in practice the coil will adopt a similar method, but this analysis will produce only one stable balance point in the drum cycle, this is discussed further in Section 4.10.

It was originally proposed to use the Successive Bi-section Method to cause
Figure 4.14 Final Calculation Procedure where Spilling Occurs
convergence on a solution, but problems were experienced in setting a stable lower limit which would work for all cases. Eventually it was decided to adopt the crude method of approaching the solution in small steps from the known upper limit.

The flow chart for the calculations is shown on Figure 4.14.

If the head difference across the first loop is significant, then the air and liquid plug length has to be recalculated according to Equation 4.25 and 4.26 respectively. The whole calculation process has to be repeated until the liquid plug length used in the calculations agrees with the one produced by the calculations, within the required tolerance of 1mm. Problems were experienced in ensuring that this convergence always worked when the pump was close to its failure point.

When the calculation procedures shown in Figure 4.13 and 4.14 are combined, the output can be used to plot the level differences and the flow rates for a particular pump.

In the case where bubbling occurs, the same method is followed except that the third box down in Figure 4.13 is now Ω(nb)=π-Ωbub and Ω(nb) replaces Ω(ns).

4.7 AIR AND LIQUID FLOW RATES

The Coil Pump pressurises both air and liquid, but the focus of this Thesis is concentrated on the liquid flows and not the air flows.

The liquid flow rate is governed by the average amount that the coil inlet scoops in, in each revolution. The inlet plug lengths, Lwi and Lai, have been discussed in Section 4.4.10.
The average liquid flow rate will be

\[ Q_t = L \cdot w_i \cdot a \cdot S \]  

Equation 4.31.

\( Q_t \) is the theoretical flow rate,
\( a \) is the cross sectional area of the helical pipe,
\( S \) is the speed of rotation.

In practice, the actual flow rate will be given by,

\[ Q_a = C_d \cdot Q_t \]  

Equation 4.32.

\( C_d \) is the discharge coefficient.

Two factors could affect the use of Equations 4.31 and 4.32 to predict the actual flow, namely drum speed and lift. If the inlet was sweeping around the drum at a very low velocity, \( C_d \) should be close to one. As the velocity increases, the dynamic losses at the inlet must start to influence the amount of liquid taken in by the coil. In general, they would reduce the amount of liquid entering the coil but it is unlikely that they would affect the air intake.

If the pressure in the first air plug in the coil, \( H(1) \), is significantly higher than atmospheric, then there must be a head difference across this first coil to counteract this. Assuming that the leading edge of the liquid plug entering the coil is at the external liquid surface then, by the time the whole of the liquid plug has entered the coil, this leading edge must be depressed. This has been discussed previously in Section 4.4.10.

The two factors affecting the flow rate which has just described are postulations, which can only be proved by experimental evidence. Attempts at achieving this have not proved very successful, but a description of this is left to Section 5.5 in the next chapter.

4.8 POWER TRANSFER MECHANISM

It is not possible to carry out a theoretical study of the efficiencies of the coil
pump, but it worth considering the power transfer mechanism at this stage. Power is applied to the axis of the drum through a mechanical drive. In the case of the laboratory pump, the power source was an electric motor turning the shaft via a toothed belt and pulley wheels. How is this mechanical power converted to hydraulic power? The transfer occurs at the end of the coil where the helical pipe turns through a right angle as it enters the outlet arm: see Figure 4.15. The force on the bend due to hydrostatic pressure, ignoring momentum changes,

\[ F_{out} = \rho g Hout a \]  

Equation 4.33.

\( \rho \) is the density of water,
\( g \) is acceleration due to gravity,
\( a \) is the cross sectional area of the helical pipe.

The torque on the shaft due to this force = \( F_{out} R \).
The power associated with this torque P = \( 2 \pi F_{out} R S / 60 \), if \( S \) is the speed of
revolution in rpm.

Substituting in Equation 4.33,

\[ P = p \cdot g \cdot \pi \cdot D \cdot a \cdot \left( \frac{S}{60} \right) \cdot H_{out} \]  

Equation 4.34.

This equation represents the mean hydraulic power for a average quantity of fluid entering (and filling) the coil per second when pressurised to a head of Hout.

The power required to pressurise the liquid

\[ P_w = p \cdot g \cdot L_{wi} \cdot a \cdot H_{out} \cdot \frac{S}{60}. \]

If the air were acting as a perfect gas,

\[ P_a = p \cdot g \cdot L_{ai} \cdot a \cdot H_{out} \cdot \frac{S}{60}. \]

If \( L_{ai} + L_{wi} = \pi \cdot D \) then from Equation 4.34, \( P = P_w + P_a. \)

The force on the bend of the coil can transfer sufficient power to supply the hydraulic requirements. In practice, energy will be lost to the system in pressurizing the air as it act as a 'real' gas, and not an 'ideal' gas. The compressed air at the pump outlet will, however, assist in lifting the water up the delivery pipe and so some of this pressure energy is effectively recovered.

4.9 DYNAMIC EFFECTS IN THE COIL

It has been assumed previously in this Chapter that the whole of a liquid level difference in a loop is available to resist the pressure change across the loop. In fact, this is not true since some of this head is dissipated in overcoming friction and other minor losses. If friction is significant then,

\[ h_e(n) = h(n) - h_f(n). \]

\( h(n) \) is the observed head difference.
\( h_f(n) \) are the dynamic losses associated with liquid plug \( n \).
\( h_e(n) \) is the effective head available to resist a pressure difference.

As the drum speed increases so \( h_f(n) \) increases. There are difficulties in assessing \( h_f(n) \) since it is doubtful whether the Colebrook-White and the
D'Arcy Equations can be used to assess the friction loss in a liquid plug because of the unusual flow regimes within the liquid plug. It is also difficult to find a realistic loss factor for the inlet as it sweeps through the external liquid.

Above about 20 rpm on the laboratory pump it is evident that the dynamic losses are starting to play an important part in the pump's behaviour. Below this figure it is felt that they can be ignored in most cases without a great loss in accuracy. The exception to this is a very large diameter drum rotating at a slow speed, here again the flow velocities may be high because of the large radius.

4.10 MULTIPLE ANALYSES WITHIN A DRUM CYCLE

Investigations have been carried out on the effects of partial plugs in the inlet and outlet, though they are not reported here. The plug configuration in the coil was analysed at time intervals of an eighth of the drum cycle time. The lengths and level differences of the plugs entering and leaving the coil at the particular time were calculated. From this it was possible to determine the approximate variation in the capability of the coil to resist pressure. In fact, the calculations suggest that there is no significant weak point on the drum cycle when the pump has a poor capability to resist the applied pressure.

4.11 THE DELIVERY PIPE

There is insufficient space to describe the delivery pipe behaviour in this Thesis, instead a brief description will be given. When the pump is working, air and liquid plugs alternately enter the delivery pipe. At the pump outlet, the pressure head is equal to the sum of the liquid plug lengths in the delivery pipe at a particular time. This sum changes as liquid plugs leave and enter the delivery pipe. In the pipe, the air plugs have a tendency to rise up through the liquid plugs and if the liquid plugs have a high velocity, the
relative velocity between the two fluid phases is less. Because of this the speed of rotation of the drum affects the outlet head.

A series of static analyses on the delivery pipe can be used to determine Hout throughout the drum cycle and this has been found to agree well with pressure transducer readings, or further details see Mortimer(1984).

The delivery pipe pressure variations affect the pump in two ways. Firstly, these changes cause the liquid plugs to oscillate about a mean in the coil. Secondly, a change in the pump speed affects the air and liquid plug ratio in the delivery pipe and, as the speed increases, so Hout increases for the same lift.

4.12 COMMENTS

The theory developed in this chapter is not mathematically concise, nor can it be. The process of calculation is spread over a number of flow charts and they have to be brought together into large computer program before the results can be produced. The source code for these programs has not been reproduced in this thesis for a number of reasons.
(a) The code would occupy a large amount of space.
(b) Complex program codes are very difficult to understand by a reader new to the subject.
(c) The aim of this thesis is not to provide a working manual for the design of lift coil pumps.
CHAPTER 5

TESTING THE THEORY
FOR THE LIFT PUMP

5.1 INTRODUCTION

The aim of this chapter is to show how the results from the practical laboratory tests on the coil pump compare with those predicted by the theory described in Chapter 4.

Over 150 test runs were carried out in the laboratories at Loughborough. 65 runs were undertaken by Annable (1982) on his postgraduate research project; the rest were undertaken by myself or by undergraduates on final year projects.

All the results used in this chapter are taken from work I carried out myself or where I acted as supervisor and checks were made during the project to assess the reliability of the results.

The comparison of the calculated and measured results falls conveniently into three main sections. The first, in Section 5.4, deals with the level differences within the loops. The second, in Section 5.5, is concerned with the comparison of measured and calculated flow rates. Thirdly, Section 5.6, describes the failure mechanism of the pump.

Of the shorter sections, 5.7 deals with the characteristic curve and 5.8 describes some of the work carried out on power consumption and efficiency.
Figure 5.1 The Layout Of the Laboratory Lift Pump
5.2 THE LABORATORY RIG

The laboratory coil pump is shown in Figure 5.1. The drum supporting the coil sits in a holding tank in which the water level can be varied. The drum is powered by a variable speed electric motor driven through a toothed belt and pulleys.

The water leaving the coil pump rises up through a vertical delivery pipe into a header tank which can be varied in height. The water then leaves this tank via a weir and pipe, and enters a secondary tank at a lower level where the flow is recorded using a measuring cylinder and stop watch. The flow then returns to the holding tank forming a closed system.

5.3 THE LABORATORY METHOD

Most of the results quoted in this chapter are taken from Annable's postgraduate work. He used two drum sizes in his experiments, 0.88m and 0.488m. In combination with these drums he used three diameters for the helical coil, 13mm, 25.4mm and 38mm. With various combinations of drums and coil sizes, he varied the rotational speed, number of loops, and lift. For each combination he measured the flow rate and liquid level differences in the coil. In most of the experiments carried out at Loughborough, the pump was discharging into a vertical, or near vertical delivery pipe.

The level differences in the loops could not be measured accurately whilst the pump was in motion so the levels were taken immediately after stopping the pump. Analysis of video pictures of the rotating pump indicated that the liquid levels in the moving pump oscillated about a mean and, when the pump was stopped, the stationary levels gave a good indication of the mean dynamic levels. When the pump was stopped, the air plugs in the delivery
pipe rose up through the water plugs and out of the pipe outlet. This left a water column in the delivery pipe which was measured. A pressure transducer in the base of the delivery pipe indicated that the pressure head just before the pump was stopped was generally within 10% of the pressure generated by the static water column after stoppage.

Any other measurements of the internal workings of the pump were considered to be too difficult to measure and transmit to the logger from a semi-submerged rotating drum. It would have been desirable to measure, say, the internal pressures at certain points in the helical coil but it was felt that the effort was not worth the anticipated results.

It was necessary to damp out fluctuations of flow in the delivery pipe, and so a substantial amount of storage was needed in the system. The flow rate was measured after the conditions became steady, this could take between 10 to 15 minutes.

On most of the examples of measured data used in this chapter, estimates have been made of the experimental errors involved. These estimates have been determined by myself and not the person who undertook the experiments.

5.4 LEVEL DIFFERENCE COMPARISONS

5.4.1 Introduction

Why compare the measured and calculated level differences? On comparison, the two sets of results, under a wide range of different conditions, should look very similar. If they do not, the theory is of little use. If they are similar, the theory should be able to predict the head limitations and attributes of the pump in a particular situation.
In comparing the results, a plot of the two sets of level differences is an obvious method. A visual assessment of the closeness of the two sets is subjective since one can only attribute a degree of closeness of the two lines which is based on personal assessment.

Correlation coefficients could be calculated for the measured and the calculated level differences though they tend to give a coarse measure of the goodness of fit. Aitken (1973) discusses this problem when attempting to compare measured runoff hydrographs against calculated ones: the hydrograph shapes are similar to the level difference curves. He describes well a number of different parameters which could be used to evaluate the systematic errors, which in this case, have caused the difference in the curves. Brief test carried out using correlation coefficients, coefficients of determination and coefficients of efficiency indicated that no one parameter performed significantly better than the others.

Because the correlation coefficient is familiar and well used it was decided to use this parameter together with the horizontal displacement of the curves to measure the closeness of the two curves.

5.4.2 Description of the Measured Level Difference Graphs

This section deals with the variation of the measured level differences in the loops under a variety of conditions. The graphs referred to in this section also include the calculated results and they should be ignored at present as this comparison is dealt with in the next section.

Figure 5.2 shows the first example of the theoretical and measured level differences. The horizontal axis denotes the loop number and the vertical axis, the liquid level difference across that loop. Figure 5.2 can also be thought of as a plot of the level difference across a plug as it moves through the coil,
Figure 5.2 Level Differences for Spilling Loops. H_{lift}=4.0m

Data:  
- D=488m  
- d=25mm  
- Dimer%=50  
- S=12rpm  
- H_{out}=2.9m

Loop No from inlet

Figure 5.3 Level Differences with Spilling Loops. H_{lift}=6m

Data as Fig 5.2 except,  
- H_{out}=4.78m

Loop No from Inlet
and in this case, the points on the bar chart would be joined to make a continuous curve.

The non-spilling region in the pump is represented by the rising part of the curve from the origin to the peak. Loop number 16 represents the last non-spilling loop and loops 17 to 20 denote the spilling region.

In Figure 5.2, the pump was lifting water a height of 4 metres but the pressure at the pump outlet was only 2.9 metres due to the buoyancy effect of the air in the delivery pipe. See Section 4.11 for further details. Under these conditions, the pump was operating well below its maximum head capacity and so a number of loops are redundant. If these redundant loops, numbers 1 to 7, were removed from the pump, the level differences in the remainder of the loops would not change significantly. In Figure 5.2, the measured level differences in some of these loops (1 to 7) have a small negative value. At present it is not possible to explain why these small negative level differences occur in practice, the theory indicates that they should fall exponentially towards zero.

As the outlet pressure increases, the sum of the level differences across the loops must increase; this is shown in Figures 5.3 and 5.4. In Figure 5.3 the lift is 6 metres (4.78 metres effective pressure head) and, in Figure 5.4, it is 8 metres (6.35 metres effective pressure head). As the lift increases, the curve moves to the left to cause an increase in the sum of the head differences. This curve is a fixed shape for a particular pump at a specified depth of immersion and this can be proved by constructing a master curve on a transparent sheet and comparing it with the displaced measured curves.

As the pump approaches its maximum lift, the curve moves further to the left and the level difference across the first loop increases. The situation in Figure 5.4 occurred at a lift of 8m, about 0.5m below the pump's maximum.
Figure 5.4 Level Differences For Spilling Loops. H_{lift}=8m

Figure 5.5 Level Differences with Spilling Loops

- Measured Values
- Calculated Values

Data as for Fig 5.2 except

D_{m}=915m
d=25mm
Dimer\%=65
S=5.4 rpm
H_{out}=7.8m
pump failure is discussed in Section 5.6.

Figure 5.5 shows a further example of a pump with a spilling region. The drum diameter is nearly twice the previous cases and the depth of immersion is 65%. The gradient of the line in the spilling region is steeper than the previous case because of the high outlet pressure of 7.8m.

Examples of the same pump experiencing bubbling when operating at high lifts, are shown in Figure 5.6 and 5.7, with a depth of immersion of 30% of D in both cases. In Section 4.4.7, it was suggested that all the level differences in a bubbling region are the same. In Figures 5.6 and 5.7(a), the measured levels are irregular in the bubbling zone, this is because the bubbling effect is produced by the rising movement of a bubble which has grown sufficiently large to break away under the low point of the loop. When the pump is stopped, the air plug length will depend on whether the bubble under formation has risen through the water plug or, it is still held in the air plug. On some occasions when the pump was stopped, a bubble of air would rise up through the water plug after it seemed that conditions had settled down, this extra bubble movement would change the two air plug lengths. Video pictures of the bubbling region on a rotating pump suggest that the level differences are, in fact, very similar.

Figure 5.7 also shows a high-spilling region adjacent to the outlet, but the discussion on this will be left to Section 5.6.

5.4.3 Comparison of Theoretical and Measured Values

As mentioned previously, over 100 sets of data exist for the measured level differences. It is proposed to show only seven sets in this thesis, but the ones chosen cover a range of operating conditions and are representative of most of
The heavily shaded blocks in Figures 5.2 to 5.4 show three examples of the calculated values of the level difference graphs for a pump subjected to an increasing lift. Figure 5.3 shows a good agreement between the two sets of results (Correlation Coefficient 0.98). Figure 5.2 has significant errors between the measured and calculated values particularly, on loops 14, 15 and 16. The theory does not allow for the development of negative level differences near the inlet. Effectively, the theoretical level differences in loops 1 to 7 in Figure 5.2 are zero. Since the measured heads are starting from a lower base line when rising to the peak difference, it is likely that this is the cause of the errors in loops 14 to 16.

Figure 5.4 shows good agreement between the measured and calculated values (Correlation Coefficient = 0.99). The only exception to this is the difference across the first loop where the measured value is substantially higher. In the theoretical calculations, the level difference across the first loop reduced the water plug length from 0.767 to 0.72m and so reduce the level difference. See Section 4.4.10 for further explanation.

On Figure 5.5 the drum diameter has been doubled and the depth of immersion increased to 65%. Here the fit between the two curves is reasonably good, with a correlation coefficient of 0.96 and now, the calculated values have a horizontal displacement of one loop to the right. If the spill angle, Ωsp, is increased by an arbitrary 5° then the level differences across the spilling coils would decrease and the fit between the two curves would improve. (Correlation Coefficient = 0.98). This might suggest that the spill angle as calculated by the method given in Section 4.4.8 is an under estimate. In Appendix 3 some reasoning is given on the factors that may affect the magnitude of Ωsp and it seems very likely that the larger drum diameters may produce values of Ωsp which will lead to lower level differences.
Figure 5.6 Level Differences with Bubbling Loops

Measured values

- D = 0.488 m
- d = 0.025 m
- Dimer% = 30
- S = 12 rpm
- H_out = 4.57 m
- H_lift = 7 m

Theoretical values

Loop No
Figure 5.7(a) Level Differences with Bubbling Loops

Data as for Fig 5.6 except Hout=6.9m

Loop No

Figure 5.7(b) Water Plugs Lengths for Fig 5.7(a)

Loop No

77
It was considered at one stage that a detailed analysis of all the measured level difference curves might have lead to guide lines which could be used to determine $\Omega_{sp}$ more accurately but it was abandoned after a sensitivity analysis suggested the results would be of little use.

Figure 5.6 shows theoretical and measured level differences where bubbling is occurring. The theoretical assumption described in Section 4.4.7 which is used to determine when bubbling starts is that $\mathcal{O}(n)=\pi$. As with the spilling case, an adjustment to $\Omega_{\text{bub}}$ will improve the fit between the two curves, but this is not considered in detail here since this is considered a matter of tuning the results. However, the factors affecting $\Omega_{\text{bub}}$ and $\Omega_{sp}$ are well worth further investigation.

Figure 5.7(a) shows level differences on another bubbling pump, but in this case, spilling has occurred adjacent to the inlet, a high-spilling region. The extent of this spilling region is clearer on a plot of the water plug lengths shown in Figure 5.7(b). The theoretical level differences and the plug lengths follow the same patterns as the measured ones but they do predict them accurately. This difference occurred on a number of high-spilling cases and it is difficult to suggest a reason why - possibly an error in the estimation of $\Omega_{sp}$.

5.4.4 Effect of Drum Speed on Level Differences

The main effect of speed is on the flow rate, this is discussed in Section 5.5. Drum speed also effects the level differences in the loops. This cannot be checked directly since only the static levels have been measured.

It is difficult to demonstrate the effect of speed directly on the level differences in the loops since a speed change also changes the ratio of air to water plugs in the delivery pipe. This changes the head at the pump outlet and, hence, the
levels differences in the loops.

Attempts were made to relate drum speed to the number of spilling loops, the spilling level difference gradient and the spilling water lengths, none proved successful. This is probably because the effects are small and are masked by experimental errors.

Speed must affect the behaviour of the pump but it is difficult to isolate. Further investigations on this problem are required, particularly at higher speeds.

5.4.5 Variation of Outlet Head

It is not proposed to discuss this head variation in any great detail. Methods are available for predicting this behaviour (Mortimer, 1986). Some of the effects of the air plugs in the delivery pipe can be seen by comparing $H_{\text{lift}}$ and $H_{\text{out}}$ in Figures 5.1 to 5.8. In general with the laboratory pumps, the head at the outlet of the pump is about 75% of the lift.

5.5 FLOW RATE COMPARISON

Only the liquid flow rates will be discussed in this section because the lift pump is primarily a liquid lifting device; it is also very difficult to measure the air flow rate accurately without disturbing the liquid flow.

The theoretical estimation of the flow rates is developed in Section 4.7. Comparisons were made between the flow rates calculated by Equation 4.31 and the measured flow rates. The recorded flow rates were measured using a stop watch and measuring cylinder. A substantial storage was provided in the header tank to damp out the pulsating nature of the flow in the delivery pipe.
Figure 5.8(a) Errors in flow rate estimation with varying depths of immersion

Error = 100 \cdot \frac{(Q_a - Q_t)}{Q_a}

Dimer = 30\%
Dimer = 40\%
Dimer = 50\%
Dimer = 60\%

S = 12 \text{ rpm}
D = 0.456 \text{ m}
d = 0.025 \text{ m}
H_{\text{lift}} = 3 \text{ to } 8 \text{ m}

Figure 5.8(b) Errors in flow rate estimation with varying speeds of rotation

Error = 100 \cdot \frac{(Q_a - Q_t)}{Q_a}

Dimer = 50\%

S = 12 \text{ rpm}
D = 0.456 \text{ m}
d = 0.025 \text{ m}
H_{\text{lift}} = 3 \text{ to } 8 \text{ m}
Figure 5.8(a) shows the variation of the depth of immersion with the percentage error between theoretical and measured values. All 32 readings were taken at a constant speed of 12 rpm, but the lift varies between 3 and 9m. Even with this extra variable hidden in these results, the mean error is close to, or within, the band of experimental error.

In Section 4.7, it was suggested that speed of rotation may have an important influence on the accuracy of estimation. Figure 5.8(b) suggests that this may not be the case for low speeds of rotation. There is no noticeable trend in the accuracy of prediction as the speed increases. One could suggest that the errors become less negative (or more positive) as the speed increases on Figure 5.8(b) but taking the experimental error band into account this rise is not significant. Flow measurements on the Coil Treatment Unit which is 1.5 metres in diameter and rotates at between 1 and 2 rpm shows the percentage error between theoretical and measured to be within ± 4%.

Stuckey and Wilson (1980) found that the theoretical flow rates did not always agree with the measured values; they suggested that 'slip' may account for the discrepancy. The slip may be positive or negative leading to a positive or negative error. They suggest a negative slip is produced by a pressure near the inlet of greater than atmospheric; a positive slip is produced by air escaping through the rotary joint. As mentioned previously, a large head difference across the first coil does reduce the flow rate. Regarding a positive slip, we made great efforts to ensure that the rotary joint did not leak, so we were not troubled by this problem.

A further factor to consider is the effect of a changing outlet head on the flow rate. Surprisingly this seems to be small. Consider Figure 5.9 where three flow rates (expressed as a flow per rev) are plotted against head. The first two are measured flow rates at 8 and 16 rpm and the third is found by averaging the first three measured water plug lengths in the coil and multiplying them
Figure 5.9 Variation of Flow rate with Lift

Theoretical flow

+ 16

□ 8rpm

◇ Q from averaged, measured plug

D = .488m

d = .25mm

Dimen% = 50
by the cross-sectional area of the pipe.

Two points are apparent from Figure 5.9. Firstly, the flow rate varies little with increasing head and, secondly, the three flow rate estimations agree well with each other and with the theoretical flow rate at low pumping heads also shown on this graph. Figure 5.9 is typical of the behaviour of the pump under a wide range of conditions.

Since Equation 4.31 generally predicted the flow rate to within + or - 4% over a wide range of conditions, it was decided that a more detailed analysis of the parameters influence on flow rates was unnecessary at this stage.

5.6 THE FAILURE MECHANISM

5.6.1 Introduction

The failure mechanism is difficult to investigate in the laboratory since the pump behaviour becomes unstable as it approaches its failure point. The pump may operate under a quasi-steady state for, say, 10 min, before it gradually starts to fail. Firstly the flow rate decreases then suddenly, blowback occurs. With blowback, a plug of air and water are ejected from the inlet with great force.

Failure of the pump occurs when the pump is unable to resist the head difference across it and the manometric structure collapses. The laboratory and theoretical investigations suggest a failure is caused by three related mechanisms. These are:

(a) the level difference across the first loop increasing to a large value;
(b) spilling or bubbling mechanism reaching the first loop;
(c) the shortening of the inlet water plug length.

Which of these phenomena actually causes the failure in a particular case is
difficult to predict. As the pumping lift increases, all three tend to become significant at about the same time, mainly because they are directly or indirectly related.

5.6.2 A Spilling Failure

This is best illustrated by an example. Take the case where, D=0.488m, d=0.025m, Dimer=50%, S=16 rpm, and N=20. The laboratory pump would lift water to a height of 9m (Hout=6.2m) and maintain a steady flow rate. At a height of 10m (Hout=7m approx), it started by lifting water to this height, but after a time, the water plug taken in by the inlet was ejected in a violent manner, we have termed this 'blowback'. For this reason the measured levels could not be plotted at this lift.

The failure can be explored in more detail in theoretical terms. The level difference across the first loop and the number of spilling loops as the pump approaches failure are shown below.

<table>
<thead>
<tr>
<th>Head at outlet (m)</th>
<th>Head diff across first loop (mm)</th>
<th>No of spilling Loops</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.2</td>
<td>240.</td>
<td>16</td>
</tr>
<tr>
<td>6.3</td>
<td>280.</td>
<td>17</td>
</tr>
<tr>
<td>6.4</td>
<td>450.</td>
<td>20</td>
</tr>
</tbody>
</table>

Above a head value of 6.4m, the hydraulic structure resisting pressure breaks down.

Though the program does not model the level differences across the first coil with a great degree of accuracy, it does identify the general trends well. The figures above show that for a small increase in the head at the outlet, there is a large increase in the head difference across the first coil and the number of spilling loops, leading to failure.
Figure 5.10 Inlet and Outlet Near Pump Failure
Another factor needs to be included. Consider Figure 5.10 which shows the inlet and outlet in at two positions in the drum cycle. It is assumed that there is an exact number of turns on the helical pipe. As the drum moves from the position shown on Figure 5.10(a) to the position shown on (c), the effective head across the outlet liquid plug decreases whilst the new inlet plug has a limited capacity to withstand a pressure difference across it. Superimposed on this is the variation of the outlet head due to the pressure variations in the delivery pipe.

As the failure point is approached with an increasing outlet head, the head difference across the first loop will be increasing and the spilling reaches the first coil. The maximum capacity for the coil to withstand a pressure head will be varying as will the outlet head. At some point, the pressure head in the first air plug will reach a level sufficient to force air out through the inlet, once this happens the manometric structure of the coil collapses.

The values of level differences in the above table also indicate why the flow rate starts to decrease only just before failure occurs and does not gradually decline. See Section 5.5.

5.6.3 Bubbling Failure

The failure mechanism by bubbling is very similar to that of spilling. As the failure point is reached, the number of bubbling coils increases rapidly and so it is assumed that failure occurs when the first air plug attempts to bubble air through the first water plug. Evidence of this can be seen in the laboratory.

5.7 PUMP CHARACTERISTIC

In many of the previous comparisons, the theoretical results have been compared with data taken from a static pump.
Figure 5.11 Pump Characteristic For Spilling Case

Spilling failure

- D=0.488m
- d=25mm
- S=8 rpm
- Dimer%=50%

- Meas H
- Calc H

Figure 5.12 Pump Characteristic Bubbling case

- Measured H
- Calculated H

- D=0.488m
- d=25mm
- S=12 rpm
- Dimer%=30
With the comparison of pump characteristics, all the measured data are taken from a working pump.

The complete measured and calculated pump characteristics for the case developed in Figure 5.2 to Figure 5.4 is shown in Figure 5.11. In this example the theoretical flow rates are greater than the measured flow rates by 6%. The theoretical failure point by spilling, was at a head of 9m compared by 8.5m(approx) in the measured case. An example for the characteristic for the bubbling case is shown in Figure 5.12. Here the measured flow rate is more variable when compared to the theoretical values. Most calculated points do lie within the variation due to experimental errors, with one exception. This could have been a poor reading.

The theoretical failure point in Figure 5.12 agrees well with the measured results.

5.8 MEASUREMENT OF POWER CONSUMPTION AND EFFICIENCY

This proved to be one of the most frustrating aspects of this whole investigation. To illustrate the problem, consider a pump 0.88 metres in diameter wound with a helical coil 25mm in diameter. The pump is rotating at 4 rpm and lifting water to a head of 4 metres. In this case, the power consumption of the electric motor driving the pump was measured in the laboratory at 19 Watts. The power was measured by an integrating ampmeter and a voltmeter. The motor characteristic curves indicate that the motor efficiency was about 70% in these conditions.

The power consumption of the motor and other mechanical parts was then measured with the drum turning, but with a bung in the inlet. This reading gave the power required to overcome the mechanical losses when the helical coil was 'dry'; this was measured at 15 Watts. The hydraulic power was
calculated to be approximately 2 Watts (assuming the air to be an ideal gas).

In this example, the power taken by the mechanical losses was six times the hydraulic power. In addition, the hydraulic power varied throughout the drum cycle. These two factors made the measurement of the hydraulic efficiencies of the pump very difficult.

The situation could have been improved by using a torque-meter between the motor and the pump and a low resistance rotary joint could have been used.

In a conversation with Stuckey at Salford, he described the efforts he had been making to measure the hydraulic efficiencies accurately and this had been proving very difficult to achieve.

Wilson (1981), Annable (1982) and Galvin (1986) all attempted to measure power input (and efficiencies) by monitoring the electrical input. Pump efficiencies varied considerably between 10 and 80%. Results from these tests and others from the Coil Treatment Unit suggest a pump efficiency of around 50% may be a reasonable value to use. The power transfer mechanism described in Section 4.8 could indicate much higher efficiencies are likely. More work needs to be carried out on this aspect and, to date, little has been published by other authors.

5.8 COMMENTS

Can the theory expounded in Chapter 4 be considered successful in its predictions? It is difficult to answer with a simple yes or no. In all the cases tested, the flow rates and failure points have come within 10% of measured values. The closeness of the level difference curves could be improved by the adjustments with \( \Delta \)sp and \( \Delta \)bub in the spilling and bubbling cases, however, more work is needed to fully understand their effects. On the weaker aspects of the theory, the prediction of power consumption and pump efficiencies is
poor but this was to be expected.

In a wide range of cases, the theoretical predictions will give a very good indication of how the pump will behave. In this sense, the theory is successful.
CHAPTER 6

PRACTICAL ASPECTS OF THE LIFT PUMP

6.1 INTRODUCTION

The lift pump has been applied to two practical applications so far, the first is a stream powered version for lifting water and the second use is in treating waste water. Both these applications stemmed from the laboratory work on the coil pump.

The stream-powered version is an obvious use of the pump which has been taken up by a number of people. The second application is more unusual and, as far as we know, it has not been considered by other investigators. It is not intended to describe this treatment process in great detail since much of the development work is not directly relevant to the subject of this thesis. The Coil Treatment Unit did evolve from the laboratory work on the pump and the prediction of its hydraulics is based on the theory developed in the Chapter 4.

6.2 STREAM POWERED LIFT PUMP

6.2.1 A Description

The lift pump has three features worthy of exploiting in a practical situation. It is simple in design, it is easy to construct and it will operate at low speeds. These three features indicated that a low-cost, stream powered version would be worth developing. The reason why the pump was to be a low cost version was because of its simplicity. It could be of use in developing countries, particularly if it was cheap to build and run.
Figure 6.1 Details of the Stream Powered Coil Pump
The design for the pump which Annable and I developed is shown in Figure 6.1. It consists of a 25mm diameter flexible plastic pipe wrapped around the inside of a 50 gallon oil drum to form the helical coil containing 26 loops. The inside of the drum is also filled with inflated automobile inner tubes; this provides buoyancy for the drum when it is working in a stream or river.

The pump is driven by the stream impinging on chevron shaped paddles which are made of scrap metal and spot welded onto the outside of the oil drum. An annular shroud is added to the blades to increase the power transfer by restricting the side ways movement of water. The chevron shape of the paddles was chosen because laboratory experiments suggested that they would produce more power than a flat paddle and would operate in a more stable manner over a wide range of depths of immersion.

The bearings and holding mechanism of the pump consisted of two short lengths of galvanized steel pipe, one attached to each end of the drum via a standard pipe flange. Loose brass pipe-support rings encompassed each steel pipe and these acted as simple bearings. Scaffolding poles were driven into the bed of the stream to form the anchorage points and these were attached to the bearings by steel cables. These cables were looped over the scaffolding poles to allow vertical movement of the drum due to river level changes.

The rotary joint was situated next to one of the bearings and it consisted two rubber lip seals placed back to back and held in a casing made up of standard pvc pipe fittings. The delivery pipe, a flexible plastic tube, was connected to the stationary side of the rotary joint. The weight of this joint caused the drum to heel over when it was floating in water, and this was corrected by adjusting the position of the inner tubes inside the drum.
Plate 6.1 Stream Powered Coil Pump in Operation
6.2.2 Field Tests

The pump shown in Figure 6.1 was tested in a local stream and it lifted 3 l/min up to a height of 8.0 metres. At this pumping rate, the stream velocity was 0.6 m/sec which caused a drum rotation of 8 rpm. With an increased stream velocity achieved by partial stream blockage, the maximum lift under these site conditions was 9.5m and the maximum flow rate obtained was 4 l/min which is sufficient for a village of about 100 people in a developing country. These two maxima could not be achieved together because of the power limitation of the water wheel. The pump operating in the field is shown in Plate 6.1.

The aim of the field test was to show that a stream powered coil pump could operate in a stream and this was achieved. It would have been desirable to test and refine the mechanical design of the pump to achieve a trouble free and robust machine, but we had neither the resources nor the expertise to do this. In fact, this work is better left to the practical engineers working in the countries in which the pump is to be used.

Other people have constructed stream powered versions of the pump. Morgan (1979) built an undershot version in Zimbabwe. This was erected in a field and fed by a channel. The wheel was 2m in diameter to give a good power output.

Sydfynsgruppen (1980)² set up a river powered pump at Wema on the River Tana in Kenya. This consisted of a 2m diameter drum made from local materials and wrapped with rubber hose. It was designed to lift 20m³/day to a height of 4m. This group produced an excellent design manual for constructing one of these pumps which included advice on cutting the paddles from the roof of a VW Beetle! (Sydfynsgruppen 1980(3) ). This same group have another pump working on the River Nile in Sudan; Danish Scouts (undated).

Stuckey and Wilson (1980) also developed a stream powered version of the
pump. The paddles were situated inside the drum and the drum's axis was parallel to the flow. As far as I am aware this pump was not tested widely in the field.

6.2.3 Limitations of the Pump

The most difficult aspect of the design of the stream powered coil pump is to provide a water wheel with sufficient power output to drive a coil pump with an acceptably low mechanical resistance.

Regarding the water wheel, it has to be matched to the stream conditions in which it will be working and to the power needed to drive the pump. The oil drum is not an efficient form of wheel because it is under-shot and, also, it has a small lever arm. Smeaton (1759) showed that over-shot wheels are nearly twice as efficient as under-shot wheels, but making use of an overshot wheel on a river or stream requires extensive civil engineering works.

Our efforts were concentrated on the development of the coil pump and not on the wheel design so there is much scope for work in this area.

The component requiring the greatest attention both in design and operation is the rotary joint. If it is to seal effectively under a high pressure, it is likely that it will exert a high frictional force on the shaft and hence have a high power requirement. If the rotary joint is too loose, the power requirement will be less but the joint may leak after some time in operation. Morgan, the Danes and ourselves did find a combination of wheel and pump which were compatible but others have been frustrated with their designs.

The stream-powered coil pump is not the simple solution to all river based, low-technology pumping schemes. With careful design and construction, it is possible, however, to produce a cheap and reliable form of pumping using only materials that are available in many developing countries.
The stream pump has few direct competitors. The Plata Pump developed in New Zealand is an effective combination of turbine and positive displacement pump, but it cannot be classed as simple low cost technology. The ram pump uses natural power sources and it can be cheaply constructed (Watt (1975), Schiller(1986).), but it does need a few metres head of water for the drive pipe. For this reason its field of application is different to the coil pump.

6.3 THE COIL TREATMENT UNIT

6.3.1 Introduction

At the outlet of the lift coil pump both air and liquid are flowing under pressure. In addition, the internal surface of the coil is subjected to contact with liquid and air alternately whilst under pressure; these three factors can be used to treat waste water (sewage).

It is not intended to explain this treatment process in great detail since it is not appropriate to this thesis. However, a short introduction to the basic methods of treatment is felt necessary before the unit is described.

Biological treatment of waste water is generally achieved by introducing naturally occurring organisms to the waste water and providing a supply of oxygen. The organisms will break down the bio-degradable material in the waste water and so treat it. Practically, one method of achieving this is to expose a moving solid surface alternately to wastewater and to air. A biological film will form on the surface which will breakdown the organic matter. In a second method of treatment, the organisms are held in suspension in the waste water by aeration; the air also supplies oxygen for the process. The aeration can be achieved by either mechanical stirring or by bubbling air through the waste water from a compressed air supply. As the waste water is fed into the tank at one end it flows out at the other. In order to ensure a healthy population
Figure 6.2 Exploded View of Coil Treatment Unit
Plate 6.2 Coil Treatment Unit Operating at Derby Works
of organisms to treat the incoming waste water, the outflow from the tank has to be passed through a settlement tank where the sludge, containing the organisms, settles to the bottom and is returned to seed the incoming waste. This treatment process is called the "Activated Sludge Process".

6.3.2 A Description of the Unit

An 'exploded' view of the Coil Treatment Unit is shown in Figure 6.2. The coil used to drive the unit is 1.4 metres in external diameter and it has eight loops. The helical coil is square (0.2 by 0.2 m) in cross-section and is made from of fibre glass. In fact the helical coil also forms the drum and each coil is made from one fibre glass moulding. This method of construction is similar to that used by Candler (1977) for archimedian screw pumps.

Large diameter coil pumps made by wrapping a pipe around a drum are difficult to manufacture since a large pipe cannot be readily bent to a tight radius. Fibre glass provides a cheap and durable solution to the problem.

The coil is filled with a random-packed plastic medium which has a high surface area but still allows the passage of water. Galvin (1987) studied this aspect in detail and found that at low speeds, the medium offered little resistance.

The coil under normal working conditions is operating with a safety margin against blowback of about 30%. It is also experiencing bubbling in many of its loops.

At the outlet to the pump, the air and waste water are separated and led into their own systems. This is achieved by a cylindrical tank which allows the air to collect at the top where it is lead into the air system. The waste water is taken off at the bottom of the tank. The separator, as the cylinder is called, though simple in concept, requires very careful design to ensure a satisfactory behaviour.
The air from the separator is lead into the activated sludge tank and the waste water is 'jetted' back into the main tank to cause additional aeration.

6.3.3 The Prototype Unit

Severn Trent Water Authority and British Technology Group financed this development and a patent was taken out to protect the idea, Mortimer & Ellis (1986). A unit was constructed at Loughborough and set up at Derby Water Reclamation Works in early 1985. It has been running continuously, barring a few breakdowns, since then. It is not proposed to discuss the results here only to say that once the unit has settled down (after about half an hour) it runs very smoothly and quietly from the hydraulic point of view. This is mainly due to the constant outlet head imposed on the pump by the separator. For further details see Mortimer & Ellis (1987).

6.4 OTHER USES

We have not exploited all the attributes of the lift coil pump yet. One important aspect of this pump is that it provides an unimpeded passage for the flow of liquid through the pump and so should be able to cope with large solids in the flow.

Only one moving mechanical part of the pump comes in contact with the flow and that is the rotary joint. This should enable it cope with liquids which are abrasive.

As described in the last section, the coil is also an air compressor as well as a liquid pump though their are few situations where this feature can be exploited fully. The coil pump does have the disadvantage that it is bulky for the amount of liquid it can pump. In many situations this makes the coil pump uncompetitive, however, where bulk is unimportant, the pump has significant attractions.
CHAPTER 7

THEORY OF THE SINGLE COIL SUCTION PUMP

7.1 INTRODUCTION

As a capricious experiment in the laboratory, the discharge pipe of the lift coil pump was dropped into a sump containing water and an air inlet was added on the discharge side of the rotary joint. The flowrate through the air inlet was controlled by a needle valve. This arrangement is shown in Figure 7.1.

When the drum was rotated in the opposite direction to that of the lift pump, air was pulled in through the inlet. Water was also sucked up from the sump and discharged into the holding tank - much to our surprise.

When the air flow rate was varied, the water flow rate changed. Generally, the higher the air flow rate, the lower the liquid flow rate. At high liquid flow rates, spilling occurred in the coil and at high air flow rates, bubbling occurred.

A rotameter (air flow measurement device) on the air inlet showed that the air flow rate was steady throughout the drum cycle. On the water suction pipe, a liquid flowmeter indicated that the flow of water was unsteady with one main pulse of flow per drum cycle.

As with the lift pump, it was felt that a theory to predict the behaviour of the pump was necessary. Since the methods used to analyse the lift pump had proved successful at low speeds of rotation, it was decided to adopt a similar form of analysis for the suction pump.
Figure 7.1 The Layout of the Suction pump
This graph shows the curves for four suction lifts at one drum speed.

Figure 7.2 General form of the Pump Characteristic
In this chapter, Section 7.2 describes the form of the pump characteristic. The calculation of the pressure changes across the coil for a single run is described in Section 7.3. Section 7.4 is concerned with the relative plug movements in the coil which have to be defined before the operating point can be established by the multiple runs; these calculation runs are described in Section 7.5.

7.2 THE PUMP'S CHARACTERISTICS

On the lift coil pump, the ratio of the air to liquid volumes per turn is fixed by the inlet conditions, namely, the depth of immersion. With the Suction Coil Pump, the depth of immersion has no effect on its operation at all. There are a range of possible air/liquid flow rate ratios for each suction lift. The ratio is varied by adjusting the air flow rate through the air entry control valve. A calculation involving the determination of the level differences in the loops provided the most useful method of finding the lift pump characteristic and this is so with the suction pump. It is felt worthwhile to show the general form of the suction pump characteristic at this stage, since this is the end result for which we are aiming, this is shown in Figure 7.2.

The family of curves shown in Figure 7.2 strictly applies to one pump speed only, though in practice, the curves can be 'multiplied up' by the ratio of the speeds. The flow rates can, therefore, be expressed as plug lengths per revolution. Each suction head will have a separate curve on the graph, so for the purpose of clarity only four suction lifts are shown on the figure.

As the suction head increases, the curve becomes shorter, centering on the peak value of the liquid plug length. When the curve reaches the shape shown by suction head Hs4 in Figure 7.2, the pump is close to its maximum suction capability.

The theory governing this pump has many features in common with that of the lift pump and a number of equations developed for the lift pump are
Air plug identification tags

OUTLET

Liquid plug identification Tags

Figure 7.3 Cascading Manometer Concept

FIGURE 7.4 Layout of Plugs in Loop n
directly applicable to the suction pump. As with the lift pump, the calculations are divided into routines and procedures.

7.3 CALCULATION ROUTINES FOR THE PRESSURE CHANGES ACROSS A COIL

7.3.1 Introduction

With the suction pump, we will not be concentrating on the pressure changes (using the level differences) across the individual loops. The level differences in the loops were very difficult to measure in the laboratory and so, no comparison can be made with the calculated values. It is proposed to focus the development of the theory directly on characteristic curves, though the level differences have to be calculated to achieve this.

Figure 7.3 shows the suction pump depicted as a cascading manometer. In this diagram no spilling or bubbling is shown, though, either may occur in this type of pump. The sum of the individual liquid level differences must equal the pressure difference across the pump, that is, if we ignore dynamic losses. Figure 7.3 shows all the level differences opposing the pressure change across the pump but, in some cases, it is possible for individual loops to generate a pressure fall across them (inlet to outlet). This unlikely situation can occur with the Double-Coil Suction Pump which is described in Chapter 10.

On Figure 7.3, the plug numbering system starts at the inlet which is the same convention used in the lift pump. For convenience, the air plug number, An for the general case, now follows the liquid plug Wn instead of preceding it as with the lift pump.

The angles $\varnothing(n)$ and $\beta(n)$ apply to the rotation of the spilling and bubbling liquid faces, respectively; see Figure 7.4.

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**Figure 7.5 Calculations Routine for Non-Spilling Region**
Unlike the lift pump, all calculations to determine the liquid level differences on the suction pump start at the inlet and progress to the outlet.

In order to determine the pressure changes across a coil (in a single calculation run), four parameters must be known. These are,
(a) the water plug length Lwi,
(b) the air plug length at atmospheric pressure, Lai,
(c) the rotation of the trailing edge of the first pair of plugs just entering the coil inlet, $\beta(1)$,
(d) the absolute suction head at the pump $H_{a}(1)$. This is equal to $H_{suc}$.

Parameters (a), (b) and (c) have to be estimated, or assumed, for a given situation and this is explained later in Section 7.5.

The Equations governing the level differences across each loop are the same as those given in Section 4.4.9 for the Lift Pump.

Since the basic equations used here are very similar to the ones for the lift pump, except the head changes across a loop are negative, we can proceed directly to the Calculation Routines.

One amendment to the lift pump equations is required to ease the calculations. In Equation 4.19 $L_{a}(n)$ has to be replaced by $L_{a}(n+1)$, the new equation becomes

$$\phi(n+1) = \frac{L_{a}(n+1)+L_{w}(n)}{R} + \phi(n) - 2\pi$$

Equation 7.0

7.3.2 Calculation Routine for the Non-Spilling Region

The non-spilling region occurs adjacent to the inlet. The calculations to determine pressure differences, therefore, start at the inlet with known values for $L_{wi}$, $L_{ai}$ and $\beta(1)$, then they proceed through the loops as shown in Figure
Relative movement of plugs in relation to pipe

Figure 7.6(a) Plug positions at time $t$

Figure 7.6(b) Plug positions at time $t+60/S$ (secs)

For Notation see Figure 4.4

Figure 7.6 Plug positions in a 'Straightened' Coil
7.5. In this case, all the parameters for the first loop are known and so the calculations move from loop to loop by pressure and geometrical considerations. Initially, $L_w(1)=L_wi$ and $L_a(1)=L_ai$ and at each loop number, the conditions are checked to see whether spilling or bubbling is occurring. The checks and the subsequent routines for these two cases are given in the following sections.

7.3.3 Calculation Routine for the Spilling Region

The conditions to cause spilling in this type of pump are very similar to those of the lift pump which are described in Section 4.4.7. For the Suction Pump spilling will occur in Figure 7.4 when,

$$\beta(n) > \pi - \Omega_{sp},$$

$\Omega_{sp}$ is defined in Section 4.4.7.

Considering Figure 7.6, if we assume that liquid plug $W_n$ is in the spilling region, then it will be losing liquid by spilling as it travels from the position where its length is $L_w(n)$ (in Fig 7.6(a)) to a position of length $L_w(n+1)$. It will also be gaining liquid from plug $W_{n-1}$.

Applying the same logic as used with the lift pump to Figure 7.6, then

$$YY-XX=\pi.D$$

since the trailing edge of the plug will remain at the same angle to the vertical.

Now from the same figure,

$$La(n+1)+Lw(n)=\pi.D.$$  \hspace{1cm} \text{Equation 7.1}

If $La(n+1)$ is known, $Lw(n)$ can be found.

The length $Lw(n)$ affects the level difference across $W_n$ which, in turn, determines the pressure in plug $A_{n+1}$, that is $Ha(n+1)$. As $Lw(n)$ changes, $Ha(n+1)$ (governed by $Lw(n)$) affects the length of the air plug $La(n+1)$ which, from Equation 7.1, determines $Lw(n)$. To solve this, an iteration process has to
\textbf{START} \\
\[ n = 0 \]

\( \beta(n) \) must be known from last non-spilling routines or initial loop data \\

\[ n = n + 1 \]

Return to Figure 7.5

\textbf{Is} \( \beta(n) < \pi - \Omega_{sp} \)?

Yes: \\
\[ L_w(n+1) = L_w(1) \]

No: \\
\textbf{If} \( \phi(n) > \pi \) \text{ abandon calculations} \\
\( \phi(n) \) from Equation 4.20 \\
\( \beta(n) = \pi - \Omega_{sp} \)

\[ L_w(n) = L_w(n) + \Delta L_w \]

\[ h(n) \) from Equation 4.17 \\
\[ H(n+1) = H(n) + h(n) \]

\[ L_a(n+1) \) from Equation 4.3 \\
\[ L_w(n') \) from Equation 7.1 \\
\[ \text{Is} \ ABS(L_w(n) - L_w(n')) < \text{Tol}_{sp} \]?

Yes: \\
\[ \phi(n+1) \) from Equation 7.0 \\

No: \\
\textbf{END} \\
\[ \text{Is} \ n < N \]?

\textbf{Figure 7.7 Calculation Routine for Spilling Loops}
be used. This is best explained on the flow diagram on Figure 7.7.

The minimum possible length for the liquid plug is \( L_{wi} \) (equal to \( L_w(1) \)) and this provides the starting value for the calculations. The liquid plug length is increased successively, by \( \Delta L_w \) each time to calculate a new value of \( L_w(n) \) denoted by \( L_w(n)' \) in Figure 7.7. This value is then used to calculate a value of \( L_w(n) \) derived from Equation 7.4. This process continues until the change in \( L_w(n) \) is less than a chosen tolerance. On the flow chart this is called 'Tolsp' and for practical purposes, it was taken as \( 0.001 \text{m} \).

The routine shown in Figure 7.7 starts when the non-spilling routine shows that spillage is occurring. The loop on the boundary between the spilling and non-spilling regions is called the last non-spilling loop and it has to be dealt with in a separate routine.

### 7.3.4 The Calculation Routine for the Last Non-Spilling Loop

The last non-spilling loop will be gaining liquid from the preceding plug but it will not be spilling itself. The minimum length of this plug will be \( L_{wi} \) and its greatest length will be given by Equation 7.1 when it is giving and receiving spillage.

The amount of spillage received will be dependent upon the air pressures in the adjacent air plugs.

If we assume that on Figure 7.6(a), plug \( W_n \) is the last non-spilling loop, \( W_{ns} \), and it is at the position of \( W_n \), then, \( W_{ns+1} \) will be spilling into it. From this figure and Figure 7.4,

\[
L_w(ns)+L_a(ns+1)=\pi \cdot D - ((\pi - \beta(ns)-\Omega_{sp})R)
\]  

Equation 7.2

The trailing edge of the spilling plug \( W_{ns+1} \) will be at an angle of \( \Omega_{sp} \) to the vertical and \( W_{ns} \) will be at an angle of \( \pi - \beta(ns) \). Hence, \( '\pi - \beta(ns) - \Omega_{sp}' \) represents the amount of swing needed before spilling occurs and this should
It is assumed that the Last non-spilling loop No has been identified.

START

\[ Lw(n) = Lw_1 \]

\[ \beta(ns) \text{ found from non-spilling calculations} \]

\[ \Delta Lw \text{ taken as } .001m \]

\[ Lw(ns) = Lw(ns) + \Delta Lw \]

If \( \beta(ns) < 0 \) or \( \Theta(ns) < \pi \) abandon calculations

\[ \Theta(ns) \text{ from Equation 4.20 rearranged} \]

\[ h(ns) \text{ from Equation 4.17} \]

\[ H(ns+1) = H(ns) + h(ns) \]

\[ L_a(ns+1) \text{ from Equation 4.3} \]

\[ Lw(ns)' \text{ from Equation 7.2} \]

Is \( \text{ABS}(Lw(n) - Lw(n)') < \text{Tolns} \)?

\[ \text{Tolns taken as } .001m \]

Yes

END

Figure 7.8 Calculation Routine for the Last Non-Spilling Loop
Figure 7.9 Calculation Routine for Bubbling Loops
happen in the drum cycle of the last non-spilling loop.

$L_w(n)$ is affected by $L_a(n+1)$ and visa versa, hence an iteration process is needed to find a solution again. The method is shown by the flow chart on Figure 7.8, where $L_w(n)$ is successively increased by a small increment $\Delta L_w$ until the equations governing the behaviour of this loop are satisfied. In this case, this is determined by the change in the estimated plug length ($L_w(n)$) and the resultant calculated value ($L_w(n)$) being less than the value 'Tolns'. 'Tolns' is taken as .001m in the calculations carried out in Chapter 8.

### 7.3.5 Calculation Routine for the Bubbling Region

Bubbling in the suction coil is very similar to that described in Section 4.4.7 for the lift pump and, with the suction pump, it will occur when, in Figure 7.4,

$$\phi(n) > \pi - \Omega_{bub}.$$ 

If on Figure 7.6(a) liquid plug $W_n$ is experiencing bubbling, then the leading edges of $W_n$ and $W_{n-1}$ will both have the same relative position, i.e. $\phi(n) = \phi(n-1)$ and

$$L_w(n) + L_a(n) = \pi D$$

Assuming that no spilling has occurred, then $L_w(n) = L_{wi}$ and so

$$L_a(n) = \pi D - L_{wi}.$$ 

As with the lift pump, the head difference across each bubbling loop is the same and so the calculations shown in Figure 7.9 for $\phi(n)$ and $h(n)$ need only to be carried out once. The calculation loop then determines the pressure head in each successive air plug for the number of bubbling loops, $N_b$.

As the suction pressure increases in a pump, bubbling starts at the outlet and progresses through towards the inlet. If it reaches the first coil, the pump is close to the point when it will cease to function.
7.3.6 The Calculation Routine for the Last Non-Bubbling Loop

If a bubbling region occurs near the outlet of a coil, then an air plug on the boundary between the bubbling and non-spilling regions will be receiving air but it will not be bubbling itself. This is the last non-bubbling loop and it is given the loop number 'nb'.

If, on Figure 7.6, An is in the last non-bubbling loop, Anb, then it will be receiving air from Anb+1 but not losing air itself, then,

\[ Lw(nb) + La(nb) = \pi \cdot D \cdot (\Theta(nb-1) - \Omega_{bub}) \cdot R \]

\[ Lw(nb) = Lwi. \] See Equation 7.2 where the reasoning is very similar. If the calculations start at the inlet and move towards the outlet, then \( \Theta(nb-1) \) will be known, hence, \( La(nb) \) can be found.

7.3.7 Problems with Pressure Difference Calculations

One difficulty that can cause problems is when the first liquid plug length is just less than half the circumference of the coil. Spilling and bubbling can start to occur at virtually the same time. If spilling starts first,

\[ \text{ABS}(\Theta(n) - \Omega_{sp}) > \text{ABS}(\Theta(n) - \Omega_{bub}). \]

\( \text{ABS}(...) \) represents the absolute value. Remember also that \( \Theta(n) \) and \( \beta(n) \) can be negative on the first estimation in these calculation routines.

With small diameter pipes, the influence of the viscosity on the walls will trigger off the spilling mechanism and, conversely, in large pipes, bubbling is likely to start first.

7.4 RELATIVE MOVEMENT BETWEEN INLET AND OUTLET PLUGS

7.4.1 Introduction

In order to find the pump characteristic for a particular pump, the pressure
differences across the coil must be found. Section 7.3 shows how these
differences, for the various hydraulic regimes, can be determined if \( L_{wi}, L_{ai} \) and \( B(l) \) are known.

It is not possible to derive values for these parameters by a direct analytical
means, they have to be found by using an iteration process which adjusts the
pressure difference across the coil until the known applied pressure is
achieved. To help in this optimization, it is essential to derive some
theoretical guide lines to ensure the process converges on a solution.

It is worth pointing out at this stage that in the following sections \( L_{ai} \) and \( L_{wi} \)
will be used to denote the length of the incoming plugs. These lengths will be
same as \( L_{w(1)} \) and \( L_{a(1)} \) in all cases except where spilling or bubbling affects
the plug as it enters the coil. In this situation, the pump is in an unstable
condition.

At the outlet end of the coil, if the loops adjacent to outlet are spilling or
bubbling then, \( L_{w(N)} \) or \( L_{a(N)} \) will not be equal to the effective plug lengths
leaving the coil. These are denoted by \( L_{wo} \) or \( L_{ao} \) on Figure 7.10. This
difference is because these plugs will change length as they leave the coil. \( L_{ao} \)
and \( L_{wo} \) therefore, represent the air and liquid plug lengths that actually
discharge from the coil.

7.4.2 Relative Movements of the Plugs

On the suction pump used in the laboratory, the air flow rate was governed by
a needle valve, hence the air flow rate could be set, but the liquid flow rate was
established by the pump.

The relationship between the liquid and air flow rates is not a simple one and
it has to be derived by an iteration process, as already mentioned. In order to
Relative movement of plugs to pipe

OUTLET WN AN

Relative movement of outlet plugs
if outlet ‘swing-in’ occurs

Lw(N) La(N)

Relative movement of inlet plugs
if inlet ‘swing-in’ occurs

Lw(1) La(1)

W1 A1 INLET

Figure 7.10 Movement Inlet and Outlet Plugs
carry out this process, the end conditions in the coil have to be studied.

For convenience, both the flow rates of the air and the liquid will be expressed as plug lengths $L_{wi}$ and $L_{ai}$. The relationship between flow rate and plug length is given by

$$Q_a = L_{ai} \pi r^2 \cdot \frac{S}{60}$$

A similar equation is valid for the liquid plugs.

Consider Figure 7.10 which is a 'straightened' coil of a suction pump. In one revolution, the effective volume of the fluid plugs leaving the coil must equal the effective volume entering.

For the total volume out per revolution, $V_{out} = L_{wo} + L_{ao}$.

$L_{ao}$ is the effective volume (expressed as a length) of air to leave the coil taking any bubbling effects into account.

$L_{wo}$ is the effective length of liquid leaving the coil taking any spilling effects into account where volumes are replaced by lengths.

For the fluid volume in, $V_{in} = L_{wi} + L_{ai}$.

Now $L_{wi} = L_{wo}$ since, in one revolution, the amount of liquid entering the coil must, by continuity, equal the amount leaving. $L_{ao}$ must be less than $L_{ai}$ due to air compression effects, since $V_{out} - V_{in}$ must equal then there must be, either, a relative movement of the pair of outlet plugs back towards the inlet or, a relative movement of the pair of inlet plugs towards the outlet. In the form of an equation,

$$\text{Relative movement} = L_{ai} - L_{ao} \quad \text{Equation 7.3}$$

This relative movement is caused by the air plugs compressing as they pass through the coil. In the lift pump, the air compression effects cause the plugs to swing back towards the inlet, here it is possible for the plugs to swing either way.
It is proposed to use the term 'swing-in' to describe the additional relative movement of the plugs into the pipe which is due, solely, to air compression. There is also the movement of the fluid plugs in relation to the helical pipe to be superimposed on this. See Figure 7.10.

The outlet plugs are free to 'swing-in' towards the inlet because the outlet provides no restraint. The inlet plugs are hydraulically connected to the air and liquid in the inlet tube, so a 'swing-in' by the inlet plugs causes additional liquid, or air, to be drawn into the coil.

The reason why an inlet swing should occur is explained in more fully Section 7.4.5. It would seem likely that the outlet 'swing-in' would occur in preference to an inlet 'swing-in' as this offers the least resistance but this is not always the case. In the analyses to be used later, a starting value of Lai is assumed and Lwi is calculated using this value. In many cases this liquid plug length is greater than that required just to fill one loop of the coil. This additional volume can be considered as an increase in the length of the liquid plug. In fact, the additional volume is probably shared between the two liquid phases.

Before proceeding to look at the 'swing-in' in more detail, it is worth considering the head difference that is generated across the first loop.

7.4.3 The Head Difference across the First Loop

As liquid plug W1 enters the coil at the inlet, it can sustain a pressure difference across it, if the trailing edge of the plug is higher than the leading edge. In other words, the plug has an inherent rotation as it enters the helical pipe and so it can sustain a level difference across it.

The reason why this initial rotation occurs is shown in Figure 7.11. In this
Figure 7.11 (a) Liquid plug entering arm

Figure 7.11 (b) Liquid in arm

Figure 7.11 (c) Liquid plug entering loop

Figure 7.11 (d) Air plug entering loop

Figure 7.11 A Pair of Fluid Plugs emerging from the Inlet
figure, four important positions are shown for the inlet arm. As the arm rotates around the inlet tube, the leading and trailing edges of the inlet plugs are controlled by this arm coil. The inlet liquid plug, effectively, is rotated in an anticlockwise direction in relation to the neutral plug position (with no pressure head gain across the plug). It is difficult to accurately estimate this rotation since the position of the leading and trailing edges of the plugs are controlled by the inlet arm emptying and filling with liquid; see Figure 7.11(d). It is not known accurately when the liquid will pour back down the arm into the tube and the emerging air plug will start to enter the coil. A similar problem occurs on Figure 7.11(b) when the liquid replaces the air in the arm. To make matters more complicated, the liquid level in the inlet tube is rising and falling throughout the drum cycle; this will be considered in Section 7.4.6.

Returning to the inlet 'swing-in', three cases are considered in the next three sections, no 'swing-in', full swing-in', and a partial 'swing-in'.

7.4.4 No Inlet 'Swing-in'

Since no inlet 'swing-in' occurs, the lengths of the air and liquid plugs just fill the volume occupied by one loop of the pipe. The equation relating the lengths is,

\[ L_{wi} + L_{ai} = \pi D \]  

Equation 7.3

The compression of the air plugs will produce a 'swing-in' at the outlet but this will not affect the inlet plug lengths.

7.4.5 Full Inlet 'Swing-in', No Outlet 'Swing-in'

Consider the case where a full 'swing-in' is occurring at the inlet with no 'swing-in' at the outlet. This means that the all the air compression movement is accommodated solely by the 'swing' of the inlet plugs.
Figure 7.12 Inlet plugs with no inlet swing

Figure 7.13 Inlet plug with inlet swing
As the liquid plug emerges from the arm it will have a relative velocity, \( U_m \), towards the outlet, the mean velocity \( U_m \) is,

\[
U_m = \frac{(L_{ai}-L_{ao})}{T_d}
\]

Equation 7.5

\( T_d \) is the time for one drum cycle.

\( L_{ai}-L_{ao} \) is the total swing of the liquid plug \( W_1 \) towards the outlet in one revolution. See Section 7.4.2.

As the arm moves from its position in Figure 7.11(c) to its position in Figure 7.11(d), the leading edge of the liquid plug will move forward a relative distance, \( L_{rel} \). See Figure 7.13. This forward movement is caused by the 'swing-in' and it allows the additional liquid to be drawn in to the coil.

Assuming the relative velocity due to this movement to be constant throughout the drum cycle, then,

\[
L_{rel} = \frac{U_m(L_x+L_{wi})}{\pi D} T_d
\]

The inlet swing causing \( L_{rel} \) is assumed to effect the incoming liquid plug immediately the liquid plug enters the inlet arm.

Substituting Equation 7.5 into the above yields,

\[
L_{rel} = \frac{(L_{ai}-L_{ao})(L_x+L_{wi})}{\pi D}
\]

Equation 7.6

If no 'swing-in' occurs then the length \( L_{wi} \) is given by \( L_{wi}=\pi D-L_{ai} \), so the 'swing-in' provides an additional volume which is, in effect, \( L_{rel} \), therefore,

\[
L_{wi}=\pi D-L_{ai}+L_{rel}
\]

Equation 7.7.

Equations 7.5 to 7.17 use \( L_{wi} \), if there is a large head difference across the first loop, \( L_{w(1)} \) must be used. \( L_{w(1)} \) is the calculated value of the plug length allowing for spillage effects.
7.4.5 Partial Inlet 'Swing-in'

It is likely that a partial inlet 'swing-in' will occur in a pump, particularly when the suction lift is significant. Looking at the upper and lower limits of the 'swing-in', we can summarise the equations developed in the two previous sections,

<table>
<thead>
<tr>
<th>Swing-in at inlet</th>
<th>Liquid plug length</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full</td>
<td>( Lw_f = 2\pi R - L_{\text{ai}} + L_{\text{rel}} )</td>
</tr>
<tr>
<td>None</td>
<td>( Lw_n = 2\pi R - L_{\text{ai}} )</td>
</tr>
</tbody>
</table>

The superscripts "f" and "n" are used to differentiate between the two limits 'full' and 'non' for 'swing-in'.

With a partial 'swing-in', the first liquid plug length will lie between the limits set out above; these limits are used in the iteration process and are discussed further in Section 7.5.2. In order to simplify the process of iteration, a new variable, \( \%_{sw} \) is introduced. This is a percentage of the full inlet 'swing-in', \( L_{\text{rel}} \), that can occur. \( L_{\text{rel}} \) is defined in Equation 7.6; \( \%_{sw} \) can be negative. The plug lengths are related to \( \%_{sw} \) and \( L_{\text{rel}} \) by the following

\[
L_{wi} = 2\pi R - L_{\text{ai}} + L_{\text{rel}} \left[ \frac{\%_{sw}}{100} \right]
\]

Equation 7.8

7.4.7 Why the Inlet Swing Occurs?

The liquid plug emerges from the inlet in such a relative position that it can sustain a substantial head difference across it. Since the head differences in all the other plugs are governed by the rotation of this first plug, all the plugs have an intrinsic ability to resist a significant head difference across the coil, and in some cases, this head difference is not required. At high suction heads,
Figure 7.14 Variations in Flow and Levels in Inlet Tube
there is a need for these head differences to counteract the suction head at the inlet to the pump.

At low suction heads, however, the potential head generated by this first 'swing-in' (and consequently, the others) is greater than that required for the pressure difference across the pump. The plugs must, therefore, swing towards the outlet reducing the sum of the head differences in the coil down to the imposed head difference across the pump.

**7.4.8 Inlet 'Swing-in' with Spilling or Bubbling**

Spilling will reduce both a positive and negative 'swing-in' unless it reaches the first loop when it will effectively increase the 'swing-in'. Bubbling pulls air through the outlet and so increases the 'swing-in'

Attempts have been made to find a suitable direct method of predicting this partial 'swing-in' due to bubbling or spilling, but all have proved unsatisfactory. This is not the obstacle it may seem, as its effects can be determined by the iteration process explained in Section 7.5.

**7.4.9 Calculation of $B(1)$**

The determination of $B(1)$ has to be based on considerations of the conditions in the inlet tube.

The approximate changes in the flow rates into, and out of, the inlet tube are shown in Figure 7.14, throughout one drum cycle. The changes in liquid level are also shown on this figure.

In the laboratory, the rotameter (air flow meter) on the air inlet indicated that the air inflow rate was almost constant throughout the drum cycle and this is
shown on Figure 7.14(a).

The water flow meter on the suction pipe suggested that one main pulse of flow occurred during one drum cycle. This is shown on Figure 7.14(b). It is not intended that the shape of the curve represents accurately what is happening in practice, since this flow variation is unknown due to the great difficulty in measuring unsteady flow.

Figures 7.14(c) and (d) show a constant outflow rate for both the air and liquid. It is very difficult to prove this assumption by measurement but, since the inlet arm is moving around the inlet tube at a constant rate and, the liquid plugs in the coil show no irregular movements during the drum cycle, this assumption appears reasonable.

The liquid level variations in the inlet tube are shown in Figure 7.14(e). The inflow and outflow considerations dictate the storage changes in the inlet tube, hence, the liquid level is at its highest when the liquid plug is about to enter the inlet arm and at its lowest when the air plug is about to enter the inlet arm. It is the fall in the liquid level that causes the additional suction head on the liquid column in the suction pipe and this in turn causes the surge in the flow. It is assumed that the air pressure in the inlet tube remains approximately constant throughout the drum cycle and this must be true if the air inflow rate remains constant.

Figure 7.15 shows the inlet positions at the maximum and minimum levels. If \( Q_{ain} \) is the air flow into the tube and \( Q_{aout} \) is the air flow, then the volume between the maximum and minimum levels is given by

\[
V_{in} = (Q_{aout} - Q_{ain}) \times T_{ain}
\]

Equation 7.9

\( T_{ain} \) is the time taken for an air plug to enter the inlet arm this is governed by the equation,
Figure 7.15 Level Changes in Inlet Tube
This again assumes that the air flow and liquid flows out of the tube are constant. If the air plug length is known then the air flow rates can be found from the equations,

\[
Q_{\text{ain}} = \frac{La(1)a}{Td}, \quad \text{Equation 7.11.}
\]

\[
Q_{\text{aout}} = \frac{La(1)a}{T_{\text{ain}}}, \quad \text{Equation 7.12.}
\]

\(a\) is the cross section of the helical pipe.

From geometrical considerations in Figure 7.15 the volume between the maximum and minimum levels in the tube is given by the equation

\[
V_{\text{it}} = \left[ \cos \omega \sin \omega \frac{d_i^2}{4} + 2 \left( \frac{\omega}{2\pi} \frac{x d_i^2}{4} \right) \right] + \left[ \cos \mu \sin \mu \frac{d_i^2}{4} + 2 \left( \frac{\mu}{2\pi} \frac{\pi d_i^2}{4} \right) \right].
\]

\(L_{\text{it}}\) is the effective length of the inlet tube. \(d_i\) is the diameter of the inlet tube.

The two additional parameters have been included in this equation, \(\omega\) and \(\mu\) (see Fig. 7.15) and they are related to each other by the following,

\[
\mu = \omega \pi + 2(L_{\text{wi}} + L_{\text{x}} - L_{\text{rel. sw}}/100)/d_i. \quad \text{Equation 7.14.}
\]

This equation is for all conditions and the term shown in the brackets in this equation is derived from Equation 7.8.

The method for finding \(B(1)\) is shown by the flow chart on Figure 7.16. An iteration process is again needed where \(\omega\) is adjusted until the volume change in the inlet tube calculated from flow rates (\(V_{\text{it'}}\)) and geometrical considerations (\(V_{\text{it}}\)) are in agreement within the required tolerance (normally .005 radians). The iteration process is relatively crude as \(\omega\) is set at high value, \(\pi/2\) and then progressively reduced. \(\omega\) can be allowed to negative though this situation has not occurred in laboratory tests.
Lwl, %sw, Lrel, Lx are known

\( \omega = \pi/2 \)

\( \mu \) from Equation 7.14

Vit from Equation 7.13

Qain from Equation 7.11

Qaout from Equation 7.12

Vit' from Equation 7.9

Is \( \text{ABS(Vit-Vit')} < \text{Tol} \)

\( \beta(1) = \mu + \pi/2 \)

If \( \beta(1) > \pi \), \( \beta(1) = \pi \)

END

Figure 7.16 Calculation Routine for Inlet Angles
In practice $\beta(1)$ must be less than $90^\circ$ since the air plug can only start to form at the coil end of the inlet arm when the water in the arm can drain out. It is assumed that this will occur when the inlet arm is approximately horizontal; in Figure 7.16, if $\beta(1)$ is less than $\pi/2$ it is made equal to $\pi/2$.

The method of calculating $\beta(1)$ is only approximate, but calculations suggest that the determination of the dosing flow rates are not highly sensitive to errors in $\beta(1)$.

7.5 ESTABLISHING THE PUMP OPERATING POINT

7.5.1 Introduction

The operating point is a position on the pump characteristic when the pump is operating under steady state conditions, in other words, it is the flow rate associated with a particular suction head on a characteristic curve.

All the required equations and the boundary conditions for calculating the level differences across a coil are now defined, the difficulty then comes in solving these equations. The problem on the suction coil pump is not as easily solved as the lift coil pump since the air flow rate and liquid flowrate are both variables in the iteration process.

One possible method is to used a formalized optimization technique such as the modified version of the Newton-Raphson Method. This method was rejected at an early stage in the research in favour of a method of multiple computer runs which varies the parameters methodically (based on hydraulic behaviour) until the boundary conditions were satisfied.

This decision was made for two main reasons. Firstly, an important aim at the early stage of theory development was to found out how each variable
affects the pump's behaviour. This need was best satisfied by the method of multiple runs. The second reason is that I have had experience of applying Newton-Raphson techniques to hydraulic problems in the past and, in some cases, the computer effort required for the Newton-Raphson technique exceeded that of the multiple run method. In the cases where the method was not converging to a solution it could also be very difficult to find out what hydraulic inconsistencies were causing the problem.

The methods of calculation shown on the flow diagrams on Figures 7.5, 7.7, 7.8 and 7.9 are orientated to the multiple run method and not the Newton-Raphson Method. The decision affecting the method to be used was made at the start of the theory development and so the techniques described in the early part of the chapter reflect this fact.

If there was to be a demand for a frequently used computer program of a suction coil pump analysis, it could be well worth while developing a more efficient computer program than the one described here. All the equations and the boundary conditions necessary are given in the early part of the chapter, though not separately listed, are specified in different sections of the text.

7.5.2 Limits on the Liquid/air Plug Lengths

In Section 7.2 it was mentioned that the most important characteristic curve is a plot of the liquid plug length against the length of the air plug, for different suction lifts.

Because the air flow rate is controllable it is assumed that this is known. The liquid flow rate, or in this case, the liquid plug length will lie between an upper and lower boundary. The upper boundary is determined by the condition where the inlet plugs undergo a full 'swing-in' due to air compression, that is \( \%sw=100 \). This situation causes the maximum amount
of fluid volume to be taken into the pump in stable operation. The lower boundary condition is produced when no 'swing-in' occurs at the inlet, where $sw\% = 0$. If a partial 'swing-in' occurs, as often is the case, the solution lies between the two boundaries. Under some conditions close to failure, the swing may become negative, here the characteristic will lie below the lower boundary.

It would be very desirable to predict the swing, $%sw$, directly. This is not possible, since the 'swing-in' determines the head difference across the coil (ignoring the fixed outlet pressure for the moment) and the head difference across the coil affects the swing. It has, unfortunately, to be determined by an iterative process.

The single characteristic curve (for one suction head) is determined by finding the known liquid plug length which will satisfy the boundary, and internal hydraulic constraints, from each of a series of air plug lengths. Experience has shown that with an air plug length of less than 20% of effective pipe drum circumference, gross bubbling occurs at most suction heads making the pump virtually inoperable. With the plug length of 80%, gross spilling occurs producing a similar effect. These two values provide a good guide for the range of air plug lengths to be used in the calculations.

7.5.3 General Procedure

The general procedure is shown in Figure 7.14. The flow chart has two main loops, the one to the right varies the length of the air plug between 80% and 20% of the effective drum circumference in steps of 5%. This will produce a number of equally spaced points (on the x-axis) on the characteristic curve. It is likely that some of the values of $L_{ai}$ will not have a stable operating point and the calculations have to be abandoned. The criteria for abandonment are not shown on Figure 7.14 because of a lack of space. Briefly they are,
Determine Hout for %sw=0 using Method below (Box A to Box B).
If Hout<Hat, High%sw=0, Low%sw=-100. If not High%sw=+100, Low%sw=0

Hout=H(n+1)  
\[ x = 80\% \quad m = 1 \]  
\[ x = x - 5\% \]  
\[ m = m + 1 \]

Box A
\[ \%sw = (\text{High}\%sw - \text{Low}\%sw)/2 \]
Lwi from Equation 7.8
\( \beta(1) \) from Figure 7.16

Non-spilling routine Figure 7.5
For exit path see Fig 7.5

Last non-bubbling routine, Section 7.36.
Bubbling routine Fig 7.9.

Box B
Is Hout-Hat<Tolo?

A
Yes

No

Is x=20?

No

If Hout>Hat High%sw=%sw
If Hout<Hat Low%sw=%sw

Is m<20 ?

Yes

No

BND

Figure 7.17 Overall Calculation Procedure
\[ \pi - \Omega_{\text{bub}} \geq \emptyset(n) > 0, \]
\[ \pi - \Omega_{\text{sp}} > \emptyset(n) > 0. \]

Each routine also has abandonment conditions and if this occurs the calculations should move to point A on the Figure 7.16. This causes the calculations to move on to the next point on the curve (the next value of x).

The loop to the left on Figure 7.16 is concerned with finding a solution that provides an outlet pressure to the coil, $H_{\text{out}}$, which is equal to the atmospheric pressure, within the required accuracy, $T_{\text{olo}}$, on the flow chart. With some calculation runs this cannot be physically achieved, hence the need to limit the number of passes around the left hand loop ($m \leq 20$).

The solution is found by using the Bi-section Method which is explained in Appendix 2. Initially, the lower and upper limits to the value of the percentage 'swing-in', $\text{Low} \%_{\text{sw}}$ and $\text{High} \%_{\text{sw}}$ have to be set. Practice has shown that the solution always lies within $\%_{\text{sw}} = +100\%$ and $\%_{\text{sw}} = -100\%$, in fact, most solutions lie within $\%_{\text{sw}} \pm 50\%$. The Bi-section Method is restricted to either positive values of $\%_{\text{sw}}$ only or, negative values only. When the lower limit was of the opposite sign to the upper limit, instabilities occurred in the convergence.

The method of setting the limits $\%_{\text{sw}}$ is shown in the top box of the flow chart where an initial calculation run is carried with $\%_{\text{sw}} = 0$. If $H_{\text{out}} > H_{\text{at}}$ then positive values of $\%_{\text{sw}}$ are used otherwise it is negative values that are used. The number of iterations are limited to 20, as experience as shown that the method will not normally find a solution after 20 iterations.

Problems at the failure points will be explained in greater detail in the next Chapter.
7.6 COMMENTS

The theory covering the single coil suction pump is similar in many ways to the lift pump theory. It is possible to draw the two theories closer by describing the theory for the lift pump more rigorously in terms of inlet and outlet swings, but this approach has no advantages.

With the suction pump, the air flow rate becomes an additional variable in the calculations, but the number of spilling/bubbling plugs can be determined directly so this not a variable in the iteration process as it is with the lift pump.

The raison d'etre of any theory rests with its ability to predict the observed behaviour; this is the subject of the next chapter.
CHAPTER 8

TESTING THE THEORY OF THE SINGLE-COIL SUCTION PUMP

8.1 INTRODUCTION

A series of laboratory tests was carried out to provide results which could be checked against the theoretical predictions using the method explained in Chapter 7. Sections 8.2 and 8.3 are concerned with the practical tests themselves and Sections 8.4 to 8.8 deal with the comparison of measured and calculated results. These latter sections also describe the mechanisms of pump failure which came to light from the laboratory tests.

8.2 LABORATORY APPARATUS

8.2.1 The Inlet

The laboratory rig used for the lift pump investigation was adapted for these tests. The pipe on the stationary side of the rotary joint was replaced by a 200mm diameter inlet tube. A window was built into the end of the tube so that the level changes could be observed. The air inlet was also added and this was situated on top of the tube and connected to the rotameter by a flexible pipe. The rotameter incorporated a needle valve to adjust the flow rate and the meter was capable of measuring the flow rate to an accuracy of 3%. After continual use for three months, the needle valve was starting to become blocked by small particles sucked in by the air, so a simple air filter was added to protect the valve.

The suction pipe dropped vertically into the sump which was directly
Figure 8.1 General Layout of Laboratory Apparatus
underneath the tank holding the coil. The sump was 3 metres square in plan and 6 metres deep. The water level in this sump could be adjusted to a pre-set level by means of electric pumps. The general layout is shown on Figure 8.1.

8.2.2. The Coil

For these tests, the first helical pipe to be used was on the drum was transparent and flexible, but it was soon found that this badly distorted by the suction pressure. Over a suction lift of about 3m, the suction pipe and the coil were flattened. To overcome this problem two solutions were adopted. The coil nearest to the inlet was made from a clear rigid plastic pipe, bent to fit the drum. The next four coils were made from steel galvanized pipe, again bent to fit the drum. The rest of the loops were made up from a flexible pipe where the suction pressures were less.

This cumbersome arrangement of pipes was used because we initially wanted to see what was happening to the water levels in the coils. Bending the clear plastic tube accurately proved to be so difficult that only one coil received this treatment. It proved a little easier to bend the steel pipe to shape but the operation still consumed over a man day per loop.

This arrangement of loops meant that we were unable to measure, or observe, the water level differences in most of the loops, which was disappointing.

In the second set of experiments, the pipe diameter was varied. In this case, a steel reinforced flexible hose was used, which was not transparent, but its deformation with suction pressure was small.
8.2.3 The Outlet

The suction coil discharged into the tank where the water level was kept at a constant level by means of an overflow weir. In all the experiments, the water level in the tank was kept below the bottom of the coil to prevent the outside water level interfering with the discharge from the coil.

The water flowing out of the holding tank dropped down a vertical pipe back into the sump forming a closed system.

8.2.4 Flow Measurement

The flow measurement proved difficult since a liquid rotameter placed in the suction pipe showed that the flow rate fluctuated significantly during the drum cycle. The outflow from the coil occurred over, approximately, a half a drum cycle as the outlet moved around the circumference of the drum. Measurement here was not possible.

The only practical method of flow measurement was to use a measuring cylinder and stop watch under the pipe on the outlet weir to the tank. The main problem with this method was that the storage available in the system meant that the flow did not settle down to a steady rate for at least an hour. In fact, the rig was normally left two hours before any reading was taken.

Estimates of the accuracy of flow measurements suggest an error of ±3%. When this flow rate was converted into a water plug length, the error increased to ±6% because of errors in measurement of the effective coil diameter and the manufacturing tolerances in the loop diameter.
8.3 RANGE OF TESTS

It was not possible to carry out the wide range of tests that had been used on the lift pump. The main reason for this was that each experiment took between 1 and 5 hours to complete. Compared with the Lift Pump, the air flow rate also introduced another variable.

The parameters used in these experiments were limited by the following factors.

(a) The drum size was fixed by the dimensions of the existing rig.
(b) It was felt that the practical operating range for a single coil suction pump would probably lie within speeds of 2 and 6 rpm.
(c) The coil pipe diameter was chosen as the largest practical pipe that could be fitted around the drum.
(d) The maximum number of loops was limited by the length the drum, particularly with the larger diameter pipes.

In the experiments, three physical pump parameters were varied; the helical pipe size, the number of loops, the speed of rotation. For each set of pump parameters, the suction head and the air flow rate were varied. The drum diameter was held constant.

The number of variations for each parameter was as follows;

Helical pipe size - 38mm and 50,
Number of loops - 6 and 11,
Speed of rotation - 2 and 6 rpm,
Suction Head - varied between 1.7 and 5m.

Not all the possible combinations of the above parameters were tested. In all 77 separate experiments were undertaken.

For each set of pump parameters, the suction head was held constant and
Figure 8.2(a) Theoretical Characteristic for \( H_{\text{Suc}} = 3.0 \text{m} \)

![Graph showing theoretical characteristic for \( H_{\text{Suc}} = 3.0 \text{m} \)]

Figure 8.2 (b) Variation of Spilling, Bubbling Loops with \( L_{\text{ai}} \)

![Graph showing variation of spilling and bubbling loops with \( L_{\text{ai}} \)]

*The air plugs lengths are measured at 20\(^\circ\)C and atmospheric pressure.*
the air flow rate was varied, normally, about eight settings spaced over the full range of water flow rates. The spacing could only be judged by practical experience.

8.4 THEORETICAL AND MEASURED CHARACTERISTICS

8.4.1 Introduction

Before carrying out a comparison between the theoretical and measured values it is worth looking at the general form of the characteristic curve.

A typical example of a theoretical characteristic curve for the pump is shown in Figure 8.2(a) where the suction lift is 3.0 m. This curve has been produced from data produced by a computer program based on the flow charts given in Chapter 7. The program was written in Basic and run on a Apple Macintosh computer. This program required a larger amount of computer processing time and, ideally, needs to be rewritten in Fortran to speed up the calculations.

The water and air flow rates are expressed as plug lengths, where,

\[
\text{Plug length} = \frac{\text{flow rate}}{\text{pipe area} \times \text{speed of rotation}}.
\]

All the air plug lengths referred to in this chapter are at ambient temperature and at atmospheric pressure.

The curve on Figure 8.2(a), which in this case is nearly a straight line, can be divided into three sections. The upper section of the curve is governed by the behaviour of the spilling plugs, in the middle section, little or no spilling occurs and in the bottom section, bubbling occurs. Figure 8.2(b) shows these conditions in the loops for a suction head of 3.0 metres. The bars below the zero line represents the number of bubbling loops, above the line, spilling loops. When the air plug length is reduced to 0.6 m, all 10 coils
Figure 8.4 Example of Water level Differences Hsuc=3.0m

D=.93m
d=50mm
N=10
S=2rpm
Lw(1)=1.61m
La(1)=1.0m

Figure 8.4 Examples of Inlet and Outlet Angles

data as for Figure 8.3

Inlet Angle = \( \beta(n) \)
Outlet Angle = \( \varnothing(n) \)
are spilling and this corresponds to the values to the left of the peak water plug length on Figure 8.2(a). At the other extreme, as the air plug length increases from 1.4 to 1.5m the number of bubbling coils increases dramatically causing the pump to fail. Hence, on Figure 8.2(a), the theoretical points for the inlet air plug lengths of 0.7 and 1.5m are not plotted because a hydraulic solution could not be found in the calculations with all the loops spilling or bubbling.

At the peak liquid flow given in Figure 8.2(a), the 6 loops nearest to the outlet are spilling. For values of \( L_a(1) \) in the range 1.0 to 1.3 only on or two coils are spilling and, above \( L_a(1) = 1.3 \), the number of bubbling coils increases rapidly.

At a suction head of 3m, the peak flow on Figure 8.2(a) is reached at \( L_{ai} = 0.8m, L_{wi} = 1.84 \), this represents a real flow rate of 4.2 litres per minute at 2 rpm.

In the laboratory, the experimental data shows a falling left hand limb to the curve as shown in Figure 8.2(a) where both the theoretical and measured results are shown. The problems with predicting this left hand falling limb are discussed in Section 8.6.

As a typical case of a mid-range value, Figure 8.3 shows the theoretical water level differences across the loops for \( L_a(1) = 1.0m \) in the example shown in Figure 8.2. Loop No 1 is the inlet and loop No 9 is the last non-spilling loop. Loop No 10 is spilling.

The theoretical inlet angles, \( \beta(n) \), and the theoretical outlet angles, \( \varnothing(n) \), are shown in Figure 8.4. The two angles are obviously dependent on each other and this is shown by the constant difference between them in the non-spilling region. It is only when spilling occurs that this difference changes because of a change in water plug length.
Figure 8.5 Characteristic Curves for $H_{suc}=3.0\text{m}$

Figure 8.6 Characteristic Curve for $H_{suc}=1.765\text{m}$

*The air plugs lengths are measured at 20°C and atmospheric pressure.
The spilling and bubbling failure points on the pump are difficult to establish in the laboratory since a failure with this pump is not as vigorous or as well defined as the lift pump. With a bubbling failure, when the air flow rate is set near to the point of failure, the pump continue to operate but the bubbling will progress back towards the inlet loop by loop. When it reaches the last loop, the flow rate will slowly decline to zero. The plugs will still appear to be stable at zero flow, but all will be experiencing bubbling. The manometric structure does not completely collapse at a bubbling failure since the pump can still maintain a suction pressure, though it does not generate a liquid or air flow.

With a spilling failure, the number of spilling loops gradually increases until spillage is occurring into the first loop, however, the pump will still operate until the spillage becomes excessive.

With the lift pump it was not possible to define theoretically when failure occurred, but it could be seen clearly in the laboratory. With the single coil suction pump the failure point can be precisely stated theoretically, but it is difficult to observe in the laboratory.

8.4.2 Comparison of Theoretical and Measured Characteristic Curves

An example of the theoretical prediction using the method described in Section 7.5.3 is shown in Figure 8.5. Not all the calculated points have been shown for reasons of clarity. These curves are for the same pump and operating conditions as the one shown in the Figure 8.2(a).

Two curves agree very well over the straight section of the curve and the theoretical curve predicts the bubbling failure point well and also the peak liquid plug length. Further measured values could have been included to the extreme of the left hand limb of the curve to extend the limb, but in this region the behaviour is unsteady. In effect, this characteristic covers the
The air plug lengths are measured at 20°C and atmospheric pressure.
The air plugs lengths are measured at 20°C and atmospheric pressure.
Figure 8.9 Characteristic Curve for Hsuc=3.35m

Figure 8.10 Characteristic Curve for Hsuc=4.22m

*The air plugs lengths are measured at 20°C and atmospheric pressure.
stable area of the pump's behaviour. See Section 8.7 for further details.

There are no theoretical points on the curve to the left of the peak, the reason for this is explained in Section 8.6.

Figures 8.6 to 8.10 show a family of theoretical and measured curves for a pump with a 38mm diameter coil forming 6 loops. The suction heads vary from 1.765m which was the smallest that could be achieved on the rig to a maximum value of 4.22 metres which is very close to the pump's limit.

On Figure 8.6, the theoretical curve predicts the peak and the bubbling failure points well but it tends to underestimate the liquid plug length in the mid-range, by a maximum of 8%. This also applies to a suction head 2.27m on Figure 8.7.

The main section of the measured and calculated curves on both Figures 8.7, 8.8 and 8.9 are not straight line, agree within the limits of experimental error.

At a suction height of 4.22 metres on Figure 8.10, the pump is close to its limit and the characteristic curve is much shorter than at the lower suction heads. In fact, it appears that, the maximum suction head for the pump is reached when the spilling failure point closes in on the bubbling failure point, though it is difficult to show this in practice.

At this suction head, the theoretical curve has a steeper gradient than the measured, though once again the maximum difference in liquid length is only about 8%. As before the theoretical curve predicts the measured peak liquid length and the bubbling failure point well.

An attempt was made to establish a theoretical curve for a suction head of
Figure 8.11 Combined Measured Characteristics

Figure 8.12 Variation of Maximum Water Plug Length with $H_{suc}$

*The air plugs lengths are measured at 20°c and atmospheric pressure.*
D = 0.93 m
d = 50 mm
N = 10
S = 2 rpm

Measured
Theoretical

Theoretical failure

Figure 8.13 Characteristic for Hsuc = 1.83 m

Figure 8.14 Characteristic Curve for Hsuc = 5.0 m

*The air plugs lengths are measured at 20°C and atmospheric pressure.
4.5m but no point could be established that satisfied all the boundary conditions.

Figure 8.11 shows the family of measured characteristics for the 38mm diameter coil in 6 loops. Through each set of points have been fitted a third or fourth order polynomial curve.

On this graph all the bubbling failure points are at a similar value for Lwi. The exception to this the curve for Hsuc=3.37m where I feel I did not take the pump close enough to failure. This feeling reinforced by Figure 8.9 at this suction head which shows a theoretical bubbling failure point significantly below the final measured point. Similarly, the peak water plug lengths are at similar Lai value. As the suction head increases on Figure 8.11 so the maximum liquid flow rate decreases. This is shown graphically in Figure 8.12, where the points are connected by a second order polynomial. The correlation coefficient equals one within two decimal places: this suggests a very good fit.

Two further characteristic curves at Hsuc=1.83m and 5.0m are shown in Figures 8.13 and 8.14, here the coil pipe diameter has been increased to 50mm and the number of loops has been set at 10. These two curves should be considered with Figure 8.2(a) which was derived using the same pump parameters. The combined characteristic for the measured values is shown in Figure 8.15. This combined characteristic is similar to the previously derived family of curves shown on Figure 8.11. An attempt to produce a feasible theoretical curve at Hsuc=5.5m proved unsuccessful. In the laboratory we were unable to establish a long term steady pumping rate with suction heads of greater than 5m. As with Figure 8.11, Figure 8.15 shows for varying suction heads, a similar inlet water plug length at which a bubbling failure occurs. With Figure 8.15 there is a more noticeable decrease of the air plug length at higher suction heads for the peak water plug length.
The air plugs lengths are measured at 20°C and atmospheric pressure.
8.5 'SWING-IN'

In order to provide an adequate working range under most steady hydraulic conditions, the theoretical 'swing-in' values, %sw, are assumed to vary between about +100 and -100%. The negative value of %sw indicates that the chain of water plugs in the coil is moving, relatively, back towards the inlet due to the expansion of the air plugs as they move along the coil. It was not possible to measure %sw in the laboratory, so the values discussed in section are based on theoretical results.

Firstly if we take the case of a low suction head, where Hsuc=1.83m shown on Figure 8.13 - the theoretical %sw values are shown on Figure 8.16. The 'swing-in', %sw for the solution that satisfies the boundary conditions is plotted against the air plug length at standard temperature and pressure. As the air plug length increases, the maximum head difference across a loop increases. Spilling reduces the head difference across a loop whereas bubbling tends to maintain the maximum head difference. At an air plug length of 1m on Figure 8.16, 9 loops are spilling and this has so reduced the total head difference across the coil that a negative 'swing-in' is required to increase this to the imposed value, 1.83m.

The positive 'swing-in' values associated with the longer air plug lengths occur because the potential head that can be generated across the coil at %sw=0 is greater than the imposed head. For example, with an inlet air plug length of 1.4m and %sw=0, the theoretical difference across the coil is 2.96 m, hence a positive 'swing-in' is required to reduce this outlet head. When the air plug length is 1.6m, the total head difference across the coil with a 'swing-in' value of zero is 4.03m; here a greater positive 'swing-in' is required to satisfy the boundary conditions. It is difficult to give a descriptive explanation of the effect the inlet air plug length on the 'swing-in' since $\xi(1)$, the water plug length, the number of spilling or bubbling
Figure 8.17 Percentage 'Swing-in' for $H_{\text{Suc}} = 5.0 \text{m}$

Figure 8.18 Characteristic Approaching a Spilling Failure

The air plugs lengths are measured at 20°C and atmospheric pressure.
loops are all changing so the explanation in the last paragraph is very much simplified.

Figure 8.17 shows the 'swing-in' values for a suction head of 5 metres, these values are associated with the points shown on Figure 8.14. With %sw=0 the coil is incapable of resisting the head difference of 5 metres across it for air plug lengths of 0.85 to 0.935 metres. The actual 'swing-in', therefore, has to be negative in order to generate the additional head and does occur at the inlet because of spillage. Take, for example, the case where Lai=0.9m, with %sw=0 the head difference across the coil is 4.67 metres. The 10% negative 'swing-in' provides the additional head to raise the outlet pressure of the coil to atmospheric.

In three cases, the water plug behaviour in the inlet loop was recorded on a video and an approximate measurement was made of the inlet angle of the water plug, (π-β(1)). This is a difficult operation to carry out since the inlet angle is continually changing as the plug enters the coil and the optical distortions of the tube and tank glass make any reading only approximate, say within 10°.

The three readings were taken on the 38mm diameter coil with six loops at a suction head of 2.27 metres (see Figure 8.7). The values are as follows,

<table>
<thead>
<tr>
<th>Lai(1)</th>
<th>Theoretical Value</th>
<th>Measured Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.9m</td>
<td>64°</td>
<td>60°</td>
</tr>
<tr>
<td>1.3m</td>
<td>88°</td>
<td>90°</td>
</tr>
<tr>
<td>1.5m</td>
<td>90°</td>
<td>&gt;90°</td>
</tr>
</tbody>
</table>

These values are not proof that the theoretical inlet angles model the real situation well, only that they are a good indication.
Figure 8.19 Variation of Outlet Pressure and Plug Length with Percentage 'Swing-in'
8.6 SPILLING FAILURE

A spilling failure occurs as the air flow rate is reduced, this causes the water plugs to increase in length which in turn causes the plugs at the outlet to spill. This spilling region starts at the outlet and moves towards the inlet until all the loops are spilling. If the air flow is reduced further, air is pulled in vigorously through the outlet into the last loop. A further reduction of the air flow rate causes the air to be pulled through more loops until all but the loop nearest the inlet is experiencing this 'vigorous bubbling'. At this point the pump ceases to work.

In the results shown up to now, this effect has not been shown on the characteristic curve because it produces a very unsteady behaviour in the pump. It was decided that this unstable section should not be included on the general characteristic. This 'vigorous bubbling' effect occurs over a short time in the drum cycle with a sudden rush of air back through the pump: surprisingly the pump continues to lift water during this action.

Flow measurements taken whilst a pump was experiencing 'vigorous bubbling' are shown in Figure 8.18. This curve shows the falling limb of the characteristic from the peak to the lowest air plug length of 0.07 m. This represents an air plug length which only subtends an angle of 9° at the drum centre, and it appears only as a small bubble of air in the inlet loop. In fact, 9 out of the 10 loops are receiving most of their air through from the outlet so the air plug length will increase towards the outlet. Under these conditions the pump could maintain its water flow rate for an indefinite period.

The decline in the water flow rate (or plug length) with a decreasing air flow rate is accompanied by an increase in the number of loops
experiencing 'vigorous bubbling'. At a water plug length of 0.7m, 4 loops are being subjected to the effects of bubbling, this increases to 6 as the air plug length reduces to 0.4m.

As already mentioned, the theory developed in the previous chapter is unable to predict this curve in its present form. One reason for this is illustrated by the example shown in Figure 8.19 where the inlet air plug length is 0.6m. Decreasing the 'swing-in', %sw, normally increases the outlet head, Hout. As the %sw falls to -250% the outlet head does increase up to a value of 9.6m (H_{at} is taken as 10.13m). A further increase in the negative 'swing-in' causes a gradual fall in the outlet head, hence the outlet boundary condition cannot be met.

A break in the line denoting the inlet water plug length is shown on Figure 8.19, this occurs when the spilling region reaches the inlet loop and the water making up the first plug partly comes from spillage at the expense of the dosing liquid. The inlet plug length shown on Figure 8.19 is in fact L_{ai} and not L_{w(1)}. The former is 0.3m shorter.

Two factors are interesting. The %sw value at the peak outlet head is about -250% which gives a value for the water plug length of 1.05 metres. The point corresponding to these values is shown by point X on Figure 8.18 which agrees reasonably well with the measured values on this figure. The computer program also shows that 3 loops adjacent to the inlet are experiencing bubbling which again is similar in number to the 4 observed. This theoretical bubbling has been caused by the excessively high negative 'swing-in' which is required to meet the outlet head boundary condition.

In Chapter 7 in Figure 7.14 it was stated that the limits to %sw should be ±100%, if these extremes conditions are to be investigated then the limits must be increased beyond this figure.
Theoretical values

Figure 8.20 Characteristic Curves at 2 and 6 rpm

Figure 8.21 Calculated Characteristic allowing for Dynamic Losses

*The air plugs lengths are measured at 20°C and atmospheric pressure.
The theoretical predictions close to the spilling failure do provide good indicators of the pumps behaviour but mathematically they are not valid since one of the boundary conditions cannot be met.

Adjustment of spilling and bubbling parameters has not lead to a method of satisfying the outlet condition, part of the problem could be associated with the violence with which the air bubbles back which cannot be accommodated in theoretical terms at present.

8.7 BUBBLING FAILURE

As the air plug length increases the water plug length decreases and this eventually causes bubbling to occur at the outlet which for a small further increase in Lai rapidly moves to the inlet. In practice, when air bubbles pass through into the first air plug of the pump at the inlet a progressive failure begins.

Air bubbling through a coil, increases the 'swing-in' at the outlet but this only pulls more air through to the outlet.

Examples of a \( \%sw \) close to a bubbling failure are shown in Figure 8.16 for a suction head of 1.83m and for a head of 5.0m in Figure 8.17m.

It is not possible to define the point of a laboratory bubbling failure precisely but, in general, the theoretical prediction of the bubbling failure point is at a slightly higher value of Lai than the measured one. See Figures 8.5 to 8.10, 8.13 and 8.14. However, it is felt that predictions are satisfactory.

8.8 EFFECT OF DRUM SPEED

The effect of drum speed has been mentioned in Section 7.2. If the flow rate
is linearly related to drum speed, then the actual air and liquid flow rates could be found by multiplying the plug lengths, the pipe area and the drum speed. The system is linear in one sense in that the unit curve (plug lengths/rev) for, say, a pump at 6 rpm is approximately twice the unit curve for that of 3 rpm. This only applies if the pressure in the inlet tube is the same in both cases. Generally the dynamic losses in the coil are small but unfortunately, the suction pressure in the inlet tube will increase as the speed increases. This is best illustrated by an example. Figure 8.20 shows the measured characteristic curves for two sets of results from the pump operating at a suction head of 3m, one was at a speed of 2 rpm and the other at 6 rpm. The curve for the higher speed is displaced to the left, though the peak values for \( Lw(1) \) is similar.

A pressure transducer in the inlet tube showed that the pressure head in the inlet tube was close to 3.0m at 2 rpm, but at 6 rpm it registered 3.21 m. Strictly it is not the suction lift that should be quoted but the pressure head in the inlet tube.

If the theoretical head losses are calculated for the suction pipe they amount to 0.11m. The velocity used in the calculation was taken as twice the mean flow rate since the pulsating flow produces significant movement for about half the drum cycle. Both friction losses and minor losses were calculated on this basis.

Figure 8.21 shows the theoretical curve recalculated at a suction head of 3.15m and the measured curve at 6 rpm. The agreement between the two curves is good, particularly on the straight section of the curve.

It is envisaged that this type of pump, generally, would be used at rotations of less than 3 rpm and so the head losses in the suction pipe are small. In the above case the suction pipe was only 38mm in diameter compared with 50mm diameter for the coil. In this case I feel that the suction pipe was
undersized.

8.9 COMMENTS

On the range of tests carried out in the laboratory, the closeness between the measured and calculated results is good. It was hoped that some simple rules might be derived to find the value of the water plug length at the bubbling failure and the approximate air plug length at the maximum water plug length - as yet they have proved illusive.

The only disappointing aspect is associated with predicting the falling limb of the curve near spilling failure. However, as already mentioned this is not considered a serious disadvantage as the pump would not operate well in this region anyway.
CHAPTER 9

PRACTICAL ASPECTS OF THE SINGLE COIL SUCTION PUMP

9.1 LIFTING LIQUIDS CONTAINING SOLIDS

To date no direct commercial applications have been developed for the single coil suction pump. One possible use is the lifting up to ground level of crude sewage from a sump below ground level. This often occurs on sewerage systems which operate mainly by gravity.

The pump would provide an unobstructed flow to the sewage passing through the coil. The rotary joint would be the only mechanically moving part in contact with the sewage which, if well designed, should give trouble-free performance as we found on the Coil Treatment Unit.

The low speed of the pump should also eliminate the common problem of abrasion by grit.

More work needs to be carried out to assess the potential of the Single Coil Lift Pump for moving sewage and other liquids containing solids.

9.2 SELF PRIMING DEVICE

One disadvantage of the single coil pump is that once the liquid level in the pump sump drops below the bottom of the suction pipe, the pump loses its prime and then it will not be able to restart pumping when the sump level rises again. This problem can be overcome by installing a self priming device on the Inlet tube. A simplified version of this device is shown in

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Figure 9.1 Sketch of The Priming Device
Figure 9.1. Whilst a suction pressure is exerted in the suction pipe the piston is held in a position that isolates the priming tank. If the suction pressure is broken by air entering the bottom of the pipe, the piston is withdrawn by a spring allowing the liquid to enter inlet tube. Liquid is then taken into the coil re-establishing the manometric effect in the loops.

This priming device has been used successfully on the laboratory coil pump. The main problem with this arrangement is that it needs a small priming tank and a source of supply (water in this case) for the priming device.
CHAPTER 10

THE THEORY OF THE DOUBLE COIL SUCTION PUMP

10.1 INTRODUCTION

If the single coil suction pump loses its prime and the pump continues to rotate, then all the liquid plugs will be ejected from the coil and none will replace them. The pump has then lost its capability to pressurise a fluid until it is primed again, that is, unless it is fitted with a self priming device.

In order to overcome this problem, a series of experiments was carried out with the pump drum supporting two helical coils, one wound clockwise and the other anticlockwise. These two coils were joined end to end and, at the joint, a third pipe was connected in. The other end of this third pipe was connected to the suction pipe through the rotary joint. See Figure 10.1 for one possible layout of the pump.

When the drum rotates, one coil pulls liquid into the pump (the inlet coil) from the inlet tank whilst the second coil takes it out (the outlet coil). If the capacity of the coil taking liquid into the pump is less than that of the coil taking the liquid out, then a suction pressure will be generated at the junction of the coils and this pressure will pull up the liquid through the suction pipe to balance the flows. The liquid being taken in through the inlet is termed the 'host liquid' whilst the 'dosing liquid' is pulled up through the suction pipe. The distinction is made between the two liquids because one of the proposed uses of this pump is to introduce a chemical into a second liquid.

The difference in pumping capacities of the inlet and outlet coils can be
Figure 10.1 General Layout of a Double Coil Suction Pump
achieved by either, using a larger diameter pipe on the outlet coil or, having a larger effective drum diameter on the outlet coil, as shown in Figure 10.1. The two coils can be wound on separate drums on a common axis or, more simply, the outlet pipe can be wound on top of the inlet pipe.

As with the two other pumps described in this thesis, the aim of the theory is to produce a characteristic curve for the pump under varying hydraulic conditions.

The air being pulled into the suction pump is not an independent variable, as with the single coil suction pump. It is the depth of immersion of the inlet coil and the speed of rotation that determines the air flow rate. These two factors also govern the liquid flow rate. Consequently, for a specified pump configuration (including the depth of immersion), the main pump characteristic consists of a plot of the dosing liquid flow rate against suction head.

The outlet coil acts in a very similar way to the single coil suction pump so very little modification is required to the theory described in Chapter 7. The inlet coil, however, is acting in a similar manner to a lift pump except that the pressure at its outlet (the pipe junction) is now below atmospheric. In the inlet coil, a new phenomenon is present where spilling and bubbling occur simultaneously. Because of this, the equations developed in Chapter 4 need to be modified; this is explained in Section 10.2

Section 10.3 describes the interaction of the two coils on the pump and Section 10.4 is concerned with method used to construct a characteristic curve itself.

Since there are two coils on this pump, two sets of variables are required. To differentiate the two sets of variables where necessary, the inlet coil will have the subscript 'i' and the outlet, the subscript 'o'.
10.2 THE BEHAVIOUR OF THE INLET COIL

10.2.1 Introduction

As already mentioned, the behaviour of the inlet coil is similar to that of the lift pump. It is proposed to follow the same reasoning as given in Chapter 4. In order to prevent excess repetition, only the differences between the two cases will be discussed in detail in this Section.

10.2.2 The Cascading Manometer

The layout of the air and liquid plugs in a typical inlet coil of a double coil suction pump is shown in Figure 10.2. For simplicity neither spilling nor bubbling is shown.

The liquid level differences in the loops now reflect the higher pressure at the inlet when compared to the outlet.

The equations developed for the lift pump covering the cascading manometer and the air compression equations, that is, Equations 4.1 to 4.4, are the same as for the inlet coil. In Equation 4.1, the liquid level differences can be taken to be negative.

10.2.3 Relative Movements of the Plugs

The swing of the liquid plugs in the inlet coil as they move through the helical pipe is caused by an air expansion and not contraction as occurs with the lift pump. The expansion of these air plugs under a reducing pressure causes the liquid plugs to 'swing' towards the outlet as shown on Figure 10.2. The plug swing must be towards the outlet in order to develop the level
Figure 10.2 Cascading Manometer for the Inlet Coil
differences across the liquid plugs that are required to resist the pressure change across the coil.

Figure 10.3 shows the plugs in a straightened pipe at the start and end of a drum rotation. This Figure is analogous to Figure 4.4 for the lift pump, except that now, for convenience, lines XX and YY are on the leading edge of the liquid plug Wn and not the trailing edge.

The leading edge of liquid plug Wn is defined by Line XX in Figure 10.3(a) and, one revolution later, this leading edge has moved to the position shown by Line YY on Figure 10.3(b). If the plugs are subjected to no other forces than those produced by the mechanical movement of the coil then,

$$\YY-XX = \pi \cdot D$$

Since air expansion does cause a relative movement or swing,

$$\YY-XX = \pi \cdot D + \text{Exp}(n)$$

Exp(n) is the relative movement of the leading edge of the liquid plug Wn towards the outlet as the water plug moves from the position shown in Figure 10.3(a) to the position shown on Figure 10.3(b). The use of Exp(n) here is very similar to that of Con(n) for the lift pump, except that, Exp(n) is for air expansion and not contraction.

Now $$\YY-XX = Lw(n+1) + La(n)$$, then

$$Lw(n+1) + La(n) = \pi \cdot D + \text{Exp}(n)$$  \hspace{1cm} \text{Equation 10.1}$$

This equation corresponds to Equation 4.6 for the lift pump.

### 10.2.4 Movement of Non-Spilling Plugs

For the non-spilling region, $$Lwi = Lw(n+1)$$, and so Equation 10.1 becomes,

$$\text{Exp}(n) = La(n) - Lai$$ \hspace{1cm} \text{Equation 10.2}$$

or more correctly, $$\text{Exp}(n) = \pi \cdot D - (Lw(n+1) + La(n))$$. This is equivalent to Equation 4.7 for the lift pump.
Figure 10.3 Plug Layout in an Unwound Pipe
The relative movement, $\text{Exp}(n)$, causes the liquid plug to swing towards the outlet which, in turn, causes the level difference across the loop. For further explanation, see Section 4.4.5 where the reasoning applied to the lift pump is applicable here.

10.2.5 Movement of Spilling Plugs

With the spilling loops, the criterion for spillage is the same as used for the lift pump and described in Section 4.4.8. The spilling angle, $\Omega_{\text{sp}}$, given in Equations 4.8 and 4.9 is the same in this case, as is the calculation for $\varrho(n)$ defined by Equation 4.10.

If the liquid plugs are spilling then the relative movement of the leading edge of the plug towards the inlet is zero i.e. $\text{Exp}(n)=0$. Equation 10.1 then becomes,

$$L_w(n+1) = \pi.D - L_a(n)$$

Equation 10.3,

or

$$L_w(n) = \pi.D - L_a(n-1).$$

If $\pi.D = L_{ai} + L_{wi}$ then $L_w(n) = L_{wi} + L_a(n) - L_a(n-1)$.

For any loop, $L_a(n)$ will be greater than $L_{ai}$ and so,

$$L_w(n) < L_{wi}$$

This is the opposite to the lift pump where the plug lengths become longer towards the outlet, here they become shorter. At first sight this phenomenon seems puzzling.

Consider Figure 10.4 which shows two adjacent loops spilling. In one drum revolution, liquid plug $W_n$ will have moved from its position shown on Figure 10.4 to the position occupied by plug $W_{n+1}$ on the same figure. During this time, the plug will be losing liquid by spillage to $W_{n+1}$ and also gaining liquid from $W_{n-1}$. Air plug $A_{n-1}$ will have increased in length due to a decrease in pressure. Assuming that the leading edge of the spilling plugs are all at an angle of $\Omega_{\text{sp}}$ to the vertical, then to accommodate the increase in air plug length, $L_w(n)$ must be shorter than $L_w(n+1)$. 

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In the spilling region of a lift pump, the liquid plug length is determined by the compression of the preceding air plug; see Equation 4.11. In this case, it is the following air plug that governs the length. This factor is evident in the argument given in the previous paragraph.

Focussing on the spillage, the same reasoning can be used here that was used for the lift pump in Section 4.4.8. Incorporating Equation 10.3 into this reasoning, the amount of spillage out of liquid plug \( W_n \) is given by the equation

\[
\text{Spillout}(n) = L_{ai} - L_{a(n-1)} \quad \text{Equation 10.4}
\]

If spilling occurs in the inlet coil it is likely to switch to a combination of spilling and bubbling within a few loops. This is because the trailing edge of a plug reaches the soffit of a particular loop as the plug shortens and any further
shortening causes bubbling to occur. Once established a spilling-bubbling plug will continue to act in this manner until it leaves the outlet.

10.2.6 Movement of the Bubbling Plugs

The same reasoning that was applied to the lift pump in Section 4.4.7 can be applied to the bubbling plug in the inlet coil, only in this case $\text{Con}(n)$ becomes $\text{Exp}(n)$. The plug movements associated with $\text{Exp}(n)$ now refer to the trailing edge of the plug, not the leading edge used for the spilling case.

As with the lift pump, there is no relative movement of the plug towards the outlet due to air volume changes and so $\text{Exp}(n) = 0$. If no spillage is occurring, then $L_w(n) = L_{wi}$ and so Equation 10.1 becomes

$$L_a(n) = L_{ai}$$

Equation 10.5.

If all the plugs lengths remain the same, then,

$$\varnothing(n-1) = \varnothing(n) = \varnothing(n+1)$$ etc.

The head differences across the plugs will all be the same in this region.

10.2.7 Movement of the Spilling-Bubbling Plugs

Consider Figure 10.5 where liquid plug $W_n$ is spilling. As it moves to the position shown by plug $W_{n+1}$ on the same figure its length will decrease as explained in Section 10.2.5. If this decrement is sufficiently large, the trailing edge of plug $W_n$ will dip into the soffit of the pipe and bubbling will start to occur. Even when the liquid plug is bubbling, it will still continue to spill. It is this phenomena that is called the 'Spilling-Bubbling'.

Why should a liquid plug spill and bubble at the same time? This is best explained by reference to Figure 10.6. It is assumed that the trailing edge of the spilling liquid plug $W_n$ has just reached the position where it is about to cause
Figure 10.5 Start of the Spilling-Bubbling Region

Figure 10.6 Plugs in the Spilling-Bubbling Region
bubbling where \( \beta(n) = \pi - \Omega_{sp} \). As it moves to the position shown by plug \( W_{n+1} \) on the same figure, air will bubble from \( A_{n-1} \) to \( A_{n} \). A smaller air bubble passing from \( A_{n-1} \) to \( A_{n} \) will cause the leading edge of \( W_{n} \) to 'fall back' causing the spilling rate to decrease, or stop. During this time \( W_{n-1} \) will have been spilling into \( W_{n} \). The trailing edge of \( W_{n} \) will also 'fall back and the bubbling rate will also decrease or stop. Further movement of \( W_{n} \) along the coil will cause the trailing edge to move back down to the soffit causing the bubbling again. The whole process just described will repeat itself again.

By the action in the spilling-bubbling plugs

\[ L_{w(n)} = L_{w(n+1)} \]  \hspace{1cm} \text{Equation 10.6.} \\
\text{and} \hspace{1cm} L_{a(n)} = L_{a(n+1)} \]  \hspace{1cm} \text{Equation 10.7.}

Using the same criteria laid down for the bubbling and spilling plugs in Sections 4.4.7 and 4.4.8 respectively, we have,

\[ L_{w(n)} = \pi + \Omega_{bub} - \Omega_{sp} \]  \hspace{1cm} \text{Equation 10.8.} \\
\text{and} \hspace{1cm} L_{a} = 2\pi R_{l} - L_{w(n)} \]  \hspace{1cm} \text{Equation 10.9.}

The spillage from the liquid plug must equal the spillage out of the plug in the spilling- bubbling region. This must also apply to the air plugs since they remain the same length.

Initially the plugs must be spilling before they move into the spilling - bubbling region. In the spilling region, \( \text{Spillin}(n) < \text{Spillout}(n) \). It seems likely, that as the plugs move into the spilling - bubbling region the difference in spillage is made up by the amount of passed by the bubbling process.

The reasoning given above provides a hydraulic justification that the spilling - bubbling region can exist and, as far as one can judge, it does occur in practice. On the inlet coil of the laboratory pump, a region of bubbling loops can be observed where the leading edges of the same liquid plugs are close to the crown of the helical pipe. It is very difficult to say with absolute certainty that spilling is occurring, since the rising air bubbles distort the liquid movement,
but it is fair to assume they are spilling.

10.3 THE CALCULATIONS ROUTINES FOR THE INLET COIL

10.3.1 Introduction

The calculations are divided into routines and procedures, as with the lift pump. Each routine is described separately in this section before the overall calculation procedure is dealt with in Section 10.3.10.

The calculations have been further separated into cases where spilling (or spilling-bubbling) predominates and where bubbling predominates. If

\[(\frac{L_{wi}}{2.R_i} - \Omega_{bub}) < (\frac{L_{wi}}{2.R_i} - \Omega_{sp})\]

then bubbling occurs, if not, spilling occurs. This assumes that one or other will occur, at low suction heads there may be only non-spilling loops in the coil and so either set of calculations can be used. Spilling loops are generally associated with depths of immersion of greater than 50% and bubbling loops occurs with values below 50%. In the following sections, depths of immersion are used to refer to the type of behaviour though, strictly, the above equation should be used.

10.3.2 The Level Differences

In order make use of the procedures developed for the lift coil pump here, the angles, \(\Theta(n)\) and \(\beta(n)\) which applied to the trailing edge and the leading edge respectively, of a liquid plug in the lift pump now are reversed. This is shown in Figure 10.6. This means that Equations 4.16 to 4.20 can be used for the inlet coil calculations.

10.3.3 Calculation Routine for Non-Spilling Region

The calculations start at the last non-spilling loop (loop number Nospill) and
progress back to the inlet. It is assumed at this stage that Nospill, the number of spilling loops, is known.

The calculation routine is very similar to the lift pump which is shown Figure 4.8. The only difference is that the eighth box down from the top is now $H_a(n-1) = H_a(n) + h(n)$ instead $H_a(n-1) = H_a(n) - h(n)$.

### 10.3.4 The Calculation Routine for Spilling Region

On the lift pump, if either spilling or bubbling is occurring, then each successive loop is assumed to be spilling air or liquid as the calculations progressed towards in the inlet. With the inlet coil of the suction pump, it is very likely that the spilling-bubbling region will occur between the outlet and the spilling region. The last spilling loop then becomes the outlet boundary to the spilling region. At the inlet end of this region, Nos+1 becomes the first loop to spill, where Nos is the number of last non-spilling loop. It is assumed at this stage that this loop number is known.

The method of calculating the level differences in the spilling region is very similar to the lift pump except that, in box number 3 from the top in Figure 4.9 where $L_w(n)$ is calculated, Equation 10.3 is used instead of Equation 4.11. The calculations start at the boundary of the spilling-bubbling region and continue through the loops towards the inlet until the liquid plug length equals, or is greater than, $L_wi$. The plug length in the spilling - bubbling region is shorter than $L_wi$ and in the non-spilling region it equals $L_wi$. The number of loops in the spilling region, therefore, is determined by the ability of the spillage to increase the plug length across the region (working towards the inlet).

The Calculation Routine for the last non-spilling loop here is the same as shown on Figure 4.11 for the lift pump.
Start

\[ n = N \]

\[ \Theta(n) = \pi \cdot \Omega_{\text{bub}} \]

\[ \beta(n) = \pi \cdot \Omega_{\text{sp}} \]

h(n) from Equation 4.17

\[ h_i = h(n) \]

\[ n = n - 1 \]

\[ H(n-1) = H(n) + h_i \]

Is \( n < N - N_{s/b} \)?

Ns/b is the number of spilling/bubbling loops

Yes

END

No
10.3.5 The Calculation Routine for Bubbling Region

Again the calculation routine for the inlet coil is very similar to the lift pump which is shown on Figure 4.10. The only difference is that in the sixth box down from the top is, \( H(n-1) = H(n) + h(n) \) and is not \( H(n-1) = H(N) - h(n) \) as shown on Figure 4.10.

The Calculation Routine for the last non-bubbling loop here is the same as described in Section 4.5.5 for the lift pump.

10.3.6 The Calculation Routine for Spilling-Bubbling Region

It is assumed at this stage that a spilling-bubbling region does exist in the coil. The calculation start at the outlet end and proceed towards the inlet. It is also assumed that the number of spilling-bubbling loops \((Ns/b)\) is also known.

The calculation routine is shown in Figure 10.7. All the loops have the same angle, \( \Theta(n) \), and level difference, hence the calculation loop reduces to a simple additive process to determine the pressure heads in the air plugs.

10.3.7 Last Spilling Loop

There will be a spilling-bubbling region on the outlet side of the last spilling plug, or in a rare case, this plug may occur in the last loop.

On Figure 10.5, \( \Theta(n) = \pi - \Theta_{bub} \) and if liquid plug \( W_n \) is spilling \( \beta(n+1) = \pi - \Theta_{sp} \). The last spilling loop can, therefore be treated as a conventional spilling coil.

10.3.8 First Spilling Loop

If the depth of immersion is greater than 50%, the first spilling loop (Loop No
ns) will be bounded by the spilling liquid region on one side and the non-spilling region on the other. The first spilling plug will be spilling itself but not receiving spillage from the following plug. It is, in fact, the equivalent of the last non-spilling plug on the lift pump and so the parameter "ns" will still be used. Since the plug rotations are now anticlockwise and not clockwise (see Figure 10.2) the plug will spill before receiving spillage.

If Wns is the first spilling loop, then initially, Lw(ns) is unknown and, at a maximum, it will equal Lwi just before it spills and the minimum will be Lw(ns+1). The method for calculating the head difference across the first spilling plug is incorporated into Figure 10.9. Initially the liquid plug length is taken as a minimum and ϕ(ns)=π-Ωsp since it is spilling. It is then possible to calculate 0(ns) and hence h(ns) using Equations 4.20 and 4.17. Lw(ns) is then adjusted to satisfy the boundary conditions. See Section 10.4.3.

10.3.9 The First Bubbling Loop

The first bubbling loop will lie between the bubbling region and the non-spilling region. In this loop, the air plug will be losing air by bubbling but not gaining any itself. The length of this air plug, Lanb, initially is unknown, its minimum length will be Lai, the length of the plug in the bubbling region. The maximum length is uncertain. In a similar method to the first spilling plug, the length is initially set at Lai, ϕ(nb)=π-Ωbub and so h(nb) can be calculated. See Figure 10.10. At a later stage, La(nb) is adjusted to satisfy the boundary conditions.

10.4 CALCULATION PROCEDURE FOR INLET COIL

10.4.1 Combinations of Calculation Routines

A number of different combinations of hydraulic conditions are possible on the inlet coil; these are shown in Figure 10.8. If the depth of immersion is less
Figure 10.8 Combinations of Hydraulic Conditions in the Inlet Coil

(a) Dimer, 50% Low Hsuc

(b) Dimer, 50% Hsuc high.

(c) Dimer > 50% High Hsuc
For low Hsuc, as (a)
than 50% and the suction head is low, all the loops will contain non-spilling liquid plugs. At higher suction heads, a region of bubbling plugs will exist at the outlet side of the non-spilling plugs.

If the depth of immersion is greater than 50% and the suction head is high, then three hydraulic conditions can exist in the coil. On the inlet side, is a region of non-spilling plugs. There is a region of spilling-bubbling plugs on the outlet side and sandwiched in between are the spilling plugs. It is unusual to have the non-spilling plugs moving directly into the spilling-bubbling region, though theoretically this transition could occur in one revolution.

10.4.2 Overall Calculation Procedure

The calculation procedure has been divided into two main sections, the first deals with depths of immersion less than approximately, 50% and the second is concerned with depths over 50%.

To be more precise, if

\[ \pi - \frac{L_{wi}}{2R} - \omega_{sp} \geq \frac{L_{wi}}{2R} - \omega_{bub} \]

then bubbling will occur in preference to spilling, if not, the opposite is true.

With the depths of immersion greater than 50%, there are two variables which are unknown, they are:

- \( N_{s/b} \), the number of spilling-bubbling coils;
- \( N_{s} \) the number of spilling coils.

Where the depth of immersion is less than 50%, the unknown variable is \( N_{b} \), the number of bubbling loops.

Investigations have shown so far that only one solution exists for each case mentioned above, that is, if the following boundary conditions are satisfied.

- \( H_{out} = H_{at} \)
- \( \pi > \Omega(n) > 0 \)
Figure 10.9 Overall Calculation Procedure. Dimer>50%

Inlet Coil
Figure 10.10 Overall Calculation Procedure Dimer<50%
\[ \pi > \beta(n) > 0. \]

The last two conditions are obvious constraints for a stable system, but most trial runs fail on one of these two conditions.

10.4.3 Depth of Immersion greater than 50%

When the depth of immersion is greater than 50%, the calculations are carried out by the method shown on Figure 10.9. The hydraulic regions present in the coil are those shown in Figure 10.8(c). In the procedure, the number of spilling-bubbling loops \((N_s/b)\) is varied from zero to \((N_i-1)\). The calculations start at the outlet and move to the inlet where the inlet pressure is calculated, or, the calculations are abandoned because \(\beta(n)\) or \(\phi(n)\) are less than zero. This adjustment process continues until the calculated inlet pressure is greater than atmosphere. As with the lift pump, the 'fine' adjustment is then made by increasing \(Lw(ns)\) by small decremental steps and re-calculating the pressure difference across the non-spilling region. This continues until the calculated inlet pressure is equal to the atmospheric pressure, within the required tolerance, \(Tol\). In the calculations, \(Tol\) is normally 0.010m.

10.4.4 Depths of Immersion less than 50%

The procedure for the calculations when the depth of immersion is less than, approximately, 50% is shown in Figure 10.10. In the upper loop on the chart, the number of bubbling loops, \(N_b\), is varied from zero to \(N_i\). For each value of \(N_b\), the bubbling and non-spilling routines are carried out (working back from the outlet) to find the outlet pressure \(H(0)\). As \(N_b\) increases so \(H(0)\) increases from a value well below \(H_{at}\). The calculations continue until \(H(0) > H_{at}\), the fine adjustment is then made by increasing \(La(ns)\), the length of the last non-spilling plug, until \((H(0) - H_{at})\) is within the required tolerance. This method is similar to that used for the lift pump and it is described in Section 4.6.6.
Figure 10.11 Coil Junction During One Drum Revolution
It was originally thought that more than one solution might exist for values of Nb greater than that obtained for the first solution, but this has not been the case. If the number of bubbling loops is increased to the number on the coil and no solution has been found this calculation run is abandoned.

Not included on Figures 10.9 and 10.10 are the adjustments to be made to Lwi and Lai in the inlet coil due to high values of "swing" in the first loop. The method to be used is very similar to the one described on Figure 4.14 for the lift pump.

10.5 INTERACTION OF INLET AND OUTLET COILS

10.5.1 Introduction

As the drum holding the two coils rotates, plugs of liquid leave the inlet coil and enter the outlet coil. Observation of the double coil pump suggests that the leading edge of the liquid plug leaving the inlet coil enters the outlet with a decrease in $B(N_i)$ i.e. swings down towards the horizontal. The dosing liquid is then pulled up into the outlet coil in a surge of flow as the trailing edge of the liquid plug passes the coil junction. This behaviour has been noted in the laboratory using a dosing liquid containing dye. The trailing edge will therefore move back as it enters the outlet coil pulling in the dye coloured tracer liquid.

10.5.2 The Rotation of the Plug as it changes Coils

Four stages of the drum cycle are shown in Figure 10.11 where a liquid and air plug are leaving the inlet coil and entering the outlet coil. On this figure, the last liquid plug in the inlet coil is denoted by $W_{Ni}$ and the same plug on the outlet coil is denoted by $W_{O1}$ where $N_i$ is the number of loops on the outlet
Figure 10.12 Manometric Arrangement of the Coils
coil and Wo1 is the first liquid plug in the outlet coil. This same system of lettering applies to the angles and plug lengths.

The dosing liquid that is pulled into the coil is shown in a lighter shading than the host liquid; see Figure 10.11 (d). On this figure, the dosing liquid is shown as a discrete plug at the trailing edge of the host plug. In fact, the dosing liquid is pulled into the host plug over a period of, say, an quarter, or more, of a drum revolution and is mixed into the host liquid by internal plug circulation.

It is also useful at this stage to show the two cascading manometers meeting at the junction. This is shown in Figure 10.12, where, in effect, the diagram is made up of the combination of Figure 10.2 for the inlet coil and Figure 7.3 for the outlet coil.

First consider the case where there is no inlet 'swing-in'. On Figure 10.11 (a), the inlet arm is assumed to be empty when at the '12 o'clock' position, the reason for this will be explained later. As the arm rotates in a clockwise direction, the leading edge of the last inlet plug will meet the arm inlet and flow into the arm. If Lx is the effective length of the inlet arm, then the leading edge of the plug will retreat by an amount Lx', where,

\[ Lx' = Lx \cdot \left( \frac{d \phi}{d \theta} \right) \]

The angle subtended by this distance will be Lx'/Ro.

The original definition for the angles subtended by the leading and trailing edges of the liquid plug provides a complication here, in that Bi(Ni) becomes \( \pi - \theta(1) \). See Figures 10.11 (a) and (c). The relationship between these two angles is given the following equation,

\[ \theta(1) = Bi(Ni) - Lx'/Ro. \]

In Figures 10.11(b) and (c), the liquid plug will move from the inlet to the outlet coil. Some dosing liquid may be taken into the plug during this time.
Figure 10.13 Bubbling at the Coil Junction
On Figure 10.11(d), the inlet arm reaches the end of the inlet liquid plug and air will bubble up the arm as well as the dosing liquid flowing down into the coil. This phenomenon has been observed in the laboratory and it will empty the inlet arm as it rises to the vertical position.

Generally, with the inlet coil of a double suction pump, the liquid plugs are bubbling close to the outlet, a low depth of immersion is required on the inlet coil to provide sufficient air for the outlet coil to develop a substantial head across the coil.

The effects shown in Figure 10.11 lessen the adverse head difference across the first outlet coil loop. This head difference can be reduced further by the bubbling effects. This is shown in Figure 10.13. As the inlet arm moves towards the leading edge of the last inlet liquid plug, air is bubbling through the plug as shown in Figure 10.13(a). As this liquid plug moves through the joint, bubbling is still occurring and this air will cause the leading edge of the plug, now in the outlet coil, to move back in an anticlockwise direction; see Figure 10.13(b). This action will stop when the 'T' piece reaches the trailing edge of the plug and the dosing liquid is introduced. This position is shown in Figure 10.13(c). Figure 10.13 is slightly misleading since the diameter of the outlet pipe will be greater than the inlet pipe or, the effective drum diameter will be greater on the outlet coil. These differences will produce the apparent shortening of the water plug in Figure 10.13 which is not fully shown on this figure.

The amount of air bubbling in one drum cycle is equal to \((L_a(N_i)-L_{ai})\): see Section 4.4.8 for reasoning. If the bubbling is assumed to occur at a constant rate throughout the drum cycle, then the amount of air bubbling (expressed as a length) through the last plug which affects the leading edge of the plug is given by

\[
L_{bub} = \frac{(L_a(N_i)-L_{ai})L_{wi}}{\pi D_i}
\]

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The rotation of the leading edge of the first outlet coil liquid plug will be affected by:

(a) the rotation of the edge produced by the inlet coil;
(b) the effects of bubbling through the junction, if they occur;
(c) the rotational changes due to inlet arm filling;
(d) the "swing-in" due to the outlet coil adjusting to the pressure difference.

If all four effects listed above are present then

$$\theta(1) = \theta_i(N_i) - L_w/R_o + L_{rel}/R_o - L_{bub}/R_o$$  
Equation 10.10

$\theta(1)$ refers to the plugs in the outlet coil. The case of the inlet coil spilling is not considered in great detail since in many practical cases, bubbling is occurring in the inlet coil. If spilling in the last inlet loop does occur (without the spilling-bubbling mechanism occurring) then $L_{bub}$ will equal zero and the spillage into this last loop will bring the liquid plug length back to $L_{wi}^0$.

10.5.3 The Calculation Procedure for the Outlet Coil

If there was no 'swing-in' of the plugs (positive or negative) in either coil, the amount of dosing liquid entering the pump should equal the difference between the volume of the inlet and outlet loops. This would mean that the length of the outlet plug in the first loop could be found from the following equation,

$$L_{wo}(1) = L_{wi}^0 + \frac{(\pi d_o^2 \cdot \pi D_o - \pi d_1^2 \cdot \pi D_1)}{\pi d_o^2}$$  
Equation 10.11

If a 'swing-in' is occurring in the outlet coil, the plug length will shortened if the same arguments are followed as explained in Section 7.4.5 for the single coil suction pump. The modified length of the outlet liquid plug would then be given by the equation,
Lwi, Lai, B(Ni) Known inlet calcs.

Bo(1) from Equation 10.10
Lbub=0

START

Lwo(1) from Equation 10.11

Lao(1) from Equation 10.13

%sw=%sw+0.5

Lwo'(1) from Equation 10.12

Is bubbling is occurring in Inlet Coil?

Yes

Bo(1) from Equation 10.10, Lbub=0

No

Non-spilling Routine Figure 7.5
For exit path see Fig 7.5

Last Non-Bubbling Routine, Section 7.36.
Bubbling Routine Fig 7.9.

Last Non-Spilling Routine, Figure7.8.
Spilling Routine Fig7.7.

Is ABS(Hout-Hat)<Tol?

Yes

Figure 10.14 Calculation Procedure for Outlet Coil

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The 'swing-in' (%sw) is only due to air expansion effects, the inlet arm filling and the bubbling through the junction are effectively causing a 'swing-in' but this does not effect the length of the liquid plug in the outlet coil as Lrel does.

The length of the air plug in the inlet coil, Lai(Ni), is passed from inlet coil calculations to the outlet calculations. Before use, the air plug length has to be modified for the change of pipe diameter, if any. The relationship is

\[ La(1) = Lai(Ni) \left( \frac{d_0^2}{d_i^2} \right) \]

Equation 10.13

Figure 10.14 shows the Calculation Routine to determine the effects of the transfer of the pairs of fluid plugs from the inlet coil to the outlet coil. To start these calculations, the characteristics of the incoming plugs from the inlet coil will already have to been calculated. For the outlet coil the value of %sw is initially set at zero then increased in steps of 0.5% until the calculated outlet pressure Hout or H(0) is equal to the ambient pressure, within the tolerance Tol which is taken as .01 metres. Varying %sw also varies the length of the liquid plug Lwi9. Figure 10.14 is very similar to Figure 7.17 used for the single coil suction pump.

Figure 7.17 uses the method of successive bi-section to converge to a solution, in this case, the process is much less efficient. Ideally it would be preferable if the successive bi-section method was applied to Figure 10.14.

10.5.4 Overall Calculation Procedure for Both Coils

It is assumed that the following parameters are known

The two drum diameters (Di,Do).
START

Is Dimer < 50%?

Yes

Procedure
Fig 10.10

No

Procedure
Fig 10.9

Transfer Values
Lwo(1)
Lao(1),
B1(Ni)
Ω1(Ni)

Calculation Procedure
Figure 10.14

Dosing Rate from
Equation 10.14

END

Figure 10.15 Double Coil Calculation Procedure
The two pipe diameters (di, do).
Drum speed (S).
Depth of immersion of inlet coil (Dimer).
Suction lift (Hsuc).

The aim of this procedure is to determine how much dosing liquid is pulled up into the outlet coil.

The calculation procedure is shown Figure 10.15, the upper part of the flow chart applies to the inlet coil and the lower part of the chart refers to the outlet coil. The results from this chart will only produce the dosing rate for one suction head. In order to produce a pump characteristic, this calculation will have to be repeated for a number of suction lifts.

This process represents a large computer program with many iteration loops. A program based on the Figure was written in Microsoft Basic on the Apple MacIntosh and because of its unwieldy size, the inlet and outlet coil calculations were kept in two separate programs.

The volume of dosing liquid lifted into the pump per drum revolution is given by

\[ Q_d = L_{wi(outlet)} - L_{wi(inlet)}. \]

It is then possible to vary the suction head to determine the characteristic of the pump. The use of this procedure is discussed further in the next chapter.

10.6 COMMENTS

The calculations for this type of pump are cumbersome. The theory could benefit from further refinements on the section on the transfer of plugs between one coil and the other. The transfer conditions used for the inlet coil only apply to bubbling plugs though the theory covering the inlet coil includes
the spilling and bubbling cases.

Since the air plugs generally shorten considerably when transferring to the outlet coil (assuming an increase in coil diameter), the inlet-coil air plug length needs to be significant length to ensure that the outlet-coil air plug length is sufficient to allow the outlet coil to generate the required head difference. If spilling does occur in the inlet coil it will change to a spilling-bubbling region within a couple of loops so the chance of spilling occurring in the inlet coil at the junction is uncommon therefore it has not been considered in detail.
CHAPTER 11

TESTING OF THE DOUBLE-COIL SUCTION PUMP THEORY

11.1 INTRODUCTION

The range of tests falls into two distinct groups. The first is based on two larger drum diameters, 0.45 and 0.91 metres and the second set is based on a much smaller drum size, 0.1m in diameter.

The larger diameter drums were used to investigate the general behaviour of the double-suction coil pump, whereas, the experiments on the smaller drum were designed to look at the possibility of using this pump as a dosing pump; see Chapter 12. The results from the smaller drum would also provide further evidence for the validity of the theory of the pump's behaviour, if it had already been tested on the larger pumps.

11.2 THE LARGE PUMPS

11.2.1 The Laboratory Experiments

The apparatus used for these experiments was the same as for the single coil suction pump. The drum was set up in a tank containing the host liquid, in this case, water. Two pipes of different diameters were wound around the outside of the drum, end to end. The two coiled pipes were secured together via a pipe 'T' piece connected to the outside of the drum. The third connection on the 'T' was joined to a pipe which lead to the rotary joint which, in turn, was connected to the sump via the suction pipe. Water was used for the dosing liquid and the host liquid.

As already mentioned the testing rig and the measuring procedures were the
same as those described in Section 8.2 for the Single Coil Suction Pump.

In this case, the depth of immersion of the inlet drum was now important and so this had to be set at the start of each set of tests. With this rig, however, the depth of immersion could not be set greater than 50%. The flow rate measured at the outlet to the holding tank was not the pumping rate from the sump since, this flow was made up of the dosing liquid and the inlet coil flow from the holding tank. Unfortunately it was not practical to measure the dosing rate directly in the suction pipe since the pulsing flow precluded the use of flowmeters.

Over 70 tests were carried out where the following parameters were varied:
- speed, 1 to 5 rpm;
- helical pipe diameters, 32, 38 and 50mm;
- depth of immersion 50, 40, and 28%;
- suction heads from a minimum of 1.7m down to the maximum of the pump.

Not all the possible combinations of the above parameters were tested. In each run, the flow rate from the outlet coil was measured.

As with the Single Coil Pump, the level differences in the loops could not be observed because of the opaqueness of the pipe material that was required to resist the suction pressures.

11.2.2 The Characteristic Curve

As with the other types of coil pump, the important characteristic curve is a plot of flow rate against head, in this case, the suction head is plotted against the flow rate of the dosing liquid.

Before comparing the calculated with the measured results, it is worth
Figure 11.1 Typical Theoretical Pump Characteristic
looking at the general form of the characteristic. This is shown on Figure 11.1 and is a theoretical prediction of the head flow relationship for a pump with 10 turns on each coil, but with a larger helical pipe on the outlet coil; not all the calculated points are shown. The drum speed was 2.5 rpm.

The curve on Figure 11.1 has been constructed using a computer program based on the flow chart shown on Figure 10.14. The program was written in Basic, run on an Apple Macintosh and required about 5 minutes computing time to establish each point on the curve.

The main part of the curve between points A and B is approximately a straight line. As the suction head increases above the value at B, the flow rate declines rapidly until, at a suction head of 4.5m, the dosing flow rate falls to zero: see point C. At this point, water is still taken in at the inlet and expelled through the outlet coil, but the capabilities of the two coils to move water are equal. At this stage gross spilling or bubbling will be occurring.

At zero suction head (point A), the flow rate equals the difference between the volumes in a loop of the outlet coil and the inlet coil. This will be discussed further in Section 11.2.3.

In all cases that have been studied so far, the curve between A and B approximate well to a straight line, but as will be shown later, the decline of the flow rate between points B and C is not always as 'sharp' as shown on Figure 11.1.

Taking the calculations for the 3m suction head on Figure 11.1 as an example of the theoretical plug configuration predicted by the computer program, the lengths of the water and air plug at the inlet are 1.24 and 1.609m respectively. In the first loop of the inlet coil, 43% of the total volume is filled with water. The level differences in the loops in both the inlet and outlet coils are shown.
Figure 11.2(a) Level Differences for a Suction head of 3m

Figure 11.2(b) Plug layout at Coil Junction

Air Plug Length = 1.25m

Outlet direction

Outlet Coil

Inlet direction

Air Plug = 2.17m

INLET COIL- LOOP 6

OUTLET COIL - LOOP1
in Figure 11.2(a). A rise in pressure across the liquid plug in the direction of flow is taken as positive. The first loop on the outlet coil has a difference of 0.22m across it and the last inlet-coil loop is bubbling.

The plug rotation of this last plug on the inlet coil is shown in Figure 11.2(b) where the leading edge of the plug in the inlet coil is at an angle of 23° to the vertical; \( \Omega_{\text{bub}} \) is taken as zero in this case. As this leading edge passes through the junction of the two coils, it will experience a negative 'swing-in' of 11% which causes the leading edge of liquid plug to swing back by 5°. The water falling back into the inlet arm causes another 45° rotation. These two cause the rotation of the leading angle of the plug to 'fall back' by 50°. In addition, the bubbling through the coil junction adds a further 27° making a total of 100° (23+50+27). With a water plug length of 1.68m (subtending angle, 212°), the trailing edge will at an angle of 48° (360-212-100°) to the vertical (12 o'clock); this angle is \( \beta(1) - \pi \) in the outlet coil calculations; see Figure 11.2(b). This 48° angle will produce a level difference in the first loop of the outlet coil of 0.3m. To summarise, as a water plug passes through the joint, it swings from a level difference of 0.87m giving a pressure fall in the direction of plug movement to a level difference of 0.3m giving a pressure rise.

The last two loops in the outlet coil are spilling. The last plug is considerably longer than the one in loop 9 since the last plug is spilling but is not recieving spillage itself. This allows the leading edge of the plug to be depressed and so increase the head difference substantially: see Figure 11.2(a).

11.2.3 Comparison of Characteristic Curves

Examples of the comparison of calculated and measured values for the 0.915m diameter drum are shown in Figures 11.3 to 11.5. In all three curves the match is good. Unfortunately it was not possible to investigate suction heads of less than 1.7 m on this rig so no comparisons could be made below
Figure 11.3 Characteristic Curve For 32mm Dia Inlet Pipe

Figure 11.4 Characteristic Curve for 38mm Dia Inlet Pipe
Figures 11.3 and 11.4 highlight two factors in the behaviour of the pump. The dosing flow rate is basically controlled by the difference between the capacities of the loops on the inlet and outlet coils. In these cases, the drum diameters are the same so that it is the difference in the helical pipe sizes that govern the flow rate. In fact, the effective drum diameters are slightly different on Figure 11.3, this is because the effective drum diameter is the sum of the drum diameter and the outside diameter of the helical pipe.

The pump shown on Figure 11.3 has a 32mm diameter pipe on the inlet coil whereas the pump in Figure 11.4 has a 38mm. In both cases, the outlet coil pipe diameter is 50mm. The ratio of the differences between the inlet and outlet pipe cross-sectional areas for the two pumps is 1.4 and this represents the ratio of the vertical ordinates on the graphs, approximately. The pump in Figure 11.4 has only six loops on the inlet coil compared with ten for the pump in Figure 11.3. This has the effect of reducing the maximum suction lift from 5.4m to 4.5m. Increasing the number of loops on the outlet coil, in this case, would have no effect on the maximum suction lift since the capacity to lift is limited by the inlet coil.

In basic terms, the dosing rate is governed by the difference between the capabilities of the two coils to move water and the applied suction head. Eventually, one of the coils will reach its limit to support a suction lift and the pump will cease to function. The water/air ratio in an outlet loop will be higher than in the inlet loop because of the addition of dosing liquid and so the maximum suction lift of each coil will be different. It cannot be assumed, therefore, that an equal number of loops on each coil will equalize the two maximum potential lifts.

Figures 11.4 and 11.5 shows a pump at depths of immersion of 40% and 28%
Figure 11.5 Characteristic Curve for Low Depth of Immersion

![Diagram](image)

Figure 11.6 Characteristic Curve for 0.47 Dia Drum

![Diagram](image)
respectively. In both these cases bubbling occurs in the inlet coils and the additional air due to the reduced depth of immersion causes a bubbling failure at a lower suction head and, though it is difficult to gauge from the figures, it also increases the gradient of the straight line. The decrease in the depth of immersion also causes a small flow reduction for the same suction head.

Figure 11.6 shows the results of tests carried out on a drum of diameter of 0.47. Unfortunately most of the tests undertaken with this pump had to be discarded since the rotary joint was found to be leaking after the end of the tests. This was only discovered because the calculated results did not tie up measured results. The joint was then checked and the leak was found. The joint was inspected at the start of the tests and it is these few early runs that are shown here. It is very difficult to know when a rotary joint on the suction pipe is leaking. With a lift pump washing up liquid will display the bubbles of leaking air, but when the air is being sucked in, the detection is much more difficult. We checked and renewed the 'O' rings at regular intervals and we used shaving cream and a large amount of patience to check for leaks, but in this case, we failed to detect them. Checking the suction pressure loss over time in a static pump can give an indication of a leak, but some leaks only manifest themselves when the pump is rotating, so this method is not certain to find all leaks.

The pump with the 0.47m. drum diameter had one coil wound over the other hence both the helical pipe diameter and effective drum diameter are both greater on the outlet coil. With this pump we also managed to reduce the suction head down to zero which was not possible with the larger drum diameter.

On Figure 11.6 the depth of immersion was a few millimetres over 50%, this meant that the water plugs close to the boundary between bubbling and spilling when they had sufficient rotation. In the theoretical calculations on
the inlet coil, because the depth of immersion was so close to 50%, the spilling region only occurred in one loop then it switch quickly to the spilling-bubbling behaviour. Up to 1.5m suction head, no spilling or bubbling occurred in the loops on the inlet coil. As the head approached 2m, the spilling-bubbling region was established and it started to move towards the inlet. At the 2m head the spilling-bubbling region covered half the loops. The theoretical failure on this pump came in the outlet coil. Except at heads below 0.5m. spilling was occurring in the outlet coil. At a head of 1.5m eight out of the ten the loops were spilling. Just before failure occurred, a bubbling region was established at the outlet and this moved rapidly towards the inlet. It seems very likely that this caused the failure.

The description of the behaviour mentioned in the last paragraph is based on the theoretical results. Again, because of the suction head, the strengthened pipe was only semi-translucent and the water levels could only be seen with great difficulty. From the observations that the levels in the inlet coil appeared to take a similar form to the theoretical predictions. Most of the loops in the outlet coil appeared to be spilling at the higher suction heads, again this agrees with the theoretical results. At greater depths of immersion it did appear that loops were spilling and bubbling at the same time.

The measured characteristic curve on Figure 11.6 appears to be much more rounded than the characteristics shown previously and also the calculated curve with which it is compared. Certainly the gradient of the measured curve is steeper than the calculated one, though, the differences between the two lie within the experimental errors. With the limited amount of data available for this drum size, it is difficult to suggest causes for the roundness of the curve unless a small leak was reducing the dosing flow rate.
Figure 11.7 Variation of Flow Rate with Speed, $H_{suc}=2.0\text{m}$

Figure 11.8 Variation of Flow Rate with Speed, $H_{suc}=3.5\text{m}$

Data as for Fig. 11.4
11.2.4 Effects of Speed Change

As with the Lift Pump and the Single Coil Suction Pump, increasing the speed reduces the maximum head difference the pump can sustain and it also reduces the flow rate. Figures 11.7 and 11.8 show the effect of speed on flow rate for a low head and at a higher head. In each case there is a linear decline, approximately, in the flow rate as the drum speed increases. It seems likely that this decline is associated with an increase in the suction head due to rise in dynamic losses. It was hoped to measure this increase, but because the inflow of dosing liquid into the pump occurred over a short part of the drum cycle, the dynamic losses were masked by the unsteady state. A similar situation occurred with the single coil suction pump but there was much greater damping of the pressure fluctuations.

It was also not possible to show experimentally that the maximum pressure difference sustainable across the pump was lessened by increasing the drum speed. This was because of the steep gradient on the characteristic near to failure made it difficult to define this part of the curve.

11.3 DOSING PUMP

11.3.1 Introduction

The dosing pump is a small scale version of the double coil suction pump. It is called the dosing pump because it is capable of introducing a small amount of liquid into a larger body of liquid. This will be discussed further in the next chapter.

Only one size of pump was tested in the laboratory and this was run at one speed only because of the amount of time required for each experiment.
11.3.2 The Experimental Rig

The rig used was a scaled down version of the one shown in Figure 10.1 except that the system was not closed in that the overflow from the holding tank went to waste and the dosing liquid, water in this case, was taken from a measuring cylinder. Hence the pump was subjected to a falling suction head in most cases.

The greatest problem in carrying out these experiments is associated with flow measurement. The flow rate was only about 2.5 cc/min and most conventional flow measuring techniques would not cope with this low flow. The measuring cylinder acting as the container for the dosing liquid (the sump in Figure 10.1) was placed on an electronic balance measuring to an accuracy of .01g. The pump was started and the weight of the cylinder and its contents was measured at regular time intervals (normally every minute). A fixed measuring scale was used to measure the suction head. The change in this reading could also be checked against the loss of water from the cylinder.

Each experiment lasted for most of a working day and so a video recorder was used to monitor the experiment and the results could be read off the screen at a later stage more quickly.

The suction head was varied from zero to the maximum of the pump could sustain.

In one set of tests, the weight of water in the measuring cylinder was allowed to drop by 5 grams. This amount of liquid was poured back into the measuring cylinder and the time taken for the pump to remove the 5 grams of water was measured. This test was repeated a number of times at different suction heads. The aim of test was to achieve a constant head, although the head did vary, it was only over a small range, less than 1%.

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Figure 11.9 Characteristic Curve for Doser

Figure 11.10 Characteristic Curves for Doser
11.3.3 The Calculated and Measured Results

The tests carried out in the laboratory are limited to one pump configuration, the only variable being the suction head. Figure 11.9 shows the measured results for the dosing rate as the suction head varied from 6cm to 27cm. These limits were set by the range the electronic balance (max reading 300g). For reasons of clarity, only alternate data points have been shown. The experimental errors appear to be high but they only represent a variation of 0.03g. This is three times poorer than the accuracy of the balance (reading to 0.01g) but there was a slight lag on the electronic readout after a weight increase and the pulsating flow in the pump added to the errors.

Below a suction head of 10cm the flow rate rises significantly but between 13 and 25 cm there is only a slight decrease in the flow rate. A line of best fit in this region suggests that it decreases less than 0.1%, though the variation from reading to reading is about 2%. The three further runs with this pump shows very similar curves, and an apparent random variation of the data points.

Is this variation in the measured flow rates a true flow variation or just random experimental errors? Until a more accurate method of measuring the flow rate has been devised this question cannot be answered. My feelings are that at these very low values of weight changes they are variations due to experimental errors and not changes in the pumping rate. No coil pump we have tested has shown unpredictable, unsteady flow rates over a time period greater than one drum cycle, though none have used such small diameter pipes.

Figure 11.10 shows the same results related to origin. Superimposed on these results are a number of 'steady state' readings, including one for a zero flow rate. In this case, the pump was allowed to lower the level in the measuring
Figure 11.11 Characteristic Curve for Doser
cylinder until the flow was reduced to zero. At this point the pump was still generating a suction head, but it was only just sufficient to hold the static water column in the suction pipe. On this figure are also shown the calculated values but they will be discussed a later in this Section.

Figure 11.11 shows the results of the second run with the steady state data superimposed, here the falling head and state values show a trend very similar to Figure 11.10..

Returning to Figure 11.10, the fit of the calculated values to the measured values cannot be considered as good. The gradient of the theoretical line is steeper than the measured one. The agreement between the two sets of results when the suction head falls away and when value for the maximum suction head (at zero flow) is good.

Why is the agreement between these two sets of results poorer than results from the larger drums? Firstly, at this scale, the errors in the pipe diameter are critical. In the original calculations, both diameters were assumed to be 6mm since this was the nominal pipe diameter when bought. This produced a curve which was vertically displaced downwards by 0.5cc/min from the present one which is based on the inlet pipe being 5.5mm in diameter. An error of 0.5mm can change the theoretical flow rates by 20%. On discovering this problem, both pipes were remeasured very carefully and we estimated that the diameter for the inlet pipe was of 5.5mm whilst the outlet pipe was still 6mm. In fact, we found that the inlet and outlet pipes had been taken from two different rolls of pipe.

It could be argued that the diameter of a stiff plastic pipe could not be measured to within 0.5mm of a millimetre and I would have to agree to a certain extent. The inlet pipe certainly seemed to have a smaller diameter than the outlet pipe and the reduction of 0.5mm in the value used was a
reflection of this. With hindsight we should have been much more careful with the choice of pipe material.

The other factor affecting the discrepancy between the two curves in Figure 11.10 is concerned with the calculation of the rotation of the last plug in the inlet coil (needed for the outlet coil calculations). The theoretical level differences which are directly related to the plug rotations produce a much shallower curve than the observed ones. Though the actual levels were not measured, only photographed, the observed plug rotations increased significantly towards the outlet whereas the calculated values did not. An example of the theoretical differences for 3cm suction are as follows:

<table>
<thead>
<tr>
<th>Loop No</th>
<th>Level Difference (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>57.6</td>
</tr>
<tr>
<td>2</td>
<td>58.6</td>
</tr>
<tr>
<td>3</td>
<td>60.8</td>
</tr>
<tr>
<td>4</td>
<td>64.1</td>
</tr>
<tr>
<td>5 (Outlet)</td>
<td>68.4</td>
</tr>
</tbody>
</table>

This represents a shallow curve which it must be when considering small theoretical air plug expansions. The greater change in levels between one loop and the next on the laboratory pump is likely to be due to a combination of friction and surface tension in the small diameter which are not considered in the theoretical model. Nakorgakov(1986) carried out work on bubbles of air rising in small diameter vertical pipes. Surface tension had a strong influence on the behaviour of the fluids, but because the pipes on the coil were not vertical, I could utilize or adapt Nakorgakov's results for this situation.

Preliminary calculations suggest that an increase in level differences in the loops due to, say, surface tension would increase the last plug rotation in the inlet coil which would, in turn, lessen the gradient of a line drawn through
the calculated points shown on Figure 11.10.

11.4 COMMENTS

The use of the theories developed for the Lift Pump and the Single Coil Suction Pump has lead to the theory for the Double Coil Pump which agrees well with the measured results. The theoretical predictions work well both at the failure points and the intermediate values. When the theory is applied to pumps with very small diameter helical pipes then, errors do occur in the theoretical predictions. More work on understanding the effects of small pipe diameters on the workings of the coil pump is required to clarify why these errors occur.

Designing a double coil suction pump to suit a particular flow rate and suction head is a cumbersome process with the need of much computer processing time. The process is basically one of trial and error though this can be made much more effective with a little experience.
CHAPTER 12

PRACTICAL ASPECTS OF THE DOUBLE COIL SUCTION PUMP

The large diameter double coil pump can be used to lift liquids containing solids in a similar way to the single coil suction pump, only in this case the pump is self priming. A double coil pump with both coils empty will prime itself, provided the pump is not operating too close to its maximum suction lift. This pump has the attraction is that it mechanically simple and provides a clear passage to the flow, but it does have the disadvantage of being bulky. Though this pump has the advantage of priming itself, it does have a lower pumping rate compared with a single coil pump having the same physical dimensions.

The small dosing pump could be used to introduce small amounts of chemical into a body of liquid. An example of this is the disinfection of water with hypochlorite solution where, for example, the pump used in the laboratory experiments could be used to dose a water supply for about 2000 people.

The flow rate of the dosing chemical will remain virtually constant (less than 1% change) over a level range of 200mm. A tank of plan dimensions of 0.5 by 0.5metres would last approximately 20 days. It is also possible to turn the pump with a water wheel (in a similar manner to the stream powered pump) and, therefore, an external power source would not be required. The water wheel could also ensure that the dosing rate would be approximately proportional to the channel flow rate, assuming it is variable. With this in mind a patent was taken out on this application (Mortimer 1986).

Another possible use for the dosing pump is that of introducing a flocculating agent in sediment laden water before it enters a settlement lagoon.
Care would have to be taken with any dosing pump to ensure that the helical pipe diameters are formed within a fine limits. With this precision, the hydraulic behaviour of the pump can be more accurately predicted. It is possible that these coils can be made from coated metal or fibre glass.

As a useful addition, it may be worth adding an adjustable scoop on the entrance to the inlet coil to allow the dosing flow rate to be varied within moderate limits.
The theory developed for the lift coil pump is based on the simple concept of multiple manometers. This basic concept has to be developed to a considerable degree, however, to provide a theoretical prediction for the pump which is valid under a wide range of operating conditions. Only when rotational speeds become high does the theory need correction factors to allow for dynamic losses.

Most investigators have ignored the spilling and bubbling phenomena in coil pumps but both are very important factors in the pump's behaviour, especially, when the pump is working near its limit. It could be argued that a coil pump that is not spilling or bubbling is under-utilized.

The theory developed in this thesis for the lift coil pump can be used to provide design guidelines for the users of the pumps in developing countries. These people will not require the detailed information that the theory can provide, but the design guidelines they use can be derived from the theory with a much greater degree of confidence.

The lift pump theory becomes essential when the pump is to be adapted for specialist uses such as the Coil Treatment Unit. For example, the computer programs for predicting the hydraulic behaviour of the pump were used extensively in the design stage of the unit and, in practice, it has behaved in a very similar manner to the theoretical predictions.

The stream powered pump we designed at Loughborough generated a world wide interest when we published our worked. We received over one hundred letters from all parts of the world asking for details of the pump and
many variations have been built in the Far East, Africa and South America.

Looking at further applications of the lift pump theory, it would be rewarding to apply the theory to the spiral version so that the advantages of each type can be clearly identify. The problems of incorporating the cumbersome geometric equations into the theory, however, will make the process very tedious, though not impossible. The spiral pump has the attraction of compactness and so it could be well worth while adapting the theory developed in this thesis for the spiral pump.

As far as I am aware, no one has devised or built a suction coil pump. The field of application for this pump has limited use in the field of low technology but there are specialist uses such as the dosing pump and sewage lifting, where the pump's advantages can be fully used.

The theoretical predictions of the pump's behaviour are very satisfactory over a wide range of conditions. More work needs to be carried out, however, on the falling limb of the characteristic curve of the single coil suction pump at low air flow rates. The present theory is unable to predict the aspect of the pump's behaviour well, though this is felt to be a only a minor disadvantage.

Work on the coil pumps has not finished at Loughborough. Talks with manufacturers regarding the commercial development of the Doser and the Coil Treatment unit are being carried out at present (Summer 1987) and work has started on a new configuration for the coil pump which could find use in sewage treatment. The possibilities for new pipe and drum configurations are numerous, but the development of new coil pumps which have a useful place in water engineering today is much more difficult to achieve.

In assessing the potential and usefulness of the coil pumps, they should not be compared with rotodynamic and reciprocating pumps since the coil pumps cannot compete on size or output, but they do have a niche, namely, low cost.
pumping and specialist uses such as sewage treatment and pumping sewage and dosing of chemicals.

Without an adequate knowledge of the coil pump's behaviour, its potential cannot be exploited. This thesis has provided a good part of that knowledge and has realized some of the pump's potential.
ACKNOWLEDGEMENTS

I would like to thank the two technicians involved in this work for the skill and enthusiasm they injected into these projects, they are Lew Benskin and Mick Longmire. Without their high level of workmanship the investigations would not have run so smoothly.

Thanks also to John Pickford for looking through the thesis and to my wife for proof reading. Finally I would like to acknowledge the effect of my children, Kate, Helen and James had on this thesis - without them it may have been finished much sooner.
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APPENDIX 1

DETERMINATION OF POWER TERM IN THE GAS EQUATION

THEORETICAL CONSIDERATIONS

The equation used to predict the volumes changes of the air plugs is

\[ PV^k = \text{Constant.} \]

See Section 4.4.3.

Theoretically, \( k \) should lie between 1 and 1.4. If the process of gas expansion or compression is so slow that the temperature of the gas remains constant with time then it is termed isothermal and the \( k \) value should be 1. If, however, the process is adiabatic then there would no loss or gain of heat during the period of expansion or contraction and, in this case, the \( k \) value would be 1.4.

The actual value of \( k \) will depend on the ability for heat to transfer into, or out of, the air plug and this is very difficult to assess practically.

EXPERIMENTAL CONSIDERATIONS

Annable(1982) carried out a series of experiments by increasing the pressure in a static plug of air trapped in a loop of a 25mm diameter pipe.

His data yielded a \( k \) value of 0.904 which is highly suspect. It is difficult to explain why he should obtained a value such as this, since the experimental errors alone could not account for this low \( k \) value.

A second approach involves using the experimental data gathered on the
liquid level differences in the coils and determine the air plug lengths and the associated air pressures. Once again this is a static test and so the heat transfer mechanism will be different to the air plugs moving through the loop.

Eight cases were picked random from the test results. A regression analysis on the air plugs lengths and pressures in the spilling region gave a \( k \) value varying from 1.07 to 1.17. The average was 1.15 with a standard deviation of 0.048. A regression analysis in the non-spilling region gave an average \( k \) value 1.33 with a standard deviation of 0.22. The higher standard deviation in the second case was due to the restricted number of non-spilling loops in any pump and the random variation of levels in redundant loops. The reason for dividing the analysis into spilling and non-spilling regions was that the less predictable nature of the non-spilling region upset the accuracy of the overall results.

OTHER FACTORS

It has been assumed that the pipe in which the air plugs expand and contract is rigid, this is not the case with a plastic pipe material. As the pressure increases so the diameter increases and this will effectively decrease the air plug length. This will have the effect of increasing the \( k \) value.

It is possible to find the theoretical modified \( k \) value taking into account the effects of the pipe volume changes. The solution of the resulting equation requires a trial and error process using a computer. Taking one typical example from the laboratory results, the effects of the pipe volume increases the \( k \) value by .05 (the calculations are not shown). In effect, this is an equivalent \( k \) which includes the effect of elasticity of the pipe within the compressibility of the air. It is felt that increase in \( k \) is not major.
influence on the calculations.

COMMENTS

What value should be used in the analysis? All the analyses in this Thesis use a value of 1.15 which is the average of the results within the spilling region. More laboratory work is needed in order to understand and predict compression/expansion effects within the air plugs.

Fortunately a sensitivity analysis on a number of lift pump cases shown in Chapter 5 suggests that a variation in k value within the range 1 to 1.4 does not affect the level difference curve significantly nor the characteristic curves for the pump.
APPENDIX 2
SOLUTION BY BISECTION METHOD

This method is used to find a solution to an equation which cannot be solved directly. In all the cases considered in this thesis, one unknown parameter P in the equation is varied until the equation is satisfied.

The following need to be known.
(a) The equation in a form such that the parameter P, which is required, is on one side of the equation A=f(P), but still associated with other variables and constants.
(b) An upper limit for parameter P. If this is unknown it can be set to a very high value.
(c) A lower limit to parameter P.
(d) The trend by which f(P) varies with A. If f(P) increases does A increase or decrease.

Initially P is set at a mean value of (Upper limit of P - Lower Limit of P)/2. f(P) is calculated and, if f(P)>A and A increases as f(P) increases, then the lower limit is increased to the current value of P. If f(P)<A the upper limit is made equal to the current value to P. This is repeated until the change in P between iterations is within the required tolerance.

Tests showed that the method of successive bi-section needed, in most cases, the fewest number of iterations to converge on the solution, within the specified tolerances in most cases.
APPENDIX 3
PARAMETERS AFFECTING SPILLING

INTRODUCTION

Attempts were made to develop a theory to predict the mechanism of spilling based on a balance between gravitational and dynamic forces of the flow over the crown, this proved unsuccessful.

The sections below attempt briefly to identify the parameters that will influence spilling and the spilling angle $\Omega_{sp}$ by considering simplified cases.

THE STATIC CASE

Consider Figure 3A.1 where water is spilling over a circular stationary drum. The flow rate over the drum is dependant on the velocity $V$ and the cross-sectional area of flow. $V$ is governed by $z$, the depth of water over the crown which, in turn, will be related to $\Omega_{sp}$, assuming a horizontal water surface.

From this diagram, $\Omega_{sp}$ decreases, if,
(a) the flowrate increases,
(b) the cross-sectional area of flow increases with the same flow rate.
(c) the drum diameter increases.

It is assumed that the line AA on Figure 3A.1 extends from the centre of the drum to the intersection of the water surface and the centre line of the helical pipe.
THE DYNAMIC CASE (Simplified)

If the drum is now rotating clockwise then the shear stress causing by the rotating pipe wall will increase fluid velocity $V$ at the crown. If the flow rate remains constant, then $z$ will decrease and $\Omega_{sp}$ will increase.

Figure 3A.2 shows the forces on an element of water approaching the crown of the drum. The resultant of the viscous and gravitational forces will produce an inclined water surface which will become steeper as the drum speed (and hence the viscous force) increases. An inclined water surface will increase $\Omega_{sp}$ if the flowrate remains constant and this agrees with the statement in the previous paragraph that an increase in speed increases $\Omega_{sp}$.

COMMENT

The discussion above only provides indicators to the behaviour of $\Omega_{sp}$ with changing conditions. It is not possible to take detail measurements of $\Omega_{sp}$ on a moving coil pump. To isolate the spilling effect on a stationary, or semi-stationary rig to observe the behaviour in detail is also very difficult to achieve.
Figure 3A.1 Spilling over a pipe crown

Figure 3A.2 Forces on Element of Water