Condition monitoring of internal leakage in modern water hydraulic cylinders using acoustic emission

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CONDITION MONITORING OF INTERNAL LEAKAGE IN MODERN WATER HYDRAULIC CYLINDERS USING ACOUSTIC EMISSION

CHEN PING

School of Mechanical and Aerospace Engineering

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Abstract

This research aims to develop a sensitive method for condition monitoring of internal leakage in a modern water hydraulic cylinder by means of acoustic emission (AE). Modern water hydraulics, which uses tap water as a hydraulic fluid, has been given increasing attention in recent years due to its inherent advantages over oil hydraulics, e.g., no environmental pollution, reduced product contamination, no fire hazard, low operating cost, etc. However, in order to benefit from all these advantages, some problems remain to be addressed. One of the most common problems of water hydraulics is leakage because the viscosity of water is low as compared with that of hydraulic oil. So far, there is no suitable method for condition monitoring of internal leakage in water hydraulic components. In this thesis, the author developed a new AE-based method for condition monitoring of internal leakage in a water hydraulic cylinder.

Specially-designed experiments were conducted to study the characteristics of internal-leakage-induced AE signals. Various parameters were extracted from AE signals and the correlations with the internal leakage rate were studied in detail. Results showed that the AE rms value is a most desirable parameter to represent quantitatively internal-leakage-induced AE signals. Also, comprehensive numerical simulations were performed to investigate the characteristics of turbulence generated by internal leakage in a water hydraulic cylinder. A parameter, called mass-weighted integral of turbulent kinetic energy (MITKE), was defined to measure the intensity of turbulence generated by internal leakage. Using the dimensional analysis method, a new quantitative AE model was then established to describe the relationship between the AE rms value and the internal leakage rate. Based on this AE model, two condition monitoring strategies, i.e., static condition monitoring strategy and dynamic

condition monitoring strategy, were proposed. Experimental results showed that using these strategies, the seal integrity and internal leakage rate could be predicted with sufficient accuracy.

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List of Symbols

A_s	Cross-section area of score on piston seal (m^2)
A_l	Piston area of a hydraulic cylinder (m^2)
A_2	Piston area minus rod area (m^2)
a_{score}	Side length of score on piston seal (m)
D_c	Diameter of chamber of hydraulic cylinder (m)
D_s	Equivalent diameter of score on piston seal (m)
D_{score}	Diameter of score on piston seal (m)
E_f	Average energy of AE signal measured in the frequency domain (V^2)
E_T	Mass-weighted integral of turbulent kinetic energy ($kg \cdot m^2/s^2$)
F	Load on the rod of a hydraulic cylinder (N)
F_f	Friction force (N)
ΔF	Frequency sample interval (Hz)
f_s	Sampling rate (Hz)
I_i	Three components of turbulence intensity
K	Stress intensity factor ($Pa \cdot m^{1/2}$)
k	Turbulent kinetic energy (m^2/s^2)
L	Characteristic length of flow (m)
M	Mass of leaking water (kg)
M_p	Magnitude of dominant peak in power spectral density (V^2/Hz)
N	Number of sampling points
N_{AE}	AE count
$\dot{N}_{AE}$	AE count rate (s^{-1})
n	Number of scores on piston seal
n_c	Critical number of scores on piston seal

P_1	Pressure at the piston end of a hydraulic cylinder (Pa)
P_2	Pressure at the rod end of a hydraulic cylinder (Pa)
$P[k]$	Power spectral density (PSD) of a discrete signal $x[n]$ (V^2/Hz)
p	Pressure (Pa)
$\bar{p}$	Mean of pressure p (Pa)
Q	Flow rate (m^3/s)
Q_1	Flow rate at the piston end of a hydraulic cylinder (m^3/s)
Q_2	Flow rate at the rod end of a hydraulic cylinder (m^3/s)
Q_c	Critical internal leakage in a hydraulic cylinder (m^3/s)
Q_e	External leakage from a hydraulic cylinder (m^3/s)
Q_i	Total internal leakage in a hydraulic cylinder (m^3/s)
q_i	Internal leakage through a score on piston seal (m^3/s)
Re	Reynolds number
R_{score}	Radius of score on piston seal (m)
r	Correlation coefficient
T	Sampling period of a discrete signal $x[n]$ (s)
t	Time (s)
U	Velocity (m/s)
$U(\mathbf{x}, t)$	Vector of fluid's velocity (m/s)
$\bar{U}(\mathbf{x}, t)$	Mean of $U(\mathbf{x}, t)$ (m/s)
U_i	Three components of fluid's velocity $U(\mathbf{x}, t)$ (m/s)
$\bar{U}_i$	Mean of U_i (m/s)
$\mathbf{u}(\mathbf{x}, t)$	Fluctuation part of fluid's velocity $U(\mathbf{x}, t)$ (m/s)
u_i	Three components of $\mathbf{u}(\mathbf{x}, t)$ (m/s)
$\bar{u}_i$	Mean of u_i (m/s)

$\overline{u_i u_j}$	Reynolds stress (m^2/s^2)
V_{in}	Mean fluid velocity at inlet (m/s)
V_s	Mean fluid velocity at the outlet of score (m/s)
V_{rms}	rms value of AE signal (V)
$\mathbf{x}$	Position vector (m)
x_i	Three components of a position vector $\mathbf{x}$ (m)
$x[n]$	A discrete signal (V)
x_{rms}	rms value of a discrete signal $x[n]$ (V)
$X[k]$	Discrete Fourier transform (DFT) of a discrete signal $x[n]$ (V)
Π_i	Pi terms
δ	Prediction interval (V)
μ	Absolute viscosity ($\text{N}\cdot\text{s}/\text{m}^2$)
ν	Kinematic viscosity (m^2/s)
ν_{eff}	Effective viscosity (m^2/s)
ν_T	Turbulent viscosity (m^2/s)
ω	Dissipation rate (s^{-1})
ρ	Mass density (kg/m^3)

CHAPTER 1

INTRODUCTION

1.1 Background

Hydraulics is defined as the technology that deals with the generation, control, and transmission of pressurized liquids for performing work. It has many attractive advantages, i.e., high power to weight ratio, ease and accuracy of control, flexibility and simplicity of design, and multiplication of force. Esposito [1] provided a detailed description on hydraulics technology. Backé [2] gave a review on the advantages and future development of this technology.

Originally, water was the only available fluid for hydraulics. In the early 20th century, mineral oil started to take the place of water due to its higher viscosity, superior lubrication and anticorrosion properties. This switch led to significant advancement in oil hydraulics in the last century. Today oil hydraulics is used in a variety of applications. In spite of the great success, oil hydraulics has some inherent disadvantages. Hydraulic oils are messy, and leakage of such oils results in environmental pollution and product contamination. In addition, most hydraulic oils can cause fire under high temperature. These have caused ever-increasing public concern, because environment, health and safety are critical issues in the world today. Consequently, water was reconsidered, which has many desirable properties, such as environment friendliness, fire proof, nontoxicity and low cost. This gave rise to modern water hydraulics, which uses tap water as a hydraulic fluid (hereinafter referred to as water hydraulics). A comprehensive description about water hydraulics could be found in Refs. [3-4].

New materials and designs have been introduced to water hydraulic components. In 1990s, hydraulic systems using tap water became commercially available. However, some problems with water hydraulics remain to be addressed. Therefore, to achieve optimal, reliable and safe operations, it is necessary to monitor the performance of a water hydraulic system. Once the system performance is reduced to an unacceptable level, some maintenance should be given. Some initial efforts at condition monitoring of water hydraulic systems have been reported [5-12].

One of the most common problems of water hydraulics is the relatively large internal leakage in water hydraulic components as compared with that of oil hydraulic components. This is due to the very low viscosity of water in comparison with that of hydraulic oil. It was reported that given the same leakage path, leakage rate of water is about 30 times that of hydraulic oil [13]. In addition, severe wear of moving parts due to the insufficient lubrication of water can also result in relatively large leakage. This leakage will degrade the performance of water hydraulic components. In the case of a water hydraulic cylinder, excessive internal leakage (piston-seal leakage) will reduce the piston speed, the permissible load and the motion-control precision. Moreover, with leakage, the cylinder cannot be locked in position, which may cause an unstable or unsafe situation. Therefore, it is important to monitor the internal leakage.

There are several methods traditionally used for condition monitoring of hydraulic systems [14]. However, although some attempts have been made at detecting internal leakage in hydraulic components as well as other industrial components, there are no effective methods to solve this problem. The difficulty arises from the fact that a leakage inside a component is often too weak a process to detect with traditional methods. For methods based on pressures or flow rates, they

are not effective because the characteristics of pressures and flow rates are not only dependent on local components but also affected by upstream components and even downstream components and therefore, the effect of internal leakage on pressure and flow rate could be drowned by effects from other components. Le et al. [15-16] investigated leakages in a hydraulic cylinder drive system. Pressures and flow rates were measured and input to artificial neural networks for leakage level prediction. Their results showed that internal leakage was more difficult to detect than external leakage and the neural networks could only detect internal leakage over 1.0 L/min (this value is considered to be high when compared with the full flow rate in the leakage-free case, i.e., 9.3 L/min and 7.1 L/min for extending and retracting, respectively).

For vibration-based methods, they can be useful in some applications but may encounter difficulties if the interference of background noise and mechanical vibration exists [17]. For temperature-based methods, although leakage in hydraulic components is accompanied by the transformation of hydraulic energy into heat, it does not always result in a discernible local hotspot because the heat could easily be dissipated by the heat transfer processes existing both in the hydraulic system and between the hydraulic system and the ambient. In addition, there are many leak detection techniques used for fluid transport systems [18], but they are usually suitable for monitoring external leakage only.

A promising method for the detection and estimation of internal leakage is acoustic emission (AE), which has been used to detect and locate leakage in various applications [19-25]. Acoustic emission is transient elastic waves that are generated by the rapid release of energy from local sources within a material [24]. The major advantage of AE-based condition monitoring methods is that AE signals are sensitive

to localized dynamic processes, while seldom affected by mechanical vibration and environmental noise due to their high frequency range (usually between 50 kHz and 1 MHz). This motivates the author to explore the capability of AE and develop a sensitive method to monitor small internal leakage in water hydraulic cylinders.

So far, using AE to detect external leakage, such as leakage from pipelines and pressure vessels, has achieved great success, whereas there is still a lot of work to establish an AE-based method to monitor internal leakage in hydraulic components. It is expected that monitoring internal leakage in hydraulic components is more complex than detecting external leakage due to the complex interior structures of hydraulic components and the existence of moving parts. Furthermore, due to the complexity of AE source mechanism and wave propagation, it is a challenge to develop a quantitative AE model. So far, researches on AE techniques found in the literature are usually experimental studies. Most of them are limited to detecting the existence and location of a leak. Nevertheless, some efforts have been made to estimate internal leakage rate in water hydraulic cylinders by means of AE. Chen et al. [8] conducted a detailed experimental study of AE signals generated by a water hydraulic cylinder drive system. They found that AE signals are sensitive to internal leakage in a water hydraulic cylinder, while not affected by background noise. Their results showed that AE is a promising technique for condition monitoring of internal leakage. In this research, the characteristics of internal-leakage-induced AE signals will be studied in detail and a quantitative model will be established to describe AE signals generated by the internal leakage in water hydraulic cylinders. Based on this model, strategies will be developed for condition monitoring of internal leakage.

1.2 Objectives

This research aims to develop a sensitive AE-based method for condition monitoring of internal leakage in a water hydraulic cylinder. It is focused on internal leakage smaller than 1 L/min. To be specific, the objectives of this research are as follows:

1. To investigate the characteristics of AE signals generated by internal leakage in a water hydraulic cylinder and determine an appropriate method to represent AE signals so that useful information about the source (internal leakage) can be conveniently extracted from AE signals;
2. To investigate the source mechanism of internal-leakage-induced AE signal and develop a method to quantitatively measure this AE source;
3. To develop a quantitative model to describe the relationship between internal leakage and AE signal thereby generated;
4. To develop new condition monitoring strategies, based on the quantitative AE model, to predict internal leakage in a water hydraulic cylinder.

1.3 Scope

In this thesis, internal-leakage-induced AE signals are studied in detail. Experiments are carefully designed to study the characteristics of various AE parameters, including parameters in the time domain and frequency domain. Their correlations with internal leakage rate are investigated so that a suitable parameter can be found to represent internal-leakage-induced AE signals.

Turbulence due to internal leakage is identified as a major AE source mechanism in a water hydraulic cylinder. A flow quantity, namely, the mass integral of turbulent kinetic energy (MITKE), is defined to quantitatively measure the

intensity of turbulence activities due to internal leakage. Numerical simulation is used to study the turbulent kinetic energy associated with internal leakage. With the help of numerical method, a dimensional analysis technique is used to model MITKE. A comprehensive study of the generality of the resulting model is given with regard to properties of water and geometry of leaking path.

An AE model is established to correlate the MITKE and AE signals. Based on this AE model, condition monitoring strategies are proposed to estimate internal leakage in a water hydraulic cylinder. The proposed strategies are also verified with experimental results.

1.4 Outline

This thesis is organized as follows. Chapter 1 presents the motivation as well as the objectives and scope of this research. Chapter 2 is a literature review on water hydraulics technology, condition monitoring techniques and AE techniques. Chapter 3 is devoted to AE signal characterization, whereby an appropriate parameter is determined to interpret internal-leakage-induced AE signals. Chapter 4 describes the characterization of turbulence as well as numerical modelling of internal leakage, whereby a suitable flow quantity is defined to measure the intensity of turbulence induced by internal leakage. In Chapter 5, a quantitative relationship between AE and internal leakage is established. In Chapter 6, condition monitoring strategies are proposed to estimate internal leakage in water hydraulic cylinders. Finally, conclusions and recommendations are presented in Chapter 7.

CHAPTER 2

LITERATURE REVIEW

In order to understand the unique characteristics of modern water hydraulics as compared with oil hydraulics, a review is given on this technology. Then, fundamentals of hydraulic cylinder are given. Third, this chapter gives a review on conventional condition monitoring techniques that have been applied to hydraulic systems and leakage detection. Finally, AE-based condition monitoring techniques are reviewed.

2.1 Modern Water Hydraulics

2.1.1 Introduction

Trostmann [26] gave a comprehensive description on water as a hydraulic fluid. Compared with hydraulic oils, water has many desirable properties, such as environment friendliness, fire proof, good heat-transfer capability, nontoxicity, low cost and abundant supply. Nowadays technology is more advanced than that in the early 20th century, which makes it possible to exploit these desirable properties and resolve the problems of water hydraulics.

In the past decade, new materials and new designs have been introduced to resolve the problems of water hydraulics, e.g., poor lubrication, severe corrosion and excessive leakage. Stainless steel, ceramics and plastics are widely used in water hydraulic components because of their desirable characteristics. Stainless steel and ceramics have high resistance to corrosion and erosion. Plastics and ceramics are usually used in contact surfaces of water hydraulic pumps and motors, because they

can form a very good tribological pair. In order to reduce external and internal leakage, water hydraulic components are usually designed with small tolerance in comparison with oil hydraulic components so that small clearances between moving parts can be achieved. Detailed description of material selection and component design for water hydraulic components can be found in [3, 27]. With the help of today's advanced technology, water hydraulics has made a success in some applications. Nevertheless, efforts are still required to improve water hydraulics technology so that comparable performance and lifetime to oil hydraulics can be achieved.

2.1.2 Advantages of Water Hydraulics

Using water as a hydraulic fluid gives water hydraulics unique advantages. The main advantages include no environmental pollution, reduced product contamination, no fire hazard. Other advantages include low long-term operating costs, no negative health effects, high hygiene level, etc.

a) No Environmental Pollution

Because hydraulic oil is dirty, and generally harmful and non-biodegradable, oil hydraulics has never overcome its inherent drawbacks, i.e., environmental pollution. It was reported that in Europe, 100 million litres of hydraulic oil per annum is not accounted for [28]. In addition, hydraulic fluids other than water used in the U.S. amounted to several million tons annually, and about 85% of these fluids eventually spilled from hydraulic systems [29], which is a serious problem on environmental protection. Increasing public awareness of environmental issues calls for effective measures to prevent the environment from pollution by hydraulic oil. In order to make pollution-free hydraulic systems, water is the best choice since it is friendly to the

environment. Water hydraulics poses no environmental pollution problems.

b) Reduced Product Contamination

When hydraulic systems are used in a production line, it is always a concern whether the quality of products are affected when hydraulic fluid spatters on them. This is an issue of product contamination. It is important for many industries, especially food, beverage, pharmaceutical and chemical industries. The use of conventional hydraulic fluids, even those designated nontoxic, cannot solve these problems to a satisfactory extent. However, for water hydraulics, product contamination is much reduced, since water spatters are usually considered as harmless. An application of water hydraulics to a mass-production packaging machine in beverage industry was reported by Sorensen [30].

c) No Fire Hazard

It is imperative that a hydraulic fluid does not initiate or support fire. Special hydraulic fluids have been designed to meet fire-resistant requirements, but they usually have some drawbacks, e.g., limited operating temperature ranges and poor compatibility with common materials [1]. In contrast, water is a cheap fire-proof fluid by nature. Water hydraulic systems can meet the requirements of many hazardous applications, e.g., coal mines, hot metal processing equipment, and marine fluid power systems.

Today, water hydraulic systems can be found in industries, especially where fire risks exist, a high level of hygiene is required, or environmental concerns are of importance. Potential applications of water hydraulics were reported by Trostmann [3] and Backé [13]. Table 2.1 provides some applications of water hydraulics as well as its main advantages.

Table 2.1 Applications of water hydraulics

Applications	Main Advantages
Food & beverage industry, pharmaceutical industry, chemical industry	Good product compatibility, easy cleaning, high level of hygiene
Hot metal processing equipment, e.g., forging, moulding and welding machine	No fire hazard, good product compatibility
Off-shore equipment, underwater equipment, mining equipment	Protection of environment, no fire hazard
Nuclear power industry	No contamination, no fire hazard, non-radioactive fluid

2.2 Fundamentals of Hydraulic Cylinder

2.2.1 Working Principle of Hydraulic Cylinder

Hydraulic cylinders can be categorized into two types: single-acting cylinder and double-acting cylinder. As the water hydraulic cylinder studied in this thesis is of the double-acting type, the following description covers double-acting cylinder only. Figure 2.1 shows schematically a double-acting hydraulic cylinder. It consists of a piston inside a cylindrical barrel. Attached to one end of the piston is a rod, which extends outside one end of the cylinder. At each end of the cylinder, there is a port for the entrance or exit of hydraulic fluid. The cylinder can exert a force in both the extending direction and the retracting direction. Figure 2.2 shows the flow in a hydraulic cylinder.

Figure 2.1 Double-acting hydraulic cylinder

Figure 2.2 Flow in a hydraulic cylinder

It can be seen that a hydraulic cylinder may suffer from leakage at the piston seal and at the rod seal, i.e. internal leakage Q_i and external leakage Q_e . Considering the principle of flow continuity gives

$$Q_1 = A_1 \cdot U + Q_i \tag{2.1}$$

and

$$Q_2 = A_2 \cdot U + Q_i - Q_e \tag{2.2}$$

where

Q_1 and Q_2 are flow rates at the two ports, respectively;

A_1 is piston area;

A_2 is piston area minus rod area; and

U is piston velocity.

Eqs. (2.1) and (2.2) are valid for both extending and retracting strokes if the sign convention of Figure 2.2 is maintained, that is to say, when the actual direction of a

variable is opposite to that shown in Figure 2.2, this variable has a negative value.

Figure 2.2 also shows a force balance at the piston and rod, given as

$$P_1 A_1 - P_2 A_2 = F + F_f \quad (2.3)$$

where F is load force, F_f is friction force, P_1 is pressure at the blank end of the cylinder, and P_2 is pressure at the rod end of the cylinder.

2.2.2 Internal Leakage in Hydraulic Cylinder

Figure 2.2 shows the internal leakage (Q_i) in a hydraulic cylinder. This internal leakage will degrade the performance of a hydraulic cylinder. It will slow down the speed of a hydraulic cylinder. From Eqs. (2.1) and (2.2), the following can be derived:

$$U = \frac{Q_1 - Q_i}{A_1} \quad (2.4)$$

$$U = \frac{Q_2 - Q_i + Q_e}{A_2} \quad (2.5)$$

The above equations indicate that the occurrence of internal leakage Q_i will reduce the extending or retracting speed of a hydraulic cylinder. Moreover, a hydraulic cylinder suffering internal leakage cannot be locked in position, which may result in an unstable or unsafe situation. Another problem is that the motion-control precision of a hydraulic cylinder is much affected by internal leakage. It can be seen from Eqs. (2.4) and (2.5) that without the occurrence of internal leakage Q_i as well as external leakage Q_e , velocity U is simply controlled by flow rates Q_1 or Q_2 . However, if internal leakage Q_i exists, the motion control of a hydraulic cylinder can be complex.

In practice, internal leakage is due to the failure of piston seals. The major cause of seal failure is contamination of hydraulic fluid with solid particulates [31]. Hobson [32] found that solid particulates led to damage of reciprocating hydraulic seals, in the

form of scores extending axially over the sealing surface of the seal. These scores were created by the particulates being forced into the sealing area during the reciprocating motion. Hobson also concluded that a seal with no score damage would not lead to leakage.

As compared with oil hydraulics, water hydraulics is subject to relatively large internal leakage. The viscosity of water is about 1/30 of that of hydraulic oil, which means given the same leaking path, the leakage rate of water is about 30 times that of hydraulic oil. Therefore, it is required to monitor internal leakage in water hydraulic cylinders. When the internal leakage exceeds an acceptable level, appropriate maintenance work should be provided.

2.3 Condition Monitoring of Hydraulic Systems

There are many methods for condition monitoring of hydraulic systems, as described by Watton [14]. All the methods involve analyses of some data measured from a hydraulic system. These data include flow rate, pressure, temperature, vibration, AE, etc. In the following, condition monitoring techniques for hydraulic systems are reviewed with regard to various data used.

2.3.1 Flow Rate Method

Flow rate is one of the most important signals used to monitor hydraulic systems. It is commonly used for measuring the steady-state condition. When this is used for condition monitoring of a hydraulic system, normal or abnormal conditions of the hydraulic system could be determined by whether or not the flow rates at some critical points are within a normal range. For example, if the flow rate at the inlet port of a hydraulic cylinder is lower than normal (the cylinder moves more slowly than normal), it can be concluded that there are some problems with the system, e.g., a

worn or damaged pump, a defective pressure relief valve, or faulty flow control valves, etc.

The main drawback of the flow rate method is that the flow rate at one hydraulic component is always influenced by the flow rates at other hydraulic components. Therefore, it is common that the flow rate at every critical point in a hydraulic circuit is measured simultaneously, and then, all the measured flow rates are processed systematically to reach a decision about the condition of the hydraulic system. This process is often accomplished by a complex expert system or fault tree analysis [33]. Thus, it would increase the complexity and costs of a condition monitoring system. Besides, flow meters need to be fitted permanently in a hydraulic circuit, which would produce a pressure drop. Correct size of flow meters must be selected to reduce as much the negative effects as possible.

In the case of a hydraulic cylinder drive system, limited stroke and reciprocating motion results in dynamic flow in a hydraulic cylinder. From this point of view, flow meters giving a steady-state measurement must be combined with other types of sensors to accomplish the task of condition monitoring of a hydraulic cylinder. Furthermore, small internal leakage in a hydraulic cylinder induce no significant changes in flow rates, which make it difficult to detect such internal leakage by the flow rate method.

2.3.2 Pressure Method

Pressure measurement is also an important means of monitoring a hydraulic system. It is usually measured by a pressure gauge or a pressure transducer. Generally, the former provides steady-state pressure information, while the latter is capable of measuring the dynamic condition as well as the steady-state condition of a hydraulic system, and hence more expensive than the former.

A great deal of information can be obtained from the way a hydraulic system behaves under dynamic conditions. This could be achieved by using pressure transducers to measure dynamic pressure signals generated by the system. These signals often contain a mean component and a fluctuating component. The fluctuating component may be due to the inherent characteristics of a hydraulic system, e.g., pump ripple and cylinder impact, or faulty condition associated with hydraulic components. With appropriate signal analysis techniques, information related to faulty condition can be extracted from raw signals. Thus, condition monitoring of hydraulic components could be achieved. The drawbacks of the pressure method are similar to those of the flow rate method.

2.3.3 Temperature Method

The operating temperature of a hydraulic system must be kept within a specified range because properties of a hydraulic fluid may change significantly when temperature is out of this range. For a water hydraulic system, it must work below 50 °C. Otherwise, cavitations may occur, which can result in deterioration of hydraulic components. Therefore, temperature of a hydraulic system is often monitored. This can be conducted with temperature probes fitted into hydraulic lines and components.

As is known, when hydraulic fluid flows through an orifice in a hydraulic component, the hydraulic energy transforms partially into heat and causes the temperature to rise. Similarly, flow through a leaking path could also increase temperature in the nearby area. Therefore, it is possible to detect leakage by monitoring temperature. An example of temperature differential across a pressure relief valve has been provided and discussed by Watton [14]. Pietola et al. [34] reported that temperature differences and distributions in a component could be

acquired by thermograph techniques and the obtained thermal images were useful for fault diagnosis of the component.

The main drawback of the temperature method is that although leakage in hydraulic components is accompanied by the transformation of hydraulic energy into heat, it does not always result in a discernible local hotspot. This is because the heat could easily be dissipated by the heat transfer processes existing both in the hydraulic system and between the hydraulic system and the ambient. When in water hydraulic systems, this heat dissipation is accelerated due to the high thermal conductivity and heat capacity of water as compared with hydraulic oil. The thermal conductivity of water is 4-5 times that of mineral oil and the specific heat of water is two times that of mineral oil [3]. Moreover, the response of temperature to changes in hydraulic components is very low.

2.3.4 Vibration Method

Vibration analysis is a well-established method for condition monitoring of machines. The vibration characteristics of a machine usually change when faults occur. Vibration signals are measured with an accelerometer and often analyzed in the frequency domain.

Watton [14] studied a vibration-based method for detecting piston seal wear. An oil hydraulic cylinder was operated under no-load condition and with induced piston seal wear. It was shown that different wear conditions resulted in difference in the power spectra of measured vibration signals. In that study, flow rate was also analyzed. Watton concluded that “the accelerometer technique is preferable to flow metering and hence more cost-effective” for detecting piston seal wear.

Tan et al. [5] studied vibration signals generated by a water hydraulic cylinder. A fundamental frequency of 123 Hz and its harmonics were found in the frequency

spectra of vibration signals. The magnitude of this fundamental frequency increased with loads. Those authors also found that when the piston seal was removed, a new frequency component (48.8 Hz) appeared in the spectra.

It is important to note that the frequencies of vibration signals are usually below 20 kHz. Within this frequency range, background noise also exists. Therefore, vibration signals of interest may be interfered by background noise, which may require complex techniques to analyze vibration signals.

In addition to the above mentioned signals, there are other signals that have been used for condition monitoring of a hydraulic system, including torque, speed, force, displacement, velocity, etc. [14].

2.4 Condition Monitoring of Leakage

Leakage in industrial fields is not desirable in most cases. So far, a variety of techniques have been developed to detect and locate leakage in fluid transport systems. Some efforts have also been put into condition monitoring of leakage in a hydraulic system. In the following, a review is given to condition monitoring of leakage in a hydraulic system as well as in other systems.

2.4.1 Leakage in Hydraulic Systems

External leakage or internal leakage can occur in a hydraulic system. For a water hydraulic system, although leakage poses no environment issues, it would affect the performance of the system, e.g., reducing the volumetric efficiency and the motion control precision. Therefore, it is necessary to monitor leakage to make sure it is within an acceptable range.

a) Simple Method Based on Flow Continuity

A basic approach to monitor leakage in a hydraulic system is to use the principle of flow continuity, which states for steady flow in a pipeline, the mass flow rate is the same for all cross sections of the pipe. For nearly incompressible hydraulic fluids, this can also be expressed as the volumetric flow rate is constant in a pipeline.

Consider the case shown in Figure 2.2. Eqs. (2.1) and (2.2) can be rewritten as

$$Q_i = Q_1 - A_1 \cdot U \quad (2.6)$$

$$Q_e = (Q_1 - A_1 U) - (Q_2 - A_2 U) \quad (2.7)$$

If the values of Q_1 , Q_2 and U are measured and the areas A_1 and A_2 are known, internal leakage Q_i and external leakage Q_e can be predicted from the above two equations. Taking into account the measurement uncertainty, for example 0.3 L/min, the following rules can be obtained [14]

Rule 1: IF $Q_1 - A_1 \cdot U > 0.3$

THEN internal leakage exists

Rule 2: IF $(Q_1 - A_1 U) - (Q_2 - A_2 U) > 0.3$

THEN external leakage exists

The above rules may be incorporated in an expert system, which is a computer program composed from the explicit decision-making expertise or compiled experience of human experts. It is usually used for condition monitoring of a whole hydraulic system. All possible failures within the system are considered. More details about expert system and its applications to leakage monitoring in a hydraulic system can be found in [14].

The flow-continuity method is quite simple, and suitable for monitoring large steady-state leakage. However, “the occurrences of internal leakages make precise

prediction virtually impossible using conventional flow continuity measurements, even with the knowledge handling ability of an expert system shell” [15]. This is because of the limitations of flow meters. Flow meters are generally used to measure flow rates under steady-state conditions. In a cylinder drive system, the flow rate is highly dynamic due to the reciprocating motion [33]. Therefore, it is difficult to capture small changes in such dynamic flow rates by flow meters.

b) Advanced Methods Based on Artificial Intelligence

Le et al. [15] investigated leakage in an oil hydraulic cylinder drive system. Their experimental setup is shown in Figure 2.3. Four system variables, including two flow rates and two pressures (Q_1 , Q_2 , P_1 , and P_2), were measured for condition monitoring. Leakages from both lines (Q_{e1} and Q_{e2}) and across the piston seals (Q_i) were simulated by the use of flow control valves. In Figure 2.3, the upper cylinder was used to load the lower cylinder. Their results showed that multilayer perceptron (MLP) type neural networks (NNs) could successfully identify all different leakage types, but internal leakage was more difficult to detect than external leakage. MLPs could only detect internal leakage above 1.0 L/min, whereas external leakage could be detected successfully even at a very small leakage level (0.2 L/min). The value of 1.0 L/min is considered as high when compared with the full flow rate in the leakage-free case, i.e., 9.3 L/min and 7.1 L/min for the extending stroke and the retracting stroke, respectively. It was also found that the MLP’s performance was adversely affected by the working temperature of the hydraulic system.

Figure 2.3 Hydraulic circuit diagram of the test rig [15]

In the above work, the validity of using flow control valves to simulate leakage requires further investigation. The introduction of these valves could simulate the steady-state flow conditions due to real leakages, but it could not provide a desirable simulation of the dynamic processes existing in a cylinder drive system subjected to leakages. Actually, those authors used the steady-state measurements of flow rates and pressures, and therefore the dynamic information was ignored. In addition, the use of two flow meters and two pressure transducers would result in a complex and hence high-cost condition monitoring system.

Another work was also done by Le et al. [16]. Again, leakages in a hydraulic cylinder drive system were artificially introduced by flow control valves. Information was extracted from two transient pressures (P_1 and P_2) and input to artificial neural networks to identify leakage type and leakage rate. The experimental setup was shown in Figure 2.4, similar to that in Figure 2.3.

Figure 2.4 Hydraulic circuit diagram of the test rig [16]

Results showed that their method could identify leakage type accurately and was a promising technique to predict leakage rate level beyond 1.0 L/min (5 percent of maximum transient system flow rate). However, for internal leakage of 1.0 L/min, the performance of the neural networks was not desirable. Furthermore, in their study, three leakage rates were selected, i.e. 1.0, 3.0, and 6.0 L/min. This large leakage rate intervals, 10 to 15 percent of maximum transient system flow rate, made it easy for the neural networks to discriminate among different leakage level. Besides, that study raised the possibility of removing expensive flow meters from the condition monitoring and fault detection strategy for fluid power systems.

2.4.2 Leakage in Other Fluid Systems

In addition to hydraulic system, other fluid systems can be found in industries, such as fluid transport systems in chemical and petrochemical industries. Leakage in such systems may result in wasted product or even the hazard of mixing toxic or explosive chemicals. Therefore, leakage detection techniques are required to reduce cost and improve safety.

Nowadays, a variety of methods are available for detecting leakage in fluid

transport systems. Paxton et al. [35] gave a survey of the leakage detection methods used in a range of industries. These methods include:

- Visual inspection
- Line flow balances using flow meters
- Adding an odorant to the fluid
- Tracer gas dissolved in the fluid
- Visual inspection using ultraviolet light
- Thermal imaging
- Moisture sensitive tape
- Moisture sensitive foam
- Acoustic monitoring
- Ultrasonic testing
- Measuring leakage flow in a pressurized circuit

The above methods are usually used to detect the location of external leakage. Effort has also been put into detecting internal leakage in fluid transport systems, which is more difficult than detecting external leakage.

Thompson and Zolkiewski [17] investigated the detection of internal leakage through valves in a gas piping system by vibration analysis in the frequency range below 20 kHz. From the results of comprehensive experiments, it was found that internal gas leakage through a valve caused the valve and adjacent downstream pipeline to vibrate at particular frequencies, which were dependent only on the internal diameter of the pipe and the properties of the gas. However, if general factory noise or local high mechanical vibration exists near the valve, the leakage-related frequency components could be obscured and hence unidentifiable. In such a situation, those authors suggested that acoustic emission analysis or a trace gas be used.

Sharif and Grosvenor [23] applied AE techniques to detect internal valve leakage in a compressed air transport system. Two AE sensors were used. One had a frequency range of 20~100 kHz, resonant frequency being 30 kHz; the other was a wide-band AE sensor with a frequency range of 0.1~1 MHz. It was found that the resonant-type sensor was more sensitive to leakage, while the sensor with higher frequency range was less affected by background noise. In their experimental results, some discernible dominant peaks were present in the frequency spectrum, whose amplitudes increased with the leakage rate. Those authors concluded that these were leakage-dependent frequency components.

The above has reviewed leakage monitoring in hydraulic systems and fluid transport systems. Note that they are different from each other. The former is to determine the extent of leakage in hydraulic components, while the latter is focused on detecting and locating leaks in fluid transport systems. Usually, there are moving parts in hydraulic components, and therefore, leakage occurs through the clearance between the moving parts. When such leakage exceeds an acceptable level, appropriate maintenance should be done.

So far, much effort has been put into leakage detection in a fluid transport system and a number of techniques are available for this type of leakage. In contrast, limited work has been reported on leakage monitoring in a hydraulic system. In fact, this type of leakage is difficult to monitor, especially internal leakage in hydraulic components. This is due to the relatively complex structures of hydraulic components, the highly dynamic flows and the existence of background noise and mechanical vibration. Also, those conventional methods used for condition monitoring of hydraulic systems are not sensitive enough to monitor small internal leakage in hydraulic components, as mentioned previously. Thus, new techniques are required

for condition monitoring of internal leakage in hydraulic components. In this thesis, a sensitive AE-based method is developed to solve this problem.

2.5 Acoustic Emission

AE is “the class of phenomena where transient elastic waves are generated by the rapid release of energy from localized sources within a material, or the transient elastic waves so generated” [36]. The generated waves can propagate through the material and cause a time varying surface displacement at the boundaries of the material. An AE sensor located on a surface can detect such waves and generate voltage signals, which are referred to as AE signals. Drouillard [37] presented a good survey of the history of AE. Detailed descriptions about AE techniques were provided by Miller and McIntire [24], Williams [38] and Scott [39].

These signals are inherently of high frequency and low energy. They can be categorized as burst AE or continuous AE. The burst AE is a qualitative description of emission signals corresponding to individual AE events, while the continuous AE qualitatively describes an apparently sustained signal from rapidly occurring AE events [47], as shown in Figure 2.5 and Figure 2.6, respectively. A burst AE signal may have a continuous background, while a continuous AE signal may have some spikes interspersed. In many cases, only continuous components or burst components are of interest. Separating these two types of components can be accomplished effectively by a threshold method. Thereafter, the component of interest can be further processed.

Figure 2.5 Burst type of AE signal

Figure 2.6 Continuous type of AE signal

2.5.1 AE Sources

A typical AE source is mechanical deformation of stressed materials, such as plastic deformation, crack growth and fracture in materials. These sources are referred to as the classic AE sources. Since the early 1960s, researchers have exploited this kind of AE to detect discontinuities in materials. This is why AE is conventionally

regarded as a nondestructive testing method.

It is also found that waves from other source mechanisms can be detectable with AE sensors. These include friction, leakage, cavitations, liquefaction, solidification, etc. Sometimes they are referred to as secondary AE sources or pseudo AE sources [24]. Some of these mechanisms are related to failures of machines, e.g., severe friction and excessive leakage, and therefore, they could be used as a tool for condition monitoring of machines and processes. In fact, many AE applications are based on secondary sources [40-46].

The AE source is essentially wide band [22]. However, frequency range from 50 kHz to 1 MHz is commonly used in practice because background noise exists at lower frequencies and high attenuation occurs at higher frequencies.

2.5.2 AE Wave Propagation

AE waves are transient elastic waves. Their propagation in solid media is similar to that of sound waves in air or other fluids but are more complex. The descriptions of AE wave propagation were provided by Miller and McIntire [24], Bray and Stanley [47] and Holford [48].

In an unbounded medium, AE waves propagate as bulk waves that contain both dilatational (longitudinal) bulk waves and torsional (transverse) bulk waves. The propagating velocity is determined by elastic properties and mass density of the medium. When a free surface is present in a medium, another wave mode may exist, called Rayleigh wave. For this wave, the energy is concentrated on the surface, and the velocity is slightly less than that of a torsional wave. In a more practical situation, i.e., an infinite plate bounded by two parallel planes, AE waves are more complex than those mentioned above, and Lamb waves may exist. The propagation velocity of Lamb waves is dependent on wave frequency and plate thickness.

Since AE waves contain a wide range of frequencies, the speeds of each frequency component may be different from each other, which will cause wave distortion during propagation from AE sources to an AE sensor. Furthermore, reflection, refraction, diffraction and scattering may occur during the course of propagation due to the complex boundaries and discontinuities in media, e.g. free surfaces and cracks. All these factors significantly change AE waveforms by causing phase changes, amplitude attenuation, wave mode changes and wave repetitions.

2.5.3 Advantages of AE Techniques

The advantages of AE-based condition monitoring techniques are described in the following.

First, the frequency range used in AE techniques is much higher than that of mechanical vibration and environmental noise. Therefore, AE signals could be less contaminated by noise than signals of low frequency. Dong [49] compared AE and acceleration signals in grinding and dressing. It was found that acceleration would be heavily interfered by background noise and hence could not be effectively used for grinding and dressing monitoring. On the other hand, the grinding/dressing condition could be clearly identified by the AE signals. Thompson and Zolkiewski [17] suggested that AE be used, instead of vibration, to detect internal leakage through valves, if high-level noise or mechanical vibration exists.

Secondly, AE techniques are sensitive to local faults. Since AE is generated by the transient of local stress distribution, it is capable of supplying information about local dynamic processes in materials. It can indicate that a process is taking place rather than that it has taken place [42]. As is known, all incipient faults appear as a localized fault. Therefore, AE is a useful tool to detect incipient faults.

With these advantages, AE techniques have been successfully used as a

nondestructive testing method to detect whether there is a failure in structures or not. However, there is still a lot of work to be done before AE techniques can be established as a quantitative condition monitoring method. It is a challenging task to model AE sources and propagation because of the complex source mechanisms and propagation of AE waves. When AE waves propagate from the source to an AE sensor, they are usually distorted by the propagation media (as described in Section 2.5.2) and the sensor frequency response. Therefore, there may be difficulties in extracting information on the AE source from the measured AE signals. Analytical techniques for AE signal analysis are still in infancy.

2.5.4 AE Modelling Methods

Although AE has been successfully used in various industrial applications since 1960s, analytical techniques for AE modelling are still in its infancy [50]. This is due to the complex AE source mechanism and wave propagation. Nevertheless, some attempts have been made to model AE signals generated by different source mechanisms. So far, most AE modelling work in the literature is related to crack or metal machining. The following is a review with emphasis on the methodology of AE modelling.

a) Modelling AE Count

AE due to crack growth has been studied by analytical methods since the early stage of AE technique. The generated AE signal is largely attributed to the plastic deformation processes ahead of the crack tip. In the pioneering work by Dunegan et al. [51-52], a theoretical model was proposed to describe AE signals from a cracked specimen under load. Since the plastic deformation in the vicinity of a crack tip is completely controlled by the stress intensity factor (K), it is reasonable to expect that

AE signals due to plastic deformation would be dependent on the stress intensity factor (note that K is an important parameter in fracture analysis, dependent on the geometry of the specimen, the shape, size and location of the crack, and the type of loading). Therefore, those authors developed a model to correlate AE signals from a cracked specimen and the stress intensity factor for the crack. The model was obtained through three steps:

1. Assume that the observed AE count rate, $\dot{N}_{AE}$, is proportional to the rate of increase of the volume of plastically deformed material ahead of the crack tip, $\dot{V}_p$, i.e., $\dot{N}_{AE} \propto \dot{V}_p$.
2. By using linear elastic fracture mechanics theories to determine the volume of plastically deformed material ahead of the crack tip, V_p , a relationship is obtained, i.e., $V_p \propto K^4$.
3. Apply the above results and obtain a relationship between the AE total count N_{AE} and the stress intensity factor K for that crack, i.e.,

$$N_{AE} = AK^m \quad (2.8)$$

where A is a constant and the exponent m has a theoretical value of four.

Those authors analyzed experimental results from notched specimens of beryllium and aluminium and found that the total AE count varied by the sixth to eighth power of the stress intensity factor, rather than the theoretically predicted exponent of four. Possible reasons for the discrepancy might be the presence of other AE sources in addition to plastic deformation [52].

Palmer and Heald [53] also developed an analytical model to describe AE counts generated by a crack. They assumed that the total AE count, N_{AE} , is directly proportional to the plastic zone size ahead of the crack, s , i.e.,

$$N_{AE} = B \cdot s \quad (2.9)$$

where B is a constant of proportionality, dependent on strain rate, temperature, specimen thickness and material microstructure. It is known from fracture mechanics that the plastic zone size depends on the applied stress at the crack. Thus, those authors obtained a model to correlate the AE count and the applied stress. In the case of small stresses, their model showed that the AE count was proportional to the square of stress intensity factor, i.e., $N_{AE} \propto K^2$. Their experimental results were in good agreement with the model.

The power-law relationship between the total AE count and the stress intensity factor, i.e., $N_{AE} = AK^m$, has gained wide acceptance [24, 54-56], though various experimental values of the exponent m were reported. This variation could be due to the nature of material and the type of crack [57].

b) Modelling AE Energy

Since AE is generated by the rapid release of energy from local sources, it is reasonable to consider the energy of AE signals as a function of the energy released from AE sources. Some models have been proposed in the literature to correlate AE energy with metal machining parameters, etc.

Dornfeld and his co-workers [58-62] developed quantitative models for AE generation in machining processes. During metal cutting processes, plastic deformation is the source of AE signals. Based on this understanding, Kannatey-Asibu and Dornfeld [58] established a theoretical relationship between AE and the process parameters of orthogonal cutting by relating the energy content of AE signal to the plastic work of deformation during orthogonal cutting. Their model includes:

1. Assume the energy source of AE signal is plastic work of deformation (W),

- which is the applied energy that results in plastic deformation.
2. Assume the energy rate (power) of AE signal is directly proportional to the rate of plastic work of deformation during metal cutting.
 3. The energy rate of AE signal is measured by $\frac{dE}{dt} \propto V_{rms}^2$, where V_{rms} is the root mean square (rms) value of AE signal.
 4. From the mechanics of metal cutting, a relationship between the rate of plastic work and the process parameters of metal cutting is obtained, i.e., $\dot{W} = f(\alpha, U, b_1, t_1, \tau_k, \phi, l, l_1)$, where α is rake angle, U is cutting velocity, b_1 is width of cut, t_1 is feed rate, τ_k is shear strength of material, ϕ is shear angle, l is chip-tool contact length, and l_1 is length of the sticking zone.
 5. From the above, it is obtained that

$$V_{rms} = C\dot{W}^{1/2} \quad (2.10)$$

where C is a constant of proportionality. Liu and Dornfeld [62] improved the model for precision machining.

Lan and Dornfeld [59] refined the above model and proposed a power law relationship for high strain rate metal cutting, i.e.,

$$V_{rms} = C\dot{W}^m \quad (2.11)$$

where C is a factor of signal attenuation and m is material dependent.

Ni and Jinen [63] investigated the relationship between the parameters of fracture mechanics and AE in carbon fiber reinforced thermosoftening plastics. A power-law relationship similar to Dunegan's model was proposed. Their results showed that the power-law relationship gave a better fit for AE energy versus stress intensity factor than total AE counts versus stress intensity factor. This was due to the fact that both the amplitude and the counts were considered in the AE energy.

Pollock and Pepper [21] made an effort to obtain an empirical quantitative relationship between air leakage rate through an orifice (diameter ranging from 0.1 mm to 1 mm) and the detected AE signal. From their experimental results, it was found that the rms value of AE signal showed an approximately linear relationship to pressure and the cross-sectional area of an orifice, i.e.,

$$V_{rms} \propto p \quad (2.12)$$

$$V_{rms} \propto A_{\text{orifice}} \quad (2.13)$$

From the gas laws and choked flow theories, those authors found that the leakage rate could be given by

$$Q = 0.7 p A c_s \quad (2.14)$$

where Q is expressed in standard mL/s (the volume flow rate is normalized to atmospheric pressure) and c_s is the speed of sound. From the above, those authors concluded that the rms value of AE signal is directly proportional to the leakage rate only, and is independent of pressure or orifice size, i.e.,

$$V_{rms} \propto Q / c_s \quad (2.15)$$

From Eq. (2.15), it can be seen that the leakage rate could be estimated directly from the AE signal without knowing the orifice size, which could much facilitate the practical application. Pollock and Pepper reported that the above simple relationship worked effectively over the whole range of pressure and orifice sizes studied. However, the application of the conclusion to incompressible fluid requires further investigation. As is known, for the same leakage rate Q , a small orifice requires higher flow velocity than a big orifice, since the small orifice has a smaller cross-sectional area. Usually, the leaking flow of higher velocity is expected to be more turbulent and lead to higher level of AE signals.

In summary, AE models are usually developed by assuming that an AE parameter is proportional to a key parameter at the AE source (such as the above mentioned $\dot{V}_p$ and $\dot{W}$). Such a key parameter indicates the intensity of AE-related activities at the source. Since this key parameter is usually difficult to measure, models are developed to describe the relationships between this key parameter and some load conditions or process parameters, which are usually known. By this means, the AE parameter could be correlated to the load conditions or process parameters.

It can be seen from the above review that most analytical modelling efforts lead to a power-law relationship between AE signals and AE sources. This is also supported by many experimental results. For leakage-induced AE, a good approximation in the form $V \propto M^s$ has been reported in many cases [19], where V is the rms or average signal level, M is the leakage rate, and s is a constant. The value of the exponent s may vary from one situation to another. Jax [64] reported s values between 0.7 and 2.0.

CHAPTER 3

CHARACTERIZATION OF AE SIGNALS GENERATED BY INTERNAL LEAKAGE

Acoustic emission is transient elastic waves that are generated by the rapid release of energy from local sources. AE signal characterization is to find a method to interpret AE signals such that useful information about the sources can be extracted from the signals.

3.1 Internal-leakage-induced AE

In general, fluid leakage may induce AE waves. Such AE waves are of wide band at the source. The detected AE signals could have frequency contents up to several hundred kilohertz.

Goodman et al. [22] reported a variety of reasons why leakage from pressure envelopes, such as vessels, tanks and pipelines, may generate AE signals:

- Turbulent flow of the escaping gas or liquid;
- Cavitation during two-phase flow (gas coming out of solution) through a leaking orifice;
- The pressure surge generated when a leak starts and stops; and
- Backwash of particles against the surface of the equipment being monitored.

In the case of hydraulic cylinder systems, the first three mechanisms (turbulence, cavitation and pressure surge) may exist. As for pressure surge, it can occur when a hydraulic cylinder starts, stops or changes the direction at the end of the stroke. In practice, AE signal measurement should avoid these transient processes. In the

following, only turbulence and cavitation are discussed in detail.

The physical origin of turbulence-induced AE is the fluctuating pressure field associated with turbulence in the fluid around the area of the leak [19]. The fluctuating pressure field bears on the containing structure and the resulting waves can propagate to a nearby AE sensor. Sharif and Grosvenor [23] claimed that for internal leakage through an industrial control valve, AE signals are generated when the leaking fluid is turbulent.

Compared with hydraulic oil, the low viscosity and high mass density of water may lead to highly turbulent flow. Koskinen and Vilenius [65] studied steady-state flow in a pipe and reported that the Reynolds number of water flow was about 50 times greater than that of mineral oil flow. Therefore, for water hydraulic systems, turbulence is an important AE source mechanism.

For cavitation, it is the formation and collapse of bubbles within fluid [3]. Gas bubbles within hydraulic fluid can be created in two ways. When hydraulic pressure drops below the vapour pressure of hydraulic fluid, bubbles of hydraulic fluid vapour will be created within the fluid. Note that vapour pressure is defined as the pressure at which a liquid starts to boil (vaporize) and thus begin changing into a vapour (gas) [1]. The vapour pressure of a hydraulic fluid increases with the fluid temperature. Gas bubbles can also be created due to dissolved air in hydraulic fluid. The amount of air that can be dissolved in hydraulic fluid increases with pressure and decreases with temperature. Thus dissolved air will come out of hydraulic fluid when hydraulic pressure decreases or fluid temperature increases. Cavitation due to the above two mechanisms are usually referred to as vapour cavitation and gas cavitation, respectively. Gas bubbles within hydraulic fluid will collapse when they are exposed to high pressure. This can result in extremely high local fluid velocities, which

impacts on internal surfaces of hydraulic components and thus induces AE signals.

Among a variety of hydraulic components, pumps and valves are most susceptible to cavitation. Bubbles are likely to form in the suction inlets of pumps due to low pressure there and in valves due to throttling effect and thus rapid reduction of pressure [13, 66].

As compared with oil hydraulic systems, water hydraulic systems are susceptible to vapour cavitation but not gas cavitation. [65-66]. This is because the vapour pressure of water is higher than that of hydraulic oil and the amount of dissolved air in water is less than in hydraulic oil at the same pressure. For example, the vapour pressure at 50 °C is 0.12 bar for water and 1×10^{-8} bar for mineral oil and the saturated amount of air dissolved in water and mineral oil is approximately 2% and 10%, respectively. In order to eliminate vapour cavitation, the working temperature of water hydraulic system is usually maintained below 50 °C by a water cooling system.

Miller and Pollock [67] observed that a water leakage of 0.25 gal/h was accompanied by continuous AE. Those authors attributed this type of AE to the mechanism of turbulence. When air was artificially introduced into the leak path, burst-type of AE signals were detected. This was attributed to bubbling (gas cavitation). A previous study [8] has shown that a water hydraulic cylinder without internal leakage did not give rise to significant AE signals and the experiment in this chapter later will show that AE signals generated by internal leakage in water hydraulic cylinders are of continuous type. These observations imply that in practice, internal-leakage-induced turbulence, rather than cavitation, is the major AE source mechanism in water hydraulic cylinders.

3.2 AE Signal Characterization

In general, direct observation of either burst or continuous AE signals reveals little information on AE signal sources. In order to extract more useful information from AE signals, the signals should first be interpreted appropriately, which usually involves characterizing them with some parameters. For quantitative AE analysis, suitable parameters must be defined before any mathematical modelling is conducted. The following describes various methods that are often used for AE signal characterization.

3.2.1 Parameters in the Time Domain

Some basic parameters can be directly obtained from AE time signals. These include typical waveform parameters, as shown in Figure 3.1.

Figure 3.1 Typical waveform parameters of an AE signal

AE events are individual signal bursts produced by local AE sources. The waveform in Figure 3.1 counts for one AE event. AE count is defined as the number of times the signal exceeds a counter threshold. This parameter is widely used as a practical measure of AE activities. As shown in Figure 3.1, a single AE event can produce several AE counts. A larger event requires more cycles to ring down to the counter threshold and will produce a large AE count than a smaller event. This provides a measure of the intensity of AE activities. For continuous AE, AE count

rate is often used to measure the intensity of AE activities. It is defined as AE count per unit time.

The root mean square (rms) value is often used to measure the energy content of an AE signal. For an AE signal consisting of N samples, $x[0]$, $x[1]$, ..., $x[N-1]$, its rms value is given by

$$x_{\text{rms}} = \left(\frac{1}{N} \sum_{n=0}^{N-1} x[n]^2 \right)^{1/2} \quad (3.1)$$

The advantage of energy measurement is that the energy content of AE signal can be directly related to important physical parameters at AE sources without having to model the complex AE waveforms [24].

Among the above parameters, AE count (or AE count rate) and AE rms value are the most widely used, such as in Refs. [68-71]. In general, AE rms value and AE count rate are appropriate for continuous AE, while AE count is suitable for burst-type AE. AE rms value has an advantage over AE count (or AE count rate) because it does not require a threshold value to be specified. On the other hand, the “averaging” nature of AE rms value makes it insensitive to short-duration events, such as cracks [72].

3.2.2 Parameters in the Frequency Domain

Besides the aforementioned parameters in the time domain, parameters in the frequency domain are also of interest. They can be obtained by spectral analysis of AE signals, which can be accomplished by the use of Fourier transform. The discrete Fourier transform (DFT) of an N -point discrete-time signal $x[n]$ is defined as [73]

$$X[k] = \sum_{n=0}^{N-1} x[n] \exp(-j2\pi kn / N) \quad (3.2)$$

$X[k]$ is a complex and usually plotted as frequency spectrum, including magnitude and

phase information. The power spectral density (PSD) is another important method for spectral analysis, as given by [74-75]

$$P[k] = \frac{T}{N} |X[k]|^2 = \frac{T}{N} \left| \sum_{n=0}^{N-1} x[n] \exp(-j2\pi kn/N) \right|^2, 0 \leq k \leq N-1 \quad (3.3)$$

where T is the sampling period of the signal $x[n]$. $P[k]$ represents the distribution of signal power over frequencies.

Some parameters can be defined in spectra to characterize AE signals. Among them, the magnitudes of dominant peaks in spectra are frequently used to study AE signals. Some applications of spectral analysis to AE signals were provided by Sharif and Grosvenor [23], Lingard et al. [40], Byrne et al. [76] and Keyvan & Pickard [77].

It is important to note that the detected AE signal spectrum is dominated by the frequency response of AE sensors and propagation path, though AE waves at the source is wide-band. Therefore, the dominant frequency components in the detected AE signal could vary from one situation to another.

3.2.3 Other AE Characterization Methods

AE signals have also been characterized by other advanced methods, including wavelet analysis, artificial neural networks, etc. [46, 78-81].

Wavelet transform is frequently used for AE signal analysis. The wavelet transform of a signal $f(t)$ with respect to a wavelet $\psi(t)$ is defined as

$$(W_{\psi}f)(b, a) = |a|^{-\frac{1}{2}} \int_{-\infty}^{\infty} f(t) \overline{\psi\left(\frac{t-b}{a}\right)} dt, \quad (3.4)$$

where a, b are constants. Wavelet transform provides a flexible time-frequency window, which automatically narrows when observing high-frequency signals and widens when inspecting low-frequency signals [82-83]. This property makes it suitable for analyzing AE signals that contain both high frequency bursts of short duration and low frequency components of long duration [80]. Niu et al. [84] studied

AE signals from cutting process. The signals consisted of continuous and burst-type AE components. The continuous components were generated in the shear zone, the tool-chip interface and the flank-workpiece interface. The burst-type components were generated by tool fracture and chip breakage. Those authors used wavelet transform to decompose the AE signals into its continuous and burst components. The burst components were then used to detect tool fracture and chipping and the continuous components were then used to determine tool wear. However, for AE signals containing only continuous components, the advantages of wavelet transform over Fourier transform are not obvious.

Usually, wavelet transform is used to extract features from AE signals, which are then input into artificial neural networks to analyze AE sources. An artificial neural network is a massively-parallel distributed processor made up of simple processing units, referred to as neurons, which has a natural ability to store knowledge and make it available for use. It resembles the human brain in two respects [85]:

1. Knowledge is acquired by the network from its environment through a learning process.
2. Interneuron connection strengths, known as synaptic weights, are used to store the acquired knowledge.

The use of artificial neural networks offers some useful advantages, as described by Haykin [85] and Maksoud et al. [86]. They have the ability to approximate any nonlinear function to a desired accuracy and can be designed to adapt their synaptic weights to changes in the surrounding environment. However, the knowledge represented in an artificial neural network and the underlying computational processes are not explicit and cannot be directly explained by human, which is the main limitation of artificial neural networks.

In general, those basic parameters in the time domain and frequency domain are widely used to characterize AE signals, since they are simple and have clear physical meaning. Most AE modelling work in the literature is based on these basic AE parameters. For the advanced signal analysis methods, they are powerful in analyzing complex AE signals, especially when the signals consist of components from different AE source mechanisms, such as in metal machining.

3.3 Experiments on Internal Leakage

In order to study the characteristics of AE signals generated by internal leakage in water hydraulic cylinders, experiments were deliberately designed and carried out, as described in the following.

3.3.1 Experimental Setup

The experimental setup included a modern water hydraulic system and an AE measurement system.

a) Water Hydraulic System

The water hydraulic system is shown in Figure 3.2. Specifications of this system are given below.

- Pressure range: 20 to 140 bar
- Max. flow rate: 17.2 L/min
- Temperature range: 3 to 50 °C

The hydraulic circuit of this system is given in Figure 3.3.

Figure 3.2 Modern water hydraulic system

Figure 3.3 Hydraulic circuit diagram of the modern water hydraulic system

From Figure 3.3, it can be seen that this water hydraulic system works just like an oil hydraulic system. The pump, driven by an electric motor, sucks water from the tank, and pushes it into the water hydraulic system. As water passes through the pump, its potential energy increases as evidenced by an increase in fluid pressure. The pressurized water then flows through pipes, valves and the water hydraulic cylinder. The valves are used to control the direction, pressure and flow rate of the water flow so that the water hydraulic system can function as required. At the water hydraulic cylinder, the hydraulic pressure energy is transferred to the load to perform useful work, lifting and lowering the weights.

As shown in Figure 3.3, the power-pack valve comprises a two-way, two-position directional valve and a pressure relief valve. The directional valve is normally opened, which means water from the pump outlet returns to the tank through this valve, and thus the pump is unloaded and the power consumption is reduced to a minimum level. When the cylinder is running, this valve is closed and the pressurized water flows to the cylinder to perform useful work. The pressure relief valve is normally closed. Its function is to limit the system pressure to a specified maximum value. When the system pressure reaches this maximum, the valve is opened and the pump flow is diverted to the tank.

The extending or retracting of the water hydraulic cylinder is accomplished by the use of the directional control valve, as in Figure 3.3. Moreover, automatic cylinder reciprocating motion can be obtained with this valve combined with two limit switches and simple logic control electrical circuits.

Figure 3.4 shows a double-acting water hydraulic cylinder that was used for experiments in the present work. The cylinder barrel is made of seamless stainless steel tubing. Specifications of this cylinder are listed in Table 3.1.

Figure 3.4 Water hydraulic cylinder

Table 3.1 Specifications of the water hydraulic cylinder

Item	Value
Max. operating pressure (bar)	140
Piston diameter (mm)	50
Rod diameter (mm)	30
Stroke (mm)	500
Temperature range (°C)	1 to 50

As shown in Figure 3.3, the flow meters and pressure transducers are installed to measure the working conditions of the hydraulic system. Their specifications are listed in Table 3.2 and Table 3.3, respectively.

Table 3.2 Specifications of the flow meter

Item	Value
Measuring range (L/min)	2.84 to 28.4
Operating pressure (bar)	<345
Repeatability	±0.1% of measuring range
Accuracy	±1% of measuring range

Table 3.3 Specifications of the pressure transducer

Item	Value
Measuring range (bar)	0~250
Permissible pressure (bar)	400
Repeatability	±0.25% of measuring range
Accuracy	±1% of measuring range

For the water hydraulic cylinder used in the experiment, the piston seals consist of two sets of V-rings and a wear ring, as shown in Figure 3.5. The function of the wear ring is to guide the piston motion in the cylinder, support side loading and prevent metal-to-metal contact between the piston and the cylinder wall. The two sets of V-rings are identical but installed in opposite directions. They serve as sealing against leakage, acting during the extending and retracting strokes, respectively. The failure of these V-rings will result in the extending-stroke or retracting-stroke internal leakage, respectively. It can be seen that the piston ring configuration is symmetric.

Figure 3.5 Piston rings

b) AE Measurement System

The AE measurement system used in the experiments is shown schematically in

Figure 3.6. A wideband AE sensor was used to measure AE signals. It features a very high sensitivity for surface (Rayleigh) waves over a frequency range of 50 to 400 kHz. Raw AE signals detected by the AE sensor were passed through a preamplifier, which has a gain of 20 dB and a passband range of 50 to 1000 kHz. The signals were then sampled by a high-performance data acquisition board installed in a personal computer. The board provides a sampling frequency of up to 5 MHz with 12-bit analogue-to-digital converters.

Figure 3.6 AE measurement system

The mounting of AE sensor was carefully designed. Because the external surface of the hydraulic cylinder is cylindrical, if the sensor is directly mounted on the cylinder, it contacts the cylinder only over a very small area, which poses a problem for AE waves to propagate from the cylinder to the sensor. Therefore, a steel saddle was designed to fit the external diameter of the cylinder and fixed in the mid-stroke position on the cylinder surface. The AE sensor was then mounted on the top of the saddle by means of a magnetic clamp, as shown in Figure 3.7. In order to assist the AE propagation, highly viscous grease was applied to the coupling surfaces, i.e., the interface between the cylinder and the saddle and the interface between the saddle and the sensor.

The effectiveness of the sensor mounting was investigated by a 2H pencil lead, 0.5 mm in thickness and 3 mm in length, which is commonly used to test the response of an AE sensor. The pencil lead was broken on the cylinder surface, 10 mm apart from the saddle. A typical response of the AE sensor to the pencil lead breakage is shown in Figure 3.8. It could be concluded that the sensor mounting can provide an

effective path for AE waves to propagate from the cylinder to the AE sensor and the dynamic process generated by the pencil lead breakage can be captured by the AE sensor.

Figure 3.7 AE sensor mounting

Figure 3.8 Response of the AE sensor to pencil lead breakage

In the following experiments, all AE signals were acquired with the above AE measurement system. According to Nyquist theorem, the sampling rate must be

greater than twice the maximum frequency component in the signal being acquired. In practice, one often uses a sampling rate five to ten times the maximum frequency component in the signal to avoid any aliasing. Noting that the passband range of the preamplifier (see Figure 3.6) is 50 to 1000 kHz, a sampling rate of 5 MHz was therefore used to record AE signals in this study. Each record of AE signal contained 4096 points. With these sampling settings, the frequency resolution that the power spectral density of the measured AE signals can achieve is given by

$$\Delta F = \frac{f_s}{N} = \frac{5000 \text{ kHz}}{4098} = 1.2 \text{ kHz} \quad (3.5)$$

where N is the number of points and f_s is the sampling rate.

3.3.2 Experimental Design

a) Simulating the Real Internal Leakage

The internal leakage in a hydraulic cylinder has been simulated by connecting a flow control valve in parallel with the hydraulic cylinder [15-16]. The artificially introduced internal leakage flowed through the valve instead of the hydraulic cylinder. The advantage of this simulation is that the leakage rate can be easily controlled. However, the validity of this simulation requires further investigation. It is believed that this cannot provide a desirable simulation of the dynamic processes existing in a hydraulic cylinder subject to internal leakage. Thus, in the present work, efforts have been made to simulate the real internal leakage in hydraulic cylinders.

As mentioned in Section 2.2.2, the internal leakage in a water hydraulic cylinder is attributed to scores on the sealing surfaces of piston seals. These scores are created by hydraulic fluid contaminated with solid particulates. In the present work, experiments have been done and verified that a piston seal with no score damage does

not lead to internal leakage, even if the seal has significant wear (i.e., reduction in the diameter of the seal). This is because V-rings are pressure-driven seals. The sealed pressure can press the rings and make them contact the cylinder wall firmly.

In order to simulate scores created by the abrasive action of solid particulates, a file was used, in the present experiment, to create scores on the piston seal surfaces of a water hydraulic cylinder. Figure 3.9 shows the scored piston V-rings used in the experiments. These rings lead to an internal leakage smaller than 1 L/min in the water hydraulic cylinder for a pressure range of 0 to 75 bar. Along the circumference of the rings, sixteen scores are equally distributed, each extending over the full axial length of the rings. The dimensions of these scores were measured with a non-contact optical measurement system. Figure 3.10 shows the profile of a score taken by the measurement system. Along the edge of the score, five key points were selected and their coordinates were measured. The width and depth of the score were then calculated. In addition, a circular arc fit to these five points was also calculated so that an approximate radius of the score could be obtained. Table 3.4 lists the measurement results for the 16 scores.

Figure 3.9 Piston rings with 16 scores

Figure 3.10 Profile of a score

Table 3.4 Dimensions of scores (also see Figure 3.10)

Dimension	Average	Standard deviation
Width b (mm)	0.7715	0.1410
Depth h (mm)	0.3141	0.1087
Radius r (mm)	0.3924	0.0768

Since the piston ring configuration is symmetric, only the extending-stroke leakage was studied in the experiment. Figure 3.11 shows the piston ring configuration used in the experiment. Scored extending-stroke V-rings were installed to introduce the extending-stroke internal leakage. Although the retracting-stroke leakage was not studied, the retracting-stroke V-rings were installed such that the piston ring groove was filled up. But these were worn rings with no sealing actions against the extending-stroke internal leakage.

Figure 3.11 Piston rings used in the experiment

b) Measuring the Internal Leakage Rate

Since the present work is to study the characteristics of AE signals with respect to the internal leakage rate in water hydraulic cylinders, it is necessary to measure the internal leakage rate accurately. However, when a hydraulic cylinder is in operation, i.e., performing a reciprocating motion, this leakage rate is difficult to measure, if not impossible. Therefore, a special experiment was designed.

In the experiment, the cylinder rod was fixed at mid-stroke, as shown schematically in Figure 3.12. When high-pressure water from the pump flows into the cylinder via the lower port, it will leak through the scores on the piston seals. Thus, the internal leakage occurs. The leakage rate can be changed by adjusting the pressure of water. The leaking water was collected and measured with a weighing machine, which can perform continuous measurement. The data of water mass, five points per second, were transmitted to a personal computer. The internal leakage rate Q_i was then calculated by the time rate of change of the mass M of the leaking water divided by the mass density of water ρ , as given by

$$Q_i = \frac{1}{\rho} \frac{dM}{dt} \quad (3.6)$$

Figure 3.12 Schematic of fixation of cylinder rod

3.4 Experimental Results

In the experiment, 100 sets of data were acquired at different internal leakage rates with each set consisting of 40 records of AE signals measured at a certain leakage rate. Each record of AE signal contained 4096 points sampled at 5 MHz, from which AE parameters were calculated. For each AE parameter, results obtained from the 40 records were then averaged. In the following, all the results are the average values.

Figure 3.13 shows a typical record of AE signal generated by the internal leakage in the water hydraulic cylinder. It can be seen that this is of continuous-type AE signal. Therefore, those parameters that are suitable for characterizing continuous-type AE were calculated from the experimental data and their correlations with the internal leakage rate were analyzed, as described in the following.

Figure 3.13 AE signal generated by the internal leakage in the water hydraulic cylinder

3.4.1 AE Count Rate

For each record of AE signal, AE counts were calculated as the number of times the signal exceeds a preset threshold and the AE count rate, denoted as $\dot{N}_{AE}$, was calculated by dividing the AE counts by the signal duration. Both fixed thresholds and floating thresholds were used for counting. So far, there is no well-defined procedure to choose the threshold value, but one rule is to choose the threshold value beyond the noise level. For the present experiments, it was found that the noise level was not greater than 0.005 volts. Therefore, a wide range of threshold values greater than 0.005 volts were studied.

Figure 3.14 shows the results for fixed thresholds. It can be seen that for the threshold of 0.005 V, the saturation of the AE count rate occurred, i.e., the AE count rate maintained at a high level no matter how the leakage rate changed. With higher threshold values, there was a clearer trend in the AE count rate. However, if the threshold value is too high, such as 0.060 V, it may exceed the signal amplitude and result in zero values of the AE count rate, which is undesirable.

Figure 3.14 AE count rate versus internal leakage rate (for fixed threshold)

For floating thresholds, the threshold value was set to be proportional to the rms value of the signal. The resulting AE count rate remained at a nearly constant level, no matter how the leakage rate varied. This is shown in Figure 3.15. It is clear that floating threshold is not an effective method to extract leakage information from internal-leakage-induced AE signals.

Figure 3.15 AE count rate versus internal leakage rate (for floating threshold)

3.4.2 AE rms

The AE rms value of a record of AE signal, denoted as V_{rms} , was calculated using Eq. (3.1). Figure 3.16 shows the AE rms value versus the internal leakage rate. It is evident that there is a strong correlation between the AE rms value and the leakage rate. Comparing Figure 3.14, Figure 3.15 and Figure 3.16 shows that the AE rms value is more suitable than the AE count rate for interpreting the AE signals generated by internal leakage in water hydraulic cylinders. This may be due to the inherent drawback of AE count rate, i.e., the method of counting against a threshold cannot completely take into account the amplitude of AE signals.

Figure 3.16 AE rms versus internal leakage rate

3.4.3 AE Power Spectral Density

Figure 3.17 shows a typical PSD spectrum of the AE signal generated by the internal leakage. The PSD was calculated using Eq. (3.3). It can be seen that in the

spectrum, there is a dominant peak near 120 kHz. Note that PSD spectra of AE signals measured during the reciprocating motion of a new hydraulic cylinder without internal leakage also show peaks similar to those in Figure 3.17 but with smaller magnitude. This indicates that the peaks are not due to the excitation spectra of turbulence associated with the internal leakage but probably relate to the frequency response of the propagation media and the AE sensor. Figure 3.18 shows the magnitude of this peak, denoted as M_p , versus the internal leakage rate. Although there is a trend in the magnitude, it suffers a great scattering. Figure 3.19 shows the average energy of the AE signal measured in the frequency domain, denoted as E_f , versus the internal leakage rate. This average energy (actually the power of the signal) was obtained by integrating the PSD over the frequency range of 0 to 400 kHz (note that the PSD outside this range is negligible). It can be seen that there exists a clearer trend in Figure 3.19 than in Figure 3.18. These results show that 1) both the peak magnitude M_p and the energy E_f seem to have a power-law dependence on the internal leakage rate; 2) although the PSD of an individual frequency component displays significant scattering, summing the PSD over a frequency band can considerably reduce the scattering.

Figure 3.17 The PSD of AE signal generated by internal leakage in the water hydraulic cylinder

Figure 3.18 The magnitude of a dominant peak near 120 kHz versus internal leakage rate

Figure 3.19 AE energy within 0-400 kHz versus internal leakage rate

3.5 Discussion

In the previous section, various AE parameters were calculated from AE signals. In practice, AE signals generated by internal leakage could be interfered by background noise arising from other sources. This noise must be identified and removed before AE parameters are calculated. Since the characteristics of noise may vary from one situation to another, they should be identified by actual tests in a real situation. This can be accomplished by measuring AE signals (including any possible background noise) in the real situation using a new hydraulic cylinder without internal leakage and a hydraulic cylinder with internal leakage. Comparing AE signals from these two cases helps to identify background noise.

In the time domain, if the noise is found to be of burst type with higher amplitude than AE signals due to internal leakage, a threshold method can be used to remove the noise effectively; in the frequency domain, if the noise is found to have

different frequency components from AE signals due to internal leakage, a proper filter can be used to remove the noise effectively. In addition, an effective remedy for noise interference is to identify the noise source and block them by using high-attenuation materials in the path from the noise source to the AE sensor. For example, using flexible hoses instead of metal tube to connect a hydraulic cylinder can block AE noise from hydraulic control valves.

In the following, the correlation between AE parameters and internal leakage rate is discussed and a model is developed to predict the internal leakage rate.

3.5.1 Correlation Between AE Parameters and Internal Leakage Rate

From the above experimental results, it is clear that there are some correlations between the AE parameters and the internal leakage rate. Therefore, the correlation coefficients between the AE parameters and the internal leakage rate were studied. The correlation coefficient, denoted as r , between two variables X and Y is defined by [87]

$$r = \frac{S_{xy}}{\sqrt{S_{xx}S_{yy}}} \quad (3.7)$$

where

$$\begin{aligned} S_{xx} &= \sum_{i=1}^n (x_i - \bar{x})^2 \\ S_{yy} &= \sum_{i=1}^n (y_i - \bar{y})^2 \\ S_{xy} &= \sum_{i=1}^n (x_i - \bar{x})(y_i - \bar{y}) \end{aligned} \quad (3.8)$$

and $\{(x_i, y_i); i = 1, 2, \dots, n\}$ represents the pairs of observations of X and Y , $\bar{x}$ and $\bar{y}$ are the average values of x_i and y_i , respectively.

The correlation coefficient r measures the linear association between X and Y .

The absolute value $|r|$ close to one implies good linear correlation between X and Y , whereas $|r|$ equal to zero implies no linear correlation between X and Y . Table 3.5 lists the calculated correlation coefficients. Note that the correlation coefficient of AE count rate resulting from floating threshold is not calculated, since it is obvious that floating threshold is not suitable for characterizing AE signals generated by internal leakage. From Table 3.5, it is clear that the AE rms value V_{rms} has a stronger linear correlation with the internal leakage rate than the AE count rate $\dot{N}_{\text{AE}}$, the peak magnitude M_p and the AE energy E_f . Comparing the units of V_{rms} , M_p and E_f indicates that the square root of M_p and E_f may be of interest. Therefore, the correlation coefficients for the square root of M_p and E_f were also calculated, as listed in Table 3.5. The results show that $\sqrt{M_p}$ and $\sqrt{E_f}$ have a stronger linear correlation with the internal leakage rate than M_p and E_f , respectively.

Table 3.5 Correlation coefficients between AE parameters and internal leakage rate

AE parameter	Correlation coefficient
$\dot{N}_{\text{AE}}$	
Fixed threshold = 0.005 V	0.5732
Fixed threshold = 0.020 V	0.8048
Fixed threshold = 0.030 V	0.8596
Fixed threshold = 0.040 V	0.8959
Fixed threshold = 0.050 V	0.9234
Fixed threshold = 0.060 V	0.9419
V_{rms}	0.9654
M_p ($\sqrt{M_p}$)	0.9050 (0.9204)
E_f ($\sqrt{E_f}$)	0.9612 (0.9671)

It can be seen from the above correlation analysis that the AE rms value V_{rms} and the AE energy E_f are closely related to the internal leakage rate. In fact, these two parameters are equivalent. According to Parseval's theorem, the time signal $x[n]$ and its discrete Fourier transform $X[k]$ satisfy the following equation [74]:

$$T \sum_{n=0}^{N-1} |x[n]|^2 = \frac{T}{N} \sum_{k=0}^{N-1} |X[k]|^2 \quad (3.9)$$

Substituting Eqs. (3.1) and (3.3) into Eq. (3.9) and rearranging yields

$$x_{\text{rms}}^2 = \sum_{k=0}^{N-1} P[k] \Delta F \quad (3.10)$$

where $\Delta F = 1/(NT)$, which is the discrete frequency sample interval. Note that the term on the left-hand side is the square of the rms value, which represents the energy measured in the time domain; and the term on the right-hand side is the sum of the power spectral density in the frequency domain, which represents the energy measured in the frequency domain. Therefore, it may be concluded that energy-based AE parameters, whether measured in the time domain or in the frequency domain, have a strong correlation with the internal leakage rate and are most suitable for interpreting AE signals generated by internal leakage in the water hydraulic cylinder.

3.5.2 Internal Leakage Rate Prediction

As has been shown above, the energy content of AE signal is closely related to the internal leakage rate in the water hydraulic cylinder, and therefore, it could be used to predict the internal leakage rate. In the following, an empirical model is built to predict the internal leakage rate based on measured AE signals. The error of prediction is then analyzed.

Due to the simplicity in calculation, the parameter V_{rms} is chosen to describe AE

signals instead of the energy E_f . From the previous 100 sets of experimental data, the relationship between the AE rms value V_{rms} and the internal leakage rate Q_i can be obtained using statistical methods. Let $\{(v_i, q_i); i = 1, 2, \dots, n\}$ denote the 100 pairs of data plotted in Figure 3.16, where v_i is the value of V_{rms} , q_i is the value of Q_i and $n = 100$. Then, a linear regression equation can be built to fit the 100 pairs of values, given by

$$Q_i = a + bV_{\text{rms}} \quad (3.11)$$

where

$$b = \frac{\sum_{i=1}^n (v_i - \bar{v})(q_i - \bar{q})}{\sum_{i=1}^n (v_i - \bar{v})^2} = 7.86 \quad (3.12)$$

$$a = \bar{q} - b\bar{v} = 0.14 \quad (3.13)$$

Here $\bar{v} = \frac{1}{n} \sum_{i=1}^n v_i$ and $\bar{q} = \frac{1}{n} \sum_{i=1}^n q_i$. Thus, an empirical model is obtained to describe the relationship between the AE rms value V_{rms} and the internal leakage rate Q_i , given by

$$Q_i = 7.86 V_{\text{rms}} + 0.14 \quad (3.14)$$

Eq. (3.14) can be used to predict the internal leakage rate based on a measured AE rms value. The error of prediction is analyzed below using the prediction interval method [87]. Suppose the measured AE rms value is v_0 , then a 95% prediction interval for the actual internal leakage rate q_0 is given by

$$\hat{q}_0 - \delta < q_0 < \hat{q}_0 + \delta \quad (3.15)$$

where $\hat{q}_0$ is the internal leakage rate predicted by Eq. (3.14), i.e.,

$$\hat{q}_0 = 7.86v_0 + 0.14 \quad (3.16)$$

and

$$\delta = t_{0.025} s \sqrt{1 + \frac{1}{n} + \frac{(v_0 - \bar{v})^2}{S_{vv}}} \quad (3.17)$$

Here $t_{0.025}$ is a value of the t -distribution with $n-2$ degrees of freedom and s is defined

as $s = \sqrt{\frac{S_{qq} - bS_{vq}}{n-2}}$. S_{vv} , S_{qq} and S_{vq} are as defined in Eq. (3.8).

Eq. (3.15) means that for the measured AE rms value v_0 , the actual value of the internal leakage rate q_0 lies within the interval $(\hat{q}_0 - \delta, \hat{q}_0 + \delta)$ with a 95% confidence. It is clear that δ is a measure of the width of the prediction interval and reflects the error in predicting the internal leakage rate. Using Eq. (3.17), the value of δ is obtained as about 0.078 L/min for internal leakage rate lower than 1.0 L/min. Figure 3.20 shows the prediction interval for the internal leakage rate. The error of 0.078 L/min is, in practice, acceptable for predicting internal leakage rates in hydraulic systems. It is important to appreciate that the empirical model, given by Eq. (3.14), may not be applicable to other situations. In fact, due to the complexity of AE phenomena, the application of AE techniques depends on a specific situation.

Figure 3.20 Prediction interval for the internal leakage rate

3.6 Conclusions

The characteristics of AE signals generated by internal leakage in a water hydraulic cylinder have been studied in this chapter. Experiments were carefully designed, including the simulation of the internal leakage across the piston seals in a water hydraulic cylinder and the measurement of the internal leakage rate. AE signals obtained from the experiments were analyzed in detail. Several AE parameters were extracted from the AE signals and their effectiveness for predicting the internal leakage rate were studied. From the analysis results, some conclusions could be made, as follows:

- AE signals are sensitive to small internal leakage in a water hydraulic cylinder and AE-based methods are able to predict the internal leakage less than 1.0 L/min.
- Energy-based AE parameters, whether measured in the time domain or in the

frequency domain, are more suitable than the AE count rate and the peak PSD magnitude to interpret AE signals generated by the internal leakage.

- The AE rms value has a good linear relationship with the internal leakage rate in a water hydraulic cylinder, and therefore, it is a most desirable parameter for developing a quantitative AE model to predict the internal leakage rate.

CHAPTER 4

CHARACTERIZATION OF TURBULENCE GENERATED BY INTERNAL LEAKAGE

4.1 Introduction

As mentioned previously, AE signals are generated by turbulence associated with leakage rather than leakage itself. This implies that internal leakage and AE signals thereby generated is linked by turbulence. In light of this physical understanding, in order to establish quantitative models to correlate AE signals and internal leakage rates, the characteristics of turbulence generated by internal leakage in water hydraulic cylinders should be studied. This chapter is devoted to address this issue. In Section 4.2, theories of turbulent flow are described; in Sections 4.3, a numerical model of internal leakage in a water hydraulic cylinder is created; Section 4.4 studies the results from numerical computation followed by discussion and conclusion in Sections 4.5 and 4.6.

4.2 Turbulent Flow

There are two basic types of fluid flow: laminar flow and turbulent flow. Laminar flow is characterized by the fluid flowing in smooth layers. In this type of flow, a fluid particle in a given layer stays in that layer. In turbulent flow, the movement of a particle becomes random and fluctuates up and down in a direction perpendicular as well as parallel to the mean flow direction. Figure 4.1 illustrates these two types of flow.

Figure 4.1 Two basic types of flow: (a) laminar flow; (b) turbulent flow

For any flow geometry, there is one (or more) parameter such that the flow is laminar when the parameter is less than a particular value, whereas the flow is turbulent when the parameter is greater than a certain value. The important parameters involved (i.e., Reynolds number, Mach number, etc.) and their critical values depend on the specific flow situation involved [88].

In the following, the mathematical description of turbulent flow of incompressible Newtonian fluid is given. Note that in water hydraulic systems, the hydraulic fluid (i.e., water) is considered as an incompressible Newtonian fluid.

4.2.1 Governing Equations

Like laminar flow, turbulent flow is governed by the mass-conservation equation (or continuity equation) and the Navier-Stokes equations. For simplicity, these equations are written in Cartesian tensors form as:

$$\frac{\partial U_i}{\partial x_i} = 0 \quad (4.1)$$

$$\frac{\partial U_j}{\partial t} + U_i \frac{\partial U_j}{\partial x_i} = \nu \frac{\partial^2 U_j}{\partial x_i \partial x_i} - \frac{1}{\rho} \frac{\partial p}{\partial x_j} \quad (4.2)$$

where U_i is the three components of the fluid's velocity $U(\mathbf{x}, t)$ in the three coordinate directions, x_i is the three components of the position vector $\mathbf{x}$ in the three coordinate directions, t is time, ρ is density, ν is kinematic viscosity, and p is the modified pressure, which includes the effect of gravity [89].

As a consequence of the random nature of turbulent flows, the direct approach of solving the above equations for turbulent flows is, in general, impossible, i.e., it is impossible to predict the velocity U at a specified position and time. However, it is possible to predict the mean behaviours of turbulent flows. The most important one of these behaviours is the mean velocity field.

Let $U(\mathbf{x}, t)$ be decomposed into its mean $\bar{U}(\mathbf{x}, t)$ and the fluctuation $u(\mathbf{x}, t)$, i.e.,

$$U(\mathbf{x}, t) = \bar{U}(\mathbf{x}, t) + u(\mathbf{x}, t). \quad (4.3)$$

By convention, throughout this thesis an overbar denotes the mean of a variable. Thus, taking the mean of the continuity equation and the Navier-Stokes equations (Eqs. (4.1) and (4.2)) gives

$$\frac{\partial \bar{U}_i}{\partial x_i} = 0 \quad (4.4)$$

$$\frac{\partial \bar{U}_j}{\partial t} + \bar{U}_i \frac{\partial \bar{U}_j}{\partial x_i} = \nu \frac{\partial^2 \bar{U}_j}{\partial x_i \partial x_i} - \frac{\partial \overline{u_i u_j}}{\partial x_i} - \frac{1}{\rho} \frac{\partial \bar{p}}{\partial x_j} \quad (4.5)$$

Eq. (4.5) is called Reynolds-averaged Navier-Stokes (RANS) equation. The velocity covariances $\overline{u_i u_j}$ are referred to as Reynolds stresses, which play an important role in that the different behaviours of $U(\mathbf{x}, t)$ and $\bar{U}(\mathbf{x}, t)$ are attributed to the effect of the Reynolds stresses.

Evidently the mean velocity field $\bar{U}(\mathbf{x}, t)$ is governed by Eqs. (4.4) and (4.5). Observing these equations shows a closure problem. For a general statistically three-dimensional flow, there are four independent equations governing the mean velocity field; namely three components of the Reynolds equation together with the mean continuity equation. However, these four equations contain more than four unknowns. In addition to $\bar{U}_i$ ($i = 1, 2$ and 3) and $\bar{p}$, there are also the Reynolds stresses $\overline{u_i u_j}$.

The above equations cannot be solved unless the Reynolds stresses are determined.

As mentioned above, there are difficulties in solving Eqs. (4.4) and (4.5). Additional equations are required to model the Reynolds stresses $\overline{u_i u_j}$ such that a closed set of equations is obtained. So far, various models have been proposed to address this closure problem, among which, the k - ε model and the k - ω model are the two most commonly used approaches due to their computational economy and reasonable accuracy. They are incorporated in most commercial computational fluid dynamics (CFD) software packages. The k - ε model is superior for simple shear turbulent flows, while the k - ω model is significantly more accurate for turbulent flows with wall boundaries existing. In this thesis, the flow in water hydraulic cylinders is wall-bounded. Therefore, the k - ω model is adopted to study the turbulence generated by the internal leakage in water hydraulic cylinders. The following gives a brief description about this model.

The k - ω model has been developed for over 20 years by many researchers. Significant contributions are due to Wilcox and others [90]. In this model, the Reynolds stresses $\overline{u_i u_j}$, which appear as unknowns in Eq. (4.5), are obtained from the turbulent-viscosity hypothesis. According to this hypothesis, the Reynolds stresses are given by

$$\overline{u_i u_j} = \frac{2}{3} k \delta_{ij} - \nu_T \left(\frac{\partial \overline{U}_i}{\partial x_j} + \frac{\partial \overline{U}_j}{\partial x_i} \right) \quad (4.6)$$

where δ_{ij} is defined by

$$\delta_{ij} = \begin{cases} 1, & \text{if } i = j \\ 0, & \text{if } i \neq j \end{cases} \quad (4.7)$$

and ν_T is turbulent viscosity, determined by turbulent kinetic energy k and specific

dissipation rate ω as:

$$\nu_T = k / \omega \quad (4.8)$$

k is the kinetic energy of the turbulent fluctuations per unit mass, given by

$$k = \frac{1}{2} \overline{u_i u_i} = \frac{1}{2} (\overline{u_1^2} + \overline{u_2^2} + \overline{u_3^2}) \quad (4.9)$$

where $u_i = U_i - \overline{U_i}$ ($i = 1, 2, 3$) is fluctuating velocity. ω is a measure of the rate at which turbulent kinetic energy is converted into thermal internal energy.

In order to obtain the turbulent viscosity $\nu_T(x, t)$, two transport equations are solved for k and ω , as given by

$$\frac{\partial k}{\partial t} + \overline{U_j} \frac{\partial k}{\partial x_j} = -\overline{u_i u_j} \frac{\partial \overline{U_i}}{\partial x_j} - \beta^* k \omega + \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_T}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] \quad (4.10)$$

$$\frac{\partial \omega}{\partial t} + \overline{U_j} \frac{\partial \omega}{\partial x_j} = -\alpha \frac{\omega}{k} \overline{u_i u_j} \frac{\partial \overline{U_i}}{\partial x_j} - \beta \omega^2 + \frac{\partial}{\partial x_j} \left[\left(\nu + \frac{\nu_T}{\sigma_\omega} \right) \frac{\partial \omega}{\partial x_j} \right] \quad (4.11)$$

where α , β , β^* , σ_k and σ_ω are model coefficients [90]. The various terms appearing in Eqs. (4.10) and (4.11) represent physical processes occurring as the turbulence moves about in a flow. In Eq. (4.10), the sum of the two terms on the left-hand side is the material derivative of k that represents the rate of change of k following a fluid particle; the first term on the right-hand side is the production of k that represents the rate at which k is transferred from the mean flow to the turbulence; the second term on the right-hand side is the dissipation of k that represents the rate at which k is converted into thermal internal energy; the third term on the right-hand side represents the diffusion of k due to molecular viscosity ν and turbulent fluctuation (indicated by ν_T). Similar explanation can be given to Eq. (4.11).

In summary, the k - ω model consists of the following components:

- i) two transport equations, Eqs. (4.10) and (4.11), are solved for k and ω ;
- ii) the turbulent viscosity ν_T is specified by Eq. (4.8);
- iii) the Reynolds stresses are obtained from the turbulent-viscosity hypothesis (Eq. (4.6));
- iv) the mean continuity equation and the RANS equations, Eqs. (4.4) and (4.5), are solved for $\bar{U}(\mathbf{x}, t)$ and $\bar{p}$.

4.2.2 Intensity of Turbulence

An important measure of any turbulent flows is how intense the turbulence is. This can be measured by means of some turbulence quantities, which are described in the following.

Reynolds number

Reynolds number is undoubtedly the most famous quantity in fluid mechanics, which is defined by

$$\text{Re} = \frac{\rho UL}{\mu} \quad (4.12)$$

where U and L are characteristic velocity and length of the flow, ρ is the mass density of the fluid, and μ is the absolute viscosity of the fluid. Reynolds number indicates the ratio of inertia force to viscous force.

Reynolds number is generally important in the study of all types of fluid flows. It is often used to indicate whether a flow is laminar or turbulent. Roughly, the higher the Reynolds number, the more turbulent the flow. For example, for pipe flow, the flow is laminar if Re is less than approximately 2100 and turbulent if Re is greater than approximately 4000. For Re between 2100 and 4000, the flow may switch between laminar and turbulent conditions.

It is important to note that the choice of characteristic velocity U and length L for the calculation of Re may vary from one flow situation to another. For example, for internal flow problems, a hydraulic diameter is used as the characteristic length; for flow over a flat plate, the dimension along the plate from the leading edge is used as the characteristic length. In some complex flows, there is no definite procedure to choose a characteristic velocity and length to calculate Re .

Turbulent viscosity ratio

Turbulent viscosity ratio is defined as the ratio of turbulent viscosity ν_T to laminar viscosity ν . This quantity can be better understood by studying the RANS equations. Substituting Eq. (4.6) into Eq. (4.5) gives

$$\frac{\partial \bar{U}_j}{\partial t} + \bar{U}_i \frac{\partial \bar{U}_j}{\partial x_i} = \nu_{\text{eff}} \frac{\partial^2 \bar{U}_j}{\partial x_i \partial x_i} - \frac{1}{\rho} \frac{\partial}{\partial x_j} \left(\bar{p} + \frac{2}{3} \rho k \right) \quad (4.13)$$

where

$$\nu_{\text{eff}} = \nu + \nu_T \quad (4.14)$$

is the effective viscosity. Note that Eq. (4.13) has the same form as the Navier-Stokes equations (Eq. (4.2)). From Eq. (4.14), it is clear that turbulent viscosity ratio can be used to measure the contribution of turbulent viscosity ν_T to the effective viscosity ν_{eff} . Consequently, a large turbulent viscosity ratio indicates that turbulent effects are dominant in the fluid flow. However, it is inconvenient to use this quantity to measure the intensity of turbulence because its value changes from position to position in the fluctuating velocity field.

Turbulence intensity

Turbulence intensity is defined as the root mean square of the velocity fluctuation divided by the mean velocity:

$$I_1 = \frac{\sqrt{u_1^2}}{U_1}, I_2 = \frac{\sqrt{u_2^2}}{U_2}, I_3 = \frac{\sqrt{u_3^2}}{U_3} \quad (4.15)$$

where I_1 , I_2 and I_3 are the turbulence intensities in the three coordinate directions, respectively. The larger the turbulence intensity, the larger the fluctuations of the velocity. Like the turbulent viscosity ratio, the turbulence intensity varies from position to position in the fluctuating velocity field. It is suitable for measuring a local intensity of turbulence in the flow field.

Turbulent kinetic energy

The intensity of turbulent flows can also be indicated by velocity fluctuations in the three coordinate directions, u_1 , u_2 and u_3 . An appropriate quantity combining these velocity fluctuations is turbulent kinetic energy k , which is defined as:

$$k = \frac{1}{2}(\overline{u_1^2} + \overline{u_2^2} + \overline{u_3^2}) = \frac{1}{2}\overline{u_i u_i} \quad (4.16)$$

It is the mean kinetic energy of the turbulent fluctuations per unit mass. The turbulent kinetic energy k is closely related to the Reynolds stress $\overline{u_i u_j}$, therefore, it is a quantity of considerable importance in turbulent flows. As k is the mean kinetic energy per unit mass, integrating k over the fluid domain of interest gives total turbulent kinetic energy, i.e.,

$$E_T = \int_V k \rho dV \quad (4.17)$$

where E_T is mass-weighted integral of turbulent kinetic energy (MITKE) and ρ is the mass density of fluid. This property makes turbulent kinetic energy most appropriate for quantitatively measuring the intensity of turbulent flows.

4.3 Numerical Simulation of Internal Leakage

Turbulent flows are complex processes, and therefore, analytical methods to solve such turbulent problems are difficult. Instead, with the help of the ever-increasing power of computers, numerical methods are often used to solve turbulence problems. Numerical solutions can provide information on turbulence quantities which are difficult if not impossible to measure experimentally and are thus invaluable for the study of turbulent flows. In this thesis, a popular commercial CFD software package is used to study the turbulence generated by internal leakage in the water hydraulic cylinder that was used in the previous experiments. Details are given below.

4.3.1 Geometry Modelling

In CFD analysis, the geometry of fluid domain must first be modelled and meshed. In this thesis, water inside the hydraulic cylinder is the fluid domain of interest. It can be represented by a three-dimensional (3D) model, as shown in Figure 4.2 (also refer to Figure 2.1 for the structures of a double-acting hydraulic cylinder). Note that some reasonable simplifications were made to facilitate the modelling. In the model, the two chambers of the water hydraulic cylinder are connected through scores, whereby internal leakage occurs. The piston rod is not modelled. This simplification is reasonable because internal-leakage-induced turbulence activities are confined to the small area near those scores and the piston rod is considered as far from that small area. Later, the numerical results, as shown in Figure 4.7 and Figure 4.10, will show that far from that small area, the value of internal-leakage-induced TKE is nearly zero.

In addition, it is assumed that scores are of uniform size and equally distributed

along the circumference of a piston seal. This assumption leads to an axisymmetric model. Thus, in a numerical computation, it is possible to solve for one sector of the model with the use of symmetry boundary conditions. In this way, the requirements for computational resources are much reduced.

Figure 4.2 Modelling the fluid domain in a water hydraulic cylinder

The assumption that scores are uniformly distributed along the circumference of piston seals is explained as follows:

- i) In practice, scores are usually created by solid particles contained in hydraulic fluid. They spread randomly along the circumference. Therefore, in a numerical simulation, one has to use the expected distribution of scores when modelling the fluid domain.
- ii) The structures of the water hydraulic cylinder are axisymmetric, and therefore, scores that are created during the operation of the cylinder is expected to be equally distributed along the circumference in a sense of probability;
- iii) The size of scores is about two orders of magnitude smaller than that of the cylinder and the internal leakage through a small score has significant effects

only in the region near that score. Therefore, the interaction between leakages from different scores could be negligible.

Figure 4.3 shows the dimensions of the fluid domain used in the numerical simulation. It includes only one score. Note that the stroke of this water hydraulic cylinder is 500 mm. The piston is in the mid-stroke position. The length of the scores l_s is 12 mm, which is determined by the axial length of one set of V-rings. As can be seen from Figure 3.5, there are two sets of V-rings installed on the piston. They seal against leakage for the extending and retracting strokes, respectively. That is, at the same time, only one set of V-rings are working. Thus, the length of the score is taken as the axial length of one set of V-rings.

The angle of the sector α is determined as

$$360 \div 16 = 22.5 \text{ degree,}$$

which is the case for the 16-score piston seal described in Section 3.3.2. R_{score} is the radius of the score. As listed in Table 3.4, the measured average score radius for the 16-score piston seal is 0.3924 mm. When those scores are compressed by hydraulic pressure, the actual score radius will be smaller than the measured value. In the following simulation, the diameter of score D_{score} (or $2R_{\text{score}}$) is taken as 0.3 mm. The turbulent flows with other score diameter values will be studied in next chapter.

Figure 4.3 Computational domain for the numerical simulation

Figure 4.4 shows the mesh of the computational domain. As the score dimensions are very small compared with the chamber dimensions, a fine mesh must be used. Therefore, the resulting mesh is very large. It consists of around 440,000 elements.

Figure 4.4 Mesh of the computational domain

4.3.2 Material Properties and Boundary Conditions

The operating temperature of water hydraulic systems is usually limited to the range of 5 to 50 °C [13]. In this numerical simulation, properties of water at 30 °C are used without loss of generality, that is, the mass density is 996 kg/m³ and the absolute viscosity is 7.98E-4 N·s/m². Numerical simulations for different operating temperatures are presented in the following chapter.

At the inlet, a velocity boundary condition (B.C.) is applied. Ten values of the inlet velocity V_{in} are selected for numerical simulation. They are 0.001, 0.002, 0.003, ..., 0.010 m/s, corresponding to flow rates of 0.12 to 1.2 L/min through the water hydraulic cylinder. Turbulence boundary conditions are also applied:

Turbulence intensity = 5% (represents fully-developed turbulence);

Hydraulic diameter = 50 mm.

At the outlet, pressure = 0 bar is applied. In addition, symmetry boundary conditions

and wall boundary conditions are applied to appropriate boundaries. Table 4.1 gives a summary of the numerical simulation.

Table 4.1 Summary of numerical simulation

Parameter	Value
Chamber length	250 mm
Chamber diameter	50 mm
Score length	12 mm
Score diameter	0.3 mm
Angle of the sector	22.5°
Mass density	996 kg/m ³
Absolute viscosity	7.98E-4 N·s/m ²
Inlet velocity B.C.	0.001, 0.002, 0.003, ..., or 0.010 m/s
Inlet turbulence B.C.	
Turbulence intensity	5%
Hydraulic diameter	50 mm
Outlet pressure B.C.	0 bar
Turbulence model	<i>k-ω</i> model

During the numerical computation, the residuals are monitored, which are commonly used to judge the convergence of a solution. The residuals must decrease to a small value for the solution to be convergent. In this study, additional quantities are also monitored for judging convergence, including flow rate at the outlet and mass weighted integral of TKE over the entire fluid domain. Thus, in this study, the solution is considered to be convergent when

- the residuals are less than 1E-3;
- the flow rate at the outlet becomes steady; and
- MITKE over the entire fluid domain becomes steady.

4.4 Numerical Results

4.4.1 Characteristics of Turbulent Flow in Hydraulic Cylinder

Typical results from the above numerical simulation are shown in Figure 4.5 to Figure 4.11. They are obtained with the inlet velocity V_{in} being 0.005 m/s. The mid-plane of the fluid domain and a cross-section plane 5 mm downstream of the score are selected for plotting various flow quantities. It can be seen from Figure 4.5, that high pressure is built up in the chamber upstream of the score. This is because water flow is constrained by the small leaking path (the score). Figure 4.6 shows that the velocity of water flow in the score is much higher than that in the chambers. When this high-velocity water flows into the chamber downstream, it generates strong turbulence, as shown in Figure 4.7. Figure 4.7 also shows that strong turbulence occurs at the entrance to the score. In Figure 4.8 to Figure 4.10, pressure, velocity and turbulent kinetic energy are plotted on the cross-section plane. It is clear that the effect of the internal leakage from the score is confined to a small region near the score. Far from that region, the value of TKE is very small, as indicated in Figure 4.7 and Figure 4.10. Figure 4.11 shows the flow path upstream and downstream of the score. It is obvious that there is a recirculation region in the downstream chamber.

Figure 4.5 Contours of pressure on the middle plane

Figure 4.6 Contours of velocity on the middle plane

Figure 4.7 Contours of turbulent kinetic energy on the middle plane

Figure 4.8 Contours of pressure on the cross-section plane downstream of the

score

Figure 4.9 Contours of velocity on the cross-section plane downstream of the score

Figure 4.10 Contours of turbulent kinetic energy on the cross-section plane downstream of the score

Figure 4.11 Flow path upstream and downstream of the score

4.4.2 Variation of Flow Quantities with Internal Leakage

Figure 4.12 shows the variation of flow quantities with the inlet velocity V_{in} increasing from 0.001 m/s to 0.010 m/s. These includes

- average pressure in the chamber upstream of the score, denoted as P_1 ;
- mean velocity at the outlet of the score, denoted as V_s ;
- flow rate through the score, i.e., internal leakage, denoted as q_i ; and
- MITKE over the entire fluid domain, denoted as E_T .

It can be seen that pressure P_1 , velocity V_s , internal leakage q_i , and MITKE E_T increase with the velocity at the inlet. Note that in the numerical simulation, the purpose of applying different velocities at the inlet is to produce different internal leakage rates and then, investigate the variation of flow quantities with this internal leakage. As indicated in Figure 4.12c, the internal leakage rate is proportional to the velocity at the inlet. Figure 4.12d shows that MITKE increases with the velocity at the inlet and therefore the internal leakage rate. Thus, MITKE could be an appropriate parameter to indicate the internal leakage rate.

Figure 4.12 Flow quantities versus inlet velocity

4.4.3 Location of Turbulence

As shown in Figure 4.7, high values of TKE are located at the inlet and outlet of the score. In order to investigate the location where most turbulence occurs, values of MITKE for the upstream chamber, the score and the downstream chamber were calculated and compared. It was found that more than 98% of MITKE was contributed by turbulent flow downstream of the score. Table 4.2 lists the percentage of MITKE in the downstream chamber to the total MITKE that is calculated over the

entire fluid domain.

Table 4.2 Percentage of MITKE in the downstream chamber

Inlet velocity B.C. (m/s)	MITKE (%)		
	Upstream chamber and score	Downstream chamber	total
0.001	0.63	99.37	100
0.002	1.97	98.03	100
0.003	1.79	98.21	100
0.004	1.56	98.44	100
0.005	1.41	98.59	100
0.006	1.32	98.68	100
0.007	1.25	98.75	100
0.008	1.20	98.80	100
0.009	1.16	98.84	100
0.010	1.13	98.87	100

Since MITKE generated in the upstream chamber and the score is insignificant, it can be ignored. In the remainder of this thesis, MITKE (denoted as E_T) refers to the mass integral of TKE over the chamber downstream of the score unless otherwise stated.

4.5 Discussion

In the numerical model described in Section 4.3, some assumptions and simplifications were made to facilitate the modelling. In the following, the validity of these assumptions and simplifications are discussed.

As mentioned previously, only one sector of the whole cylinder was modelled, which results from the assumption that scores are uniformly distributed along the circumstance of the piston seal. This may not be true in practical situations. However, the numerical model described in Section 4.3 could still be applicable unless there is a

strong interaction between scores. From the results shown in Figure 4.8 to Figure 4.10, it can be seen that the effect of internal leakage from each score is really confined to a small region and hence, the interaction between scores, in general, would not be strong.

In the above numerical simulation, the reciprocating motion of piston in the water hydraulic cylinder was not taken into account. This is the case when the piston rod is fixed, as in those experiments in Chapter 3. In general, the piston speed of a water hydraulic cylinder is much lower than the water velocity in the leaking path (i.e., score). For example, the maximum piston speed that the water hydraulic cylinder studied in this thesis can achieve is about 0.146 m/s, which is about two orders lower than the flow velocity shown in Figure 4.12b. Therefore, the piston motion has little influence on the internal leakage and hence the internal-leakage-induced TKE. It is also important to appreciate that if ultra low internal leakage rate (and hence ultra low flow velocity in the leaking path) is studied, the influence of piston speed may be significant in the numerical simulation.

It was assumed that the flow in the cylinder is turbulent in the above numerical simulation. Effort has been made to solve the flow problem with the laminar flow assumption. But the numerical computation encountered convergence difficulties (the residuals never reduced to less than $1E-3$). The resulting solution (note that it was not a converged solution) was carefully investigated. As shown in Figure 4.13, there is almost no diffusion downstream of the score due to the laminar assumption. Comparing Figure 4.6 and Figure 4.13 indicates that the turbulent flow assumption can result in a more rational solution than the laminar flow assumption. This may imply that turbulence is inevitable in this flow situation.

Figure 4.13 Contours of velocity of laminar flow

4.6 Conclusions

In this chapter, turbulent flow has been described and numerical modelling of internal leakage in a water hydraulic cylinder has been conducted. Results show that the numerical model could give useful information on the characteristics of turbulent flow in a water hydraulic cylinder. Some conclusions could be drawn, as follows:

- internal leakage could generate high intensity of turbulence in the downstream flow, while no strong turbulence occurs in the upstream chamber of the water hydraulic cylinder;
- the influence of internal leakage from a score is really confined to a small region near the outlet of that score, far from which TKE is almost zero; and

- MITKE increases with the internal leakage rate and it could be an appropriate parameter to correlate the internal leakage and the turbulence thereby generated.

This above numerical model will be used in Chapter 5 to study MITKE under different flow conditions.

CHAPTER 5

ESTABLISHING QUANTITATIVE RELATIONSHIP BETWEEN AE AND INTERNAL LEAKAGE

5.1 Introduction

In the previous two chapters, the characteristics of AE signals and turbulence associated with the internal leakage in a water hydraulic cylinder have been studied in detail. This chapter is then to establish a quantitative model to describe the relationship between the internal leakage and AE signals thereby generated.

5.2 Modelling Turbulence Generated by Internal Leakage

5.2.1 Dimensional Analysis Method

Although many fluid mechanics problems can be solved using analytical methods, there remain a large number of problems whose solution relies on data obtained from experiments and/or numerical simulations. For example, direct numerical simulation (DNS) data of Kim et al. [91] are frequently referred to in the study of turbulent flows. Thus, it is always desirable to interpret the data in such a way that some general knowledge could be obtained from these experimental/numerical results and then applied to other similar problems. Dimensional analysis provides such an approach to analyze the data that general formulations could be established based on the data. Historically, dimensional analysis plays an important role in the study of turbulent flows.

The basis for dimensional analysis is the Buckingham pi theorem [92], which states if an equation involving k variables is dimensionally homogeneous, it can be

reduced to a relationship among k minus r independent dimensionless products, where r is the minimum number of reference dimensions required to describe the variables. An equation is dimensionally homogeneous means that the dimensions on the left hand side of the equation are the same as those on the right hand side. Dimensional homogeneity is an essential requirement for any physically meaningful equation. The steps to perform a dimensional analysis are described below [88].

1. **List all the variables involved in the problem.** These variables must be independent of each other. Generally, a good understanding of the problem is required so that extraneous variables are not included and important variables are not omitted.
2. **Express each of the variables in terms of basic dimensions.** For a wide variety of fluid mechanics problems, only three basic dimensions are required, i.e., length L , time T , and mass M . Instead, length L , time T , and force F could be used.
3. **Determine the required number of pi terms.** This can be accomplished by means of the Buckingham pi theorem. The reference dimensions usually correspond to the basic dimension (there may be rare occasions in which the minimum number of reference dimensions is less than the number of the basic dimensions).
4. **Select a number of repeating variables, where the number required is equal to the number of reference dimensions.** All of the required reference dimensions must be involved in these repeating variables, and each repeating variable must be dimensionally independent of the others. Note that the dependent variable should not be used as a repeating variable.
5. **Form a pi term by multiplying one of the non-repeating variables by the**

product of the repeating variables, each raised to an exponent that will make the combination dimensionless.

- 6. Repeat Step 5 for each of the remaining non-repeating variables.**
- 7. Check all the resulting pi terms to make sure they are dimensionless.**
- 8. Express the final form as a relationship among the pi terms.** The final form could be written as

$$\Pi_1 = \phi(\Pi_2, \Pi_3, \dots, \Pi_{k-r})$$

where Π_1 contains the dependent variable, and the function ϕ could be determined with data from experiments or numerical simulations.

5.2.2 Modelling Turbulence Generated by Internal Leakage

From the results of the numerical simulation in the previous chapter, it is clear that internal leakage through a score produces strong turbulence downstream. It is of great interest to model the MITKE under different flow conditions. This is achieved by means of dimensional analysis, as described in the following.

First, all relevant variables are listed, as follows:

- D_s , the equivalent diameter of a score on the piston seal;
- D_c , the diameter of the chamber of the water hydraulic cylinder;
- ρ , the mass density of the fluid;
- μ , the absolute viscosity of the fluid;
- V_s , the mean velocity of flow at the outlet of the score; and
- E_T , MITKE.

Note that the length of the score does have influence on the MITKE but is not included in the above list. In fact, it influences the MITKE through V_s in the list. It follows that the number of variables in the problem is six, i.e., $k = 6$. Among these

variables, D_s and D_c describe the geometry of the system, ρ and μ define the fluid properties. The total turbulent kinetic energy E_T is defined as the mass-weighted integral of turbulent kinetic energy over the fluid domain within the chamber downstream of the score, as given by

$$E_T = \int_V k \rho dV \quad (5.1)$$

where k is defined by Eq. (4.16) and V refers to the fluid domain within the downstream chamber. E_T has the unit of energy, i.e., $\text{kg}\cdot\text{m}^2/\text{s}^2$.

Note that the equivalent diameter of a score D_s is defined such that a circle of the equivalent diameter has the same area as the cross-section area of the actual score (whether the cross section of the score is circular or noncircular). As the internal leakage rate is of great importance, the equivalent diameter could be more appropriate than the hydraulic diameter. The effects of the equivalent diameter and the hydraulic diameter will be examined later. It is also important to note that the pressure at the outlet of the score is not in the variable list, which follows the conventions of describing a jet from a nozzle. As is known, the jet flowing into an ambient of the same fluid is completely defined by the flow velocity at the nozzle, the diameter of the nozzle and the properties of the fluid [89].

Next all the variables are expressed in terms of basic dimensions, as follows:

$$D_s \doteq L \quad (5.2)$$

$$D_c \doteq L \quad (5.3)$$

$$\rho \doteq FL^{-4}T^2 \quad (5.4)$$

$$\mu \doteq FL^{-2}T \quad (5.5)$$

$$V_s \doteq LT^{-1} \quad (5.6)$$

$$E_T \doteq FL \quad (5.7)$$

where the notation $\doteq$ is used to indicate dimensional equality. It can be seen that all the three basic dimensions are required to describe the above variables, i.e., $r = 3$.

According to Buckingham pi theorem, three ($k - r = 3$) pi terms can be formed.

D_s , V_s , and ρ are selected as repeating variables, whose dimensions are independent of each other. The first pi terms could be formed as follows:

$$\Pi_1 = E_T D_s^a V_s^b \rho^c \quad (5.8)$$

Then,

$$(FL)(L)^a (LT^{-1})^b (FL^{-4}T^2)^c \doteq F^0 L^0 T^0 \quad (5.9)$$

It follows that $a = -3$, $b = -2$, and $c = -1$. Therefore, the first pi term is given by

$$\Pi_1 = \frac{E_T}{D_s^3 V_s^2 \rho} \quad (5.10)$$

The second pi term could be formed as follows:

$$\Pi_2 = \mu D_s^a V_s^b \rho^c \quad (5.11)$$

Then,

$$(FL^{-2}T)(L)^a (LT^{-1})^b (FL^{-4}T^2)^c \doteq F^0 L^0 T^0 \quad (5.12)$$

It follows that $a = -1$, $b = -1$, and $c = -1$. Therefore, the second pi term is given by

$$\Pi_2 = \frac{\mu}{D_s V_s \rho} \quad (5.13)$$

which could be rearranged and expressed as

$$\Pi_2 = \frac{\rho V_s D_s}{\mu} \quad (5.14)$$

Note that Π_2 has the same form as Reynolds number.

The above procedure is repeated for the third pi term.

$$\Pi_3 = D_c D_s^a V_s^b \rho^c \quad (5.15)$$

Then,

$$(L)(L)^a (LT^{-1})^b (FL^{-4}T^2)^c = F^0 L^0 T^0 \quad (5.16)$$

It follows that $a = -1$, $b = 0$, and $c = 0$. Therefore, the third pi term is given by

$$\Pi_3 = \frac{D_c}{D_s} \quad (5.17)$$

From the above dimensional analysis, a relationship among the pi terms is obtained as follows:

$$\frac{E_T}{D_s^3 V_s^2 \rho} = \phi \left(\frac{\rho V_s D_s}{\mu}, \frac{D_c}{D_s} \right) \quad (5.18)$$

The above equation provides a model to describe the MITKE under different internal leakage conditions. However, the actual form of the function ϕ cannot be obtained by means of dimensional analysis. It could be determined by experiments or numerical simulations. In the remainder of this thesis, Eq. (5.18) is referred to as the MITKE model.

5.2.3 Uniqueness of the Dimensional Analysis

It can be seen that the pi terms obtained depend on some arbitrarily selected variables. In the following, the uniqueness of Eq. (5.18) is examined.

The previous procedure is repeated with D_s , V_s , and μ (instead of D_s , V_s , and ρ) being repeating variables. The first pi terms could be formed as follows:

$$\Pi_1' = E_T D_s^a V_s^b \mu^c \quad (5.19)$$

Then,

$$(FL)(L)^a (LT^{-1})^b (FL^{-2}T)^c \doteq F^0 L^0 T^0 \quad (5.20)$$

It follows that $a = -2$, $b = -1$, and $c = -1$. Therefore, the first pi term is given by

$$\Pi_1' = \frac{E_T}{D_s^2 V_s \mu} \quad (5.21)$$

The second pi term could be formed as follows:

$$\Pi_2' = \rho D_s^a V_s^b \mu^c \quad (5.22)$$

Then,

$$(FL^{-4}T^2)(L)^a (LT^{-1})^b (FL^{-2}T)^c \doteq F^0 L^0 T^0 \quad (5.23)$$

It follows that $a = 1$, $b = 1$, and $c = -1$. Therefore, the second pi term is given by

$$\Pi_2' = \frac{\rho D_s V_s}{\mu} \quad (5.24)$$

For the third pi term, it could be formed as

$$\Pi_3' = D_c D_s^a V_s^b \mu^c \quad (5.25)$$

Then,

$$(L)(L)^a (LT^{-1})^b (FL^{-2}T)^c \doteq F^0 L^0 T^0 \quad (5.26)$$

It follows that $a = -1$, $b = 0$, and $c = 0$. Therefore, the third pi term is given by

$$\Pi_3' = \frac{D_c}{D_s} \quad (5.27)$$

Finally, the relationship among the pi terms is in the form:

$$\frac{E_T}{D_s^2 V_s \mu} = \phi_1 \left(\frac{\rho V_s D_s}{\mu}, \frac{D_c}{D_s} \right) \quad (5.28)$$

It can be seen that the resulting pi terms are the same as the previous ones except for the first pi term. Investigating Eqs. (5.10) and (5.21) shows that the pi term

Π_1' in Eq. (5.21) could be obtained by multiplying Π_1 in Eq. (5.10) by Π_2 , i.e.,

$$\frac{E_T}{D_s^2 V_s \mu} = \frac{E_T}{D_s^3 V_s^2 \rho} \cdot \frac{\rho V_s D_s}{\mu} \quad (5.29)$$

Thus, multiplying Eq. (5.18) by Π_2 gives

$$\Pi_1' = \phi(\Pi_2, \Pi_3) \cdot \Pi_2 = \phi_1(\Pi_2, \Pi_3) \quad (5.30)$$

Therefore, although Eqs. (5.18) and (5.28) have different forms, both would lead to the same final equation for E_T .

Next, instead of the equivalent diameter D_s , the cross-section area of the score A_s is selected as a variable, which has a dimension of L^2 . The above dimensional analysis procedure is repeated with A_s , V_s , and ρ being repeating variables. The first pi terms could be formed as follows:

$$\Pi_1'' = E_T A_s^a V_s^b \rho^c \quad (5.31)$$

Then,

$$(FL)(L^2)^a (LT^{-1})^b (FL^{-4}T^2)^c \doteq F^0 L^0 T^0 \quad (5.32)$$

It follows that $a = -3/2$, $b = -2$, and $c = -1$. Therefore, the first pi term is given by

$$\Pi_1'' = \frac{E_T}{A_s^{3/2} V_s^2 \rho} \quad (5.33)$$

The second pi term could be formed as follows:

$$\Pi_2'' = \mu A_s^a V_s^b \rho^c \quad (5.34)$$

Then,

$$(FL^{-2}T)(L^2)^a (LT^{-1})^b (FL^{-4}T^2)^c \doteq F^0 L^0 T^0 \quad (5.35)$$

It follows that $a = -1/2$, $b = -1$, and $c = -1$. Therefore, the second pi term is given by

$$\Pi_2'' = \frac{\mu}{A_s^{1/2} V_s \rho} \quad (5.36)$$

The above procedure is repeated for the third pi term.

$$\Pi_3'' = D_c A_s^a V_s^b \rho^c \quad (5.37)$$

Then,

$$(L)(L^2)^a (LT^{-1})^b (FL^{-4}T^2)^c \doteq F^0 L^0 T^0 \quad (5.38)$$

It follows that $a = -1/2$, $b = 0$, and $c = 0$. Therefore, the third pi term is given by

$$\Pi_3'' = \frac{D_c}{A_s^{1/2}} \quad (5.39)$$

Thus,

$$\frac{E_T}{A_s^{3/2} V_s^2 \rho} = \phi_2 \left(\frac{\rho V_s A_s^{1/2}}{\mu}, \frac{D_c}{A_s^{1/2}} \right) \quad (5.40)$$

Noticing that $A_s = \frac{\pi D_s^2}{4}$, it can be obtained:

$$\Pi_1'' = \frac{E_T}{A_s^{3/2} V_s^2 \rho} = \left(\frac{\pi}{4} \right)^{-3/2} \frac{E_T}{D_s^3 V_s^2 \rho} = \left(\frac{\pi}{4} \right)^{-3/2} \Pi_1 \quad (5.41)$$

$$\Pi_2'' = \frac{\mu}{A_s^{1/2} V_s \rho} = \left(\frac{\pi}{4} \right)^{-1/2} \frac{\mu}{D_s V_s \rho} = \left(\frac{\pi}{4} \right)^{-1/2} \Pi_2 \quad (5.42)$$

$$\Pi_3'' = \frac{D_c}{A_s^{1/2}} = \left(\frac{\pi}{4} \right)^{-1/2} \frac{D_c}{D_s} = \left(\frac{\pi}{4} \right)^{-1/2} \Pi_3 \quad (5.43)$$

Therefore, although Eqs. (5.18) and (5.40) have different forms, both would lead to the same final equation for E_T .

5.2.4 Generality of the MITKE Model

As mentioned previously, dimensional analysis could lead to some general formulations based on experimental/numerical data. In the following, the generality of the MITKE model is examined. This is to check whether the model can represent the internal-leakage-induced turbulence in a general form regardless of properties of water and dimensions of the cylinder.

a) Effect of Properties of Water

As is known, the properties of water vary with temperature. Specifically, the mass density and viscosity of water decrease with temperature. Since the operating temperature of a water hydraulic system is usually limited to the range of 5 to 50 °C, five temperatures in this range were selected to study the effect of properties of water on MITKE, as listed in Table 5.1. A series of numerical simulations were conducted using these properties. Table 5.2 summarizes the input to these simulations. In Table 5.2, the score diameter is arbitrarily selected as 0.2 mm, which is different from the score diameter of 0.3 mm in Table 4.1. The effect of score dimensions on MITKE will be studied later.

Table 5.1 Properties of water at different temperatures [93]

Temperature (°C)	Density (kg/m ³)	Absolute viscosity (N·s/m ²)
10	1000	1.31E-3
20	998	1.00E-3
30	996	7.98E-4
40	992	6.53E-4
50	988	5.47E-4

Table 5.2 Summary of numerical simulation for different properties of water

Parameter	Value
Chamber length	250 mm
Chamber diameter	50 mm
Score length	12 mm
Score diameter	0.2 mm
Angle of the sector	22.5°
Mass density	See Table 5.1
Absolute viscosity	See Table 5.1
Inlet velocity B.C.	0.001, 0.002, 0.003, ..., or 0.010 m/s
Inlet turbulence B.C.	
Turbulence intensity	5%
Hydraulic diameter	50 mm
Outlet pressure B.C.	0 bar
Turbulence model	$k-\omega$ model

Results of the numerical simulations are shown in Figure 5.1 to Figure 5.4. These figures show that the pressure P_1 decreases with temperature; the MITKE increases with temperature. This is because the viscosity of water decreases with temperature. Figure 5.2 shows that the relationship between water velocity at score outlet V_s and the internal leakage q_i does not change with the properties of water. In fact, this relationship is

$$q_i = A_s \times V_s \tag{5.44}$$

Where A_s is the cross section area of the score.

From the numerical results, the three pi terms, Π_1 , Π_2 and Π_3 , were also calculated using Eqs. (5.10), (5.14) and (5.17). Figure 5.4 shows the results of Π_1 versus Π_2 . It can be seen that all the five curves superpose each other, which implies that the function ϕ in the MITKE model, Eq. (5.18), may have a general form, whatever the properties of water are.

Figure 5.1 Pressure versus internal leakage rate for different water temperatures

Figure 5.2 Velocity at score outlet versus internal leakage rate for different water temperatures

Figure 5.3 MITKE versus internal leakage rate for different water temperatures

Figure 5.4 Effect of water temperatures represented by the MITKE model

b) Effect of Size of the Geometric Model

In the following, the generality of the MITKE model for different sizes of the geometric model (see Figure 4.3) is examined. Since the axial dimensions of the cylinder are not of importance, the MITKE model does not comprise parameters representing the axial dimensions of the cylinder. Thus, only the dimensions in the cross section of the cylinder are studied. Five sets of numerical simulations were designed. The numerical simulation described in Table 4.1 was used as a base. Other simulations were designed by scaling the cross section of the geometry of the base simulation. Four scale factors were used, i.e., 0.8, 0.9, 1.1 and 1.2. The scale factors 0.8 and 0.9 indicate that the diameter of the cross section was reduced to eight tenths and nine tenths of that of the base simulation, respectively, while the scale factors 1.1 and 1.2 indicate that the diameter of the cross section was increased by a factor of 1.1 and 1.2, respectively. In addition, the inlet velocities for these four simulations were modified by dividing the inlet velocities of the base simulation by the four scale factors, respectively. This adjustment is to make all the five sets of simulations take the same values for Π_2 , which would facilitate the comparison of the numerical results. For example, consider a scale factor of 0.8. The resulting value of Π_2 is given by

$$\Pi_2 = \frac{\rho(V_s/0.8)(0.8D_s)}{\mu} = \frac{\rho V_s D_s}{\mu}. \text{ Note that the pi term } \Pi_3 = D_c/D_s \text{ is the same for all the five sets of simulations.}$$

Results are shown in Figure 5.5 to Figure 5.8. From Figure 5.5, it can be seen that for the same internal leakage rate, as the scale factor increases, the pressure P_1 decreases. This is because that the cross-section area of the score increases with the scale factor and hence leads to smaller resistance to water flow through the score. Figure 5.6 shows that for the same q_i , V_s decreases with the scale factor. This can be

explained by Eq. (5.44), in which V_s is in inverse proportion to A_s , given the same q_i . In Figure 5.7, E_T decreases with the scale factor for the same q_i . This results from the fact that V_s decreases with the scale factor, as in Figure 5.6. When the numerical results are represented with the MITKE model, all the five curves superpose each other, as shown in Figure 5.8. This result indicates that for different scales of the fluid domain, the MITKE model could give a uniform representation of MITKE.

Figure 5.5 Pressure versus internal leakage rate for different model scales

Figure 5.6 Velocity at score outlet versus internal leakage rate for different
model scales

Figure 5.7 MITKE versus internal leakage rate for different model scales

Figure 5.8 Effects of model scale represented by the MITKE model

c) Effect of Score Diameter

In the previous section, the effects of scale of the geometric model have been examined, where the diameters of the score and the chamber were changed with their ratio being a constant. In order to study the effects of individual diameters, a series of numerical simulations were designed in which one diameter varied, while holding the other diameter constant. Results for the effects of score diameter are described in this section and results for the effects of chamber diameter are described in Section e).

Three sets of numerical simulations were designed. They are the same as the one described in Table 4.1 except that different score diameters D_{score} were used, i.e., 0.2 mm, 0.3 mm and 0.4 mm. Results are shown in Figure 5.9 to Figure 5.12. It can be seen that for the same internal leakage q_i , a small score diameter leads to a high pressure P_1 , a high flow velocity V_s , and hence, a large MITKE. Figure 5.12 shows the pi terms. As the values of Π_3 are different for different D_{score} , the three curves

would not superpose. However, it can be seen that all the curves exhibit the same trend, i.e., Π_1 increases with Π_2 as well as Π_3 .

Figure 5.9 Pressure versus internal leakage rate for different score diameters

Figure 5.10 Velocity at score outlet versus internal leakage rate for different score diameters

Figure 5.11 MITKE versus internal leakage rate for different score diameters

Figure 5.12 Effects of score diameter represented by the MITKE model

d) Effect of Score Cross-section Profile

In the previous section, the cross section profile used in the numerical simulation is a semicircle. In this section, an equilateral triangle is used for the cross section profile. The numerical results will be compared with those in the previous section to investigate the effects of cross section profile of the score on the resulting turbulent flow. Figure 5.13 illustrates the two types of cross section profiles. Smooth semicircle and sharp triangle are two extremes of the cross section profile. In practice, the shape of the cross section usually lies in between these two extremes.

Figure 5.13 Cross section profiles of a score

Since the shapes of the two profiles are different, a consistent method should be selected to calculate the characteristic length of a score. As mentioned in Section 5.2.2, an equivalent diameter, instead of the hydraulic diameter, is used to characterize the size of the cross section of a score. The equivalent diameter is defined such that a circle of the equivalent diameter has the same area as the cross-section area of the score. The calculations of these diameters are given below.

	Equivalent diameter	Hydraulic diameter
Semicircle	$(\sqrt{2}/2)D_{\text{score}}$	$\pi D_{\text{score}}/(2 + \pi)$
Equilateral triangle	$\sqrt{\sqrt{3}/\pi} a_{\text{score}}$	$(\sqrt{3}/3)a_{\text{score}}$

Numerical simulation was conducted using equilateral-triangle score. Details are as listed in Table 4.1 except score diameter. Instead, three values of side length,

a_{score} , were used. They had the same equivalent diameters as those of semicircular scores in the previous section, as listed in Table 5.3.

Table 5.3 Equivalent diameter and hydraulic diameter of score

Profile	Size (mm)	Equivalent diameter (mm)	Hydraulic diameter (mm)
Semicircle	$D_{\text{score}} = 0.2000$	0.1414	0.1222
	$D_{\text{score}} = 0.3000$	0.2121	0.1833
	$D_{\text{score}} = 0.4000$	0.2828	0.2444
Equilateral triangle	$a_{\text{score}} = 0.1905$	0.1414	0.1100
	$a_{\text{score}} = 0.2857$	0.2121	0.1649
	$a_{\text{score}} = 0.3809$	0.2828	0.2199

Figure 5.14 to Figure 5.18 show the results for equilateral-triangle score. For comparison, the results for semicircular score (as in the previous section) are also plotted in these figures. It is clear that for the two cross section profiles, the characteristics of flow quantities, i.e., P_1 , q_i and E_T , are the same. Figure 5.17 shows the effects of score size represented by the MITKE model. Results for the two profiles are close. For the same value of Π_3 , results for triangular score are about 2% higher than those for semicircular score. The equivalent diameter and hydraulic diameter were also compared with respect to the MITKE model. Figure 5.18 shows the effects of score size on MITKE represented by the MITKE model, in which hydraulic diameter was used instead of equivalent diameter. It can be seen that in Figure 5.18, there is no consistent relationship between Π_1 and Π_3 . It is then concluded that the equivalent diameter is a more suitable parameter for characterizing the cross section of a score.

Figure 5.14 Pressure versus internal leakage rate for different score sizes

Figure 5.15 Velocity at score outlet versus internal leakage rate for different
score sizes

Figure 5.16 MITKE versus internal leakage rate for different score sizes

Figure 5.17 Effects of score profile represented by the MITKE model

(D_s = equivalent diameter)

Figure 5.18 Effects of score profile represented by the MITKE model
(D_s = hydraulic diameter)

e) Effect of Chamber Diameter

In Section c), the effects of score diameter have been studied. This section will study the effects of chamber diameter. Three sets of numerical simulations were designed. The simulation described in Table 4.1 was used as a base, in which the chamber diameter D_c was 50 mm. Another two values of D_c were studied, i.e., 40 mm and 60 mm. Results are shown in Figure 5.19 to Figure 5.22.

Figure 5.19 and Figure 5.20 show that the characteristics of P_1 and V_s are the same, regardless of the chamber diameter. That is true because that P_1 and V_s are determined by score dimensions. Figure 5.21 shows that the chamber diameter D_c has some effects on the MITKE. Figure 5.22 shows Π_1 increases with Π_2 and Π_3 .

Figure 5.19 Pressure versus internal leakage rate for different chamber diameters

Figure 5.20 Velocity at score outlet versus internal leakage rate for different chamber diameters

Figure 5.21 MITKE versus internal leakage rate for different chamber diameters

Figure 5.22 Effects of the chamber diameter represented by the MITKE model

In the above, a lot of numerical simulations have been conducted to study the effects of properties of water and geometry of the cylinder on the MITKE. The results show that using the proposed MITKE model, it is possible to obtain a general form to describe the internal-leakge induced turbulence. The functional form of ϕ in the model (see Eq. (5.18)) will be determined in next section.

5.2.5 Determining the MITKE Model

In the previous section, a total of nineteen cases of numerical simulations have been performed, among which, there are sixteen different cases, as listed in Table 5.4. Figure 5.23a shows the relationship among the three pi terms for all the nineteen cases. In this section, a regression method was used to determine the form of the function ϕ from these results. As some of the curves superpose, only four typical curves (corresponding to No. 5, 8, 13 and 19 in Table 5.4) were selected for regression, as shown in Figure 5.23b. These four curves cover the range of Π_3 in the data. Thus, a total of forty data points were used for regression, as listed in Table 5.5.

Table 5.4 List of all numerical simulation cases

No.	D_s (mm)	D_c (mm)	Scale	Temperature (°C)	Remarks
1	0.2	50	1.0	10	Effects of properties of water
2	0.2	50	1.0	20	
3	0.2	50	1.0	30	
4	0.2	50	1.0	40	
5	0.2	50	1.0	50	
6	0.3	50	0.8	30	Effects of geometry scale
7	0.3	50	0.9	30	
8	0.3	50	1.0	30	
9	0.3	50	1.1	30	
10	0.3	50	1.2	30	
11 (= No. 3)	0.2	50	1.0	30	Effects of score diameter
12 (= No. 8)	0.3	50	1.0	30	
13	0.4	50	1.0	30	
14	0.1905*	50	1.0	30	Effects of score cross-section profile
15	0.2857*	50	1.0	30	
16	0.3809*	50	1.0	30	
17	0.3	40	1.0	30	Effects of
18 (= No. 8)	0.3	50	1.0	30	chamber
19	0.3	60	1.0	30	diameter

* Side length of equilateral triangle

Figure 5.23 Variation of Π_1 with Π_2 and Π_3 : (a) all curves; (b) selected 4 curves.

Table 5.5 Data of pi terms for regression

No.	Π_1	Π_2	Π_3
1	29	693	177
2	36	1385	177
3	37	2079	177
4	38	2773	177
5	39	3466	177
6	40	4159	177
7	41	4851	177
8	42	5544	177
9	43	6238	177
10	44	6931	177
11	40	923	236
12	46	1847	236
13	48	2771	236
14	51	3695	236
15	53	4618	236
16	54	5542	236
17	56	6466	236
18	57	7390	236
19	58	8314	236
20	59	9238	236
21	51	1329	283
22	57	2659	283
23	60	3989	283
24	63	5319	283
25	66	6650	283
26	68	7980	283
27	70	9310	283
28	71	10640	283
29	73	11970	283
30	74	13300	283
31	64	2004	354
32	72	4009	354
33	78	6014	354
34	82	8018	354
35	85	10023	354
36	88	12029	354
37	90	14034	354
38	92	16038	354
39	93	18043	354
40	95	20048	354

From Figure 5.23, it is obvious that a power function is suitable for ϕ . Let ϕ take the form:

$$\Pi_1 = b_0 \Pi_2^{b_1} \Pi_3^{b_2} \quad (5.45)$$

Taking natural logarithm of the above equation gives

$$\ln(\Pi_1) = \ln b_0 + b_1 \ln(\Pi_2) + b_2 \ln(\Pi_3) \quad (5.46)$$

Let

$$y = \ln(\Pi_1) \quad x_1 = \ln(\Pi_2) \quad x_2 = \ln(\Pi_3)$$

$$a_0 = \ln b_0 \quad a_1 = b_1 \quad a_2 = b_2$$

The above equation can be written as

$$y = a_0 + a_1 x_1 + a_2 x_2 \quad (5.47)$$

which is a linear equation. The least squares fitting to those data in Table 5.5 gives

$$a_0 = -2.1447 \quad a_1 = 0.1639 \quad a_2 = 0.8655$$

The R^2 value is 0.998, which is sufficiently high to be confident that the model fits the data reasonably well.

Then,

$$b_0 = 0.1171 \quad b_1 = 0.1639 \quad b_2 = 0.8655$$

Therefore, an estimated form of the MITKE model is given by

$$\Pi_1 = 0.1171 \Pi_2^{0.1639} \Pi_3^{0.8655} \quad (5.48)$$

Figure 5.24 plots the resulting MITKE model against the data points from numerical results. Figure 5.25 shows the residuals, which is the difference between the numerical data and the prediction by Eq. (5.48). It can be seen that the residuals are small. Note that the relatively large residuals on the left side of Figure 5.25 are for numerical results of low internal leakage rates. They can be explained by the fact that low internal leakage rates result in weak water flow in the cylinder and such a weak process is susceptible to numerical errors compared with a strong process.

Figure 5.24 Determination of the function ϕ for Π_1 , Π_2 and Π_3

Figure 5.25 Residuals of curve fitting

Substituting Eqs. (5.10), (5.14) and (5.17) into Eq. (5.48) gives

$$\frac{E_T}{D_s^3 V_s^2 \rho} = 0.1171 \left(\frac{\rho V_s D_s}{\mu} \right)^{0.1639} \left(\frac{D_c}{D_s} \right)^{0.8655} \quad (5.49)$$

By rearranging, the above equation can be written as

$$E_T = 0.1975 q_i^{2.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \quad (5.50)$$

The relationship $q_i = \frac{1}{4} \pi D_s^2 V_s$ is used to obtain Eq. (5.50). This model indicates that

E_T increases with q_i , ρ and D_c but decreases with μ and D_s . It is important to note that:

- q_i and D_s have stronger influence on E_T than the other factors, as indicated by their exponents;
- A small D_s usually leads to a small q_i , and therefore, E_T for a small D_s may not greater than that for a large D_s . But given the same q_i , E_T does decrease as D_s increases;

It is important to note that the MITKE model given by Eq. (5.50) describes the MITKE generated by internal leakage through one score only. In practice, more than one score is usually observed on piston seals. The total turbulent kinetic energy (i.e., MITKE) generated by internal leakage through all the scores is the sum of the MITKE generated by internal leakage through each score, since superposition principle is valid for energy. From the assumption made in Section 4.3.1 that all the scores along the circumference of a piston seal are of uniform size, it can be obtained that the internal leakage through each score, and hence the generated MITKE, are also the same. Therefore, the total MITKE can be given by

$$E_T = 0.1975 n q_i^{2.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \quad (5.51)$$

where n is the number of scores on the piston seal.

5.3 Modelling Internal-Leakage-Induced AE

It is well known that the actual AE waveform generated by leakage is difficult to predict [22]. This is largely due to the complex propagation of AE waves. Furthermore, AE signal propagation is much dependent on specific conditions, such as propagation paths and materials, which impedes the development of a general model for AE propagation. In the literature, little effort has been reported on modelling AE waveforms generated by leakage. Alternatively, parameters of AE signals are often studied with correlation to leakage rate and it has been found that some parameters have clear relationships with leakage rate [12]. Therefore, this thesis does not intend to model AE waveforms. In the following, an AE parameter is modelled.

In this research, to model AE signals generated by the internal leakage in water hydraulic cylinders, it is assumed that the rate of AE energy generated by internal leakage is related to the intensity of turbulence induced by that internal leakage. In the previous chapters, it has been shown that the rms value of AE signal is a suitable parameter to characterize the power of AE signal and MITKE is a suitable parameter to measure the intensity of turbulence. Also, it is noticed that the power-law form of relationship has been widely used to model AE signals in various applications (see Section 0). Thus, the relationship between the AE rms value and MITKE is modelled as

$$V_{rms} = A \cdot E_T^m \quad (5.52)$$

where A is a factor used to indicate the radiation efficiency of AE energy from an AE source, the attenuation of AE waves during propagation from an AE source to an AE

sensor and the amplification of AE signal by the measurement system. Such a factor has been widely used in AE modelling [51, 53, 58, 59, 62, 94, 95].

Substituting Eq. (5.51) into Eq. (5.52) gives

$$V_{rms} = A \left(0.1975 n q_i^{2.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \right)^m \quad (5.53)$$

It is important to appreciate that in Eqs. (5.51) and (5.53), q_i is the internal leakage rate through one score but not the internal leakage rate through a water hydraulic cylinder (denoted as Q_i). The internal leakage rate through a hydraulic cylinder is the sum of internal leakage rates through all the scores. In practice, q_i can be regarded as the average leakage rate through a score, i.e.,

$$q_i = Q_i / n \quad \text{or} \quad Q_i = n \cdot q_i \quad (5.54)$$

Substituting Eq. (5.54) into Eq. (5.53) gives

$$V_{rms} = A \left(0.1975 n \left(\frac{Q_i}{n} \right)^{2.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \right)^m \quad (5.55)$$

or

$$V_{rms} = A \left(0.1975 Q_i q_i^{1.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \right)^m \quad (5.56)$$

The values of A and m in the above equation could be determined by experiments. As the values of A and m should be positive, the above equations show that V_{rms} increases with ρ and D_c , while decreases with μ . Given the same internal leakage rate q_i , V_{rms} increases with n as well as Q_i , while decreases with D_s .

It is important to note that Eqs. (5.53), (5.55) and (5.56) are established for standard SI units.

CHAPTER 6

CONDITION MONITORING OF INTERNAL LEAKAGE IN WATER HYDRAULIC CYLINDERS

6.1 Introduction

From Chapter 3 to Chapter 5, AE signals and turbulent flows associated with internal leakage in water hydraulic cylinders have been characterized and an AE model has been established to describe internal-leakage-induced AE signals. This chapter deals with the implementation of AE-based condition monitoring of internal leakage in water hydraulic cylinders.

Observing the AE model (Eq. (5.53)) shows that if those parameters (q_i , ρ , μ , D_s , D_c , A and m) are known, n can be solved at an observed value of V_{rms} , and hence, Q_i can be obtained by Eq. (5.54). Note that n is unknown, because it keeps changing as new scores are generated during the operation of a water hydraulic cylinder. For convenience, those parameters are divided into two groups:

- Hydraulic-system-related parameters (q_i , ρ , μ , D_s , and D_c).
- AE-related parameters (A and m);

As the parameters vary from one application to another, they must be determined for each application before the AE model can be used to predict seal conditions and internal leakage rates in that application. In the following, the process to estimate values of the model parameters for a specific situation is referred to as parameter estimation.

It is important to note that A and m are affected by the following conditions:

- the AE sensor used;

- the propagation path from the AE source to the AE sensor; and
- the data acquisition system.

If anyone of them is changed, the values of A and m must be re-determined. For hydraulic-system-related parameters, they may take different values for different hydraulic systems.

In the remainder of this chapter, the estimation of values of the model parameters is first described, and then, the implementation of AE-based condition monitoring of water hydraulic cylinders is described.

6.2 Parameter Estimation

As mentioned above, values of the model parameters must be determined for each application before the AE model can be used to monitor seal conditions and internal leakage rates in that application. This section describes how to estimate values of the parameters in the AE model.

6.2.1 Hydraulic-System-Related Parameters

Hydraulic-system-related parameters include q_i , ρ , μ , D_s , and D_c . The values of ρ and μ at different water temperature are available in handbooks; the value of D_c can be found in the specifications of a hydraulic cylinder. However, the values of q_i and D_s are not readily available. They must be obtained by investigating failed piston seals. Besides, an average number of scores observed on failed piston seals can be used as the critical value defining the failure of a piston seal (denoted as n_c).

a) Estimation of D_s

It is important to note that score sizes measured on failed piston seals cannot be directly used in the AE model, because scores on a piston seal will deform when the

piston seal is compressed by hydraulic pressure. Ideally, the score size on a piston seal should be measured with that piston seal installed on the piston and compressed by hydraulic pressures. However, it is difficult to conduct such a measurement. A more feasible method is to estimate score sizes based on the relationship between pressure drop across scores and leakage rate through those scores. The following is an example of this method.

In Chapter 3, the score sizes of the 16-score piston seal were measured, as listed in Table 3.4. When the scores are compressed by hydraulic pressure, the actual score sizes should be smaller.

Figure 6.1 shows the experimental data and numerical results of pressure drop across scores versus internal leakage rate through scores. The experimental data are from the experiments described in Section 3.3. The data consist of one hundred points, corresponding to the one hundred data of internal leakage rate in Figure 3.16. Note that the experimental data of internal leakage rate were divided by sixteen to obtain the average internal leakage rate through a score. Thus, they can be compared with numerical results. The numerical results in Figure 6.1 were calculated by numerical simulation described in Section 4.3 but four values of the score diameter D_{score} were used, i.e., 0.1 mm, 0.2 mm, 0.3 mm and 0.4 mm.

Figure 6.1 The relationship between pressure and internal leakage for 16-score piston seal

From Figure 6.1, it is clear that the experimental data are much closer to the numerical results for 0.2-mm score than the others. Therefore, the actual value of the score diameter is estimated as

$$D_{\text{score}} = 0.2 \times 10^{-3} \text{ m}$$

Then, the equivalent diameter is

$$D_s = (\sqrt{2}/2) D_{\text{score}} = 0.1414 \times 10^{-3} \text{ m}$$

In Figure 6.1, the pressure range of experimental data is about 40 to 80 bar. The above value of D_s can be taken as the average diameter for this pressure range. In practice, if a wide pressure range is of interest, it can be divided into small sub-ranges and a score size can be obtained for each sub-range.

As mentioned previously, scores on the circumference of a piston seal are generated due to solid particles in hydraulic fluid. Therefore, the score size depends

on the size of solid particles. These particles are usually generated due to the rubbing between moving parts in hydraulic components. The particle size is characteristic of a hydraulic system (including hydraulic cylinders as well as other components in the hydraulic circuit). It is dependent on materials and surface roughness of moving parts of hydraulic components. Thus, if the above procedures are repeated for a number of failed piston seals, a good estimation of the characteristic score size of that hydraulic cylinder system can be obtained.

b) Estimation of q_i

q_i is the average leakage rate through a score on a failed piston seal. It depends on the characteristic score size of that piston seal rather than the number of scores on that piston seal. It also depends on hydraulic pressure drop across that piston seal.

The values of q_i at different pressures are available from the experimental data of pressure versus leakage rate that are used to estimate D_s in Section a). These data are obtained by the experiment described in Section 3.3.2b. For example, the experimental data in Figure 6.1 shows that for the 16-score piston seal, q_i is around 0.05 L/min at 75 bar.

6.2.2 AE-related Parameters

For a specific water hydraulic cylinder system, after the hydraulic-system-related parameters are determined, the values of AE-related parameters (A and m) can be estimated experimentally. Experiments described in Section 3.3.2 can be conducted, where actual failed piston seals (the number of scores on these seals is known) should be used instead of artificially scored piston seals. In the experiments, internal leakage rates and internal-leakage-induced AE signals can be measured. By curve fitting of the AE model (Eq. (5.55)) to these experimental data, the values of A

and m can be obtained. The following is an example to show how to obtain the values of A and m .

In Section 3.3, a quantitative measurement of internal leakage rate through a 16-score piston seal and AE signals were conducted, in which the piston rod was fixed. Figure 3.16 shows the resulting data of the AE rms value against the internal leakage rate. Note that the leakage rate in Figure 3.16 is the total internal leakage Q_i through the water hydraulic cylinder. These results were obtained under the following conditions:

- mass density of water, $\rho = 996 \text{ kg/m}^3$;
- absolute viscosity of water, $\mu = 7.98 \times 10^{-4} \text{ N}\cdot\text{s/m}^2$;
- inner diameter of the water hydraulic cylinder, $D_c = 50 \times 10^{-3} \text{ m}$;
- average diameter of scores, $D_s = 0.1414 \times 10^{-3} \text{ m}$ (as determined in Section 6.2.1); and
- number of scores on the piston seal, $n = 16$.

Substituting the above information into Eq. (5.55) gives

$$V_{rms} = A q_1^m \quad (6.1)$$

where

$$\begin{aligned} q_1 &= 0.1975 n \left(\frac{Q_i}{n} \right)^{2.1639} \frac{\rho^{1.1639} D_c^{0.8655}}{\mu^{0.1639} D_s^{2.0294}} \\ &= 0.1975 \times 16 \times \left(\frac{Q_i}{16} \right)^{2.1639} \frac{996^{1.1639}}{(7.98 \times 10^{-4})^{0.1639}} \frac{(50 \times 10^{-3})^{0.8655}}{(0.1414 \times 10^{-3})^{2.0294}} \\ &= 1.5253 \times 10^{11} \times \left(\frac{Q_i}{16} \right)^{2.1639} \end{aligned}$$

Taking the natural logarithm of Eq. (6.1) gives

$$\ln(V_{rms}) = \ln(A) + m \ln(q_1) \quad (6.2)$$

Let $\ln(V_{rms})$ be the dependent variable, $\ln(q_1)$ the independent variable. Eq. (6.2) can be regarded as a linear equation. By fitting Eq. (6.2) to the experimental data, as in Figure 3.16, using the linear regression method [87], the unknown parameters A and m are computed as

$$A = 1.4319$$

$$m = 0.6227$$

Figure 6.2 shows the fit of the AE model to the experimental data. Two axis scales are used: (a) logarithmic scale; (b) linear scale. It can be seen that the AE model gives a good fit to the experimental data.

Figure 6.2 Curve fitting of the AE model to experimental data: (a) logarithmic scale; (b) linear scale

The 95% prediction interval (see Eq. (3.17)) is also plotted in Figure 6.2. Due to the nonlinear exponential transform from Eq. (6.2) to Eq. (6.1), the nearly constant interval in Figure 6.2a transforms into a taper shape in Figure 6.2b. Therefore, it is not suitable to measure the goodness of the fit. Alternatively, the quality of the fit can be measured by the residual, which is the difference between the experimental data and

the AE model. Figure 6.3a shows the fit residuals of the AE model to the experimental data. The maximum absolute value of the residuals is 0.011 V. Figure 6.3b shows the percentage of residuals to the observed values in the experiment. It is found that the average residual is 9.6%. This is reasonably small. It is important to appreciate that the residuals are mainly due to the variation in the experimental data rather than the AE model. From the above analysis, it is evident that the AE model gives a good fit to the trend in the experimental data.

Figure 6.3 The fit residuals of the AE model to the experimental data: (a) magnitude; (b) relative residual

By substituting the above values of parameters into Eq. (5.55), a determined AE model can be obtained for the above example, given by

$$V_{rms} = 1.4319 \left(0.1975 n \left(\frac{Q_i}{n} \right)^{2.1639} \frac{996^{1.1639}}{(7.98 \times 10^{-4})^{0.1639}} \frac{(50 \times 10^{-3})^{0.8655}}{(0.1414 \times 10^{-3})^{2.0294}} \right)^{0.6227} \quad (6.3a)$$

$$= 2.3441 \times 10^6 \left[n \left(\frac{Q_i}{n} \right)^{2.1639} \right]^{0.6227}$$

or

$$V_{rms} = 2.3441 \times 10^6 (n q_i^{2.1639})^{0.6227} \quad (6.3b)$$

$$V_{rms} = 2.3441 \times 10^6 (Q_i q_i^{1.1639})^{0.6227} \quad (6.3c)$$

In summary, the experiment described in Section 3.3.2 can be carried out to determine values of parameters in the AE model. The experimental data of pressure against internal leakage rate can be used to estimate the value of D_s , while the experimental data of AE rms value against internal leakage rate can be used to determine values of A and m .

Besides, it is interesting to compare the simple AE model (Eq. (3.14)) and the complex AE model (Eq. (6.3)). Note that Q_i in Eq. (3.14) is in the unit of L/min, while Q_i in Eq. (6.3) is in the unit of m^3/sec . Since Eq. (3.14) is established for the 16-score piston seal, let $n = 16$ in Eq. (6.3). Figure 6.4 shows the comparison between these two models. It is clear that Eq. (3.14) is a close approximation to Eq. (6.3) in the leakage range of 0.4 to 0.8 L/min. Note that when the leakage rate is small enough, the AE rms value predicted by Eq. (3.14) could be negative, which is not rational. Therefore, the AE model given by Eq. (6.3) is more accurate than the simple model given by Eq. (3.14).

Figure 6.4 Comparison between Eq. (3.14) and Eq. (6.3a)

6.3 Verifying the AE Model for Different Score Numbers

To verify the validity of Eq. (6.3) for a different number of scores, experiments described in Section 3.3.2 were repeated, but a 12-score piston seal and an 8-score piston seal were used instead of the 16-score seal. The numbers of scores, i.e., 12 and 8, were arbitrarily selected. Figure 6.5 shows the 12-score piston seal and the 8-score piston seal. Table 6.1 lists the measured score dimensions in comparison with those of the 16-score piston seal.

Figure 6.5 Piston rings: (a) 8 scores; (b) 12 scores

Table 6.1 Score dimensions

Dimension	16 scores	12 scores	8 scores
Radius (mm)			
Average	0.3924	0.3963	0.4319
Standard deviation	0.0768	0.0562	0.0209
Depth (mm)			
Average	0.3141	0.2553	0.3599
Standard deviation	0.1087	0.0532	0.0740

It can be seen that the score dimensions of the 12-score and 8-score piston seals

are close to those of the 16-score piston seals. Therefore, D_s of the 12-score and 8-score piston seals is estimated as 0.1414×10^{-3} m, the same as that of the 16-score piston seals.

In the experiments, fifty data points of internal leakage rate through the water hydraulic cylinder and AE rms values thereby generated were measured for the 12-score and 8-score piston seals, respectively. Figure 6.6 and Figure 6.7 shows the experimental data as well as the prediction by the AE model. The prediction is calculated using Eq. (6.3a) with n equal to 12 and 8, respectively. From these figures, it is evident that the AE model gives a reasonably accurate prediction of internal-leakage-induced AE signals for various numbers of scores.

Figure 6.6 AE rms versus internal leakage rate for the 12-score piston seals

Figure 6.7 AE rms versus internal leakage rate for the 8-score piston seals

Figure 6.8 and Figure 6.9 show AE rms value versus internal leakage rate for different numbers of scores. They are calculated with Eqs. (6.3a) and (6.3b), respectively. It is clear that given the same internal leakage rate Q_i , the AE rms value decreases with n (this is because the flow velocity through a score decreases as n increases, given the same Q_i); while given the same internal leakage rate q_i , the AE rms value increases with n .

Figure 6.8 AE rms value versus internal leakage rate Q_i for different number of scores

Figure 6.9 AE rms value versus internal leakage rate q_i for different number of scores

6.4 Condition Monitoring of Internal Leakage

After the values of parameters in Eq. (5.53), (5.55) or (5.56) have been determined for a specific water hydraulic cylinder system, the AE model is ready for use for condition monitoring of internal leakage and piston seal integrity of that water hydraulic cylinder system. The following describes the implementation of AE-based condition monitoring of internal leakage in water hydraulic cylinders. Two strategies are proposed, namely, static condition monitoring and dynamic condition monitoring.

6.4.1 Static Condition Monitoring

In the strategy of static condition monitoring, the rod of a water hydraulic cylinder is fixed and a static hydraulic pressure is applied to one of the chambers of that water hydraulic cylinder. The procedures to conduct static condition monitoring are proposed, as follows:

1. Fix the cylinder rod of a water hydraulic cylinder at mid-stroke such that the piston can sustain hydraulic pressures (it is unnecessary to disconnect the water hydraulic cylinder from the hydraulic system);
2. Apply a constant hydraulic pressure to one chamber of that hydraulic cylinder (this pressure is referred to as test pressure in the following);
3. Measure AE signals and calculate AE rms values using Eq. (3.1);
4. Use the established AE model (such as Eq. (6.3)) to estimate the internal leakage rate as well as the integrity of piston seals.

Consider an example of Eq. (6.3). Suppose the test pressure is 75 bar. Then, q_i is equal to 0.05 L/min ($8.3333 \text{ m}^3/\text{s}$), as shown in Figure 6.1. Substituting this value into Eqs. (6.3b) and (6.3c) gives

$$V_{rms} = 0.0151n^{0.6227} \quad (6.4a)$$

$$V_{rms} = 92.0501Q_i^{0.6227} \quad (6.4b)$$

Eq. (6.4) can be re-written as

$$n = 841.6612V_{rms}^{1.6059} \quad (6.5a)$$

$$Q_i = 7.0138 \times 10^{-4} V_{rms}^{1.6059} \quad (6.5b)$$

Using Eq. (6.5), the number of scores on a piston seal and the internal leakage rate at the test pressure can be predicted based on the measured AE rms value. Furthermore, let n_c be the critical number of scores and Q_c the critical internal leakage rate defining the failure of a piston seal. If the predicted value is larger than the critical value, i.e.,

$$n > n_c \text{ or } Q_i > Q_c,$$

a piston seal is suspected to have failed and some maintenance work should be done. Note that the values of n_c and Q_c depends on applications. Some applications have stringent requirements on n_c and Q_c , while the others do not.

It is noted that the value of n predicted by Eq. (6.5a) is unlikely to be an integer. Nevertheless, the value of n predicted by Eq. (6.5a) can still be a quantitative index of a piston seal condition.

In summary, by using the strategy of static condition monitoring, a quantitative prediction of internal leakage and piston seal integrity can be achieved. This strategy provides an efficient and sensitive method for condition monitoring of a water hydraulic cylinder.

6.4.2 Dynamic Condition Monitoring

In contrast to the static condition monitoring, a dynamic condition monitoring is to monitor the condition of a water hydraulic cylinder during its reciprocating motion.

Using this strategy, the operation of a water hydraulic cylinder is not interrupted.

a) Dynamic Condition Monitoring Strategy

In the static condition monitoring, the values of D_s and q_i are obtained experimentally, as described in Section 6.2.1 (details of the experiments are given in Section 3.3.2b). However, these values should not be used for a dynamic condition monitoring, because the loading conditions used in Section 3.3.2b are different from those in a dynamic condition monitoring. In Section 3.3.2b, the piston is at rest and a steady hydraulic pressure is applied. In a dynamic condition monitoring, the piston is in reciprocating motion. Due to limited strokes, the flow conditions (pressure and flow rate) are quite dynamic. Therefore, it is rational to expect that even for the same piston seal, the values of q_i and D_s may be different for the two loading conditions. Note that the other parameters (ρ , μ , and D_c , A and m) are not affected by loading conditions.

Since it is difficult to measure the values of q_i and D_s under dynamic loading conditions, a completely determined AE model (such as Eq. (6.5)) cannot be obtained for the dynamic condition monitoring. To overcome this issue, a strategy different from the static condition monitoring strategy is proposed.

Consider two piston seals with different number of scores, namely, Seal 1 and Seal 2. From Eq. (5.53), the AE models for these two seals are given by

$$V_{rms1} = A \left(0.1975 n_1 q_i^{2.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \right)^m \quad (6.6)$$

$$V_{rms2} = A \left(0.1975 n_2 q_i^{2.1639} \frac{\rho^{1.1639}}{\mu^{0.1639}} \frac{D_c^{0.8655}}{D_s^{2.0294}} \right)^m \quad (6.7)$$

In Eqs. (6.6) and (6.7), since q_i and D_s vary with flow conditions, i.e., hydraulic

pressure p and flow rate Q , V_{rms1} and V_{rms2} are a function of p and Q . Suppose the two seals work under the same flow conditions, which implies that they take the same values of q_i and D_s . Then, Eq. (6.6) divided by Eq. (6.7) gives

$$\frac{V_{rms1}(p, Q)}{V_{rms2}(p, Q)} = \left(\frac{n_1}{n_2} \right)^m \quad (6.8)$$

Let Seal 2 be a reference seal, for which the number of scores is known. The AE rms values generated by this reference seal under dynamic flow conditions can be obtained by experiments. Thus, from Eq. (6.8), the number of scores on a piston seal being monitored can be predicted, as given by

$$n = n_0 \left(\frac{V_{rms}(p, Q)}{V_{rms0}(p, Q)} \right)^{1/m} \quad (6.9)$$

where n_0 is the number of scores on the reference seal; $V_{rms0}(p, Q)$ is the AE rms value generated by the reference seal under the flow condition (p, Q) ; and $V_{rms}(p, Q)$ is the AE rms value generated by the seal being monitored under the flow condition (p, Q) .

In practice, a failed piston seal can be used as a reference seal. Experiments can be repeated for a number of failed piston seals to obtain a good estimation of

$n_0 \left(\frac{1}{V_{rms0}} \right)^{1/m}$. Then, Eq. (6.9) can be used to predict the number of scores on a piston

seal based on the measured AE rms value V_{rms} .

Note that the flow condition (p, Q) is affected by externally applied loads to the water hydraulic cylinder, and therefore, it may vary during the operation of the water hydraulic cylinder. In contrast, constant flow conditions are used in the static condition monitoring strategy. Since it is not feasible to conduct experiments to measure $V_{rms0}(p, Q)$ for the reference seal under all these flow conditions, an appropriate matrix of hydraulic pressures and flow rates, covering the working range

of the hydraulic cylinder, should be used to collect data of $V_{rms0}(p, Q)$. The value of V_{rms0} at other pressures and flow rates may be obtained by interpolation.

By using Eq. (6.9), condition monitoring of seal integrity can be conducted under dynamic flow conditions. This method is referred to as dynamic condition monitoring strategy in this thesis. Because information on q_i and D_s under dynamic flow conditions is not available, this strategy cannot give a quantitative estimation of the internal leakage rate. However, n can be used as an index to indicate the level of internal leakage rate. Besides, this strategy can be used for on-line condition monitoring, that is, condition monitoring can be conducted continuously without interrupting the operation of a water hydraulic cylinder.

b) Experimental Verification

Experiments were conducted to verify the proposed dynamic condition monitoring strategy. The experimental setup is the same as described in Section 3.3.1. Internal leakage was artificially introduced by the use of scored piston seals, as described in Section 3.3.2. The previously mentioned piston seals were used, namely, 16-score piston seal, 12-score piston seal and 8-score piston seal. In addition, a completely new piston seal ($n = 0$) was also included. Mass weights were used to load the water hydraulic cylinder, as shown in Figure 6.10. Different hydraulic pressures were achieved by applying different amount of weights, while the flow rate remains constant, which is controlled by a proportional valve.

Figure 6.10 Loading of water hydraulic cylinder

During the reciprocating motion, AE signals were measured when the piston was at mid-stroke during the extending stroke (this is the stroke moving against gravity load). Forty sets of AE signals were collected at different hydraulic pressures for each piston seal. Each set consisted of 40 records of AE signals measured at a certain hydraulic pressure. And each record of AE signals contained 4096 points sampled at 5 MHz, from which an AE rms value was calculated. Then, the AE rms values calculated from the 40 records were averaged.

Figure 6.11 shows this average AE rms value versus the hydraulic pressure drop across the piston. The flow rate was 8.4 L/min for all the data. It can be seen that there are obvious trends in the experimental data: 1) the AE rms value increases linearly with hydraulic pressure; 2) the larger the number of scores on a piston seal, the higher the AE rms value. It is important to note that the AE rms value for $n = 0$ is mainly attributed to AE noise, such as that generated by friction between piston and cylinder

wall. As shown in Figure 6.11, this noise does not change significantly as the hydraulic pressure increases.

Figure 6.11 AE rms value versus hydraulic pressure during the reciprocating motion of the water hydraulic cylinder

Let V_{16} , V_{12} , V_8 and V_{new} denote the AE rms values for the 16-score seal, 12-score seal, 8-score seal and new seal, respectively. Since AE noise is not related to internal leakage, it should be removed from AE signals. This can be accomplished by

$$V'_i = \sqrt{V_i^2 - V_{new}^2}, i = 16, 12, 8 \quad (6.10)$$

where V'_i is the AE rms value after AE noise is removed. Because superposition can be applied to AE energy but not AE rms value, AE noise is removed by subtracting the square of V_{new} from the square of V_i . Note that the square of AE rms value represents the AE energy measured in the time domain.

Let the 16-score seal be a reference seal. From Eq. (6.9), it can be obtained that

$$n_i = 16 \left(\frac{V_i'(p)}{V_{16}'(p)} \right)^{1/0.6227}, i = 12, 8 \tag{6.11}$$

In Eq. (6.11), n_i is predicted based on the ratio of V_i' to V_{16}' at a hydraulic pressure (for all the experimental data in Figure 6.11, the flow rate is a constant, V_i' and V_{16}' vary with hydraulic pressure only). Note that $m = 0.6227$ obtained in Section 6.2.2 is used in Eq. (6.11), since the value of m is not affected by loading conditions.

The results predicted by Eq. (6.11) are shown in Figure 6.12 and Figure 6.13. The abscissa in the figures indicates that forty sets of data were collected for the 12-score seal and the 8-score seal, respectively. Table 6.2 lists the details of the prediction. It can be seen that the proposed dynamic condition monitoring strategy is able to predict the seal integrity with sufficient accuracy.

Figure 6.12 Prediction of number of scores by the dynamic condition monitoring strategy ($n = 12$)

Figure 6.13 Prediction of number of scores by the dynamic condition monitoring strategy ($n = 8$)

Table 6.2 Prediction by the dynamic condition monitoring strategy

Piston seal	Actual score number	Predicted score number	
		mean	Standard deviation
12-score seal	12	11.0	0.8
8-score seal	8	7.9	0.6

In summary, the procedures to implement the dynamic condition monitoring strategy are proposed, as follows:

1. a water hydraulic cylinder is operated as usual, conducting its reciprocating motion;
2. measure AE signals, flow rate and hydraulic pressure drop across the piston when the piston is at mid-stroke, either during extending or retracting stroke

- (the working stroke should be selected, which is the stroke moving against loads);
3. calculate an AE rms value using Eq. (3.1);
 4. remove AE noise using Eq. (6.10);
 5. compare the AE rms value to that of a reference seal under the same flow conditions and calculate their ratio; and
 6. predict the integrity of piston seal using Eq. (6.9).

6.5 Discussion

In the above, two strategies have been proposed for condition monitoring of a water hydraulic cylinder. One is the static condition monitoring strategy and the other is the dynamic condition monitoring strategy. Table 6.3 compares these two strategies.

Table 6.3 Comparison between condition monitoring strategies

Item	Static strategy	Dynamic strategy
Flow conditions	Static	dynamic
Measure AE signals	Yes	Yes
Measure hydraulic pressure	No*	Yes†
Measure flow rate	No	Yes†
Predict seal integrity	Yes	Yes
Predict internal leakage rate	Yes‡	No
Conduct on-line monitoring	No	Yes

* It is required to apply a known hydraulic pressure.

† Since the flow conditions are dynamic, hydraulic pressure and flow rate may change during the operation of a water hydraulic cylinder. To predict seal integrity, it is necessary to measure hydraulic pressure and flow rate.

‡ It can predict internal leakage rate under the static conditions but not the dynamic conditions.

From Table 6.3, it is clear that the static strategy is relatively simple, while the dynamic strategy requires much effort (AE signals, hydraulic pressures and flow rates are all required) but can be used for on-line condition monitoring.

For practical implementation, both the static condition monitoring and the dynamic condition monitoring can be conducted without dismantling the hydraulic cylinder system. The static condition monitoring strategy requires the operation of the hydraulic cylinder to be interrupted, which implies potential down time. To reduce down time, this test may be scheduled in the regular maintenance time, if any. In fact, the test can be completed with a few operations in a short time, as listed below.

- Stop the hydraulic cylinder being monitored (it is unnecessary to stop the other components in the hydraulic system, such as the pump);
- Fix the cylinder rod (this can be accomplished quickly with the help of some jigs);
- Mount an AE sensor on the hydraulic cylinder, if a permanent mounting is not adopted for the AE sensor;
- Apply a test pressure to the hydraulic cylinder (the test pressure is supplied by the pump); and
- Record and analyze AE signals (this can be easily done by a computer program).

Therefore, it does not necessarily lead to an increase in down time. As for the dynamic condition monitoring strategy, it can be conducted continuously without interrupting the operation of the hydraulic cylinder being monitored. However, due to the complex dynamic flow conditions during the operation of the cylinder, the accuracy that the dynamic condition monitoring can achieve may be lower than that of the static condition monitoring.

In practice, if close condition monitoring of the internal leakage is required, the two strategies can be used together to achieve this. The dynamic condition monitoring is used for on-line monitoring of piston seal integrity. Once deterioration of piston

seal is detected, the static condition monitoring can be conducted to give a more accurate estimation of internal leakage rate and seal integrity. If close condition monitoring of the internal leakage is not required, the static strategy can be a suitable choice. It can be conducted on a regular basis. This will save the cost due to the dynamic condition monitoring.

Compared with conventional condition monitoring methods for hydraulic systems, such as pressure method, flow rate method and vibration method, the AE method is sensitive, accurate and does not require complex signal analysis techniques. As mentioned in Chapter 2, conventional methods usually involve complex techniques to extract features from original signals. This is due to the limitation of conventional signals. Their characteristics are largely determined by the whole hydraulic system. On the contrary, AE are sensitive to a local source mechanism. AE signals detected at a hydraulic cylinder are mainly due to small internal leakage across the piston seals. With this advantage, the AE method developed in this thesis is able to monitor internal leakage rate or piston seal integrity even when the internal leakage is smaller than 1 L/min, whereas those conventional methods have difficulty in monitoring such small internal leakage.

Note that the dynamic condition monitoring strategy described in this chapter is not used for quantitative prediction of internal leakage rate. It is designed such that the difficulty in measuring q_i and D_s under dynamic flow conditions can be avoided. Thus, it will be easy to implement this strategy for an on-line inspection of seal integrity. Nevertheless, if the values of q_i and D_s under dynamic flow conditions are given, the AE model (Eq. (5.53)) can also be used for on-line quantitative condition monitoring of internal leakage rate without interrupting the operation of the hydraulic cylinder being monitored. In next chapter, some future work is recommended for

studying the method for measuring q_i and D_s under dynamic flow conditions.

In summary, two AE-based condition monitoring strategies were proposed in this chapter and verified by experimental results. This provides a new method for condition monitoring of internal leakage rate and piston seal integrity. With the promising results obtained, some future work will be recommended in next chapter for further improvement of this AE method for condition monitoring of internal leakage in water hydraulic cylinders.

CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

Modern water hydraulic cylinder systems are subject to internal leakage due to the very low viscosity of water as compared with hydraulic oil. Therefore, it is desirable to monitor such internal leakage to make sure it is within an acceptable level. The author has developed a new AE-based method for condition monitoring of internal leakage in a water hydraulic cylinder. Specially-designed experiments were conducted to study the characteristics of internal-leakage-induced AE signals and comprehensive numerical simulations were performed to investigate the characteristics of turbulence generated by internal leakage in a water hydraulic cylinder. A new AE model was established to describe internal-leakage-induced AE signals. Based on this AE model, two condition monitoring strategies were proposed and experimental results showed they could predict the seal integrity and/or internal leakage rate with sufficient accuracy. Below are conclusions and recommendations for future work.

7.1 Conclusions

1. Sensitivity of AE to small internal leakage in water hydraulic cylinders

Experimental results showed that small internal leakage (less than 1 L/min) in water hydraulic cylinders can generate significant acoustic emission waves and a clear correlation exists between internal leakage rate and AE signals thereby generated. Therefore, AE could be a sensitive method to monitor small internal leakage as compared with conventional methods (pressure, flow rate, temperature, vibration,

etc.). An AE-based method could be developed to predict the internal leakage rate in a water hydraulic cylinder.

2. Characterization of Internal-leakage-induced AE signals

Special experiments were designed to study the characteristics of AE signals generated by internal leakage in a water hydraulic cylinder. Various parameters were extracted from AE signals and their correlations with internal leakage rate were studied in detail. Results showed that energy-based AE parameters, whether in the time domain or in the frequency domain, have a clear relationship with the internal leakage rate. Especially, the AE rms value is a most desirable parameter to represent quantitatively internal-leakage-induced AE signals.

3. Characterization of Internal-leakage-induced turbulence

MITKE was defined in this thesis to quantitatively measure the intensity of turbulence due to internal leakage. It has an advantage over the other turbulence-related quantities. The other turbulence-related quantities usually measure a local intensity of turbulence in a fluid domain, but MITKE is capable of measuring the intensity of turbulence for the entire fluid domain.

A numerical model was created to study the characteristics of turbulent flow generated by internal leakage in a water hydraulic cylinder. Results showed that MITKE increases with the internal leakage rate and it could be an appropriate parameter to describe internal-leakage-induced turbulence.

4. Modelling the relationship between internal leakage rate and AE signals

To build a quantitative relationship between internal leakage rate and AE signals, this thesis proposed a new method based on the physical insight into the mechanism of internal-leakage-induced AE. It is important to appreciate that AE is not directly related to internal leakage. Instead, AE is generated by turbulence, which,

in turn, is induced by internal leakage. Therefore, turbulence is a link between internal leakage and AE.

By dimensional analysis, a general model was built to describe the relationship among dimensions of water hydraulic cylinder, properties of water, velocity of leaking flow and the resulting MITKE. Comprehensive numerical computations were performed to investigate the effects of various flow conditions on MITKE. From the numerical results, a MITKE model was determined, which gives a relationship between the internal leakage rate and the MITKE thereby generated.

Finally, the AE rms value was modelled as the power function of the MITKE, i.e., $V_{rms} = A \cdot E_T^m$. By combining this power-law relationship with the above MITKE model, a quantitative AE model was obtained to correlate the internal leakage rate and the AE rms value. Experimental data showed that this AE model can describe the internal-leakage-induced AE rms value with reasonable accuracy.

5. Condition monitoring of internal leakage or piston seal integrity in water hydraulic cylinders

Condition monitoring of internal leakage or piston seal integrity in a water hydraulic cylinder is difficult to achieve using conventional methods. In this thesis, an AE-based method was developed to solve this problem. Two strategies were proposed for condition monitoring of internal leakage and seal integrity under static conditions and dynamic conditions, respectively. Experimental results showed that using these strategies, the internal leakage rate or the seal integrity could be predicted with sufficient accuracy.

7.2 Recommendations for Future Work

Based on the results from this research, some future work is recommended, as described in the following.

1. Further study for the dynamic condition monitoring

As mentioned previously, in the dynamic condition monitoring, the values of D_s and q_i are unknown and therefore a strategy is proposed to overcome this problem. Using that strategy, the number of scores on the piston seal could be predicted with sufficient accuracy, which could then be used as an indicator of the internal leakage level. Further studies are required, if one wants to predict the actual internal leakage rate besides the number of scores. In essence, this requires a method to estimate the actual values of D_s and q_i when a water hydraulic cylinder is in operation, i.e., in reciprocating motion. The following experiment is recommended to achieve this.

The water hydraulic system, as in Figure 3.3, can be modified such that the double-acting water hydraulic cylinder can work as a single-acting one. The cylinder extends when water from the pump enters the piston end of the cylinder. Retraction is accomplished by mass weights. Leaking water will be collected at the rod end, as shown in Figure 7.1. Leakage from multiple strokes of extension may be collected and used to calculate an average leakage for one stroke. Various hydraulic pressures can be obtained by adjusting the load (mass weights) and various flow rates can be obtained by adjusting the flow control valve. From the experimental data, the actual values of D_s and q_i under different flow conditions can be estimated using the procedures described in Section 6.2.1.

Figure 7.1 Further study for the dynamic condition monitoring

2. Application of the AE method to other internal leakage problems

The core of the AE method developed in this thesis is to use MITKE to link internal leakage rate and AE signals. The relationship between MITKE and internal leakage rate is obtained by the combination of dimensional analysis and numerical simulation, and the relationship between MITKE and AE rms values is modelled by Eq. (5.52).

Since the validity of the above methodology is not limited to internal leakage in water hydraulic cylinders, it is recommended to study the AE method with application to other leakage problems, which involve internal-leakage-induced turbulent flow of incompressible Newtonian fluid (this is an assumption made in Chapter 4). An example may be internal leakage in hydraulic valves. Such work helps to understand the generality of the AE method developed in this research.

3. Study of AE signals during transient processes

In this thesis, AE signals were all measured when the piston was at mid-stroke. These AE signals were of continuous type. It is also interesting to study the characteristics of AE signals that are generated when a water hydraulic cylinder starts, stops or changes the direction at the end of the extending or retracting stroke. During these transient processes, flow conditions are highly dynamic, the resulting AE

signals may be of burst type with higher amplitude than those obtained in this thesis. The correlation between the AE signals with piston seal integrity and internal leakage can be studied to see whether these AE signals are more sensitive to piston seal integrity and internal leakage. In addition, the characteristics of these AE signals and background noise can be studied to see whether it is easier to distinguish these AE signals from background noise.

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