EXPERIMENTAL STUDY OF TRANSFORMER LIQUID FLOW AND TEMPERATURE DISTRIBUTIONS

A thesis submitted to The University of Manchester for the degree of Doctor of Philosophy in the Faculty of Science and Engineering

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Muhammad Daghrak

School of Electrical and Electronic Engineering
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### Abbreviations

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<th>Description</th>
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<tbody>
<tr>
<td>A&lt;sub&gt;c&lt;/sub&gt;</td>
<td>Channel cross sectional area (m&lt;sup&gt;2&lt;/sup&gt;)</td>
</tr>
<tr>
<td>β</td>
<td>Thermal expansion coefficient (1/K)</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>CT</td>
<td>Current transformer</td>
</tr>
<tr>
<td>CJC</td>
<td>Cold junction compensation</td>
</tr>
<tr>
<td>c&lt;sub&gt;p&lt;/sub&gt;</td>
<td>Heat capacity (J/kgK)</td>
</tr>
<tr>
<td>CHT</td>
<td>Conjugate heat transfer</td>
</tr>
<tr>
<td>D</td>
<td>Hydraulic diameter (m)</td>
</tr>
<tr>
<td>D&lt;sub&gt;th&lt;/sub&gt;</td>
<td>Winding model depth in (m)</td>
</tr>
<tr>
<td>E&lt;sub&gt;AB&lt;/sub&gt;</td>
<td>Open voltage produced by thermocouples</td>
</tr>
<tr>
<td>FEM</td>
<td>Finite Element Method</td>
</tr>
<tr>
<td>FOTS</td>
<td>Fibre optic temperature sensors</td>
</tr>
<tr>
<td>f</td>
<td>Friction coefficient</td>
</tr>
<tr>
<td>FR</td>
<td>Inlet flow rate (lpm)</td>
</tr>
<tr>
<td>FR&lt;sub&gt;PIV&lt;/sub&gt;</td>
<td>Inlet flow rate measured by summing PIV measurements in radial cooling ducts (lpm).</td>
</tr>
<tr>
<td>FR&lt;sub&gt;M&lt;/sub&gt;</td>
<td>Flow rate recorded by the positive displacement flow meter (lpm).</td>
</tr>
<tr>
<td>g</td>
<td>Temperature gradient between Δθ&lt;sub&gt;w&lt;/sub&gt; and Δθ&lt;sub&gt;om&lt;/sub&gt; (K)</td>
</tr>
<tr>
<td>Gr</td>
<td>Grashof number</td>
</tr>
<tr>
<td>HST or θ&lt;sub&gt;h&lt;/sub&gt;</td>
<td>Hot spot temperature (°C)</td>
</tr>
<tr>
<td>NHTM</td>
<td>Network hydraulic thermal model</td>
</tr>
<tr>
<td>HV</td>
<td>High voltage winding</td>
</tr>
<tr>
<td>H</td>
<td>Hot spot factor</td>
</tr>
<tr>
<td>HWA</td>
<td>Hot wire anemometry</td>
</tr>
<tr>
<td>ΔH</td>
<td>Height difference between radiator centre and transformer winding centre (m)</td>
</tr>
<tr>
<td>h&lt;sub&gt;p&lt;/sub&gt;</td>
<td>Pump head (m)</td>
</tr>
<tr>
<td>h&lt;sub&gt;f&lt;/sub&gt;</td>
<td>Friction head loss in a hydraulic system (m)</td>
</tr>
<tr>
<td>h</td>
<td>Convective heat transfer coefficient (W/m&lt;sup&gt;2&lt;/sup&gt;.K)</td>
</tr>
<tr>
<td>k&lt;sub&gt;s&lt;/sub&gt;, k&lt;sub&gt;r&lt;/sub&gt;</td>
<td>Pipe absolute roughness, pipe relative roughness</td>
</tr>
<tr>
<td>K&lt;sub&gt;loss&lt;/sub&gt;</td>
<td>Loss coefficient related to pipe fitting type</td>
</tr>
<tr>
<td>k&lt;sub&gt;th&lt;/sub&gt;</td>
<td>Thermal conductivity (W/m.K)</td>
</tr>
</tbody>
</table>
Experimental Study of Transformer Liquid Flow and Temperature Distributions

LV  Low voltage winding
LDV  Laser Doppler velocimetry
lpm  Litres per minute
ṁ  Inlet mass flow rate (kg/s)
Nu  Nusselt number
ONAN  Oil natural air natural cooling
ODAF  Oil directed air forced cooling

\[ p_i \]  Fluid static pressure at location \( i \)
\[ P_c \]  Channel wet perimeter (m)
\[ P_e \]  Channel wet perimeter (m)
\[ Pr \]  Prandtle number
\[ P_L \]  Total pressure loss in the transformer cooling loop (Pa)
\[ P_{loss} \]  Resistive losses stated as either W/plate or as W/m²
\[ p_i \]  Fluid static pressure at location \( i \)
\[ Q \]  Factor represents eddy loss distribution in transformer winding
\[ (H = S \times Q) \]
\[ q_{\text{cond-x}} \]  Heat transfer through conduction (W/m²)
\[ q_{\text{conv}} \]  Convective heat transfer rate (W/m²)
\[ Re \]  Reynold number
\[ S \]  Cooling efficiency factor of transformer winding \((H = S \times Q)\)
\[ T_{top}, T_{out} \]  Top, outlet, oil temperature (°C)
\[ T_{bottom}, T_{in} \]  Bottom, inlet, oil temperature (°C)
\[ V_{av} \]  Average velocity in a channel (m/s)
\[ V_{in} \]  Winding inlet velocity (m/s)
\[ \nu \]  Kinematic viscosity (m²/s)
WTI  Winding temperature indicator
\[ \gamma_a \]  Paper ageing factor
\[ \Delta \theta_o \]  Top oil temperature rise over ambient temperature (K)
\[ \Delta \theta_w \]  Average winding temperature rise over ambient temperature (K)
\[ \Delta \theta_h \]  Hot spot temperature rise over ambient temperature (K)
\[ \Delta \theta_b \]  Bottom oil temperature rise over ambient temperature (K)
\[ \Delta \theta_{om} \]  Mean oil temperature rise over ambient temperature (K)
\[ \rho \]  Density (kg/m³)
\[ \mu \]  Dynamic viscosity (N.s/m²)
Experimental Study of Transformer Liquid Flow and Temperature Distributions

Abstract

Determination of the temperature distribution and the so-called hot spot temperature within the transformer winding is crucial for both thermal design in the factory and thermal rating during the operation. In oil-immersed power transformers, oil acts as the coolant in addition to its role of insulation. The oil circulates within the winding either naturally in Oil Natural (ON) cooling modes or forced to circulate in a zig-zag fashion in Oil Directed (OD) cooling modes. Temperature distribution is heavily dependent on the oil flow distribution. This thesis aims to experimentally investigate the effects of a wide range of operational and geometrical parameters on the transformer oil flow and temperature distributions.

An experimental setup based on disc type transformer winding models was established where winding model geometries such as radial cooling duct height, axial cooling duct width, and number of discs per pass can be adjusted and operational conditions such as winding inlet oil velocity, winding inlet oil temperature, and electrical loss distribution within the winding model can be varied. Thermocouple arrays were used to measure temperature distribution. A Particle Image Velocimetry (PIV) system was used to capture the oil flow distribution including the detailed phenomenon of reverse flow.

Under OD cooling modes, experimental validations were conducted under isothermal conditions for the application of dimensional analysis on oil flow proportion in and pressure drop coefficient over the winding model. It was verified that if the dimensionless controlling parameters, pass inlet Reynold number and the ratio of radial duct height over axial duct width, are matched, both oil flow proportion and pressure drop coefficient are matched. Under non-isothermal OD cooling modes, it was found that the existence of electrical losses does not affect the oil flow distribution. In addition, extra high pass inlet oil velocity causes oil reverse flow and oil stagnation to occur and hence the hot spot temperature is not necessarily reduced with further increase of inlet oil velocity.

Under ON cooling modes, more distorted flow distribution which leads to oil stagnation and hence higher hot spot temperature were observed under higher loading levels or lower winding inlet oil velocities. Inlet oil temperature and non-uniform losses in the winding model showed minor impact on the oil flow distribution. In terms of geometric parameters, higher radial cooling duct or higher number of discs per pass makes oil stagnation easier to occur which could significantly increase the hot spot temperature.

Thermal performances of conventional mineral oil and alternative liquids including gas-to-liquid oil and synthetic ester oil were compared using the zig-zag disc type winding model under both OD and ON cooling modes. Mineral oil and gas-to-liquid oil showed similar behaviours under both cooling modes. Under OD cooling conditions, the synthetic ester is more resistant to oil reverse flow due to its high viscosity, which leads to more uniform oil flow distributions and hence lower hot spot temperature but with the cost of higher pressure loss. Under ON cooling modes, retro-filling scenarios were investigated and it was shown that synthetic ester causes lower inlet flow rates due to its higher viscosity and hence increases the hot spot temperature compared to the other oils.
Declaration

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The author

Muhammad Daghrarah finished his BEng at Birzeit University in Palestine in 2010. He worked for two years at the same university as a teaching assistant till 2012. He joined the University of Manchester for MSc in Electrical Power System Engineering from 2012 to 2013 at which he graduated with distinction. Starting January 2014, he enrolled as a PhD candidate in electrical and electronic engineering at the University of Manchester.

His research interests involve thermal modelling of power transformers, asset management, condition monitoring of assets, smart grid technologies, renewables, and system transient studies.
Dedication

{Say, “Indeed, my prayer, my rites of sacrifice, my living and my dying are for Allah, Lord of the worlds. No partner has He. And this I have been commanded and I am the first of the Muslims”} Al-An’am, 162

I dedicate this work to my mother Khitam Ibrahim Harb and to my father Mustafa Hamoud Daghrah who gave me unbounded love.

I also dedicate this work to my twin sister Asma, she always keeps me in her thoughts and gives me endless love.

I dedicate this work to a hopeful peace for the Palestinians at which they one day would enjoy their basic human rights with dignity and equality without racism and discriminations.

I thank The University of Manchester and The British Council for awarding me with the HESPAL scholarship which gave me the chance to study a master degree at the University of Manchester.

I thank my supervisors Prof. Zhongdong Wang and Dr. Qiang Liu for their technical guidance and contributions to this work. Their utmost dedication and excellence enhanced the quality of this work. I thank my friends and colleagues within the transformer group for the life enhancing experience I had. In particular, I would like to thank my colleague Xiang Zhang for the fun, interesting and mind-stimulating discussions during our cooperation.

I would like to express my gratitude to M&I Materials, National Grid, Scottish Power, Shell, TJ|H2b Analytical Services, UK Power Networks and Weidmann Electrical Technology for their financial and technical contributions to the Transformer Research Consortium at The University of Manchester through which this PhD was conducted.

Finally, I dedicate this work to my future wife and children. I already want them to know that I love them…
Chapter 1 Introduction

1.1 Motivation

Electricity is the wheel driving the development of modern societies. The process of delivering electrical power to customers goes through generation, transmission, and then distribution. To reduce the losses occurring during long distance electricity transmission, transformers are used to step up the voltage and reduce the transmitted current. As such, the resistive losses are reduced. A single-phase transformer has a core and two windings. One winding is called the primary winding and it is constructed by wrapping an insulated conductor of a specified number of turns around the magnetic core of the transformer. The secondary winding is constructed following the same procedure. Ampere’s law states that a current-carrying wire produces a magnetic field around it. The basic function of transformer core is to link the mutual magnetic flux produced from the primary winding to the secondary winding [1].

Electrical losses within the transformer include no-load losses and load losses. No load losses are from core losses or iron losses, which are due to eddy currents in the core and magnetic hysteresis phenomenon. The load losses are mainly due to copper loss which is a type of resistive loss and is equal to $I^2R$. Load losses also include stray losses caused by leakage flux in the windings and tank.

Electrical losses are converted to heat which create thermal stresses inside the transformer. The temperature profile within the transformer is not uniform because the non-uniform oil flow distribution and the uneven distribution of electrical losses within transformer cooling structure. This leads to the definition of Hot Spot Temperature (HST) as the hottest temperature within the transformer. The location inside the transformer near the HST is constantly under higher thermal stress than other parts of the transformer. Hence, the insulation near the HST is weakest which usually determines the thermal lifetime of the transformer. Accurate transformer thermal modelling is required to help asset managers predicting with better accuracy the expected life time of the transformer and so create reliable transformer replacement priorities. Understanding the location of the HST enables its more accurate measurement using the emerging fibre optics temperature sensors technologies.

The majority of thermal models of the hot spot temperature were conducted using simulation based approach in which a section of the transformer is modelled and
simulated to find the location and value of the HST under different operating and winding geometrical conditions. It was noticed in published work the limited amount of experimental based studies on hot spot temperature modelling and the lack experimental verifications of simulation based models.

1.2 Research Objectives

The aim of this PhD research study is to perform experimental work to document and better understand the influence of contributing factors on oil flow distributions and on the HST in oil directed and oil natural cooling modes in disc type transformer windings. The associated objectives are:

1. To build an experimental setup which allows adjustment of disc type winding model geometries and the measurement of temperature and oil flow distributions in the disc-type winding model.
2. To experimentally identify the influencing parameters and their effects on oil flow distributions and on the HST under oil directed cooled transformers.
3. To experimentally identify the influencing parameters and their effects on oil flow distributions and on the HST under oil natural cooled transformers.
4. To compare the thermal behaviour of alternative transformer insulating liquids under oil directed and oil natural cooling modes.

1.3 Thesis Layout

Chapter 1 Introduction
This chapter introduces the PhD research motivations, aim and objectives, and also provides the layout of the thesis.

Chapter 2 Background Knowledge
In this chapter, basic knowledge required to execute this project is presented. First section covers the construction and geometries of disc type power transformers. The second section covers thermal tests defined by a related standard to evaluate the thermal behaviour of power transformers. In the third section, ways of measuring the temperature inside the transformer are presented. Finally, basic concepts of transformer cooling from both fluid mechanics and heat transfer theorem are presented.

Chapter 3 Literature Review
In this chapter, literatures related to this study are reviewed and reported. Relevant literatures can be categorized as either simulation based or experimental based. Since
this work is entirely experimental based, more emphasis is placed on experimental based literature. Techniques of measuring oil flow rates in transformer cooling ducts used in previous literatures are highlighted.

**Chapter 4 Experimental Descriptions**
In this chapter, design and selection of individual components of the experiment setup are presented. These are split into three main components. First is the hydraulic loop. Second is the winding model. Third presents the measurement tools for temperature, flow rates, and pressure drop. Repeatability tests are presented.

**Chapter 5 Oil Flow Measurement using PIV System**
In this chapter, the application of a PIV system is presented to record oil flow rates in radial cooling ducts under both isothermal and non-isothermal conditions. Using the PIV system, oil stagnation and oil reverse flow were documented. Also, observations of oil recirculation at duct entrances were captured.

**Chapter 6 Oil Flow and Temperature Distributions under OD Cooling Mode**
In this chapter, experimental studies are carried out under isothermal and non-isothermal testing conditions to document the influence of winding geometrical parameters and transformer operational conditions on the oil flow distribution, on the HST, and on the pressure drop over the disc type winding model.

**Chapter 7 Oil Flow and Temperature Distributions under ON Cooling Mode**
In this chapter, experimental studies are conducted to document the influence of individual geometrical and operational parameters on oil flow distribution and on HST under oil natural cooling conditions.

**Chapter 8 Comparisons of Thermal Performance of Alternative Transformer Oils**
In this chapter, the thermal performance of alternative transformer liquids are experimentally studied under both oil directed and oil natural cooling conditions. Three types of transformer oils are used which are mineral oil, gas-to-liquid oil, and synthetic ester oil. Retro-filling scenarios are examined.

**Chapter 9 Conclusions and future work**
In this chapter, conclusions are drawn and emphasized. Possible future work is raised.


Chapter 2 Background Knowledge

2.1 Power Transformer Structure

Power transformers are built with complex structures. Transformer design is a sophisticated process, which involves electrical, mechanical and thermal design considerations. Iteration process is required to meet electrical, thermal and mechanical requirements in order to have reliable operation within the expected transformer lifetime. The transformer life expectancy is around 40 to 50 years [2]. Thermal stress occurs due to the electrical losses generated within the transformer and the imperfect cooling system of the transformer. Thermal stress accelerates the ageing of the transformer by deteriorating the insulation system. Figure 2.1 shows an overall picture of typical components of power transformers [2]. These components can be classified into internal components and external components. All components outside transformer main tank are the external components. These include cooling radiators, protection and relaying devices, transformer bushings, the conservator, external pumps and fans. Internal components are all components inside the transformer main tank. These include transformer steel core, transformer windings with their supporting structure and finally the transformer insulations. Cooling of the transformer depends both on the external and internal components working together. The heat generated in the winding is transferred to the insulation liquid through both convection and conduction. The insulation liquid circulates, either naturally or driven by external pumps, from inside the tank through the winding and the core to the external radiators. The radiators have large surface area to maximize heat dissipation to surrounding air or water.

Figure 2.1 Depiction of various components in a power transformer [2]
The first internal component of the transformer system is the steel core. A laminated steel core links the primary and the secondary windings through the magnetic field. As shown in Figure 2.2, axial cooling ducts, sometimes referred to as vertical cooling ducts, are provided within the core to cool it down. Usually, core losses account for 10% of total transformer losses [2]. The circular shape of the core is provided by constructing it from steel sheet strips with different widths.

Temperature rise limits are not usually specified for the core because it has lower temperatures than other parts such as the Low Voltage (LV) winding or the High Voltage (HV) winding. Nonetheless, it is stated in the standard IEC 60076-2:2011 [3] that the temperature rise of the core should not exceed a point of which it can degrade other parts of the transformer. For three phase power transformers, three or five limbs core might be used. The choice of the core structure is not only dependent on the electrical requirements but also on the transport logistics from the factory to the site.

The second transformer component is the winding. One winding structure of interest which is being used widely in power transformers is the disc type winding. The winding consists of wrapped copper turns over the core. A complete set of turns around the core is called a disc. Discs are separated by spacers to allow for radial cooling ducts sometimes referred to as horizontal cooling ducts. Vertical cooling ducts are designed and placed within the core, the LV winding and the HV winding in such a way to create an efficient distribution of oil flow for better cooling of each component [4].

![Figure 2.2 Cooling ducts in disc type winding][2]
Figure 2.3 Disc type winding structure [2]

Figure 2.3 shows a depiction of disc type winding [2]. The winding consists of many winding sections, shown between the two red lines representing two set of stacked spacers. Winding sections are hydraulically independent. A pass is defined as a set of discs between two directing washers. The oil is pushed from the bottom of the winding to the top in a zig-zag fashion through both the vertical cooling ducts and the horizontal cooling ducts. The number of discs per pass can vary depending on the design. Also, a variety of options for the washer locations, the horizontal duct height, and the vertical duct width are available to optimize the cooling of power transformers. Cooling of the transformer can be either under Oil Natural (ON) or Oil Directed (OD). ON mode occurs when oil circulate naturally under the influence of thermosiphon force. OD mode occurs when oil is being forced to circulate using external pumps.

The third main internal component of the transformer is its insulation. Here, only oil immersed power transformers are considered. The traditional coolant is mineral oil and the traditional solid insulation material is the Kraft paper. The transformer insulation should guarantee a reliable operation of the transformer during off peak, peak, and emergency loading situations. Among other factors, the temperature of the insulation greatly affects its expected lifetime. Therefore, the definition of the HST came into existence to refer to the hottest temperature of the transformer winding. Despite the fact that the exact location of the HST is not precisely determined, the general location is usually at the top part of the middle-phase winding of a three phase transformer between discs 2 and 3 counting from the top [5]. Finite Element Methods (FEM) can be used to analyse the eddy loss distribution. Computational Fluid Dynamics (CFD) can also be
used to determine both the flow distribution and the temperature profile within the winding section.

The critical HST at which normal ageing rate occurs is estimated to be 98 °C [6]. The ageing rate of the paper insulation is said to double each 6 degrees rise over 98 °C as shown by equation 2.1 [6].

\[ y_a = 2^{([\theta_h - 98]/6)} \]  \hspace{1cm} (2.1)

where \( y_a \) represents the ageing rate factor and \( \theta_h \) is the HST

### 2.2 Transformer Temperature Rise Test

#### 2.2.1 Procedure of the temperature rise test

Transformer manufacturers perform a thermal type test on a transformer representing a specific design to verify that it behaves according to the standards. The IEC 60076-2: 2011 [3] standard specifies temperature rise limits between active parts of the transformer and the external cooling medium. Table 2.1 defines the adopted temperature rise terminologies. \( \Delta \theta_o \) should not exceed 60 K during rated loading conditions [3]. \( \Delta \theta_w \) should not exceed 65 K for ON and OF cooling conditions while it should not exceed 70 K for OD cooling conditions under rated loading cycles [3]. \( \Delta \theta_h \) should not exceed 78 K [3].

The temperature rise test is the test performed by the transformer manufacturer to determine temperature rises which characterise the thermal diagram. In conducting the temperature rise test, the recommendations in the standards are adhered to.

**Table 2.1 Temperature rises measured in temperature rise test**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>( \Delta \theta_b )</td>
<td>Bottom oil temperature rise over ambient temperature</td>
</tr>
<tr>
<td>( \Delta \theta_{om} )</td>
<td>Mean oil temperature rise over ambient temperature</td>
</tr>
<tr>
<td>( \Delta \theta_o )</td>
<td>Top oil temperature rise over ambient temperature</td>
</tr>
<tr>
<td>( \Delta \theta_w )</td>
<td>Average winding temperature rise over ambient temperature</td>
</tr>
<tr>
<td>( \Delta \theta_h )</td>
<td>HST rise over ambient temperature</td>
</tr>
<tr>
<td>( g )</td>
<td>Temperature gradient between ( \Delta \theta_{om} ) and ( \Delta \theta_w )</td>
</tr>
<tr>
<td>( H )</td>
<td>Hot spot factor</td>
</tr>
</tbody>
</table>
The IEC 60076-2:2011 [3] specifies the following procedures in measuring the temperatures required to construct the thermal diagram of Figure 2.4 as follows. The thermal diagram represents the relationships between different temperature rises [3]. Two parallel lines are shown. The first line, left hand line, is the oil temperature rise from bottom to top of the transformer winding. The second line, right hand line, is the winding temperature rise from bottom to top of the winding. The thermal diagram assumes a linear rise of temperatures from bottom of the transformer to top of the transformer. The objective of the temperature rise test is to make sure that all temperature rise limits specified by the standards are met when loading the transformer at rated conditions.

Figure 2.4 Thermal diagram of power transformers from IEC 60076-2:2011 [3]

To execute the temperature rise test, the transformer needs to be energized at rated conditions. For low and medium power transformers, the transformer can be brought to rated conditions by connecting a suitable load to the transformer and exciting it from a source with the rated voltage and current. In case suitable voltage source and load are not available; a back to back method can be used to excite the tested transformer with the rated conditions. In this method, the transformer under test and another transformer are connected in parallel. With proper turns ratio and applied voltage, rated current can be obtained in the transformer under test. The previous two methods are sometimes not feasibly possible because of the lack of required energizing facilities. In that case and for large power transformers, a short circuit test method can be used instead. In this case, the transformer is injected with the total rated losses instead of the rated operating power which include both the no load losses and the load losses at rated conditions. The
objective is to evaluate $\Delta \theta_o$, $\Delta \theta_{om}$, $\Delta \theta_w$, and $\Delta \theta_h$. The testing procedure is as follows [3]:

1. The total losses are injected to the transformer by subjecting it to a current which is slightly higher than the rated current to accommodate for both load losses and magnetization losses at rated voltage. Once $\Delta \theta_o$ reaches a steady state with variation equals or less than 1 K/hour; then the second step should be commenced immediately.

2. The injected current is reduced from the value used in step 1 to rated value for 1 hour to measure $\Delta \theta_w$. After the 1 hour, the winding resistance should be measured as in step 3.

3. The winding resistance should be measured to infer $\Delta \theta_w$ either by rapid disconnection of the winding and then measuring its resistance or by using the superposition principle in which a small DC voltage and a small DC current are injected to the winding and then used to measure the DC resistance of the winding at rated conditions.

4. The HST can then be estimated by measured temperature rises from the temperature rise test. $\Delta \theta_h$ is given by equation 2.2

$$\Delta \theta_h = \Delta \theta_o + g \times H \quad (2.2)$$

where $H$ is the hot spot factor.

2.2.2 Measurement recommendations of key temperatures

1) Measuring ambient temperature [3]

   The ambient temperature should be measured with at least four sensors and the average of their reading should be taken as good representative of the ambient temperature. The sensors should be placed at least 2 m away from the tank and cooling system main body. For forced air cooled transformers, the sensors should be placed 0.5 m above the ground level.

2) Measuring top oil temperature [3]

   The top oil temperature is measured by immersing the temperature sensors at the top of the transformer or in oil pockets. The number of sensors installed is determined in agreement between customer and manufacturer. The average of the sensors is then used as a representative of the top oil temperature. The sensors should be placed near the top of the winding as much as possible.
3) Measuring average oil temperature [3]

The bottom oil temperature is identified as the oil temperature entering the winding. Because this is not easily measured; the temperature of the returned oil from the cooler is usually assumed as the bottom oil temperature. The average oil temperature can be assumed the average of the measured top oil temperature and the bottom oil temperature.

2.2.3 The hot spot factor (H)

In the IEC 60076-2: 2011 [3] standard, the HST rise is calculated using equation 2.2. The $H$ factor is defined through the $Q$ factor and the $S$ factor as $H = Q \times S$. The $Q$ factor is drawn from the knowledge of electric loss distribution within the transformer. Figure 2.5 shows the range of the $Q$ factor depending on the transformer rating and on the copper winding strand width [3]. The $S$ factor depends on the efficiency and the cooling type of the winding structure. For disc type windings without diverting washers, the $S$ factor can be considered 1.0. However, for disc type winding with diverting washers, the $S$ factor could vary from 1.0 to 1.2 [3].

![Figure 2.5 Range of the Q factor for different conductor strand width in mm [3]](image)

2.3 Direct Temperature Measurement in Power Transformers

The main objective of the temperature rise test is to estimate the expected HST at rated conditions to accurately assess the degradation level of the transformer insulation system. Three main methods are used to measure or estimate temperatures inside the transformer tank. The first one is the Winding Temperature Indicator (WTI) to estimate the HST of the winding at different loading levels. The second one is the Fibre Optics Temperature Sensors (FOTS) which are directly used to measure the HST and are installed inside the transformer winding. The third one is thermocouples which are used
to measure the temperature of the core, the supporting structure of the winding and the bottom oil temperature. In the following, the basic operating principle of each method is given.

2.3.1 Winding temperature indicator [2]

WTI is the traditional method to estimate the HST of the transformer winding under operation. A Current Transformer (CT) is connected to the transformer LV or HV winding. The secondary of the CT is connected to a heating coil. The heating coil is placed in an oil pocket at the location of hottest top oil temperature inside the transformer tank. The oil pocket will then have the same temperature as the surrounding top oil. The heating coil heats the oil inside the oil pocket to a value higher than the top oil temperature. The setting of the CT and the heating coil are specified in such a way that the temperature of the oil pocket matches the expected calculated HST. Uncertainties are present from the HST estimation methods and so the calibration process of the WTI might not be accurate. The temperature of the oil pocket can then be measured using any reliable temperature sensors. Temperature of the oil pocket represents the HST over the top oil temperature. Figure 2.6 depicts a transformer with a WTI placed to estimate the HST of phase A.

2.3.2 Fibre optics temperature sensors

FOTS are part of a general category of fibre optic sensing technologies. Fibre optic probes are means to transmit light. Their physical properties can be altered or changed to create sensing abilities for different measured parameters such as force, temperature, strain, current and voltage [7, 8]. FOTS are of interest for power engineering measurements because they are immune to electromagnetic interference. They found many applications in distributed temperature measurements in underground cables [8-10], temperature measurements in power transformers and in large synchronous generators [11].

FOTS can be classified either to distributed-based FOTS or to point-based FOTS. Distributed-based FOTS provides multiple sensing elements along a single fibre channel. The sensing elements are distant from each other according to the total length of the fibre channel. This could be from multiple millimetres in relatively short fibre optics up to hundreds of meters in long fibre channels. Point-based FOTS has only one sensing element with each fibre optic channel.
Two main point-based FOTS technologies are used in transformer applications. The first technology uses a sensor which is made of rare earth materials such as phosphor. The rare earth material illuminates if excited by a light pulse. The illumination decay time depends on the temperature of the sensor tip. The temperature can be inferred by measuring the decay time of the illumination using a predetermined look up table. The look up table is stored and obtained through a calibration process [10].

The second technology uses a Gallium Arsenide crystal (GaAs), which has the property that its band gap energy changes with temperature. A broadband light pulse is sent to the crystal through the fibre channel. Based on the crystal temperature, part of the light pulse is absorbed and the other part is reflected from the GaAs crystal location. A light detector is located at the receiver of the FOTS instrumentation to spot the missing wavelengths from the transmitted broadband light pulse. The missing wavelengths can then be linked to the temperature of the GaAs sensor [10].

Figure 2.6 depicts a demonstration for typical installations of FOTS, WTI, and top oil temperature sensors. FOTS probes are usually placed at the top of each winding where the HST is believed to be. The FOTS probes are routed carefully outside the transformer tank using a special tank feed-through to prevent oil leak. Once the location of the HST is predicted, the installed FOTS can give reliable results of the real HST of the transformer under different loading levels. Several factors should be considered while installing FOTS. First, the FOTS probe is inherently fragile and it should be handled with care. Second, the location of the FOTS probe should be as close as possible to the location of the expected HST. Multi-channels of FOTS probes are usually installed to get the best guess of the HST [5].

---

**Figure 2.6 Depiction of FOTS and WTI installed in a transformer**
2.3.3 Thermocouples [12]

Thermocouples are self-generating sensors which are used for temperature measurements. They do not need an external excitation source. They have a wide temperature measurement range which reaches to more than 1000 °C.

Historically, Thomson J. Seebeck in 1822 was the first to observe a developed current circulating in a loop which consisted of two metals connected at two different junctions [12]. The junctions were at different temperatures. Figure 2.7 shows a depiction of two metals A and B connected at two junctions. The first junction is the measuring junction while the second junction is the reference junction. The measuring junction is placed where the desired temperature is to be measured. The reference junction is kept at a known temperature. Because of temperature differences between the measuring junction and the reference junction, a current circulates in the closed loop. Conversely, if the loop is open; then a voltage difference appears between the two junctions. The voltage difference $E_{AB}$ between the two junctions is given by equation 2.3 [12]. To infer the temperature of the measuring junction $T_1$, the temperature of the reference junction $T_2$ must be known. Traditionally, the reference junction was kept in an ice bath in which ice cubes and water are mixed together to keep the temperature at 0 °C. However, with the immergence of electronic temperature sensors, the temperature of the reference junction is recorded continuously and then fed to the conditioning circuit. This is known as Cold Junction Compensation CJC. From equation 2.3, $C_2$ should be negligible and very small to obtain a linear relationship between $E_{AB}$ and $(T_1 - T_2)$. This requirement limits the useful metals which can be used in thermocouple construction.

$$E_{AB} \approx (T_1 - T_2)[C_1 + C_2(T_1 + T_2)]$$

(2.3)

where the temperature of the measuring junction is $T_1$, the temperature of the reference junction is $T_2$, and both $C_1$ and $C_2$ are material dependent constants.

![Figure 2.7 Basics of thermocouple construction and operation](image-url)
2.4 Basics of Transformer Cooling Principles

2.4.1 Introduction to fluid mechanics

In this section the basic theory and definitions, related to this project of fluid mechanics are presented. Fluid mechanics is concerned about the study of the fluid behaviour exposed to external forces. This includes studying the flow behaviour of the fluid, the velocity of the fluid, the pressure distribution within the fluid, fluid transport from one location to the other, and the required external machinery to do the required work for fluid transportation. Figure 2.8 shows a simple hydraulic system consisting of two tanks, a fluid, a pump, and the connecting pipe network [13]. The conservation of energy determines the fluid status at positions 1 and 2. Equation 2.4 is called the energy equation which relates the pressure $p$, the velocity $V$ and the elevation of the fluid $z$ in position 1 to that of the fluid in position 2 [13].

In equation 2.4, $\gamma$ is the specific weight of the fluid. The specific weight is defined as the gravitational force of the fluid per its unit volume. $\gamma$ can be obtained by multiplying the fluid density by the gravitational acceleration. $\alpha_1$ and $\alpha_2$ are constants which depend on the flow type at positions 1 and 2 respectively. The flow type can either be laminar flow or turbulent flow. $\alpha_1$ and $\alpha_2$ range from 1.05 for turbulent flows and 2 for laminar flows [13]. The pump represents the external energy source to overcome the friction losses and the gravitational forces. All terms in equation 2.4 are normalized to have the unit meters. The pump head $h_p$ represents the external power the pump needs to provide to make the change in the fluid energy status from position 1 to position 2. $h_l$ represents the system head loss due to friction within the hydraulic network.

$$\frac{p_1}{\gamma} + \alpha_1 \frac{(V_1)^2}{2 \times 9.8} + z_1 + h_p = \frac{p_2}{\gamma} + \alpha_2 \frac{(V_2)^2}{2 \times 9.8} + z_2 + h_l$$

(2.4)

Figure 2.8 Simple hydraulic system [13]
To understand the system hydraulic losses, the viscosity of the fluid should be defined first. The viscosity represents a characteristic of the fluid to how easily it can flow. For example, honey is a high viscous fluid which tends to move very slowly compared to water which is a low viscous fluid. Two types of viscosities are defined for fluids and both are related to each other. The first one is the dynamic viscosity $\mu$ defined as the ratio between the shear stress to the velocity gradient [13]. The dynamic viscosity has the units $N \cdot s/m^2$. The second one is the kinematic viscosity $\nu$ which is related to the dynamic viscosity $\mu$ by $\nu = \mu/\rho$ where $\rho$ is the fluid density. The head loss and flow type are both affected by the fluid viscosity.

A dimensionless number is given to characterize the flow type to laminar or turbulent. The number is called Reynolds number $Re$. It is named after its discoverer Osborne Reynolds in 1883 [13]. $Re$ is given in equation 2.5 [13]. The flow is characterised as laminar flow if the calculated $Re$ is equal or less than 2000 [13]. Laminar flows have a parabolic velocity profiles as demonstrated in Figure 2.9 (a). Turbulent flows are characterized by mixing within the fluid layers and by the developed eddy currents exemplified in Figure 2.9 (b). Turbulent flow occurs when the calculated $Re$ is higher than 3000. If the calculated $Re$ is between 2000 and 3000; then the flow could be either laminar or turbulent and it cannot be easily determined.

$$Re = \frac{V_{av} \times D}{\nu}$$

where $V_{av}$ is the average flow velocity in the channel and $D$ is defined as the hydraulic diameter of the flow channel. The hydraulic diameter is given by $D = 4A_c/P_c$ where $A_c$ is the cross sectional area of the channel and $P_c$ is the wetted perimeter of the flow channel [13].

Figure 2.9: Velocity profiles of (a) laminar flow and (b) turbulent flow [13]

The head loss of the system can be obtained by summing the head loss in each component of the hydraulic loop. The hydraulic loop of the system used in this PhD study consists of pipe networks, pipe joints, radiator, a flow meter and the winding model. The head loss in each component is proportional to the square of the velocity.
Equation 2.6 is a general equation to calculate the pipe head losses. It is named as the Darcy-Weisbach equation [13].

$$h_{t-pipe} = f \times \frac{L}{D} \times \rho \times \frac{V_{av}^2}{2} \quad (2.6)$$

where $f$ is the pipe friction coefficient which depends on the flow type along with the roughness of the pipe surface. $L$ is the pipe length.

For laminar flows, the friction coefficient can simply be calculated using the $Re$ as shown by equation 2.7 [13]. If the flow is turbulent; the pipe is identified either as smooth pipe or as rough pipe. For smooth pipes, equation 2.8 is used to calculate $f$ [13].

For rough pipes, an empirical approach is usually taken to identify the friction coefficient. First, the pipe absolute roughness $k_s$ is identified from the pipe type and material. Then, the relative roughness $k_r$ is calculated using $k_r = k_s / D$; then, the $Re$ is calculated. After evaluating both $k_r$ and $Re$, empirical curves are used to find the corresponding $f$ [13].

$$f = \frac{C}{Re} \quad (2.7)$$

where $C$ is a constant which depends on the pipe type

$$\frac{1}{\sqrt{f}} = 2 \times \log(Re \times \sqrt{f}) - 0.8 \quad (2.8)$$

The head loss in the pipe network can also be induced at the inlets, outlets, and fittings. It can be represented as shown in equation 2.9 [13].

$$h_l = K_{loss} \times \rho \times \frac{V_{av}^2}{2} \quad (2.9)$$

where $K_{loss}$ is the loss coefficient related to the fitting type.

![Figure 2.10: Differences between centrifugal and positive displacement pumps](image-url)
Once the total head loss in the system is calculated; then the proper pump can be selected. There are two main types of pumps; the first is a centrifugal pump. The second is a positive displacement pump. Centrifugal pumps are more suitable for applications where variable flow rates are required. Positive displacement pumps are usually used in applications where the flow rate is required to be constant. Figure 2.10 shows the differences in pump curves for both pump types [13]. The system curve consists of the static head loss and the dynamic head loss. The static head is merely the elevation height from point 1 to point 2 of Figure 2.10. It is called static because it does not depend on the flow rate. The dynamic head consists of the friction loss, inlet loss, outlet loss, and all losses from the fitting of the pipe work and external components such as valves, coolers and the flow meter. Once the system curve is established; the pump can then be selected. The intersection between the pump curve and the system curve gives the expected operating point.

2.4.2 Introduction to heat transfer [14]

The cooling of the transformer is governed by heat transfer laws. Heat is a form of energy. From the first law of thermodynamics, the conservation of energy applies as a universal rule. Energy is only converted from one form to the other. Energy can be transferred to or from a control volume system by the interaction of that system to its surrounding. The transformer is usually equipped with a heat exchanger, which is almost always referred to as a radiator to increase its cooling efficiency. Three modes of heat transfer occur between the transformer cooling system and its surrounding. These modes are conduction, convection and radiation.

The heat transfer equation that governs the conduction mode is known as Fourier’s law. Conduction happens whenever two bodies of different temperatures are in contact. The heat transfer rate per unit area has the units of \( W/m^2 \). For a one-dimensional problem in the \( x \) direction, the heat transfer rate \( q_{\text{cond}-x} \) is given by equation 2.10 [14]. The minus sign in equation 2.10 indicates that the heat transfer is in the direction of decreasing temperature. Conduction is usually high between metals because they have free electrons that help transfer energy along with the lattice vibration. Liquids and gases usually have lower thermal conduction rate than metals.

\[
q_{\text{cond}-x} = -k_{\text{th}} \times \frac{dT}{dx}
\]
where $k_{th}$ is the thermal conductivity of the contact surface of the two bodies given in $W/(m.K)$.

Convection is the term used to describe the heat transfer that occurs between a boundary surface and a moving fluid over that surface when the fluid and the surface are at different temperatures. Two main mechanisms cause the heat transfer from the boundary surface to the fluid. The first mechanism is heat transfer through conduction. The second mechanism is through the bulk motion of the fluid over the surface. The heat transfer increases as the velocity of the fluid increases. Convection heat transfer could be either natural convection or forced convection. Natural convection is caused due to buoyancy forces that cause the oil to circulate naturally within the transformer. Forced convection happens in case there is an external pump to force the fluid to circulate. Equation 2.11, known as the Newton’s law of cooling, is a general equation for the rate of heat transfer through convection $q_{\text{conv}}^*$ in $W/m^2$ [14]. Effectively, any attempt to solve equation 2.11 is more or less an attempt to find the respective $h$ which is greatly dependent on the investigated geometry.

$$q_{\text{conv}}^* = h \times (T_s - T_f) \quad (2.11)$$

where $T_s$ is the boundary surface temperature of cooled body and $T_f$ is the fluid temperature. $h$ is the convective heat transfer coefficient in $W/(m^2.K)$.

Radiation is caused by the emitted thermal energy from matters which are at finite temperature. Radiation contributes to the energy dissipated by the radiator and the tank of the transformer. Equation 2.12 is a general heat transfer equation through radiation [14].

$$q_{\text{rad}} = \varepsilon \times A \times \sigma \times (T_s^4 - T_{\text{surf}}^4) \quad (2.12)$$

where $\varepsilon$ is the emissivity with values from 0 to 1. $A$ is the radiating surface area in $m^2$. $\sigma$ is the Boltzmann constant and $T_{\text{surf}}$ is the temperature of the surrounding surface.

In heat transfer and fluid dynamics, dimensionless numbers are used to characterize the cooling and fluid behaviours. The $Re$ was mentioned before and it was used to deviate between laminar flows and turbulent flows.

The Prandtl number ($Pr$) describe how easily a fluid can thermally mix [15]. $Pr$ can be obtained by equation 2.15 [14].
Experimental Study of Transformer Liquid Flow and Temperature Distributions

\[ Pr = \frac{c_p \times \mu}{k_{th}} \]  \hspace{1cm} (2.15)

where \( c_p \) is the heat capacity (J/kgK).

The Grashof number \((Gr)\) is used to represent the ratio between buoyancy forces to viscous experienced by the fluid and can be obtained using equation 2.16 [14]

\[ Gr = \frac{9.8 \times \beta \times (T_s - T_f) \times D^3}{v^2} \]  \hspace{1cm} (2.16)

where \( \beta \) is the coefficient of thermal expansion of the fluid (1/K).

The Nusselt number \((Nu)\) is another important dimensionless number which is used to find \( h \). \( Nu \) indicates the level of heat convection for the cooled surface. Equation 2.13 relates \( Nu \) to \( h \) [14]. \( Nu \) is also related to other dimensionless numbers as shown in equation 2.14 [14]. Different expressions of the \( Nu \) based on equation 2.14 have been used in the literature to estimate \( h \) for the transformer application [16, 17].

\[ Nu = \frac{h \times D}{k_{th}} \]  \hspace{1cm} (2.13)

\[ Nu = f(Re, Pr, Gr) \]  \hspace{1cm} (2.14)

The ratio \( R_i = Gr/Re^2 \), referred to sometimes as the Richardson number, indicates whether the cooling mode is ON dominated (if \( R_i \gg 1 \)) or OD dominated (if \( R_i \ll 1 \)) [18].
Chapter 3 Literature Review

3.1 Introduction

Different transformer thermal modelling approaches have been pursued in published literatures to advance the understanding of oil flow distribution and the HST. One path is the electrical thermal analogy in which the transformer internal components and insulation system are modelled as lumped electrical circuit elements [19-21]. These models offer a global perspective of the HST and its response in time using exponential equations. The electrical thermal analogy based models are not relevant to this PhD and so are not reviewed here. Another modelling approach is the Network Hydraulic Thermal Modelling (NHTM) approach which was pioneered by A. J. Oliver in 1980 [16]. The NHTM approach relies on modelling the disc type winding as a hydraulic network which consists of nodes and paths. Three main conservation principles are applied in NHTM models which are the conservation of mass, conservation of momentum, and the conservation of energy. NHTM models are still favoured by the industry as they require less time in simulations. However, the pressure loss and heat transfer coefficients used in the NHTM models are usually taken from other applications and hence may induce uncertainties. With the emergence of Computational Fluid Dynamics (CFD) software and fast computational processing capabilities, coefficients required in the NHTM model are sometimes calibrated using CFD simulations [22, 23]. CFD simulations provide more accurate predictions of the HST compared to the NHTM models but with much higher cost of computational resources. Published literatures related to NHTM and CFD simulations provide a good understanding of the transformer thermal performance and hence they are reviewed in section 3.2. Another modelling approach is through experimentations on laboratory-scale winding models. Such experiments are often used to verify the predictions from both NHTM and CFD models. Published literatures on the experimental work are most relevant to this research and hence they are reviewed in detail in section 3.3.

3.2 Simulation Based Studies

Driven by the desire to pinpoint the location of the HST, a software programme was developed to estimate fluid flow rates, fluid temperatures, and boundary temperatures of any network of hydraulic paths [16]. The software was named as TEFLOW and became a popular tool for transformer thermal design and the foundation for other NHTM
models [24-28]. The mathematical equations were obtained by applying the conservation of mass at each node, the conservation of thermal energy at each node, pressure drop equation applied along each hydraulic path, and the heat transfer principles applied along each path. The application of TEFLOW was demonstrated in a case study of 250 MVA transformer on the LV winding which operated with 22 kV [16]. The winding had 19 discs per pass in 5-pass winding segment. Vertical duct width was 15 mm and the horizontal duct height was 5 mm. It was observed that the location of the HST is in the middle of a pass where duct velocities are the lowest [16]. In other NHTM investigations, the HST location was also found to be in the middle disc of simulated passes [28, 29].

Hydraulic resistances for various winding geometries were calculated [4]. Oil flow portion in parallel transformer windings is affected by the winding hydraulic resistance. It was deemed important to estimate oil split between windings to estimate the HST using NHTM models [24].

A joint global-internal HST prediction model was proposed [24]. The global model calculates the expected oil flow portion in each transformer component while the internal model calculates the HST location and value in each winding following the input flow rate provided by the global model. Model predictions were compared with field data obtained from 22 transformers under 46 loading conditions and with FOTS measurements. Good agreement was reported between the simulated and measured values of both top oil temperature and HST [24].

A flexible NHTM was proposed which provided the options to test the effects of the number of discs per pass, the number of turns per disc, the insulation paper thickness and the individual horizontal cooling ducts heights on developed thermal profile within a winding model [25]. The model allowed the ability to simulate using non-uniform loss distribution within the winding as newly incorporated in NHTM models [25].

Both flow distribution and pressure losses were calculated for ON cooling modes using NHTM modelling [26]. It was found by [26] and [30] that minor pressure losses should be considered in any NHTM modelling attempts as they influence flow distribution. A comprehensive simulation was conducted to understand effect of geometrical parameters on flow distribution and on total pressure losses within winding model [30]. It was found that lower discs per pass would improve flow uniformity but would increase pressure loss within the pass [30, 31]. Larger horizontal cooling duct height
was found to have lower pressure loss but the flow distribution is found to be more distorted [27, 30]. The influence of temperature dependence of oil properties and the non-uniform distribution of losses were considered in NHTM modelling [32]. Model predictions indicated that it is essential to consider the direction of the flow in the hottest pass of the winding to reduce the HST [29, 32].

It was observed that lower number of discs per pass reduces the pass HST [27]. It was observed that higher pumps do not enhance the cooling under oil forced as the oil bypass the active parts of the transformer [33]. Higher oil flow rates can be achieved in the transformer active parts under OD cooling and the limit would be the point of reaching the static electrifications at 50 cm/s [33]. Subsequently, the NHTM model was used to simulate the thermal performance of a 750 MVA, 230 kV/380 kV transformer under ONAN, ONAF, and ODAF cooling conditions [34]. Model predictions were compared with temperature rise test data and with measurements from embedded fibre optic temperature sensors in LV and HV windings [34]. Paper bulging and oil sealing in OD cooling modes were raised as important considerations which should be incorporated under simulation models [34].

As CFD based thermal modelling of power transformers emerged, [15, 35] observed and documented the hot streak phenomenon which is caused mainly because the high Prandtl number of transformer oil. It was also highlighted the importance of including buoyancy forces in any modelling attempts of power transformer as these forces affect the flow distribution and so the HST [15, 35].

The effect of axi-symetrical modelling in power transformer modelling was investigated under ON cooling mode through both experimental and simulation work [17]. It was concluded that the axi-symmetric modelling is sufficiently accurate for thermal modelling and simulation [17]. Adversely, numerical 2D and 3D CFD simulation were conducted to study modelling accuracy [18]. It was concluded that axi-symmetric simulations, 2D, does not represent the real situation accurately. A correction method was proposed to correct results obtained using 2D simulations to match results from 3D simulations [18]. First, corrections were applied to both Re and Gr numbers used in 2D simulations to match that which was used in 3D simulations. Second, corrections were applied to the surface area of discs simulated in 2D case to match that of 3D case. After applying these corrections, both the average winding and the HSTs were compared between 2D and 3D cases and they were found to be within 1 °C [18].
A comparison between NHTM and CFD modelling was conducted under both uniform and non-uniform loss distributions [36]. It was found that both NHTM and CFD models compare well under uniform loss distribution but larger discrepancies exist when non-uniform losses were considered [36].

It was recommended that a full and detailed Conjugate Heat Transfer (CHT) model to be used in order to obtained accurate CFD simulations [35]. A full CHT model consider detailed meshing of both the fluid and the solid volumetric domains and take into account the variation of oil density with temperature [35]. Using the full CHT model, it was reported that higher inlet flow rates causes more efficient cooling of the winding and so causes lower HST [35].

NHTM models were assessed and calibrated using CFD simulations to enhanced pressure drop coefficients [23, 37]. Using NHTM simulations, the hot spot factor was investigated and it was concluded that the S and the Q factors cannot be decoupled [38]. The S factor depends on cooling mode, oil viscosity, and it does not describe the non-uniform flow distribution within winding passes [38].

In the most recent publications, dimensional analysis was used to simplify the problem and to identify factors which influence oil flow proportion in radial cooling ducts and pressure drop coefficient over winding pass under OD isothermal simulations [39]. The most influential dimensionless parameters were found to be the pass inlet Re number and the ratio between radial cooling duct height to the axial cooling duct width (α). It was emphasized that if the Re and α are matched, then both the flow proportion in cooling ducts and the pressure drop coefficient over the winding pass are matched irrespective of how the dimensionless parameters are composed of.

### 3.3 Experimental Based Studies

#### 3.3.1 Isothermal investigations

One of the main experimental studies on a physical winding model to measure oil flow distribution within winding cooling ducts was conducted under isothermal conditions [40]. It was driven by the desire to reduce temperatures within a disc type winding by knowing and equalizing oil flow distribution in the winding structure [40].

A test setup was built and the oil speed was measured within the ducts for different geometrical parameters. Parameters such as the width of the inlet vertical duct, the width of the outlet vertical duct, and the number of discs per pass were varied in certain
measures. A Perspex winding model was built to simulate a section of disc type winding. The Perspex winding model consisted of only one pass holding up to 20 discs. Each disc was made using a perspex block with radial length of 100 mm and thickness of 16 mm and no heating was applied. Both vertical and horizontal duct widths were made to be varied from 2 mm to 10 mm in 2 mm steps. For each case, oil velocities within the ducts were measured using Hot Wire Anemometry (HWA). HWA probes were inserted through drilled holes to the horizontal ducts to measure the oil speed at 85 mm in the duct radial direction. It was indicated that the number of starved ducts increased as the number of discs per pass increased and the starvations were in pass middle cooling ducts as demonstrated in Figure 3.1 [40].

![Figure 3.1](image)

A study was conducted on SF6 gas cooled power transformer both through experiments and simulations [31]. Three measures were considered to increase the performance of the cooling system. First, the pressure of the SF6 gas was increased to 0.5 MPa to increase its heat capacity. Second, the flow rate of the gas was increased by developing a large volume high pressure head blower. Third, they worked using numerical simulation and experimental setup to optimize the geometry of the winding cooling structure [31]. Figure 3.2 shows the used experimental setup to study the effects of different geometries on the flow distribution within the winding. A total of 28 discs distributed into 4 passes were used and the study was conducted under isothermal conditions [31].

Water was used instead of SF6 gas and this was justified by the fact that the kinematic viscosity of SF6 at pressure of 0.5 MPa is almost the same as that of water [31]. To
make sure that the studied flow characteristic of water resembled that of SF6; the Re number for both the real situation and the experimental model were matched. Also, inlet velocity used in the experiment was 1/3 of that of an actual transformer while the dimensions of cooling ducts triple that of an actual transformer. Both the velocity distribution within the horizontal ducts and the pressure distribution within the vertical ducts were measured. The velocity distribution was measured using tracer particles seeded into the water. The movement of seeding particles was photographed with a CCD camera to estimate the velocity. The static pressure was measured through 1 mm drilled holes which allowed probes to be inserted to the vertical duct at various locations. Parameters which were varied are both the horizontal duct height and the number of plates per pass. Reverse flow was observed in the first horizontal cooling duct and it was reported that reverse flow shifts to upward ducts when testing under higher discs per pass arrangements. They observed that pressure loss over the winding model increased as the number of disc segments per pass decreased. More uniform oil flow distribution was observed when testing under smaller number of discs per pass on the cost of higher pressure losses [31].

Figure 3.2 Experimental setup used in [31]
Figure 3.3 Oil flow measurement compared to simulation models [31]. Measurement indicated the occurrence of oil reverse flow [31].

An experimental study to verify the CFD simulation on a one-pass model of a disc type winding structure was conducted [41]. The pressure difference was measured between ducts and over the pass. CFD simulations were verified by comparing the measured pressure difference with the simulated one. This verification acted as a footprint to have the confidence to proceed with the CFD simulation to infer both the oil flow distribution within the ducts and the temperature distribution within the pass. Figure 3.4 shows the experimental layout and the positions of the pressure sensors. The inlet flow rate ranged from 0 to 25 litre/minute. Then the pressure difference over the entire pass was measured and compared to the CFD simulations.

**3.3.2 Non-isothermal investigations**

Experiment study was conducted to estimate the developed oil velocity in ON cooled transformers [42]. This was achieved by equating the pressure loss developed within the
Experimental Study of Transformer Liquid Flow and Temperature Distributions

transformer hydraulic loop \((P_L)\) with the developed thermal driving force \((p_T)\). To calculate \(p_T\), equation 3.1 was used [42]. Equation 3.1 was considered valid for the following two assumptions; first, the heat dissipation through the tank and the piping network was considered negligible. Second, the temperature increase from the bottom to the top of the winding was assumed linear with the winding height.

\[
P_T = 9.8 \times \rho_0 \times \beta \times \Delta T \times \Delta H
\]  
(3.1)

where \(\rho_0\) is the oil density at a specific temperature, \(\Delta T\) is the oil temperature difference between the outlet and the inlet of the radiator, and \(\Delta H\) is the height difference between the centre line of the transformer winding and that of the radiator.

The experimental setup used by [42] consisted of three main components as shown in Figure 3.6; the winding, the pipe network, and the radiator. The winding consisted of 20 mimic coils each constructed using a heater covered with a stainless-steel plate. The winding front and rear walls were constructed from acrylic plates and the other two side walls were thermally insulated as well as the pipe network. Laser Doppler Velocimetry (LDV) was used to measure the oil flow at the winding inlet without disturbing the flow and for this purpose an acrylic window was constructed and mounted at the winding inlet. It was found that the developed oil flow rate increased proportionally to the square root of the total electrical losses within the winding and to \(\Delta H\) as shown in Figure 3.5.

Figure 3.5 Developed inlet flow rate increased linearly with the square root of heat generation and with \(\Delta H\) [42]
Figure 3.6 Experimental setup used to study ON transformers [42]

Experiment study was performed to verify numerical model prediction for disc type winding which consisted of two columns of 6 blocks each as shown in Figure 3.7 [43]. There was a vertical cooling duct in between the two winding columns with 5 cm width and the blocks were spaced apart with 1.4 cm distance in between. Thermocouples were used to measure the temperature distribution. LDV was used to record the velocity field within the structure at various locations and so the front wall of the rig was made of plexiglas to permit optical access. Water was used as coolant during testing. During the tests, it was reported that buoyancy forces dominated the inertia forces. In addition, It was concluded that a fine grid of the used CFD model is required to improve the simulation accuracy and to minimize the differences between the predicted and the measured quantities.
An experimental was executed to determine convective heat transfer coefficient for a transformer application [17]. The experimental setup is shown in Figure 3.9 which consists of a winding model immersed in oil in a test tank. The test tank was connected to a top tank. The top tank fed the test tank with oil flow rate which was controlled by a valve. The top tank elevation was designed such that it can give a maximum flow rate of 7.5 litres/minutes. Oil circulated from the top tank to the test tank; then it circulated through the test winding and back to the main oil tank. Oil level was maintained constant in the top tank by pumping oil from the oil tank to the top tank using a pump. The surface area of the pipe work and tanks acted as cooling area for the oil. The winding model was a 1:1 scaled to a real disc type winding section. Composite discs were constructed which were wrapped with transformer insulating paper and heating was performed using a 0.1 mm constantan heating foil. The winding model hosted 16 discs arranged in two passes, 8 discs per pass. It was reported that the conferential disc temperature gradient can be neglected and that a 2 D axisymmetric modelling is sufficiently accurate to present the winding disc temperatures. Comparison of measurements and simulations are shown in Figure 3.8.
Experimental Study of Transformer Liquid Flow and Temperature Distributions

Figure 3.8 Comparison of disc radial temperatures between simulations and measurements [17]

Figure 3.9 Experimental setup used to check the validity of proposed model by [17]

Experimental validation was conducted of the accuracy of 2D and 3D CFD simulation of disc type transformers under OD cooling conditions [44]. The experimental setup used, shown in Figure 3.11, consisted of three passes in which the first and the third passes acted as conditioning passes to create typical boundary conditions for the second pass to match that in a transformer. The winding model was immersed in oil tank and the oil tank temperature was fixed to 80 °C to reduce heat dissipation through the winding model walls. Discs in the winding model were constructed using four
conductors and each conductor was heated independently. Solid blocks of plastic were used to provide thermal insulation between conductors. In the winding model, both vertical and horizontal cooling ducts dimensions were 6 mm. The flow rate within horizontal cooling ducts was visualized and velocity measured by tracking the movement of particles using camera and high-power LED light source. By knowing both the spatial displacement of the tracking particles and the time between two captured images, the velocity of each particle can be calculated. The camera was focused on the centre of each horizontal duct. It was reported that the measured mean values of velocities agree well with CFD simulations as shown in Figure 3. 10 (b). It was reported that a strong axial non-uniform oil flow distribution exists in the investigated pass which directly influence the position of the HST. It was mentioned that because higher oil flow rates exist in pass upper cooling ducts, the location of the HST is in the pass bottom conductors.

Figure 3. 10 (a) Oil flow distribution under $\dot{m} = 18.0 \text{ kg/s}$ and $T_{\text{in}} = 80 ^\circ \text{C}$ (b) Simulations and measurements comparison with $\dot{m} = 9.0 \text{ kg/s}$ and $T_{\text{in}} = 80 ^\circ \text{C}$ [44].

Figure 3.11 Experimental setup used in [44]
3.3.3 Comparison of alternative transformer liquids

To test the thermal cooling performance of alternative transformer liquids, [45] retro-filled a single phase, 15 MVA, 154/22.9 kV transformer with natural ester and compared temperature rise tests data to the transformer temperature rise tests data when it was filled with mineral oil under ON cooling conditions. Measurements were conducted using 14 installed fibre optic temperature sensors in both LV and HV winding. Table 3.1 presents a key comparison of the temperature rise test data between the two tested liquids. From Table 3.1, it can be observed that for the natural ester, the bottom liquid temperature is lower compared to that for the mineral oil. At the same time, the top liquid temperature is higher for natural ester compared to that for the mineral oil. This leads to the conclusion that the overall inlet flow rate is lower for the natural ester compared to that for the mineral oil. Lower inlet flow rate caused much higher HST in both LV and HV windings. Equation 3.2 can be used to infer the expected relative increase in inlet flow rate when using mineral oil compared to when using natural ester given that the specific heat for both oils are known and the tests were conducted under the same electrical losses.

Table 3.1 Temperature rise test data comparing mineral oil to natural ester thermal performances [45]. All temperature rises are over ambient temperature.

<table>
<thead>
<tr>
<th>Content</th>
<th>Tests under mineral oil</th>
<th>Tests under natural ester</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>HV winding</td>
<td>LV winding</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total loss current</td>
<td>2.576 A</td>
<td>2.576 A</td>
</tr>
<tr>
<td>Rated current</td>
<td>2.410 A</td>
<td>2.410 A</td>
</tr>
<tr>
<td>Injection current</td>
<td>2.580 A</td>
<td>2.577 A</td>
</tr>
<tr>
<td>Ambient Temperature</td>
<td>20 °C</td>
<td>20 °C</td>
</tr>
<tr>
<td>Bottom oil temperature rise</td>
<td>22.3 °C</td>
<td>18.4 °C</td>
</tr>
<tr>
<td>Top liquid temperature rise</td>
<td>49.0 °C</td>
<td>55.1 °C</td>
</tr>
<tr>
<td>Average liquid temperature rise</td>
<td>37.0 °C</td>
<td>38.7 °C</td>
</tr>
<tr>
<td>Average winding temperature rise</td>
<td>58.1 °C</td>
<td>57.3 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>63.0 °C</td>
</tr>
<tr>
<td>HST rise</td>
<td>73.6 °C</td>
<td>72.7 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>83.4 °C</td>
</tr>
<tr>
<td></td>
<td></td>
<td>88.4 °C</td>
</tr>
</tbody>
</table>

Assuming the specific heat for natural ester is 2100 J/kg.K and for mineral oil is 1700 J/kg.K as stated in [45], the ratio between inlet mass flow rate under mineral oil, \( \dot{m}_{\text{mineral}} \), to that under natural ester \( \dot{m}_{\text{natural}} \) given the data shown in Table 3.1 can be calculated as \( \dot{m}_{\text{mineral}} = 1.7 \times \dot{m}_{\text{natural}} \).
\[ P_{\text{loss}} = \dot{m} \times c_p \times (T_{\text{top}} - T_{\text{bottom}}) \] 3.2

where \( P_{\text{loss}} \) is the transformer total losses in Watts, \( \dot{m} \) is the inlet mass flow rate in kg/s, and \( T_{\text{top}} \) and \( T_{\text{bottom}} \) are the top oil and bottom oil recorded temperatures in K respectively.

A retro-filling study was conducted on a 50 MVA, 141 kV/13 kV transformer filled with mineral oil and then retro-filled with natural ester and the key temperatures were measured such as top oil temperature, bottom oil temperature, and the HST [46]. Tests were conducted under two cooling modes of ONAN with 30 MVA rating and of ONAF under 50 MVA rating. Table 3.2 summarizes test results. The HST was 20 °C higher when testing under natural ester compared to mineral oil under ONAF mode while top oil temperature was only 6 °C higher. As reviewed earlier in [45], results indicate that the developed inlet mass flow rate while using mineral oil is around 1.6 to 1.8 higher compared to the developed inlet mass flow rate while using natural ester mainly because natural ester has higher viscosity which increases pressure loss in the transformer hydraulic loop.

Table 3.2 Comparison of temperature rise test data using mineral oil and natural ester [46]

<table>
<thead>
<tr>
<th>Temperature rise above ambient (°C)</th>
<th>ONAN (30 MVA)</th>
<th>ONAF (50 MVA)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Mineral oil</td>
<td>Natural ester</td>
</tr>
<tr>
<td>Bottom oil temperature rise</td>
<td>40</td>
<td>33</td>
</tr>
<tr>
<td>Top oil temperature rise</td>
<td>52</td>
<td>53</td>
</tr>
<tr>
<td>HST rise</td>
<td>62</td>
<td>70</td>
</tr>
<tr>
<td>Oil viscosity (cSt) at 40 °C</td>
<td>10</td>
<td>33</td>
</tr>
<tr>
<td>Oil specific heat at 40 °C (assumed like Gemini X and MIDEL 7131)</td>
<td>1800</td>
<td>1950</td>
</tr>
<tr>
<td>Calculations of ( \dot{m} ) ratio using equation 3.2</td>
<td>( \dot{m}<em>{\text{mineral}} = 1.8 \times \dot{m}</em>{\text{natural}} )</td>
<td>( \dot{m}<em>{\text{mineral}} = 1.6 \times \dot{m}</em>{\text{natural}} )</td>
</tr>
</tbody>
</table>

3.4 Summary

Through reviewing literatures, techniques used to simulate the thermal performance of disc type windings were identified as the thermal hydraulic network models and computational fluid dynamics modelling. It was recognized that the network model, though gives quick answers for transformer designers, lacks the accuracy in used...
coefficients of pressure loss and heat transfer. On the other hand, computational fluid
dynamic simulations require large computing powers and longer processing time
hindering them unfeasible for day to day design requirement despite their accurate
predictions. Instead, CFD simulations are used to derive correlation equations, pressure
drop, and heat transfer coefficients to enhance the accuracy of network models.

Parameters influencing the temperature and oil flow rates in cooling ducts are identified
from reviewed literatures as the horizontal cooling duct height, the vertical cooling duct
width, the number of discs per pass, inlet oil velocity, inlet oil temperature, resistive and
eddy current losses in the transformer discs, and the type of insulation liquid used. It
was reported that higher vertical duct width or lower horizontal duct height create more
uniform oil flow distribution. Lower number of discs per pass causes more uniform oil
flow distributions and hence lower average winding temperatures with the cost of
increased pressure losses. It was reported that inlet oil temperature causes a linear
increase of the HST and higher inlet oil velocity enhances the transformer cooling. It
was recommended for ON simulations to use a full detailed conjugate heat transfer
model which gives detailed descriptions of both the solid and the fluid domains taking
buoyancy forces into considerations. For both ON and OD simulations, iterative process
solving hydraulic and thermal problems is used until convergence is achieved. Finally, it
was observed that ester based liquids caused increased HSTs compared to mineral oils
in reviewed cases of disc type windings.

It was observed the limited experimental work published to study the factors influencing
the HST and oil flow distribution in disc type windings. Moreover, few published
literatures included the measurement of oil flow rates in horizontal cooling ducts [40],
[31], [43], and most recently [44]. Three main techniques were used to record oil flow
rates in radial cooling ducts. The Hot Wire Anemometry (HWA) was used under
isothermal conditions and it is prone to positioning errors as it is a point based
measurement technique. Laser Doppler Velocimetry (LDV) was used under both
isothermal and non-isothermal testing conditions and through it the characterization of
inlet flow rates in ON transformer was facilitated. Finally, tracking of seeding particles
in radial cooling ducts was used both under isothermal and non-isothermal testing
conditions and through it validations of CFD simulations were achieved. In this study,
Particle Image Velocimetry (PIV) system is used to document oil flow rates in radial
cooling ducts under ON and OD testing conditions.
Chapter 4 Experimental Descriptions

4.1 Experimental Setup

The experimental setup is designed to fulfil two main objectives. The first is to allow testing under various winding model geometrical and operational conditions that include ON and OD cooling modes. The second is to allow the measurements of temperature and oil flow rates in radial cooling ducts. A schematic diagram of the experimental setup is shown in Figure 4.1. In general, it consists of a disc type winding model, a flow meter, a pump, an external heating unit, a radiator, an expansion vessel, and pipe works. Air is allowed to escape through two air bleeds above the winding model and from the radiator. The oil is filled in the system using an oil tank fixed above the system through the lowest point near the pump. The flow meter records oil flow rates in litres per minute (lpm) while the external heating unit is used to control the bottom oil temperature as desired. The winding model is designed to represent a section of disc type winding, and the radiator is used to cool the oil down. A multi-channel thermocouple system is used to record plate temperatures and a PIV system is used to record oil flow rates in radial cooling ducts. A photo of the experimental setup in the lab is shown in Figure 4.2.

Figure 4.1 Hydraulic loop of the experimental setup
Table 4.1 summarizes both geometrical and operational parameters that are investigated in this PhD research. The geometrical parameters include variations of radial duct height, vertical duct width and the number of discs per pass. The radial cooling duct height tested are 4 mm and 6 mm. Both the inlet and outlet axial cooling ducts widths were fixed to have the same widths of 8 mm, 10 mm, or 12 mm as desired. The number of discs segments, plates, per pass can be 6 arranged in 5 passes, 10 arranged in 3 passes, or 15 arranged in 2 passes.

The operational parameters are selected to represent both ON and OD cooling modes within the experimental setup testing capabilities. Power losses in each plate are controlled using variable transformers and tests are conducted under a wide range, from 20 W/plate to 70 W/plate which are equivalent to 600 W/m² to 2814 W/m². Different types of insulating oils are tested, of which the thermal performances are compared. Three oil types are used which are mineral oil (Nytro Gemini X), gas-to-liquid oil (Diala S4 ZX-I), and synthetic ester oil (MIDEL 7131).

This chapter consists of three main sections. Section 4.1 offers detailed descriptions of main components of the experimental setup and their selection considerations. Section 4.2 focuses on temperature and oil flow measurements within the winding model. Section 4.3 shows repeatability tests for temperature, flow rates, and pressure drop over the winding model.

Table 4.1 Range of testing parameters

<table>
<thead>
<tr>
<th>Geometrical Parameters</th>
<th>Operational Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial duct height (mm)</td>
<td>Axial ducts width (mm)</td>
</tr>
<tr>
<td>4*</td>
<td>(8,8)</td>
</tr>
<tr>
<td>(10,10)*</td>
<td>10,3*</td>
</tr>
<tr>
<td>(12,12)</td>
<td>15,2</td>
</tr>
</tbody>
</table>

*Inlet flow rate in m/s is calculated assuming the reference geometries tagged by *
4.1.1 Disc type winding model

The winding model is considered the main component in the experimental setup. The winding model is designed to represent a segment of a disc type transformer winding between two adjacent sticks or spacers as highlighted in Figure 4.3. Implications of model assumptions are described as follows.

1. The cylindrical shape of disc segments is substituted by a rectangular shape as this would make the sealing and construction of the winding model, from a practical point of view, more feasible. The 2D axi-symmetric assumption [17, 18] is considered valid under experimentation. As will be shown in Chapter 6, tests comparison between 2D CFD simulations and oil flow rate measurements using the PIV system prove that modelling degree and assumptions are acceptable under isothermal OD testing conditions for pressure and oil flow measurements. There are several reasons to justify the selection of a rectangular test rig rather than a cylindrical segment as in real transformers. First, for a rectangular test rig, the liquid velocity is not affected by the varied cross sectional area along the flow as the cross sectional area is the same for all locations, toward the duct entrance or toward the duct exist. This would aid in making more accurate velocity measurements within radial cooling ducts. Second, it was reported that 2D axi-symmetric modelling is sufficiently accurate if the appropriate correction factors are made [18] and so conclusions drawn from a rectangular test rig should be
representative to 2D CFD simulations under OD cooling conditions as discussed in Chapter 6.1. Finally, winding model assembly and sealing can be conducted more frequently and more easily with a rectangular test rig.

2. Disc segments are modelled as solid aluminium plates. In a transformer, each disc segment consists of several turns each turn wrapped with paper insulation. Turns are both thermally and electrically insulated using oil impregnated paper insulation. The solid aluminium plate simplification and plate heating method affect the radial plate temperature. It is expected that the radial temperature gradient along the flow direction is lower when using solid plates compared to when using composite insulated plates. Also, the lack of paper insulation reduces the plate average temperature and so create overall lower average winding temperature compared to when plates are insulated with paper. However, it is believed that the presence of paper insulation and composite plates do not affect the axial temperature distribution along the winding model. It is also believed that the presence of composite plates does not influence oil flow distribution. Oil impregnated paper, in real transformers, are used to enhance the dielectric strength in between strands and to provide mechanical support. With time, paper relaxes and bulges and so effectively blocking portion of the cooling ducts. Hence, wrapping the oil impregnated paper should be done appropriately such that bulging is avoided as much as possible. In the experimental setup with the aluminum plates, lab-based manual wrapping of paper would always result in paper bulging and hence blocking a portion of the cooling duct. This effect would increase the uncertainty and add another contributing factor to the oil flow distribution in cooling ducts. Thus it was decided not to include paper and to focus on the axial oil flow and temperature distributions instead of the radial temperature distribution.

3. Washers provide thermal insulation between passes. In transformers, washers are made of pressboards with few millimetres thickness. In the winding model, washers are made of acrylic material with 10 mm thickness and placed between passes as shown in Figure 4.3. Washers would fulfil their basic function of directing the flow in a zig-zag fashion and to create thermal insulation between passes.

Figure 4.4 shows temperature profile of a heated plate along with key plate dimensions. Each plate is made using aluminium block with 100 mm × 104 mm × 10 mm (as depth, radial length, and thickness) dimensions. Two thermocouples are embedded in each plate to record pass inlet-side temperature and pass outlet-side temperature. The average
of the two thermocouple readings is considered as the plate average temperature. Each plate is heated using two resistive cartridge heaters inserted into the plate through cylindrical holes allocated 30 mm from each edge as shown in Figure 4.3.

The temperature profile of the plate surface was taken under air natural cooling conditions using thermal camera when the plate was heated with 40 W by applying 100 V to both cartridge heaters. The temperature is slightly hotter on top of the cartridge heaters. The temperature profile is symmetrical.

Figure 4.3 Winding model construction

The temperature profile of the plate surface was taken under air natural cooling conditions using thermal camera when the plate was heated with 40 W by applying 100 V to both cartridge heaters. The temperature is slightly hotter on top of the cartridge heaters. The temperature profile is symmetrical.
Table 4.2 Cartridge heater resistance temperature-dependent factor [47]

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>20</th>
<th>93</th>
<th>204</th>
<th>315</th>
<th>427</th>
<th>538</th>
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</thead>
<tbody>
<tr>
<td>Factor</td>
<td>1.000</td>
<td>1.016</td>
<td>1.041</td>
<td>1.054</td>
<td>1.066</td>
<td>1.070</td>
</tr>
</tbody>
</table>

Table 4.2 shows the cartridge heater temperature dependent factor [47]. The resistance of the cartridge heater can be obtained at higher temperatures by multiplying the factor of the desired temperature by the cartridge heater resistance at room temperature [47]. By applying AC voltage from 10 V to 230 V and measuring the current in each cartridge heater, power injection can be calculated. Mean values and standard deviations of calculated power of the three cartridge heaters are shown in Table 4.3. Figure 4.5 shows measured voltage against the measured current for the three cartridge heaters. A total of 60 cartridge heaters are used. The mean resistance of all cartridge heaters are 540 ohms with ± 2.5% deviations between different cartridge heaters.

Table 4.3 Characterizations data of three cartridge heaters 1, 2, and 3

<table>
<thead>
<tr>
<th>Voltage (V)</th>
<th>Current (mA)</th>
<th>Power (W)</th>
<th>Mean value (W)</th>
<th>Standard deviation (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.5</td>
<td>19.57</td>
<td>20.4</td>
<td>0.2</td>
<td>0</td>
</tr>
<tr>
<td>30.3</td>
<td>54.8</td>
<td>58</td>
<td>1.7</td>
<td>1.7</td>
</tr>
</tbody>
</table>
The winding model is made using a polycarbonate material, namely Lexan® 9030 which has a rated temperature of 145 °C. Lexan® 9030 has 85% light transmission capabilities and its thermal conductivity is 0.21 W/m°C. Lexan® 9030 can be processed in workshop using milling machines. Figure 4.6 shows the dimensions of a side wall with 4 mm radial duct height. Two side walls are needed and only one host all the holes for thermocouples and wires for cartridge heaters. Front walls are supported by grooves in side walls to fix the desired width of axial cooling ducts. Front walls are also made of Lexan® 9030 material. Washers have the dimension of 100 mm × (104 + axial duct width) mm ×10 mm. Winding top and winding bottom are designed to host side and front walls of the winding model and their dimensions are shown in Figure 4.7.

<table>
<thead>
<tr>
<th></th>
<th>59.2</th>
<th>59.3</th>
<th>60.1</th>
<th>105.7</th>
<th>106.5</th>
<th>110.3</th>
<th>6.3</th>
<th>6.3</th>
<th>6.6</th>
<th>6.4</th>
<th>0.17</th>
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<tr>
<td>90.4</td>
<td>90.6</td>
<td>90.6</td>
<td>161.5</td>
<td>161.5</td>
<td>165.9</td>
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<td>120.0</td>
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<td>25.8</td>
<td>25.6</td>
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<td>26.0</td>
<td>0.47</td>
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<td>149.1</td>
<td>151.0</td>
<td>149.5</td>
<td>266.8</td>
<td>266.8</td>
<td>273.5</td>
<td>39.8</td>
<td>40.3</td>
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<td>40.3</td>
<td>0.55</td>
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<td>178.8</td>
<td>180.0</td>
<td>179.7</td>
<td>320.1</td>
<td>319.2</td>
<td>327.5</td>
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<td>57.5</td>
<td>58.9</td>
<td>57.87</td>
<td>0.91</td>
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<td>376.1</td>
<td>371.7</td>
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<td>1.26</td>
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<tr>
<td>230.5</td>
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<td>230.5</td>
<td>411</td>
<td>411</td>
<td>420</td>
<td>94.7</td>
<td>94.7</td>
<td>96.8</td>
<td>95.4</td>
<td>1.2</td>
<td></td>
</tr>
</tbody>
</table>

Figure 4.5 Characterizations of three cartridge heaters
Figure 4.6 Side walls dimensions in mm

Figure 4.7 Winding model a) top b) bottom

Figure 4.8 provides a demonstration of winding model assembly. The winding model consists of side walls, front walls, winding top and winding bottom. The assembly of the winding model is done with the following order. First, the two side walls are placed next to each other and physically supported using the supporting structure. Washers and plates are then inserted one by one into the allocated grooves of the side walls. Once all the plates are inserted, the supporting structure is tightened and the plates are manually aligned in their desired positions. Cartridge heaters and thermocouple wires are then inserted starting from the most bottom plate in the winding model through the allocated holes in one of the side walls.
Experimental Study of Transformer Liquid Flow and Temperature Distributions

Figure 4.8 Winding model assembly
Silicone sealant is then used to seal the holes and wires. The process of inserting thermocouples and cartridge heaters and sealing is continued until all plates are embedded and holes are sealed. The alignment of plates is checked again and the system is left overnight for the silicone to cure. Once the silicone is cured, winding bottom washer and winding top washer are inserted along with thermocouples to measure bottom and top oil temperatures as shown in Figure 4.9. Front walls are then inserted and washers are arranged and sealed using silicone to create the desired arrangement of zig-zag flow.

Once washers are sealed, the grooves in the winding bottom are filled with silicone and the entire assembled structure is placed over the winding bottom. Also, the winding top grooves are filled with silicone and then placed on top of the winding model. Silicone beads are spread between the front walls and the side walls and along any intersection to guarantee sealing. The winding model is left for 24 hours until silicone is cured and then leak tests are conducted. If any leak locations are spotted, sealing is fixed until no leaks from the winding model are observed.

### 4.1.2 Radiator

Radiators are heat exchangers which are used to maximize heat dissipation from the transformer oil to the external cooling medium. Transformer oil carries the heat generated by electrical losses from within the transformer winding to the radiator. The transferred heat through the radiator depends on oil flow rate, total surface area of the
radiator, external cooling medium, ambient temperature, and mean oil temperature. The heat transfer coefficient from the radiator surface to the external cooling medium determines the required total radiator surface area. Figure 4.10 shows a depiction of a typical transformer radiator [48]. Transformer radiators are constructed using many panels. Panels are connected together in parallel from inlet and outlet using pipes to distribute the hot oil through the panels. In this study, a domestic radiator is used instead of an actual transformer radiator. The selected domestic radiator is a double panel radiator with 2.0 m length and 0.7 m height. The datasheet of the radiator indicates that under normal domestic heating conditions with mean water temperature of 50 °C, the radiator is able to dissipate 4000 W. Large size radiator is selected to allow conducting tests at lower bottom oil temperatures. External fans can be used if needed to increase the dissipated heat through the radiator. The room hosting the experimental setup is equipped with an extraction fan which allows the room ambient temperature to stay around normal ambient temperatures of 25 °C as recorded during conducted tests.

Figure 4.10 Depiction of a transformer radiator [48]

4.1.3 Pump

The pump is used to drive oil flow rate under both ON and OD tests. The reason why the pump is needed under ON tests is because the hydraulic losses in the experimental setup are large enough to suppress the ability of the developed thermosiphon forces within the winding model to create a total circulating inlet flow rate. Under ON tests, the pump is used to specify the inlet oil flow rate as a boundary condition which is selected such that inside the winding model the buoyancy forces are dominating over inertia forces. In the experimental setup, head losses are caused by the pipe work, the radiator, the flow meter, the external heating unit, and the winding model. In this study,
it is desired to operate at a wide range of flow rates and so a centrifugal pump was used and deemed suitable to provide flexible range of flow rates.

To control the pump speed, two main methods can be used. The traditional method is called throttling in which a valve ahead of the pump is partially closed to increase the head loss of the system and so the operating point of the pump is shifted to the left of the pump curve. The second method is done through using an AC drive in which the drive reduces the applied frequency and voltage to the pump in the same ratio and so the entire pump curve is shifted. Two main laws describe the performance of a pump under the control of an AC drive and these are called the affinity laws. Equation 4.3 shows the first law which states the relationship between the operating frequency \( f \) of the pump and the discharge rate \( Q \) of the pump. It can be seen that the law states a linear relationship between \( f \) and \( Q \). The second law relates the operating frequency of the pump and the head \( H \) that can be provided by the pump and it is shown by equation 4.4. The relationship is not linear; if the frequency drops to 0.5 of its rated value then the head drops to 0.25 of its rated value.

Table 4.4 Properties of selected pump

<table>
<thead>
<tr>
<th>Maximum head (m)</th>
<th>Maximum flow rate (lpm)</th>
<th>Rated current (A)</th>
<th>Input power, single phase (W)</th>
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<tr>
<td>7.2</td>
<td>117</td>
<td>1.3</td>
<td>300</td>
</tr>
</tbody>
</table>

\[
\frac{Q_1}{Q_2} = \frac{f_1}{f_2} \quad (4.3)
\]

\[
\frac{H_1}{H_2} = \frac{f_1^2}{f_2^2} \quad (4.4)
\]

The actual system curve and the calculated system curve are shown on Figure 4.11 along with the pump curve provided in the pump datasheet. The actual system curve is constructed by operating the pump at lower frequencies using an AC drive. By using equations 4.3 and 4.4 to construct the pump new curve with measured oil inlet flow rate, actual system curve can be estimated.

It can be observed that there is a significant difference between the actual system curve and the calculated system curve. In general, the system curve consists of head losses in the winding model, radiator, pipe network and flow meter. In the calculated system curve, one or more of these components were underestimated in terms of their head loss.
To achieve higher inlet flow rates, a bigger pump is required or the head loss in the experimental setup should be reduced. The achieved maximum inlet flow rate of 24 lpm, equivalent to 0.4 m/s under axial duct width of 10 mm, is considered sufficient for the present study.

![Graph comparing actual and calculated system curves](image)

**Figure 4.11 Actual system curve compared to calculated system curve**

### 4.1.4 Flow meter

Inlet oil flow rate is an important parameter and it should be measured and controlled accurately. For the selection of the flow meter, the range of the inlet oil temperature should be taken into consideration as of its effects on the meter reading accuracy and reliability. The variation of the inlet oil temperature affects the oil properties such as the viscosity and density. The ideal flow meter should only relate the quantity of interest to be measured, oil flow rate, with the output signal of the meter such as angular velocity or the number of pulses per minute. Tests are planned to be conducted under a wide range of inlet oil temperatures and using different oils. Also, tests are conducted from very low inlet flow rates of 1 lpm to the maximum flow rate provided by the pump of 24 lpm. For these reasons, Positive Displacement (PD) flow meter is selected as the most suitable flow meter as it is minimally affected by oil density and viscosity. Also it provides flow rate measurements accurately under very low flow rates. PD flow meters are designed to allow the fluid to pass through them in pre-specified units of volume which is fixed by their structure. They have high accuracy and they are able to measure the flow in small quantities. The flow does not need to be conditioned with long pipes ahead of the meter and they can measure intermittent flows. They are almost insensitive to viscosity variations. An oval gear flow meter was used in this study. The flow meter
has a measurement range from 0.5 lpm to 100 lpm with 0.5% accuracy of reading. The inlet flow rate (FR) in lpm can be converted to inlet velocity \( V_{in} \) using equation 4.1

\[
FR = V_{in} \times D_{th} \times W_d \times 60000 \quad (4.1)
\]

\[
\dot{m} = V_{in} \times D_{th} \times W_d \times \rho \quad (4.2)
\]

The winding model depth \( D_{th} \) is fixed in all tests to 0.1 m and \( W_d \) is substituted in m. Also, \( \dot{m} \) can be linked to \( V_{in} \) using equation 4.2.

4.2 Temperature and Oil Flow Rate Measurements

4.2.1 Multi-channel temperature system

As described before, each plate is embedded with two thermocouples. The winding model can host up to 55 plates and so a multi-channel temperature measurement system is required. The selected system can provide up to 128 thermocouple channels. As shown in Figure 4.12, the system consists of two 32 × 2 CIO-EXP32 multiplexers. The main CIO-EXP32 is connected to a PCI-DAS08 data acquisition card inserted inside a computer and the PCI-DAS08 is controlled using LABVIEW. Each CIO-EXP32 consists of two 16 × 1 multiplexers and each channel is addressed using 4 digital bits supplied by the PCI-DAS08 as shown in Figure 4.13. The output of each 16 × 1 multiplexer is connected to an analog input in the PCI-DAS08 unit.

![Figure 4.12 Components of used multi-channel temperature measurement system](image)

System calibration was performed using a controlled temperature water bath as follows.

Water bath temperature is checked using a commercial thermocouple and a mercury thermometer. The accuracies of the mercury thermometer and the external thermocouple were checked at 0 °C in an ice bath and at 100 °C in boiling water. The
water bath temperature indicator was compared to both the external thermocouple and to the mercury thermometer and results are summarized in Table 4.5.

Using the water bath, the generated voltages from thermocouples connected to the multi-channel temperature system were recorded at different water bath temperatures from 10 °C to 100 °C in 10 °C steps. Least square curve fitting was used to map generated voltages by thermocouples at fixed water bath temperatures in the form of $T(E) = A \times E + B$ where $A$ and $B$ are constant and $E$ is the generated voltage by each thermocouple.

![Figure 4.13 Detailed view of analog and digital inputs and outputs of the multi-channel temperature system](image)

**Table 4.5 Water bath controlled temperature accuracy check (°C)**

<table>
<thead>
<tr>
<th>Set Temperature</th>
<th>Water bath indicator</th>
<th>Mercury thermometer</th>
<th>External thermocouple measurement</th>
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<td>100 (boiling water)</td>
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Calibration equations are generated for all thermocouples and accuracy and mapping checks are performed. Calibrated measurements are shown in Table 4.6. As a conclusion, the accuracy of used thermocouples can be considered as ± 1 °C.

Table 4.6 Calibrated temperatures of used 64 K-type thermocouples, (°C)

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4.2.2 Particle image velocimetry

PIV is a method used extensively in aerospace and fluid dynamic studies to document flow patterns and capture flow fields in different applications. Tracking seeding particles within the flow using high speed camera is the basic principle behind a PIV operation. The application of PIV to record oil flow rates in radial ducts is presented in chapter 5.

4.2.3 Pressure drop measurement

It is desired to measure the static pressure drop over the winding model to verify the prediction of pressure drop coefficient using dimensional analysis offered in [39]. A differential pressure measurement instrument was used. The pressure instrument has ±10 Pa pressure accuracy. Pressure drop measurements were conducted only under isothermal OD cooling conditions. Pressure measurement ports were created in front walls of the winding model at bottom pass inlet washer and at top pass outlet washer.

4.3 Repeatability Tests

Initial tests are conducted to check measurement repeatability of temperature, oil flow rates in radial cooling ducts, and pressure drop over the winding model. All tests are conducted under radial cooling duct height of 4 mm and 10 plates per pass with a total of three passes in the winding model. Oil flow rates are measured only in the winding model top pass. The cooling ducts in the winding top pass, third pass, are named from duct 1 to duct 11 from bottom to top of the pass. Similarly, plates in the third pass are named from plate 1 to plate 10 from bottom to top of the pass. Conducted repeatability tests are categorized as tests under isothermal conditions or as tests under non-isothermal conditions as discussed in the following.

4.3.1 Results under isothermal conditions

Under isothermal conditions, the oil has a uniform temperature. Repeatability of oil flow rates and pressure drop measurements are presented here. Both tests are conducted
Experimental Study of Transformer Liquid Flow and Temperature Distributions

and repeated three times in three different days to check repeatability of PIV isothermal measurements. The first test was conducted under bottom oil temperature of 20 °C and inlet oil velocity of 0.2 m/s or equivalently 12 lpm. Axial cooling duct width was 10 mm. Test results are shown in Figure 4.14 (a). On the figure, the inlet flow rate measured by the inlet flow meter (FR<sub>M</sub>) is compared to the inlet flow rate from PIV measurement (FR<sub>PIV</sub>) and measurements agree well within 5% difference from each other. The second test is conducted under bottom oil temperature of 47 °C and inlet oil velocity equals to 0.17 m/s or equivalently 12 lpm with axial cooling duct of 12 mm. Results are shown in Figure 4.14 (b). It can be concluded that oil flow rates in radial cooling ducts can be measured with sufficient accuracy using the PIV system.

Figure 4.14 PIV repeatability tests under isothermal conditions

The repeatability of pressure drop measurements is conducted under isothermal conditions with oil temperature of 50 °C. Tests are conducted under winding model with

Figure 4.15 Repeatability of pressure drop measurement
Experimental Study of Transformer Liquid Flow and Temperature Distributions

axial cooling duct of 12 mm. Results are shown in Figure 4.15. Pressure drop repeatability is within the stated accuracy of the pressure instrument ± 10 Pa.

**4.3.2 Results under non-isothermal conditions**

Temperature and PIV measurements are conducted three times under axial cooling duct width of 12 mm and inlet oil velocity of 0.17 m/s or equivalently 12 lpm. The inlet oil temperature for the three tests was 43 °C ± 1 °C. In each plate, 50 W or equivalently 2010 W/m² of heat flux density were injected. Figure 4.16 shows test results of both temperature and oil flow rates in radial cooling ducts.

![Figure 4.16 Temperature and PIV measurement repeatability tests under non-isothermal testing conditions](image)

Measurements of temperature and oil flow rates were conducted in the third pass only. Repeatability of temperature measurement is within the thermocouple accuracy of ± 1 °C. The deviation of PIV measurement from the inlet flow meter is within 5% the same as that found earlier under isothermal testing.

In addition, comparison of PIV measurement and pressure measurement with CFD simulations are presented in Chapter 6.
4.3.3 Quantifying power dissipation from the winding model

Estimating the power dissipation, $P_{\text{diss}}$, through the winding model can be easily done by calculating the difference between the total power injection in W, $P_{\text{inj,total}}$, and the total power transported by the oil in W, $P_{\text{oil}}$, using $P_{\text{diss}} = P_{\text{inj,total}} - P_{\text{oil}}$. $P_{\text{inj,total}}$ is known and controlled in all conducted tests while $P_{\text{oil}}$ can be calculated using the measured quantities of inlet mass flow rate, $m$, and the inlet, $T_{\text{in}}$, and outlet, $T_{\text{out}}$, oil temperatures, and the specific heat of oil $c_p$ as in $P_{\text{oil}} = m c_p (T_{\text{out}} - T_{\text{in}})$. In the experimental setup, 8 K-type thermocouples were used to measure $T_{\text{out}}$ and 3 K-type thermocouples were used to measure $T_{\text{in}}$ in each test. Uncertainty of temperature measurement of ± 1 °C makes estimating $P_{\text{diss}}$ challenging as the uncertainty in temperature increases the uncertainty in the estimated $P_{\text{diss}}$. Table 4.7 shows the estimated power dissipation through the winding model at several representative operating conditions. It is expected, naturally, that the power dissipation is higher when operating at higher average oil temperature and when operating at lower $m$. Nonetheless, measuring $T_{\text{out}}$ is always challenging especially with the intrinsic existence of hot streaks.

Table 4.7 Estimation of power dissipation through the winding model

<table>
<thead>
<tr>
<th>Winding model geometries: $H_d = 4$ mm, $W_d = 10$ mm, $n_d = 10$, Gemini X</th>
<th>$V_{\text{in}}$ (m/s)</th>
<th>$m$ (kg/s)</th>
<th>$c_p$ (J/kgK)</th>
<th>$T_{\text{in}}$ (°C)</th>
<th>$T_{\text{out}}$ (°C)</th>
<th>$P_{\text{inj,total}}$ (W)</th>
<th>$P_{\text{oil}}$ (W)</th>
<th>$P_{\text{diss}}$ (W)</th>
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</thead>
<tbody>
<tr>
<td>0.017</td>
<td>0.0139</td>
<td>1950</td>
<td>61.2</td>
<td>87.7</td>
<td>900</td>
<td>719</td>
<td>180</td>
<td></td>
</tr>
<tr>
<td>0.021</td>
<td>0.0174</td>
<td>1950</td>
<td>60.6</td>
<td>83.7</td>
<td>900</td>
<td>783</td>
<td>116</td>
<td></td>
</tr>
<tr>
<td>0.025</td>
<td>0.0209</td>
<td>1950</td>
<td>60.6</td>
<td>80.4</td>
<td>900</td>
<td>805</td>
<td>94</td>
<td></td>
</tr>
<tr>
<td>0.1</td>
<td>0.0835</td>
<td>1950</td>
<td>70</td>
<td>78</td>
<td>1500</td>
<td>1302</td>
<td>197</td>
<td></td>
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<tr>
<td>0.2</td>
<td>0.167</td>
<td>1950</td>
<td>70.3</td>
<td>74</td>
<td>1500</td>
<td>1204</td>
<td>295</td>
<td></td>
</tr>
<tr>
<td>0.3</td>
<td>0.2505</td>
<td>1950</td>
<td>70.2</td>
<td>73</td>
<td>1500</td>
<td>1367</td>
<td>132</td>
<td></td>
</tr>
</tbody>
</table>

4.4 Summary

The designed experimental setup was modelled to represent a section of a disc type winding model. Solid aluminium plates were used to model disc segments and they were heated with resistive heaters. A positive displacement inlet flow meter was selected to record winding inlet oil flow rates while an external heating unit was used to control winding model inlet oil temperatures. Axial temperature distributions in the winding plates were measured using a multi-channel temperature system with 64 K-type thermocouples which were calibrated to have ± 1 °C accuracy. Oil flow rates in radial cooling ducts are measured using a PIV system. The setup allows conducting tests under
a range of inlet oil temperatures, inlet oil velocities, different loss profiles and loading levels, and various combinations of winding model geometries such as radial cooling duct height, axial cooling duct width, and the number of discs per pass. Tests under OD cooling conditions are presented in Chapter 6. Tests under ON cooling conditions are presented in Chapter 7. Finally, comparison of alternative transformer oils is presented in Chapter 8.
Chapter 5 Oil Flow Measurement using PIV System

In this paper, presentation of a Particle Image Velocimetry (PIV) system is given to record and characterize oil flow rates in radial cooling ducts. Special phenomena such as oil reverse flow, oil recirculation at duct entrances, and 3D characterization of oil flow velocities in radial cooling duct are given. PIV was used to record oil flow rates in radial cooling ducts under both oil natural and oil directed cooling conditions and under both isothermal and non-isothermal testing conditions. The PIV system proved to be a strong tool to record oil flow rates in radial cooling ducts.

Co-authors provided feedback and comments on paper draft and through feedbacks during regular meetings each three months.

1. Introduction

Ageing of insulating paper in power transformers depends on their operating temperatures. Oil-immersed disc type winding power transformers are cooled using insulation oil which flows through radial and axial cooling ducts. The cooling performance of the winding determines substantially the hot spot temperature which is essential for estimating the life expectancy of transformer insulation system. The oil is either pumped through the winding or allowed to flow naturally based on developed buoyancy forces. The winding geometries influence the flow distribution of transformer oil within the winding. The thermal design of power transformers is mainly based on past experiences and the use of simulation models such as Thermal Hydraulic Networks Models (NHTM) [1]-[4] and more recently Computational Fluid Dynamics (CFD) based models [5], [6]. These models are used to predict temperature distribution within the winding under different geometrical and operational conditions. Experimental verifications of the developed models are always desired. Recently, dimensional analysis was applied to highlight the influential parameters that affect the oil flow distribution in and pressure loss over a disc type transformer winding [7]-[8].

In a transformer, the temperature is affected by both the loss distribution and the oil flow distribution. The temperature in a transformer can be measured using fibre optic temperature sensors at preinstalled locations where the hot spot temperature is expected. However, recording the oil flow distribution within a transformer remains challenging. Therefore, small scale experimental test rigs are commonly established to verify simulation based models. In the literature, some experimental studies were reported in which flow rates were measured using techniques such as Hot Wire Anemometry (HWA) [9], Laser Doppler Velocimetry (LDV) [10], [11] and tracking of tracer particles [12] which is a basic form of Particle Image Velocimetry (PIV) technique.

In this article, a thermal test rig is introduced which allows investigating temperature and flow distributions in a disc type winding model. A PIV technique is used to record and characterize oil flow rates and distribution within cooling ducts of the winding model. Phenomena such as oil recirculation at duct entrances and oil reverse flow are captured. A 3D characterization of oil flow in the cooling duct is carried out. Finally, demonstration examples of both temperature and oil flow distributions under oil directed (OD) and oil natural (ON) cooling modes are given.
2. Thermal test rig

A thermal test rig used to experimentally study transformer thermal behaviour is shown in Figure 1. It consists of a disc type winding model, a pump, an external flow meter, and a radiator. The winding model consists of three passes and each pass contains 10 disc segments (plates). Pass 3 plates are numbered from plate 1 to plate 10 from bottom to top of the pass as shown in Figure 1 (b). Each plate is heated using two heating elements and power injection is controlled by variable transformers. Thermocouples are used to measure the temperature of each plate and a multi-channel thermometer is implemented to provide a total of 64 channels for recording plates and oil temperatures. PIV is used to measure oil flow rates in the pass 3 cooling ducts. Pass 3 ducts are numbered from duct 1 to duct 11 from bottom to top of the pass as shown in Figure 1 (b). The construction of the thermal test rig was discussed in detail in [13].

![Thermal test rig with detailed view of pass 3.](image)

2.1 Particle Image Velocimetry

Figure 2 shows the working principle of a PIV system. A PIV system usually consists of a dual pulsed class IV laser source, laser optics, a camera, seeding particles, acquisition software, and a synchronizer to synchronize the operation of different PIV components.

The operational procedure of the PIV system used in this work is given as follows: first, seeding particles are added to the flow. Silver coated hollow glass spheres with diameter range from 9 to 14 micrometres were added to the transformer oil. Second, through the acquisition software the synchronizer sends synchronized control pulses to both the laser and the camera. The camera should be placed perpendicular to the laser sheet plane. Third, the camera opens its frame to capture any reflected laser by the seeding particles and then the laser source fires the first laser pulse, Pulse A. The laser pulse passes through the laser sheet optics to become a laser sheet which illuminates the seeding particles in the flow. The camera captures the first image called Frame A. Fourth, after a specified time delay $\Delta t$ the second laser pulse, Pulse B, is fired and the camera captures the second image called Frame B. During the time delay $\Delta t$, the seeding particles would have moved from their initial position in Frame A to a new position in Frame B. Using the time delay $\Delta t$ and the position displacement of the seeding...
particles, from Frame A to Frame B, the velocity profile within cooling ducts can be calculated. Finally, the software divides the raw images into small interrogation areas and processes them to infer the velocity profile. Interrogation areas are usually squares or rectangular with $32 \times 32$ pixels or $32 \times 64$ pixels. The selection of the interrogation area size depends on the flow pattern and seeding density.

The key to obtain reliable PIV data is to obtain good quality of raw images, which depends on the density and the distribution uniformity of seeding particles within the oil flow. A proper seeding density can only be achieved through references and trials. Three main recommendations are reported in [14] for obtaining good and reliable raw images. First, there should be from 5 to 15 seeding particles in each interrogation area. Second, a seeding particle should occupy from 3 to 5 pixels in the captured image. Third, seeding particles displacement should not exceed 25% of interrogation window length. The software used in the PIV system tracks the movement of seeding particles in captured pair of images, Frame A and Frame B, through statistical means to find the most likely displacement of a group of particles.

3. Results and Discussions

3.1 Measurement of oil velocity

Calibration of the raw images is necessary for deriving the oil velocity. The raw images can be calibrated using a target with known dimensions. The camera is zoomed and focused on the target. Calibration images are then taken and used to convert captured image pixels into actual distance (in meter). In this work, the known height of the winding model plates and radial cooling ducts are used as a calibration target. Two types of measurements are compared; the camera field of view of the first type is focused on one radial cooling duct, called one-duct measurement, and the camera field of view of the second type is focused on two radial cooling ducts, called two-duct measurement. Figure 3 shows comparison between one-duct measurement and two-duct measurement. In the one-duct measurement, the camera is focused on only one duct hence the measured velocity profile has higher resolution. However, the calibration distance is smaller as only one duct height is used as the calibration
distance which in turn induces, relatively, larger calibration error. Nonetheless, the one-duct measurement enables measurement of velocity data near duct edges and hence capturing a more-complete parabolic profile. In the two-duct measurement, lower resolution of each duct velocity profile is obtained. In this case, the reflected laser by seeding particles near the duct walls might not reach the camera due to the field of view of the camera setup. Nonetheless, large distance between two ducts is used as the calibration distance and so produces relatively smaller calibration error compared to the one-duct measurement. Under both measurement types, the maximum duct velocity ($V_{\text{max}}$) is captured accurately given that image calibration is reliable and accurate. The velocity profile is a laminar profile and it can be characterized by $V_{\text{max}}$. Figure 3 shows an example of each measurement type. Both measurement types were performed on duct 7 under the same testing conditions of 0.2 m/s winding model average inlet velocity ($V_{\text{in}}$). The maximum duct velocity for the one-duct measurement is 0.077 m/s while the maximum duct velocity for the two-duct measurement is 0.074 m/s. In this work, the two-duct measurement is used and $V_{\text{max}}$ was extracted from processed PIV images. Hence the laminar profile is a parabolic profile, the average duct velocity ($V_{\text{av}}$) can be related to $V_{\text{max}}$ using $V_{\text{av}} = (V_{\text{max}}/1.5)$. The field of view of the two-duct measurement is raised up duct by duct so that the velocity profile in ducts 2-10 is measured twice.

![Figure 3 Comparison between (a) one-duct measurement and (b) two-duct measurement under $V_{\text{in}} = 0.2$ m/s and oil temperature kept at ambient conditions of 20 °C.](image)

**3.2 Observations of Oil Recirculation and Reverse Flow phenomena**

**3.2.1 Oil recirculation**

At duct entrances, the existence of oil recirculation was reported and observed using CFD simulations [15]. Oil recirculation is a contributing factor to the so called minor pressure losses within the transformer winding and as mentioned in [15] it influences the flow distribution. The PIV system in the present thermal test rig can be used to observe the possible oil recirculation at radial cooling duct entrance. As an example, Figure 4 shows velocity fields at the entrance of two radial cooling ducts, duct 6 and duct 7, under $V_{\text{in}} = 0.27$ m/s and inlet oil temperature kept at ambient conditions of 20 °C.
temperature of 70 °C. It is shown that oil recirculation occurred at the duct entrance, highlighted by the red arrows on Figure 4. It was observed under conducted tests that oil recirculation is more apparent under OD cooling mode compared to ON cooling mode. It is observed that higher inlet velocity or higher inlet oil temperature leads to larger oil recirculation zone at duct entrances.

![PIV raw images and processed images](image)

Figure 4 Observation of oil recirculation at the entrances of radial cooling ducts 6 and 7 under $V_{in} = 0.27$ m/s and oil inlet temperature = 70 °C.

### 3.2.2 Oil reverse flow

In OD disc type transformers, blocking washers, demonstrated in Figure 1, are used to direct the oil flow into a zig-zag fashion and the flow is assumed to be in the same direction in a pass. However, it was reported that the flow in the pass bottom radial cooling ducts might reverse its direction under certain operational and geometrical conditions [7], [12]. This could happen especially at higher inlet flow rates, higher inlet oil temperatures and under higher number of discs per pass [12]. Using the PIV system in the present thermal test rig, oil reverse flow is also observed. An example is shown in Figure 5 under $V_{in} = 0.27$ m/s and inlet oil temperature of 70 °C. Duct 1 and duct 2 velocity profiles are plotted at the indicated profile line shown in Figure 5. It is clear that oil flow reversed its direction in duct 1. Oil reverse flow might cause overheating of discs especially if oil stagnations occur in a duct due to reverse flow in the pass.

![Profile line and velocity profiles](image)

Figure 5 Observations of the occurrence of reverse flow in the pass bottom cooling duct under $V_{in} = 0.27$ m/s and oil inlet temperature = 70 °C.
3.3 Characterization of oil flow in 3D dimensions

Oil flow within the radial cooling ducts is often assumed as a 2D flow for simplification. However, under some operational conditions the oil radial flow might not be a 2D flow and that a Z-component, defined on Figure 6, would exist within some ducts. For the used experimental setup with the aluminum solid plates, the existence of 3D flow is not expected to have an observable influence over plate temperature because the average heat convection from the plate depends on the average oil mass flow rate through cooling ducts around the plate especially with the fact that the plate is solid aluminum and there would be excellent heat conduction within the plate preventing any significant temperature gradient to exist. However if the plate is modelled using insulated aluminium strands, then the strands where the velocity is lowest due to the 3D effect would have higher temperature. In these cases, the 3D velocity component may affect the location of the HST.

![PIV application](image)

![Top view](image)

Figure 6 a) Overview of PIV application defining expected flow direction in a pass b) 3D characterization of oil flow using 9 measured locations within cooling ducts.

The Z-component makes the ability to record a representative average velocity of the cooling duct challenging. To investigate the Z-component, measurements are conducted at 9 different locations within duct 2 and duct 3 of the winding model. As shown in Figure 6 (b), three locations are defined for the laser beam named as A, B, and C and three locations are defined for the camera field of view named as Entrance, Middle and Exit of the cooling duct. To refer to each location, the camera position is tagged with the laser position (i.e Entrance_A refers to camera position Entrance and laser position A). Two examples of measurements are conducted and discussed as follows:

**Case 1:** Characterizations of oil flow in duct 2 and duct 3 are conducted under isothermal conditions with inlet velocity of 0.1 m/s and inlet oil temperature of 20 °C. Analysed images from 9 different locations are shown in Figure 7. For each duct on Figure 7, a representative maximum velocity $V_{max}$ is extracted and shown. As a demonstration example, by comparing $V_{max}$ of duct 2 at the Entrance position between laser positions A, B, and C, it can be noticed that the differences of the recorded $V_{max}$ between Entrance_A and Entrance_B is 0.015 m/s and the difference between Entrance_A and Entrance_C is 0.018 m/s. However, the difference between Entrance_B and Entrance_C is only 0.003 m/s. As the oil progresses toward the exit of the duct at the Exit position, the difference between Exit_A and Exit_B is reduced to 0.004 m/s and the difference between Exit_A and Exit_C is reduced to only 0.001 m/s. It can be concluded that under case 1, the Z-component subsides down...
towards the exit of the cooling duct. In this case, the Z-component is small mainly because the oil inlet velocity and oil temperature are low.

![Flow Characterization Images](image)

**Figure 7** Case 1 flow characterization under inlet velocity = 0.1 m/s and inlet oil temperature = 20 °C in duct 2 and duct 3. Each image has its own colour indexing to velocity.

**Case 2**: Characterizations of oil flow in duct 2 and duct 3 are provided under isothermal conditions with inlet velocity fixed to 0.27 m/s and inlet oil temperature fixed to 70 °C. This case is considered on the extreme high range of operating conditions. Results are shown in Figure 8. Large differences of extracted \( V_{\text{max}} \) are observed when comparing results taken at the Entrance positions of A, B, and C for both ducts. The flow direction of duct 2 at the Entrance_A position is reversed compared to the Entrance_C position with a velocity difference of 0.042 m/s. For duct 3, the difference between Entrance_A and Entrance_C is 0.066 m/s. Like case 1, as the oil progresses towards the exit of the ducts, the Z component subsides down. For duct 2, the difference between the recorded \( V_{\text{max}} \) between positions Exit_A and Exit_C is only 0.003 m/s and for duct 3 the difference between Exit_A and Exit_C is 0.014 m/s. Compared to case 1, the Z-component is more apparent. This indicates that higher inlet velocity and higher inlet oil temperature cause a stronger Z-component to be developed. While using a PIV system or a point based flow measurement technique, such as Hot Wire Anemometry (HWA) or Laser Doppler Velocimetry (LDV), it is important to take the 3D effect into considerations. Overall, it is observed that the Z-component is more apparent in the top most and in the bottom most cooling ducts of a pass. The Z-component is partly caused by geometrical imperfections of the winding model; however, the Z-
component may as well be intrinsic in top most and bottom most cooling ducts due to washers force the oil to divert. More investigations are needed to characterize the Z-component.

Figure 8 Case 2 flow characterization under inlet velocity = 0.27 m/s and inlet oil temperature = 70 °C in duct 2 and duct 3. Each image has its own colour range.

3.4 Examples of temperature and flow distributions under OD and ON cooling modes

The fundamental difference between OD and ON cooling modes is that in OD cooling mode an external pump is the prominent force driving the oil flow while in ON cooling mode the thermosyphon force caused by oil temperature differences is the prominent force driving the oil flow. Using the present thermal test rig, extensive studies can be conducted to characterize both cooling modes through oil flow and temperature measurements under a wide range of operational and geometrical conditions. An example for each cooling mode, is given by measuring the temperature and flow distribution in the third pass of the winding model.

OD cooling mode example: Figure 9 shows the temperature profile and oil flow distribution under OD cooling mode with \( V_{in,meter} \) recorded by external flow meter, fixed to 0.2 m/s and uniform power loss equivalent to 1000 W/m², or 50 W/plate. Inlet oil temperature, \( T_{in} \), and outlet oil temperature, \( T_{out} \), are recorded using thermocouples. \( V_{in} \) can be calculated as well from the PIV measurements in all ducts (\( V_{in,PIV} \)). For the conducted case, \( V_{in,PIV} = 0.23 \) m/s which is slightly higher than the recorded \( V_{in,meter} \). The discrepancies between \( V_{in,PIV} \) and \( V_{in,meter} \) are partly caused by the accumulative error from PIV measurements in each cooling duct. Also, the Z-
component, mentioned earlier, causes more deviations in the top most and bottom most cooling ducts. Under OD cooling mode, more oil flows in top cooling ducts in a pass which enhances the cooling of individual plates top of the pass as can be seen by the temperature profile. Higher inlet velocity causes more distorted flow distribution which may lead to oil flow stagnation or oil reverse flow in duct 1 as demonstrated earlier and discussed in [7].

![Temperature profile and flow distribution graphs](image)

**Figure 9** Example of OD temperature and flow distributions under $V_{in}$ of 0.2 m/s and losses equivalent to 2010 W/m$^2$. Under OD mode, more oil flow in pass top cooling ducts and the hot spot temperature is located in the pass bottom plate.

**ON cooling mode example:** Figure 10 shows temperature distribution and oil flow distribution under ON cooling mode with $V_{in}$ fixed to 0.021 m/s and uniform loss injection equivalent to 1200 W/m$^2$ or 30 W/plate. Under ON cooling mode, more oil flows in bottom cooling ducts of the pass which in effect reduces the temperature of pass bottom plates. Under ON cooling mode, oil flow distribution has an opposite shape compared to OD cooling mode. The hot spot temperature is located toward the top of the pass under ON conditions while under OD conditions the hot spot temperature is located toward the bottom of the pass. In both cooling modes, oil stagnation could occur, however location wise, it is either in the top cooling ducts under ON mode or in bottom cooling ducts under OD cooling mode.
Conclusion

In this article, a disc type transformer winding model based thermal test rig was introduced. Particle Image Velocimetry (PIV) system was used to record oil flow rates within radial cooling ducts. The established thermal test rig allows extensive studies on both OD and ON cooling modes under a wide range of winding model geometries and operating conditions. Operating conditions can be the winding inlet velocity, winding inlet temperature, uniform or non-uniform loss injection and finally different types of transformer insulating liquids. Geometrical conditions can be different radial cooling duct heights, different axial cooling duct widths, and various arrangement of the number of discs per pass.

The PIV technique proved to be a powerful tool to study transformer thermal behaviour. The technique can be used under both isothermal and non-isothermal conditions allowing the detailed study of flow distributions under both OD and ON cooling modes. Phenomena such as oil recirculation at duct entrances and the occurrence of oil reverse flow were documented. A 3D flow in cooling ducts was observed at high inlet oil velocities and inlet oil temperatures. The 3D flow was more apparent in top most and bottom most cooling ducts in a pass and only under OD cooling modes. It was observed that the 3D flow subsides down towards the exit of the cooling duct. Further investigations are needed to identify and characterize the causes of the 3D flow. Examples of OD and ON cooling modes using the test rig were provided. Under OD cooling conditions, more oil flows in top cooling ducts of the pass causing the hot spot temperature to be located toward bottom discs of the pass. Under ON cooling mode, less oil flows in top cooling ducts of a pass causing the hot spot temperature to be located toward the top plates of the pass.

Reference


Chapter 6 Oil Flow and Temperature Distributions under OD Cooling Mode

The first paper presents experimental validation of derived correlation equations from CFD simulations using dimensional analysis under isothermal conditions. The dimensional analysis was used, originally in [37], to simplify the problem and to determine the most influential dimensionless parameters that affect the oil flow distribution in and the pressure drop over disc type windings.

The second paper presents an experimental study of the influence of transformer operating conditions and winding geometrical parameters on oil flow and temperature distribution in OD cooled transformers. Experimental parametric sweeps were performed to individually study the influence of each parameter.


6.1 Isothermal Study of OD Cooled Transformers

*Experimental Verification of Dimensional Analysis Results on Flow Distribution and Pressure Drop for Disc Type Windings in OD Cooling Modes*

Xiang Zhang, Muhammad Daghrah, Zhongdong Wang, Member, IEEE, Qiang Liu, Member, IEEE, Paul Jarman, Massimo Negro

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Muhammad Daghrah contributed 50% to this paper through the experimental work while Xiang Zhang contributed 50% to the paper through simulation based work. Industrial co-authors provided feedback on paper draft and through regular meetings each three months.
Experimental Verification of Dimensional Analysis Results on Flow Distribution and Pressure Drop for Disc Type Windings in OD Cooling Modes

Xiang Zhang, Muhammad Daghrarah, Zhongdong Wang, Member, IEEE, Qiang Liu, Member, IEEE, Paul Jarman, Massimo Negro

Abstract— Oil flow distribution in the winding has a direct impact on the cooling performance. In addition, static pressure drop over the winding determines oil split among windings connected hydraulically in parallel. In this paper, experimental verifications are provided to support computational fluid dynamics (CFD) simulations for disc-type windings in oil forced and directed (OD) cooling modes. Oil flow distribution in and pressure drop over disc-type winding models are measured using a particle image velocimetry (PIV) system and a differential pressure instrument, respectively. Dimensional analysis is adopted to analyze the relationship between flow distribution, or pressure drop, and the controlling parameters. CFD parametric sweeps of the dimensionless parameters obtained from the dimensional analysis are conducted and the CFD results are then correlated with the dimensionless parameters. The comparisons between measured results and the corresponding results obtained from the correlations demonstrate constant consistency, proving the validity of both the method of dimensional analysis and the correlations. Finally, comparisons of experimental results from isothermal and nonisothermal conditions in OD cooling modes are executed, which show that the isothermal conclusions can be extended to nonisothermal cases because of the effects of buoyancy force and hot-streak dynamics are proved to be negligible.

Index Terms—CFD, dimensional analysis, disc-type winding, flow distribution, OD, PIV, pressure drop, transformer

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<td>$P$</td>
<td>Perimeter of the vertical duct (m)</td>
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<td>$P_{fl}$</td>
<td>Volumetric flow proportion in horizontal duct $i$</td>
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<tr>
<td>$\Delta P$</td>
<td>Static pressure drop (Pa)</td>
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<tr>
<td>$Re$</td>
<td>The Reynolds number ($\rho \cdot v_{in} \cdot D_h / \mu$)</td>
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<tr>
<td>$Ri$</td>
<td>The Richardson number ($Gr / Re^2$)</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature in Kelvin (K)</td>
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<tr>
<td>$T_{in}$</td>
<td>Temperature at pass 3 inlet in Kelvin (K)</td>
</tr>
<tr>
<td>$T_{out}$</td>
<td>Temperature at pass 3 outlet in Kelvin (K)</td>
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<tr>
<td>$\bar{v}_1$</td>
<td>Average oil velocity in duct $i$ (m/s)</td>
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<tr>
<td>$\bar{v}_{in}$</td>
<td>Average pass-inlet oil velocity (m/s)</td>
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<tr>
<td>$W_{drc}$</td>
<td>Vertical duct width (m)</td>
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</tr>
<tr>
<td>$\beta$</td>
<td>Dimensionless plate axial height ($H_{plate} / W_{drc}$)</td>
</tr>
<tr>
<td>$\beta_T$</td>
<td>Volumetric thermal expansion coefficient</td>
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<tr>
<td>$\gamma$</td>
<td>Dimensionless plate radial width ($W_{plate} / W_{drc}$)</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Oil density (kg/m$^3$)</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Oil dynamic viscosity (Pa·s)</td>
</tr>
</tbody>
</table>

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Xiang Zhang, Muhammad Daghrarah, Zhongdong Wang and Qiang Liu are with the School of Electrical and Electronic Engineering at The University of Manchester, Manchester, M13 9PL, UK (e-mail: zhongdong.wang@manchester.ac.uk).

Paul Jarman is with National Grid, UK

Massimo Negro is with Weidmann Electrical Technology AG, Switzerland

I. INTRODUCTION

The oil flow distribution in a disc-type transformer winding has a direct impact on the cooling performance which, in conjunction with power loss distribution, determines the position and magnitude of the hot-spot temperature in the winding. In addition, the static pressure drop over a disc-type transformer winding determines the selection of pump for oil forced and directed (OD) cooling modes and more importantly oil split among windings connected hydraulically in parallel, for example LV winding and HV winding in the same phase. Different models have been developed to predict oil flow distribution in and pressure drop over disc type transformer windings. These models can be grouped into two categories: network models [1-4] and computational fluid dynamics (CFD) models [5-7]. These two types of model share the same physical principles of conservation of mass, momentum and energy. Endeavors of combining the merits of both methods have been made by using CFD results to calibrate the correlation equations used in network models [8-10]. CFD...
modelling is usually conducted in axisymmetric 2D geometries to reduce computational requirements. Compared to 3D CFD simulations, 2D simulations cannot capture detailed fluid flow and heat transfer phenomena in the vicinity of the spacers. However, for an ON cooling mode, comparisons of fluid flow and temperature distributions between 2D axisymmetric and 3D CFD models show that 2D results are representative of the 3D results as long as the governing dimensionless parameters \(Gr/Re^2\) are matched [7]. In addition, key 3D results, e.g. hot-spot temperature, can be derived from 2D CFD results based on the improvement strategies proposed in [7].

Only a few experimental studies were reported to record flow rates within horizontal cooling ducts to verify simulation models. Hot wire anemometry was used in [11] to record flow distribution within an isothermal winding model. It was observed that the lowest flow rate occurred in the central ducts of the tested pass of the winding model and more uniform flow distribution was achieved with wider inlet vertical cooling duct. A CCD camera was used in [12] to track the movements of tracer particles within a zig-zag disc type winding model under isothermal conditions. Pressure drops were measured along winding vertical ducts. It was observed in [12] that flow distribution is more uniform for lower number of discs per pass at the cost of higher pressure losses over the winding model. Reverse flow was observed in the first horizontal cooling duct under higher number of discs per pass. It was also reported in [12] that wider vertical ducts would produce more uniform flow distribution in and lower pressure drop over the winding. Laser doppler velocimetry was used to measure flow velocities within vertical cooling ducts in [13] to verify a proposed numerical model on a non-directed winding under nonisothermal conditions. Pressure drop over one pass of a winding model was measured in [14] to verify a 2D CFD model and a network model with calibrated correlation equations from [9].

Systematic simulation studies of oil flow distribution in and pressure drop over disc-type transformer windings were conducted by adopting dimensional analyses in [15]. From dimensional analyses, oil flow distribution and pressure drop were transformed to dimensionless parameters of flow proportion in each horizontal cooling duct and pressure drop coefficient over the winding, respectively. The controlling parameters were transformed to the Reynolds number at the winding pass inlet and dimensionless geometrical parameters (\(\alpha, \beta, \gamma\) in [15]). The quantitative relationships between flow proportion, or pressure drop coefficient, and the dimensionless controlling parameters were obtained by conducting CFD parametric sweeps and correlating the CFD results with the controlling dimensionless parameters.

In this paper, experiments are designed to verify the method of dimensional analysis and the correlations obtained from CFD parametric sweeps. The experimental set-up and the implementation of the measurements of flow distribution and pressure drop are illustrated in section II. Theoretical analyses and numerical modeling of the flow distribution in and pressure drop over the winding model are presented in section III. In section IV, experimental verification of the modeling results are presented. Finally, the comparisons of isothermal and nonisothermal measurements are shown in section V, followed by discussion and conclusion in sections VI and VII, respectively.

II. EXPERIMENTAL SET-UP

Experimental setup is shown in Fig. 1. It consists of a winding model resembling a disc-type winding structure as shown in Fig. 2, a radiator to cool oil down, a pump to circulate oil and a flow meter to record flow rates. An external heating unit is used to heat up and control winding inlet oil temperature. The experimental setup is described as follows.

A. Winding Model

The rectangular winding model consists of 3 passes. Each pass is fitted with 10 disc segments. The disc segments are modeled as rectangular aluminum plates with geometrical dimensions of 100 mm × 104 mm × 10 mm. Each plate sits on 3 mm grooves of the side walls of the winding model making the actual depth of horizontal cooling ducts 94 mm. The winding model walls are made of lexan® 9030 polycarbonate sheets. Washers are made of acrylic material with geometrical dimensions of 100 mm × (104 + vertical duct width) mm × 10 mm. Table I provides a summary of the geometrical dimensions of the winding models. Inner and outer vertical duct width are made equal and two vertical duct widths, 10 mm and 12 mm, are tested. The heights of horizontal ducts are uniform being 4 mm or 6 mm. Thermocouples are fitted at the inlet and outlet of the winding model to record winding inlet and outlet oil temperatures, respectively. Accuracy of the thermocouples used is ±1 °C.
However, during winding model assembly minimal geometrical deviations still occur, e.g. a plate is either slightly inward or outward. It was observed, after conducting the tests, that this slight geometrical deviation gives a signature, or a footprint, to the flow distribution within the pass as will be highlighted later. More details of the winding model can be found in [16].

Fig. 2. Geometry of the winding model. (a) 3D geometry; (b) 2D geometry

TABLE I

<table>
<thead>
<tr>
<th>Horizontal Duct depth (cm)</th>
<th>Vertical duct height (mm)</th>
<th>Vertical duct width (mm)</th>
<th>Aluminum plates</th>
</tr>
</thead>
<tbody>
<tr>
<td>94 mm</td>
<td>4 mm, 6 mm</td>
<td>10 mm, 12 mm</td>
<td>10 mm, 104 mm</td>
</tr>
</tbody>
</table>

B. Pressure Measurement

Two pressure ports, as shown in Fig. 2, were used to record static pressure drop over the 3-pass winding model with a differential pressure instrument. The pressure instrument has an accuracy of ± 0.2% of the full scale of 10 kPa or equivalently the accuracy is ± 20 Pa and a repeatability of ± 10 Pa in this range. Each pressure measurement was taken 10 times. Since the pressure readings were stable, the average of the 10 measurements was used as a representative value.

C. Flow Measurement

Total oil flow rate is measured using a positive displacement flow meter, as it is less affected by viscosity variations, with accuracy of 0.5% of its reading. Within horizontal cooling ducts, a Particle Image Velocimetry (PIV) system is used to record oil flow rates. The PIV system consists of a laser source, light sheet optics and a camera. A synchronizer is used to synchronize the laser, the camera and the acquisition computer in order to obtain the velocity profile.

The accuracy of PIV measurements depends on the quality of the raw images photographed which is affected by both the density of seeding particles and their distribution within the duct. Silver coated hollow glass spheres with 9-14 micrometer diameter were used as seeding particles. In all PIV measurements, the laser sheet plane was aligned with the flow direction and perpendicular to the winding plate in each duct and was fixed only at one plane that is 3 cm from the winding model wall close to the camera. The camera field of view was focused on the 2 cm of the horizontal duct towards its exit as indicated in Fig. 2. PIV measurements were taken in steady state and in a hydraulically developed region in horizontal ducts. Twenty raw images were taken for each measurement and the statistical average of these images was used to extract the final result. The reason for using 20 raw images rather than using more is that a sweep of the number of raw images processed —1, 5, 10, 20, 30, 40, 50—shows that the case using 20 raw images gives results that are as accurate as those obtained from cases of larger number of images. More detailed description of the application of PIV to record oil flow rates in radial cooling ducts is given in [17]. Finally, the accuracy of PIV measurements was checked by comparing the total flow rate obtained from all the PIV measurements in pass 3 to that recorded by the flow meter. Discrepancies between the PIV results and the flow meter readings were within 7%.

Flow distribution in a pass is hardly affected by the number of passes in the winding [15]. Therefore, only the flow rates within the horizontal cooling ducts of the third pass were measured using the PIV system. Fig. 3 shows the results of an experiment repeated 3 times in 3 different days, under the same operating conditions, to confirm the repeatability of the PIV measurements. In the repeatability tests, the total flow rate, recorded by the flow meter, was 12.0 litres per minute (lpm) while the inlet oil temperature was maintained within the range of 46 °C to 48 °C. It can be seen from Fig. 3 that the total flow rates derived from the PIV measurements are constantly slightly higher than that from the flow meter. This is partially because only one plane in the horizontal duct is measured and the end-wall effects are neglected.

D. Oil Properties

The tested oil is a mineral oil of which the variations of density and dynamic viscosity with temperature in Kelvin are shown in (1) and (2). These equations are from least-square curve fittings of the measured data provided by the oil manufacturer.

\[ \rho = -0.65683 \times T + 1063.6 \]  

(1)
\( \mu = 7.8630 \times 10^{-5} \times \exp \left( \frac{631.96}{T - 176.03} \right) \) \hspace{1cm} (2)

### III. ANALYTICAL ANALYSES ON FLOW DISTRIBUTION AND PRESSURE DROP FOR THE WINDING MODEL

The methodology of conducting dimensional analyses, performing CFD parametric sweeps and correlating the CFD results with the identified dimensionless parameters was detailed in our previous publication [15]. This paper follows the same methodology for the winding model tested.

#### A. Dimensional Analysis on Pressure Drop over and Flow Distribution in the Winding Model

In dimensional forms, the average oil velocity in a horizontal cooling duct \( i \) (\( \bar{V}_i \)) and the pressure drop over the 3-pass winding model (\( \Delta P \)) can be expressed as functions shown in (3) and (4):

\[
\bar{V}_i = f(n_i, \bar{V}_{in}, \rho, \mu, \bar{H}_{duct}, \bar{H}_{plate}, \bar{W}_{plate}, \bar{W}_{duct}) \quad (3)
\]

\[
\Delta P = g(n_i, n_2, \bar{V}_{in}, \rho, \mu, \bar{H}_{duct}, \bar{H}_{plate}, \bar{W}_{plate}, \bar{W}_{duct}) \quad (4)
\]

These dimensional forms can be transformed to their dimensionless forms by choosing pass inlet velocity (\( \bar{V}_{in} \)), oil density (\( \rho \)), and vertical duct width (\( \bar{W}_{duct} \)) as the repeating parameters. The dimensionless forms are (5) and (6):

\[
\frac{\bar{V}_i}{\bar{V}_{in}} \cdot \frac{H_{duct}}{W_{duct}} = P_i = f'\left(n_i, Re, \alpha, \beta, \gamma\right) \quad (5)
\]

\[
\frac{\Delta P}{\rho \bar{V}_{in}^2} = C_{pd} = g\left(n_i, n_2, Re, \alpha, \beta, \gamma\right) \quad (6)
\]

#### B. Parametric Sweeps

In the winding model, the number of discs per pass and the number of passes are 10 and 3, respectively, as shown in Fig. 2. In such a case, flow proportion in a duct \( i \) (\( P_i \)) and pressure drop coefficient over the 3-pass winding model (\( C_{pd} \)) are functions of four dimensionless parameters: \( Re, \alpha, \beta \) and \( \gamma \).

According to the practical ranges of the total oil flow rate and the geometrical dimensions of disc-type transformer windings [15, 17], the four dimensionless parameters are set to be in the ranges shown in Table II. All the combinations of these discrete parameters were simulated by CFD simulations using COMSOL Multiphysics 5.2. In total, 720 CFD simulations were conducted involving 80 winding geometries which is the number of combinations of \( \alpha, \beta \) and \( \gamma \). Mesh refinement studies were conducted to guarantee mesh-independent results following the same procedures as presented in [15].

#### TABLE II

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Discrete parameter value</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \text{Re} )</td>
<td>20, 50, 100, 200, 400, 600, 800, 1000, 1200</td>
</tr>
<tr>
<td>( \alpha )</td>
<td>0.25, 0.3, 0.4, 0.5, 0.6</td>
</tr>
<tr>
<td>( \beta )</td>
<td>0.67, 1, 2, 3</td>
</tr>
<tr>
<td>( \gamma )</td>
<td>6, 10, 12, 15</td>
</tr>
</tbody>
</table>

Discrete values are used to cover the practical range of each parameter.

#### C. Correlations

The results of flow proportion in the third pass and pressure drop coefficient over the 3-pass winding model were extracted and correlated with the four dimensionless controlling parameters by a multi-layer least square curve fitting strategy.

1) Correlation of Flow Distribution

The volumetric flow proportion in duct \( i \) in the third pass (\( P_i \)) was correlated with \( Re, \alpha, \beta \) and \( \gamma \) by equation set (7):

\[
P_i = a_i \frac{Re}{b_{ij}^{\alpha_j}} + a_{ij} \ln \frac{Re}{c_{ij}^{\beta_j}} + d_{ijk}^{\gamma_j}; \quad i = \{1, 2, \ldots, 11\}
\]

\[
a_i = b_{ij} \alpha_j^3 + b_{ij} \alpha_j^2 + b_{ij} \alpha_j + b_{ij}; \quad j = \{1, 2, 3, 4\}
\]

\[
b_{ij} = c_{ijk} \beta_j^6 + c_{ijk} \beta_j^5 + c_{ijk} \beta_j + c_{ijk}; \quad k = \{1, 2, 3, 4\}
\]

\[
c_{ijk} = d_{ijkl} \gamma_j^3 + d_{ijkl} \gamma_j^2 + d_{ijkl} \gamma_j + d_{ijkl}; \quad m = \{1, 2, 3, 4\}
\]

where \( a_i, b_{ij}, c_{ijk} \) are dummy correlation coefficients, which are determined from \( \alpha, \beta, \gamma \) and the 256 coefficient \( d's \) from the last equation in (7).

2) Correlation of Pressure Drop Coefficient

The pressure drop coefficient over the 3-pass winding model (\( C_{pd} \)) were correlated with \( Re, \alpha, \beta \) and \( \gamma \) by equation set (8):

\[
C_{pd} = a_i \frac{1000}{Re} e^{a_i Re/b_{ij}^{\alpha_j}}
\]

\[
a_i = b_{ij} (4\alpha_j)^{b_{ij}} e^{4\alpha_j}; \quad i = \{1, 2\}
\]

\[
b_{ij} = c_{ijk} \beta_j^6 e^{c_{ijk} \beta_j}; \quad j = \{1, 2, 3\}
\]

\[
c_{ijk} = d_{ijkl} \gamma_j^3 + d_{ijkl} \gamma_j^2 + d_{ijkl} \gamma_j + d_{ijkl}; \quad k = \{1, 2, 3\}
\]

where \( a_i, b_{ij}, c_{ijk} \) are dummy correlation coefficients, which are determined from \( \alpha, \beta, \gamma \) and the 72 coefficient \( d's \) from the last equation in (8).

The aforementioned dimensional analyses offer ways to reduce the relationships between flow distribution, or pressure drop, and the controlling parameters to the simplest forms as expressed by (5) and (6). The CFD parametric sweeps and the subsequent correlations of the CFD results quantify the relationships as shown in (7) and (8).

#### IV. EXPERIMENTAL VERIFICATION OF DIMENSIONAL ANALYSES AND CFD CORRELATIONS

From the dimensional analyses presented in section III, oil flow proportion in a duct and pressure drop coefficient over the 3-pass winding model are functions of the Reynolds number at the winding inlet, \( \alpha, \beta \) and \( \gamma \). To validate the method of dimensional analysis and the correlations obtained, measurements of flow distribution in and pressure drop over the winding model were implemented and compared with the theoretical predictions from (7) and (8).
The experimental tests of the winding model were designed to fulfil two objectives:
1. To verify that $P_i$ and $C_{pd}$ are controlled by $Re$ itself instead of the components it is composed of.
2. To verify that results from the correlation equation sets (7) and (8) fit well with measurements in the practical range of $Re$ for two chosen winding geometries: vertical duct width 10 mm and vertical duct width 12 mm. Since vertical duct width is the denominator of the three dimensionless geometrical parameters, even the other geometrical dimensions keep unchanged each vertical duct width would lead to a new set of $\alpha$, $\beta$ and $\gamma$.

### A. Verification of Objective 1

Since the hydraulic diameter is defined as $Dh=4A/p$, for a given oil type, the Reynolds number at the winding pass inlet is mainly controlled by the total oil flow rate and the oil temperature. The tested three cases with similar $Re$ for two vertical duct widths 10 mm and 12 mm are shown in Table III.

#### TABLE III THREE CASES WITH SIMILAR RE

<table>
<thead>
<tr>
<th>Pass inlet oil flow rate (lpm)</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil temperature (°C)</td>
<td>77</td>
<td>48</td>
<td>36</td>
</tr>
<tr>
<td>$Re (W_{ax}=10 \text{ mm})$</td>
<td>541</td>
<td>536</td>
<td>547</td>
</tr>
<tr>
<td>$Re (W_{ax}=12 \text{ mm})$</td>
<td>531</td>
<td>526</td>
<td>537</td>
</tr>
</tbody>
</table>

#### 1) Flow Distribution

For the winding model of vertical duct width 10 mm, the comparisons of average oil velocities and flow proportions in the third pass from PIV measurements are shown in Fig. 4 (a) and (b), respectively. The comparisons for vertical duct width 12 mm are shown in Fig. 5. It can be seen from both Fig. 4 and Fig. 5 that although the average velocities for the three cases are different, the flow distributions in terms of volumetric flow proportion in each duct are almost identical.

#### 2) Flow Distribution

The comparisons of pressure drops and pressure drop coefficients over the 3-pass winding model for vertical duct width 10 mm and 12 mm are shown in Table IV. For each geometry, the absolute static pressure drops are very different but the corresponding pressure drop coefficients are similar.

#### TABLE IV COMPARISON OF PRESSURE DROP AND PRESSURE DROP COEFFICIENT

<table>
<thead>
<tr>
<th></th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta P$ (pa)</td>
<td>43</td>
<td>200</td>
<td>466</td>
</tr>
<tr>
<td>$C_{pd}$</td>
<td>9.1</td>
<td>10.4</td>
<td>10.6</td>
</tr>
<tr>
<td>$\Delta P$ (pa)</td>
<td>37</td>
<td>148</td>
<td>303</td>
</tr>
<tr>
<td>$C_{pd}$</td>
<td>11.29</td>
<td>11.04</td>
<td>9.96</td>
</tr>
</tbody>
</table>

From these comparisons for flow distributions and pressure drop coefficients from cases with similar $Re$, objective 1 is acceptably achieved.

#### B. Verification of Objective 2

#### 1) Flow Distribution

To verify that the flow distribution correlation equation set (7) holds valid in the practical range of $Re$, flow distributions of two more cases were tested for both geometries. The tested cases are shown in Table V.

#### TABLE V THREE CASES TO COVER THE PRACTICAL RANGE OF RE

<table>
<thead>
<tr>
<th>Pass inlet oil flow rate (lpm)</th>
<th>Case 4</th>
<th>Case 2</th>
<th>Case 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil temperature (°C)</td>
<td>20</td>
<td>48</td>
<td>70</td>
</tr>
<tr>
<td>$Re (W_{ax}=10 \text{ mm})$</td>
<td>97</td>
<td>536</td>
<td>1402</td>
</tr>
<tr>
<td>$Re (W_{ax}=12 \text{ mm})$</td>
<td>95</td>
<td>526</td>
<td>1375</td>
</tr>
</tbody>
</table>

For the geometry of vertical duct width 10 mm, the comparisons of average oil velocities and flow proportions obtained from PIV measurements and (7) are shown in Fig. 6 (a) and (b), respectively. The comparisons for vertical duct width 12 mm are shown in Fig. 7. As can be seen from both Fig. 6 and Fig. 7, results from both methods share the same...
varying trends. With the increase of Re, flow distribution gets less uniform.

For case 4 and case 2, flow distribution curves from the equation set (7) follow closely the PIV results for both geometries. For case 5, the flow distribution curves from the two methods deviate more from each other.

For case 5 of the 10 mm vertical duct geometry, both PIV measurement and the flow correlation equation set (7) show the occurrence of reverse flow at the bottom of the pass; for the geometry of vertical duct width 12 mm, the correlation shows a nearly stagnated flow in the horizontal duct at the bottom of the pass, while the PIV measurement shows an average velocity of 18 mm/s. Apart from the difference in total oil flow rate obtained from PIV measurements and that from the flow meter, which is the input for (7), and the geometrical deviations of the winding model as is mentioned in section II part A, another reason for the increased discrepancy could be that Re for case 5 (around 1400) is beyond the upper boundary of Re (Re=1200) in the CFD parametric sweeps from which the correlation equation sets were derived. It is worth mentioning that case 5 should be avoided in real transformers because the occurrence of reverse flow can seriously jeopardize the cooling performance.

Comparing Fig. 6 and Fig. 7, we can see that with the increase of vertical duct width (equivalently due to the decrease of α) oil flow distribution gets relatively more uniform. To further verify the point that the decrease of α brings a relatively more uniform flow distribution, a comparison between two cases with different horizontal duct heights is executed. The test conditions of the two cases are shown in Table IV and the comparison of flow distribution is shown in Fig. 8. As can be seen from Fig. 8, the flow distribution for the case of α being 0.4 is more uniform than the case of α being 0.6. In fact, reverse flow occurs for the case of α being 0.6. In addition, the predicted results obtained from (7) follow the measured results closely. The velocity profiles in the bottom ducts of case 6 and case 7 are shown in Fig. 9. Half of the PIV images and velocity profiles are shown in Fig. 9 because they are symmetric to the central axes. The velocity of the reversed flow in case 7 is designated as negative velocities.

Fig. 6. Comparison of flow distribution for Re in the range in Table V with vertical duct width being 10 mm. (a) Average velocity in each duct of pass 3. (b) Flow proportion in each duct of pass 3 where the total oil flow rate is regarded as one unit. The legend ‘equ’ refers to correlation equation (7).

![Fig. 6](image)

Table IV

<table>
<thead>
<tr>
<th>TEST CONDITIONS OF CASE 6 AND CASE 7</th>
<th>Case 6</th>
<th>Case 7</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pass inlet oil flow rate (lpm)</td>
<td>12</td>
<td>12</td>
</tr>
<tr>
<td>Oil temperature (°C)</td>
<td>70</td>
<td>70</td>
</tr>
<tr>
<td>Vertical duct width (mm)</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Horizontal duct height (mm)</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>α</td>
<td>0.4</td>
<td>0.6</td>
</tr>
<tr>
<td>Re</td>
<td>878</td>
<td>878</td>
</tr>
</tbody>
</table>

Fig. 7. Comparison of flow distribution for Re in the range in Table V with vertical duct width of 12 mm. (a) Average velocity in each duct of pass 3. (b) Flow proportion in each duct of pass 3 where the total oil flow rate is regarded as one unit. The legend ‘equ’ refers to correlation equation (7).

![Fig. 7](image)

Fig. 8. Comparison of flow distribution for two horizontal duct heights in Table VI. (a) Average velocity in each duct of pass 3 where the total oil flow rate is regarded as one unit. (b) Flow proportion in each duct of pass 3. The legend ‘equ’ refers to correlation equation (7).

![Fig. 8](image)
Variation of pressure drop coefficient against Re for both geometries in log-log scale is a straight line at the beginning and then levels off in the region of high Re. When the measured pressure points are transformed to pressure drop coefficients, they fall close to the predicted curves from (8) and in the same way with maximum relative error being 16.9%.

From the comparisons for flow distributions and pressure drops from cases covering the practical range of Re, objective 2 is acceptably achieved.

V. COMPARISON OF ISOTHERMAL AND NONISOTHERMAL CASES

In practice, buoyancy force and hot-streak dynamics are involved in determining flow distribution in and pressure drop over the winding, especially for oil natural (ON) cooling modes [5-7]. To verify that for OD cooling modes the influences of buoyancy force and hot-streak dynamics are negligible and therefore the foregoing isothermal results hold valid for practical cases, comparison between isothermal and nonisothermal cases are shown in this section.

A. Power Injection in Each Plate

The current density in the copper conductor of a disc-type power transformer is in the range of 2-4 A/mm² [19]. Assuming the plates in the winding model were made of copper and a current density of 4 A/mm² ran through each plate with copper resistance at 85 °C, then the resistive power generated in each plate would be approximately 35 W. To account stray losses and eddy current losses in the winding and to provide some margin, a power injection of 50 W per plate was selected which was delivered by two cartridge heaters embedded in each plate as detailed in [16].

B. Comparisons of Flow Distribution and Pressure Drop

The comparisons of flow distribution and pressure drop between isothermal and nonisothermal conditions were implemented on the winding geometry of vertical duct width 10 mm. For an OD cooling mode, typical oil velocity in horizontal ducts range from 0.075 m/s to 0.15 m/s [18]. Therefore, average oil velocity at the winding pass inlet would...
be larger than 0.1 m/s corresponding to a total oil flow rate of approximately 6 lpm. In the nonisothermal tests when plates were heated, two flow rates of 6 lpm and 12 lpm are tested, of which the related parameters are shown in Table VIII. For comparison purposes, two isothermal flow cases of 6 lpm, 45 ºC and 12 lpm, 47 ºC were conducted.

The comparisons of flow distributions in pass 3 from PIV measurements are shown in Fig. 12. As can be seen, flow distributions from isothermal and nonisothermal conditions are almost identical. The comparisons of pressure drops for these isothermal and nonisothermal cases are shown in Table III. The discrepancies on pressure drop between the isothermal and nonisothermal cases are negligible. The negligible Ri’s (Gr/Re²) for case 6 and case 7 (much smaller than 1) shown in Table VII is responsible for the negligible differences in flow and pressure drop results between the isothermal and nonisothermal cases.

It can be concluded that for OD cooling modes, the influences of buoyancy force and hot-streak dynamics are negligible. Therefore, the aforementioned results obtained from isothermal flow conditions are valid for OD cooling modes.

### TABLE VII

<table>
<thead>
<tr>
<th>Parameters Related to Nonisothermal Tests</th>
<th>6 lpm</th>
<th>12 lpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pass 3 inlet temperature (ºC)</td>
<td>45</td>
<td>47</td>
</tr>
<tr>
<td>Pass 3 outlet temperature (ºC)</td>
<td>48</td>
<td>48.5</td>
</tr>
<tr>
<td>Ri (Gr/Re²)</td>
<td>0.046</td>
<td>0.006</td>
</tr>
</tbody>
</table>

The thermal expansion coefficient is 7.8×10⁻⁴ [1/K] and the temperature gradients in the definition of Ri (Gr/Re²) are taken as the differences between pass 3 outlet temperatures and pass 3 inlet temperatures.

Fig. 12. Comparison of flow distribution for isothermal and nonisothermal flow conditions with vertical duct width being 10 mm. (a) average velocity in each duct; (b) flow proportion in each duct where the total oil flow rate is regarded as one unit.

### TABLE VIII

<table>
<thead>
<tr>
<th>Comparison of Pressure Drop and Pressure Drop Coefficient</th>
<th>6 lpm 45 ºC</th>
<th>12 lpm 47 ºC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isothermal</td>
<td>ΔP (pa)</td>
<td>91</td>
</tr>
<tr>
<td>Nonisothermal</td>
<td>ΔP (pa)</td>
<td>90</td>
</tr>
</tbody>
</table>

Each pressure drop result is the average of ten repeated tests.

VI. DISCUSSION

The method of dimensional analysis, which can simplify a physical problem to its simplest form prior to obtaining a quantitative relationship, has been widely used in fluid mechanics. However, it has not yet become the common practice for transformer thermal modeling.

In this paper, the methodology presented in [15] was verified experimentally on disc-type transformer winding models. It is worth emphasizing that none of the experimental scenarios were simulated directly by CFD, yet the experimental results are in line with the results generated from the correlation equation sets obtained from 2D CFD parametric sweeps. Therefore the 2D modeling, the method of dimensional analysis, the CFD simulations, and the correlations of the CFD results are all verified to be valid for the investigation of flow distribution and pressure drop. In addition, comparisons between several typical experimental tests and CFD simulations of the same conditions were also made, of which the fluid flow and pressure drop results from the CFD simulations and experimental tests were as consistent as those generated from the correlation equation set.

Since the correlation equation sets have been verified experimentally, some insight into oil flow distribution in and pressure drop over the winding model can be obtained conveniently from the correlation equation sets. It was found that the geometrical parameter α is more influential than the other two geometrical parameters β and γ [15].

![Fig. 13. Variation of maldistribution with Re and α. In region 1, maldistribution is smaller than 5, whereas in region 2 maldistribution is larger than 5.](image-url)
the total oil flow rate and the larger the ratio of horizontal duct height to vertical duct width, the more distorted the flow distribution would be. For design purpose, if maldistribution is desired to be smaller than 5, then region 1 indicates the feasible combinations of Re and α, as shown in Fig. 13.

It was observed in the experiments that geometrical deviations lead to non-smooth flow distribution which could cause some horizontal cooling ducts suffering from smaller flow rate than expected. Although the trend of flow distribution will not change significantly, the reduction of flow rate in some specific ducts may still cause noticeable winding temperature rise. Geometrical deviations during the transformer manufacturing process are difficult to avoid and therefore the effects of some typical geometrical deviations are worth further investigation.

VII. CONCLUSION

Dimensional analyses were conducted to simplify the physical problems of oil flow distribution in and pressure drop over a winding model to the simplest forms. CFD parametric sweeps of the identified dimensionless controlling parameters were performed and then the CFD results obtained were correlated to form predictive correlation equation sets.

Measurements of flow distribution in pass 3 using a PIV system and recording of pressure drop over the 3-pass winding model were implemented to verify the method of dimensional analysis and the correlation equation sets obtained from CFD parametric sweeps. The fact that it is the Reynolds number itself instead of the way how it is composed of that controls oil flow distribution and pressure drop coefficient, was verified experimentally for a fixed winding geometry in OD cooling modes.

It was also satisfactorily verified that the correlation equation sets can generate results that are in line with the measurements in the practical range of Re. Therefore, the 2D CFD simulations and the strategies of correlating the CFD results to form predictive correlation equation sets are also valid. The application of the correlations to real 3D cylindrical windings needs further validation.

The conclusions on flow distribution in and pressure drop over the winding obtained from isothermal conditions can be extended to nonisothermal conditions in OD cooling modes, as the experiments on nonisothermal conditions confirmed that the effects of buoyancy force and hot-streak dynamics over flow distribution and pressure drop are negligible for the tested OD conditions. However, the effects of buoyancy forces and hot-streak dynamics on heat transfer in the winding need further investigation.

Acknowledgment

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REFERENCES


6.2 Non-isothermal Study of OD Cooled Transformers

*Experimental Characterization of Oil Flow Distribution and Hot Spot Temperature in Disc Type Windings in OD Cooling Modes*

Muhammad Daghrah, Zhongdong Wang, Qiang Liu
**Experimental Characterization of Oil Flow Distribution and Hot Spot Temperature in Disc Type Windings in OD Cooling Modes**

Muhammad Daghrah¹, Zhongdong Wang¹, Qiang Liu¹

¹The University of Manchester, M13 9PL, UK
zhongdong.wang@manchester.ac.uk

**Abstract:** In oil directed and cooled disc type power transformer, oil is forced to flow through the winding in a zig-zag fashion. Distribution of undesired electrical losses within the winding along with oil flow distribution within winding cooling ducts determine the location and value of the so-called hot spot temperature. In this paper, experimental tests were conducted under oil directed cooling modes using lab-scale disc type winding model where particle image velocimetry was used to measure oil flow rates within cooling ducts. The influence of winding inlet flow rate, winding inlet oil temperature, loss distribution and different winding geometries on the flow distribution, pressure losses and hot spot temperature within the winding model were investigated. It was found that increasing inlet flow rate does not necessarily reduce the hot spot temperature. It was observed that resistive and eddy current losses do not affect oil flow distribution. Higher ratios of radial to axial cooling duct dimension cause higher hot spot temperatures. Higher radial cooling duct height aids in creating oil reverse flow in pass bottom cooling ducts.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>HST</td>
<td>Hot spot temperature (°C)</td>
</tr>
<tr>
<td>H_d</td>
<td>Radial duct height (mm)</td>
</tr>
<tr>
<td>W_d</td>
<td>Axial duct width (mm)</td>
</tr>
<tr>
<td>n_d</td>
<td>Number of discs in a pass</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>V_in</td>
<td>Pass inlet velocity (m/s)</td>
</tr>
<tr>
<td>D_h</td>
<td>Axial duct hydraulic diameter (m)</td>
</tr>
<tr>
<td>D_mw</td>
<td>Winding model depth (fixed to 0.1 m)</td>
</tr>
<tr>
<td>V</td>
<td>Kinematic viscosity (m²/s)</td>
</tr>
<tr>
<td>T_in</td>
<td>Winding inlet oil temperature (°C)</td>
</tr>
<tr>
<td>T_out</td>
<td>Winding outlet oil temperature (°C)</td>
</tr>
<tr>
<td>H</td>
<td>Hot spot factor</td>
</tr>
<tr>
<td>g</td>
<td>Temperature gradient between T_mw and T_mo</td>
</tr>
<tr>
<td>T_mw</td>
<td>Mean winding temperature (°C)</td>
</tr>
<tr>
<td>T_mo</td>
<td>Mean oil temperature (°C)</td>
</tr>
<tr>
<td>P_loss</td>
<td>DC winding losses (W/m²)</td>
</tr>
<tr>
<td>Q_eddy</td>
<td>A number to represent eddy losses profile</td>
</tr>
<tr>
<td>V_max</td>
<td>Duct maximum velocity (m/s)</td>
</tr>
<tr>
<td>V_av</td>
<td>Duct average velocity (m/s)</td>
</tr>
<tr>
<td>Gr</td>
<td>Grashof number</td>
</tr>
<tr>
<td>β</td>
<td>Volumetric thermal expansion coefficient (1/K)</td>
</tr>
<tr>
<td>HST_w</td>
<td>Hot spot temperature rise over T_in (K)</td>
</tr>
<tr>
<td>FR</td>
<td>Winding inlet flow rate in litres/minute</td>
</tr>
</tbody>
</table>

1. Introduction

Oil immersed power transformers are cooled by circulating the oil through their windings and hence transferring the heat generated within the windings to external cooling radiators. The oil is either circulated naturally in Oil Natural (ON) cooled transformers or forced to circulate with external pumps in Oil Forced and Directed (OD) cooled transformers. By forcing the oil to circulate, better cooling of a transformer winding is achieved on the account of more maintenance, risk of pump failure, and pumps associated running costs. The hot spot temperature (HST), defined as the hottest temperature within the winding, is influenced by oil flow distribution within the winding structure and by the distribution of electric losses in winding discs. The zig-zag nature of oil flow direction through OD cooled disc type transformer windings adds to the complexity of estimating oil flow distribution in winding cooling ducts. Network based models were used to estimate both flow and temperature distributions within winding structure [1-4]. However, with the emergence of Computational Fluid Dynamics (CFD) commercial software and improved computational capabilities, calibrations of network model parameters were performed using CFD simulations to provide more accurate flow and temperature predictions [4, 5]. Dimensional analysis theory was applied to simplify the problem and to identify the dimensionless parameters that influence oil flow distribution within the winding passes [6, 7]. Experimental based studies were conducted to verify simulation based models [2-4, 8, 9]. Measurements of oil flow rate within lab based winding models were performed using hot wire anemometry [8], laser doppler velocimetry [3, 9], and by tracking the movement of seeding particles [2, 7, 10]. In addition, fibre optic temperature sensors were imbedded in real transformers to provide more accurate measurements of the HST and to verify simulation based models [11].

Winding geometries such as radial duct height (H_d), axial duct width (W_d), and number of discs in a pass between two diverting washers (n_d) influence oil flow distribution within the winding structure. Higher H_d causes more distorted oil flow distribution but lower pressure losses over winding structure [2, 12]. Lower n_d causes more uniform flow distribution and more oil to flow in each radial cooling duct [2, 12, 13]. As n_d increases, the pass bottom cooling duct suffers and oil flow may even reverse its direction in the duct [2]. Using dimensional analysis in OD cooled transformers, the pass inlet Reynolds number (Re) and the ratio H_d/W_d were identified as influential dimensionless parameters which affect oil flow distribution and pressure drop over winding passes [6]. Experimental validation under isothermal conditions were provided in [7] and it was shown that oil flow proportion in a pass is matched if both Re and H_d/W_d are matched irrespective of how these dimensionless parameters were composed of. Higher Re causes more
distorted flow distribution and increases pressure losses. Lower $H$/$W$ causes more uniform flow distribution.

The hot spot factor ($H$), as defined in the standard [14], is used to derive the transformer HST. It was suggested in the standard [14] that $H$ can be represented as $H = Q \times S$ where $Q$ is affected by loss distribution within the winding and $S$ is affected by the cooling structure of the transformer winding. It was shown in [15, 16] that both the $Q$ and the $S$ factors are dependent and cannot be decoupled.

This paper aims to experimentally study the influence of winding operating conditions and geometries on the oil flow distribution, pressure drop over transformer winding structure and the HST, in disc type transformer windings under OD cooling modes. The transformer operating conditions, such as $V_{in}$, $T_{in}$, resistive winding losses ($P_{res}$), and eddy loss distribution ($Q_{eddy}$) are investigated. $V_{in}$ lumps the effect of installed pumps and the hydraulic losses in the transformer cooling loop [17]. $T_{in}$ is affected by ambient temperature, loading level, capacity of radiators bank, and is higher toward winding top passes. $P_{res}$ depends on the transformer loading level. In addition, the winding geometrical parameters $W_d$ and $H_d$ are investigated. To study the influence of each parameter, an experimental setup was designed to allow measuring temperature, flow distribution in radial cooling ducts, and pressure losses over a disc type winding model under a wide range of testing conditions. The experimental setup and its measurement capabilities are presented in section 2. Section 3 presents results of recorded temperatures, oil flow distributions, and pressure losses. Discussions and conclusions are presented in sections 4 and 5 respectively.

2. Experimental Setup and Measurements

The experimental setup was designed to have versatility in conducting tests under a wide range of OD testing conditions and various winding geometries. The setup, as shown in Fig. 1, consists of a disc type winding model, a radiator, a pump, a flow meter, and a heating unit. The winding model was made using polycarbonate material and was designed to host up to 55 disc-segments. The disc segments can be arranged to fit in different discs per pass arrangements. Disc segments are modelled as aluminium blocks with 100 mm × 104 mm × 10 mm dimensions, referred to as plates, [2, 3, 8]. Each plate was heated using two cylindrical heaters embedded into the plate. Tests can be conducted under $H_d$ of either 4 mm or 6 mm. $W_d$ can be changed to 8 mm, 10 mm, and 12 mm. Table 1 summarizes winding model geometries that can be implemented. Case 2 was chosen as the base case for investigating the effect of operational condition on oil flow distribution and hot spot temperature. Mineral oil was used as the transformer liquid in the tests.

<table>
<thead>
<tr>
<th>Case</th>
<th>$H_d$</th>
<th>$W_d$</th>
<th>Number of passes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>4</td>
<td>8</td>
<td>3</td>
</tr>
<tr>
<td>Case 2</td>
<td>4</td>
<td>10</td>
<td>3</td>
</tr>
<tr>
<td>Case 3</td>
<td>4</td>
<td>12</td>
<td>3</td>
</tr>
<tr>
<td>Case 4</td>
<td>6</td>
<td>10</td>
<td>3</td>
</tr>
</tbody>
</table>

Temperature was measured using K-type thermocouples. Two thermocouples were embedded in each plate near both entrance and exit of radial cooling ducts and their average is considered representative of plate temperature. Calibrations of thermocouples were performed and ± 1 °C accuracy was achieved. $T_{in}$ was measured using the average recordings of 3 thermocouples fixed at the entrance of winding model inlet washer and $T_{out}$ was measured using the average of 8 thermocouples fixed at the exit of the upper pass outlet washer as shown on Fig. 1. The flow meter used is a positive displacement flow meter with accuracy 0.5% of reading. $V_{in}$ was controlled using a throttling valve. Pressure drop over the winding model was measured using a differential pressure instrument with accuracy ± 20 Pa. Pressure measurement ports were fixed to the middle of the first pass and to the outlet of the third pass as indicated in Fig. 1 using red arrows. Oil flow velocities in radial cooling ducts were measured using Particle Image Velocimetry (PIV). The PIV system consists of a dual head laser source, a camera, a synchronizer, and a software. The PIV works, as exemplified in Fig. 2, via tracking the movement of seeding particles doped into the flow using two consecutive pulses and two corresponding images. By knowing the time difference between the two pulses and the spatial displacement of the seeding particles in captured images, the duct velocity profile can be measured. In radial cooling ducts, the velocity profile is a laminar profile and can be characterized by its maximum velocity ($V_{max}$) which is related to duct average velocity ($V_{avg}$) using $V_{avg} = V_{max}/1.5$; hence, the radial ducts $V_{max}$ are extracted from processed images [10]. Flow rates were measured only in radial cooling ducts of the top pass of the winding model. As demonstrated on Fig. 1, top pass radial ducts were numbered from duct 1 to duct 11 from bottom to top of the pass and top pass plates were numbered from plate 1 to plate 10 from bottom to top of the pass. Application of the PIV system to record oil flow rates in cooling ducts was discussed in detail in [10]. Measurements of flow rates using the present setup were compared to CFD simulations and the deviations between each other are within 10% [7].
Experimental Study of Transformer Liquid Flow and Temperature Distribution

3. Results

Three categories of tests were conducted. The first category, presented in subsection 3.1, is to study the influence of $V_{\text{in}}$, on flow distribution, the HST, and pressure losses over the winding model. The second category, presented in subsection 3.2, is to investigate the effect of loading level on oil flow distribution and the HST. Loading level would affect $P_{\text{loss}}$, $Q_{\text{loss}}$, and $T_{\text{in}}$, hence, three sets of tests were conducted to individually study the effect of $P_{\text{loss}}$, $Q_{\text{loss}}$, and $T_{\text{in}}$ on oil flow distribution and the HST. The third category was conducted to compare oil flow distribution and the HST under different winding model geometries specified in Table 1 as presented in subsection 3.3.

3.1 Effects of $V_{\text{in}}$ on oil flow distributions and HST

Tests were conducted under uniform loss distributions of 50 W/plate equivalent to $P_{\text{loss}} = 2010$ W/m². $T_{\text{in}}$ was fixed to near 60 °C. Tests were conducted under winding model geometries of case 2 in Table 1. Fig. 3 shows measured temperatures and oil flow distributions in the third pass of the winding model under $V_{\text{in}} = 0.1$ m/s, 0.2 m/s, 0.3 m/s, and 0.4 m/s. Higher $V_{\text{in}}$ causes more distorted flow distributions but the HST is only slightly reduced with further increase in $V_{\text{in}}$. The trend of the HST and H factor with $V_{\text{in}}$ are captured and shown in Fig. 4. The H factor is calculated using the definition in the standard [14]. The HST is related to H by (1) where $HST_{\text{rise}}$ is the HST rise over $T_{\text{in}}$. $T_{\text{out, rise}}$ is the outlet oil temperature ($T_{\text{out}}$) rise over $T_{\text{in}}$, and $g$ is the temperature gradient between the mean winding temperature ($T_{\text{mw}}$) and the mean oil temperature ($T_{\text{mo}}$).

$$HST_{\text{rise}} = T_{\text{out, rise}} + g \times H$$  \hspace{1cm} (1)

It can be observed that a significant reduction occurs in the HST when $V_{\text{in}}$ increases from 0.05 m/s to 0.1 m/s then the HST fluctuates within 2 °C. A slight increase in the HST occurs when reverse flow starts to occur in duct 1 under

0.267 m/s. Likewise; a sharp increase in H occurs when $V_{\text{in}}$ is increased from 0.2 m/s to 0.267 m/s but after that H remains almost constant. The occurrence of oil stagnation or oil reverse flow in cooling ducts near plate 1 reduce the cooling efficiency even with higher $V_{\text{in}}$, as reflected of the fluctuating temperature of plate 1, and so the reduction in the HST with higher $V_{\text{in}}$ is not linear. The HST is affected by local cooling factors that are influenced by oil flow distribution in winding pass. It is observed that H, for fixed winding geometries, is relatively higher for more distorted temperature profiles which result from higher $V_{\text{in}}$. The same behaviour of the HST with $V_{\text{in}}$ was observed under different geometrical cases presented in Table 1 (i.e. increasing $V_{\text{in}}$ does not significantly reduce the HST).

Fig. 2 Typical PIV system with an acquired experimental example of the parabolic velocity profile in a radial cooling duct

Fig. 3. Oil flow and temperature distributions under increased $V_{\text{in}}$ with $P_{\text{loss}} = 2010$ W/m², $T_{\text{in}} = 60$ °C, and $Q = 1.0$.

Fig. 4. The trend of HST and H with increased $V_{\text{in}}$ under fixed winding model geometries of case 2.
3.2 Influence of loading conditions on oil flow distributions and HST

Three set of tests were conducted to study the effect of $P_{\text{loss}}$, $Q_{\text{eddy}}$, and $T_{\text{in}}$ on the HST and oil flow distribution in the winding model. All tests were conducted under winding model geometries of case 2 shown in Table 1.

3.2.1 $P_{\text{loss}}$ effects on oil flow distributions and HST

Tests were conducted under $V_{\text{in}}$ fixed at 0.3 m/s and $T_{\text{in}}$ fixed at 60 °C. The base $P_{\text{loss}}$ was 50 W/plate, equivalent to 2010 W/m$^2$, which was considered as a rated loading level of 1 p.u. Tests were conducted under $P_{\text{loss}}$ ranging from 0.4 p.u to 1.4 p.u. Results of temperature and flow distributions are shown in Fig. 5. It was observed that the flow distribution was not affected by $P_{\text{loss}}$, if both $T_{\text{in}}$ and $V_{\text{in}}$ are fixed which lead to fixed Reynold number ($Re$) as defined in (2) where $D_h$ is the axial duct hydraulic diameter in (m) and $\nu$ is oil kinematic viscosity in (m$^2$/s) calculated using oil properties at $T_{\text{in}}$. To facilitate this conclusion further, the ratio $Gr/Re^2$ is calculated for each test in which $Gr$ is the Grashof number given in (3) where $\beta_{th}$ is the volumetric thermal expansion coefficient.

As reported in [19], the ratio $Gr/Re^2$ gives an indication as of which cooling mode is dominating. Under all tests, $Gr/Re^2 << 1$ which facilitates that under OD modes and within the range of tested conditions, resistive losses do not influence the flow distribution and the effect of buoyancy force is negligible. Higher $P_{\text{loss}}$ causes higher HST; the HST increases almost linearly with the increase in $P_{\text{loss}}$.

The H factor is calculated from tests data and results are shown in Table 2. The H factor was slightly reduced with the increase in loading level despite the significant increase in the HST.

3.2.2 $Q_{\text{eddy}}$ effects on oil flow distributions and HST

Tests were conducted under $V_{\text{in}}$ fixed at 0.3 m/s, $T_{\text{in}}$ fixed at 60 °C. Tests were conducted under eddy loss distribution profiles, $Q_{\text{eddy}}$, as defined in Table 3. $Q_{\text{eddy}}$ is defined as the maximum losses in the winding model, losses in plate 10, over resistive base losses, $P_{\text{loss}}$. Extra losses were applied only in the top 5 plates of the winding model, plate 6 to plate 10. Losses in the rest of the plates were assumed the resistive base losses of 2010 W/m$^2$.

![Fig. 5. Oil flow distribution and temperature distributions under increased $P_{\text{loss}}$, $V_{\text{in}} = 0.3$ m/s, $T_{\text{in}} = 60\,\degree\mathrm{C}$, and $Q = 1.0$.](image1)

![Fig. 6. Oil flow and temperature distributions under increased $Q$ with $V_{\text{in}} = 0.3$ m/s, $T_{\text{in}} = 60\,\degree\mathrm{C}$, and $P_{\text{loss}} = 2010$ W/m$^2$.](image2)

<table>
<thead>
<tr>
<th>$Q_{\text{eddy}}$ (W/m$^2$)</th>
<th>Plate 1-5</th>
<th>Plate 6</th>
<th>Plate 7</th>
<th>Plate 8</th>
<th>Plate 9</th>
<th>Plate 10</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>2010</td>
<td>2010</td>
<td>2010</td>
<td>2010</td>
<td>2010</td>
<td>2010</td>
</tr>
<tr>
<td>1.3</td>
<td>2010</td>
<td>2010</td>
<td>2010</td>
<td>2211</td>
<td>2412</td>
<td>2613</td>
</tr>
<tr>
<td>1.5</td>
<td>2010</td>
<td>2211</td>
<td>2412</td>
<td>2613</td>
<td>2814</td>
<td>3015</td>
</tr>
<tr>
<td>1.7</td>
<td>2010</td>
<td>2211</td>
<td>2613</td>
<td>2814</td>
<td>3015</td>
<td>3417</td>
</tr>
</tbody>
</table>
and so it was not affected by $Q_{\text{eddy}}$. In case of a more uniform temperature profile resulting from lower operating $V_{\text{in}}$ the HST location might shift to plate 10 depending on the flow distribution. Hence, higher $Q_{\text{eddy}}$ may cause higher HST as shown in Fig. 9. The temperatures of overheated plates were higher for higher $Q_{\text{eddy}}$.

3.2.3 $T_{\text{in}}$ effects on oil flow distributions and HST

Oil flow distribution within a pass is influenced by the pass $T_{\text{in}}$ as it affects oil viscosity and so the Re [6, 7]. Fig. 8 represents tests results conducted under $V_{\text{in}} = 0.3$ m/s and uniform $P_{\text{loss}} = 2010$ W/m². Higher $T_{\text{in}}$ causes less oil to flow into duct 1 and then oil stagnation occurs at $T_{\text{in}} = 55^\circ$C. Oil reverse flow in duct 1 causes further increase in the HST rise in plate 1 due to the hotter oil flowing back from duct 2 to duct 1.

3.3 Effect of winding geometries on oil flow distributions and HST

Two geometrical parameters, $W_d$ and $H_d$, were changed to investigate their influence on both the HST and oil flow distributions. Tests were conducted under $T_{\text{in}} = 70^\circ$C and uniform loss distribution $P_{\text{loss}} = 2010$ W/m² in each plate. All tests were conducted under three values of winding inlet flow rate ($\text{FR}_{\text{in}}$) of 6 litres/minute (lpm), 12 lpm, and 18 lpm but only results under $\text{FR}_{\text{in}} = 18$ lpm are shown as they are representative of the overall tests. Measured HST rise from all conducted tests are presented in Table 4. Tests were conducted under cases of geometrical conditions as summarized in Table 1.

Table 4 Summary of HST rise for tests conducted under different geometrical cases and winding inlet flow rates. All temperatures are in K. Description of cases are given in Table 1.

<table>
<thead>
<tr>
<th>Inlet flow rate* (lpm)</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
<th>Case 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>19.7</td>
<td>17.5</td>
<td>18.9</td>
<td>23.5</td>
</tr>
<tr>
<td>12</td>
<td>15.8</td>
<td>14.3</td>
<td>15.5</td>
<td>25.2</td>
</tr>
<tr>
<td>18</td>
<td>15.9</td>
<td>14.2</td>
<td>17.7</td>
<td></td>
</tr>
</tbody>
</table>

* Inlet flow rate can be converted to $V_{\text{in}}$ using \((\text{Inlet flow rate} = V_{\text{in}} \times D_{\text{th}} \times \pi \times W_d \times 6000)\) where $W_d$ is substituted in [m] and $D_{\text{th}}$ is the winding model depth

3.3.1 $W_d$ effect on the oil flow distributions and HST

Tests were conducted under three values of $W_d$ of 8 mm, 10 mm, and 12 mm. Temperatures and oil flow distributions are shown in Fig. 9. Higher $W_d$ slightly causes more uniform flow distribution. However, the improvement in the flow distribution is not reflected on the HST. Measured HST are within 2°C and so the influence of $W_d$ on the HST can be considered minor. According to [6, 7], bigger ratio of $H_d/W_d$ would cause less uniform oil flow distribution. In the conducted tests, the ratio $H_d/W_d$ varies from 0.5, 0.4, to 0.3. More investigations are needed to extend the observation to a wider range of $W_d$.
3.3.2 \( H_d \) effect on oil flow distributions and HST

Tests were conducted under two values of \( H_d \) of 4 mm and 6 mm. Fig. 10 shows both the temperature and oil flow distributions. \( H_d \) significantly affected both temperature and oil flow distributions. Under \( H_d = 6 \) mm, lower oil flow rates were caused to flow into pass bottom cooling ducts; reverse oil flow occurred in duct 1 and oil stagnation occurred in duct 2. Lower flow rates in pass bottom cooling ducts caused a significant increase in the HST. Smaller \( H_d \) would create smaller \( H_d/W_d \) ratio and so causes more uniform oil flow distribution in the pass while larger \( H_d \) would create larger \( H_d/W_d \) ratio which would cause a more distorted oil flow distribution and more likely oil reverse flow to occur [6, 7].

![Fig. 10: Oil flow and temperature distributions under varied \( H_d \) with \( V_{in} = 0.3 \) m/s, \( Q = 1.0 \), \( P_{loss} = 2010 \) W/m\(^2\), and \( T_{in} = 70 \)°C. Flow distribution data under \( H_d \) were taken from [7].]

### 3.3.4 Pressure drop over winding model under different geometries

Winding model geometries influence pressure losses over the winding structure and determine oil flow split in hydraulically parallel transformer windings. The pressure losses over the winding model were measured under geometrical cases defined in Table 1 and results are shown in Fig. 11. All tests were conducted under isothermal conditions with oil temperature fixed at 70 °C. Higher \( H_d \) and \( W_d \) reduce the pressure losses over the winding model.

![Fig. 11: Pressure loss over winding model under different geometrical cases]

### 4. Discussions

The HST is affected by both loss distributions within the winding and by oil flow distributions into radial cooling ducts. In OD cooled transformers, the two factors can be decoupled as the existence of losses does not affect oil flow distribution. The flow distribution is governed by dimensionless parameters [6, 7] and matching the dimensionless parameters would lead to a matched flow distribution in winding passes. However, the distortion level of oil flow distribution may not reflect the reduction or increase in the HST. The HST is affected by the cooling efficiency of the disc hosting it. Increasing \( V_{in} \) may lead to oil reverse flow to occur and in these circumstances the benefit of having higher \( V_{in} \) diminishes. Higher \( V_{in} \) would cause higher pressure losses over the transformer winding and so more mechanical strain on washer sealing. Higher \( V_{in} \) is achieved with large pumps capacities which may require more maintenance and higher initial costs.

Power transformers are usually operated as two transformers in parallel with 0.5 p.u loading level, each under normal operating conditions. Under contingency conditions, transformers may have to be overloaded to levels higher than their rated conditions. At the same time, switching on more pumps does not increase cooling efficiency as discussed earlier.

Winding geometries directly influence the oil flow distribution and so the HST. The axial cooling duct width \( W_a \) within the tested range, had a minor influence on the HST. However, Higher \( H_d \) and higher \( H_d \) caused less pressure drop over the transformer winding. Higher radial cooling duct height \( H_d \) increased the HST and caused oil stagnation and oil reverse flow to occur in pass bottom cooling ducts. As demonstrated in [6, 7], smaller ratio of \( H_d/W_d \) would lead to more uniform oil flow distribution while higher ratios would subsequently lead to oil reverse flow occurring in the pass bottom cooling duct.

Geometrical deviations, resulted during winding model assembly in which a plate or a washer protrudes by 1 to 2 mm toward axial cooling duct, affected the temperature profile and oil flow distributions [20]. If the plate protrudes into the axial cooling duct, the radial duct above the plate suffers and hence plate temperatures near the deviation increases. Geometrical deviations give a footprint to the temperature profile. It is important to control the geometrical deviations in order to compare the influence of different winding model geometries on both oil flow and temperature distributions.

### 5. Conclusion

Experimental studies on the thermal performance of OD cooled transformers were conducted using an experimental setup which was designed to allow the measurement of oil flow and temperature distributions in a disc type winding model. The setup allowed testing under a range of parameters such as inlet oil velocity, transformer loading level, and winding model geometries. It was concluded that higher inlet oil velocity may not enhance the transformer thermal performance and it may cause oil reverse flow to occur. The hot spot temperature increased almost linearly with the increase in the transformer loading level. It was observed that under OD cooled transformers,
the flow distribution is not affected by either resistive losses or eddy current losses. In the tested range, the axial cooling duct width slightly influenced the hot spot temperature while higher radial duct height significantly increased the hot spot temperature.

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References


Chapter 7 Oil Flow and Temperature Distributions under ON Cooling Mode

In this work, experimental study is conducted to document the influence of transformer operating conditions and winding geometries on the HST and oil flow distribution under ON cooling modes in disc type transformers. The results are written in a paper format.

Experimental Study of the Hot Spot Temperature in Oil Natural Cooled Power Transformers

Muhammad Daghrorah, Zhongdong Wang, Member, IEEE, and Qiang Liu, Member, IEEE
Experimental Study of the Hot Spot Temperature in Oil Natural Cooled Power Transformers

Muhammad Daghrarah, Zhongdong Wang, Member, IEEE, and Qiang Liu, Member, IEEE

Abstract—Ageing rate of paper insulations in disc type power transformer is accelerated when the transformer is operated above its rated temperature. In this paper, an experimental study is carried out on a disc type winding model to study the factors affecting oil flow and temperature distributions under oil natural cooling modes. Particle image velocimetry is used to measure oil flow rates in cooling ducts. The effect of winding model operating conditions and geometries are studied to document their influence on oil flow and temperature distributions. More distorted flow distribution which leads to oil stagnation and hence higher hot spot temperature were observed under higher loading levels or lower winding inlet oil velocities. Inlet oil temperature and non-uniform losses in the winding model showed minor impact on oil flow distribution. In terms of geometric parameters, higher radial cooling duct or higher number of discs per pass make oil stagnation easier to occur which could significantly increase the hot spot temperature.

Index Terms—Experiment, PIV, Power Transformer, Hot Spot Temperature, Flow Distribution, ON Cooling

I. INTRODUCTION

THERMAL modelling of a power transformer aims to pinpoint the location and the expected value of the Hot Spot Temperature (HST) defined as the hottest temperature in the transformer winding. Transformer operating conditions and its winding geometries affect the HST. Transformer operational conditions such as average winding inlet velocity ($V_{in}$), loading level of a transformer ($P_{loss}$), winding inlet oil temperature ($T_{in}$), and eddy loss distribution ($Q_{eddy}$) are factors to be considered to influence oil flow and temperature distributions [4-7]. Higher $V_{in}$ causes a more distorted oil flow distribution within the winding pass under Oil Forced and Directed (OD) cooling modes [3, 6]. In terms of geometrical parameters of disc type windings, the radial cooling duct height ($H_{d}$), the axial cooling duct width ($W_{d}$), and the number of discs between two adjacent blocking washers, a pass, ($n_{d}$) are usually considered [1-2]. It was reported that lower values of the ratio of $H_{d}/W_{d}$ cause more uniform flow distribution within the winding under OD cooling modes [1, 3].

Thermal Hydraulic Network Models (NHTMs) were used in the literature to study transformer thermal behaviour [8-10]. It was reported that flow distribution within different passes is almost identical under OD cooling conditions [8]. Computational Fluid Dynamics (CFD) modelling is used to simulate the transformer thermal problem [3, 11]. Though CFD simulations require large computational resources, they provide more detailed and potentially more accurate results than NHTM based models [11, 12]. It was shown that under uniform losses both NHTM models and CFD simulations predicted the HST accurately whereas larger discrepancies existed under non-uniform loss distributions [11]. CFD simulations capture more detailed phenomena such as the occurrence of oil reverse flow in cooling ducts [3]. The hot streak phenomenon was documented using CFD simulation, which influences the temperature profile within a transformer under ON cooling conditions [12, 13]. Using 3D CFD simulations, it was concluded that flow is not an axisymmetrical flow in the horizontal cooling ducts under ON cooling regime and that large discrepancies are observed for the estimated HST under 3D and 2D models [14].

Different techniques were used to record oil flow rates

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The authors are with the School of Electrical and Electronic Engineering at The University of Manchester, Manchester, M13 9PL, UK (e-mail: zhongdong.wang@manchester.ac.uk).
within cooling ducts. Hot Wire Anemometry (HWA) was used to record oil flow rates within a constructed experimental winding model under OD isothermal conditions [16]. Laser Doppler Velocimetry (LDV) was used to record the developed inlet oil flow rate under ON cooling conditions [17]. LDV was also used to record oil flow rates within winding cooling ducts under ON conditions [18]. Tracking the movements of tracer particles was performed and the occurrence of oil reverse flow in pass bottom cooling ducts was documented under OD isothermal conditions [1]. Recently, oil flow rates were measured under nonisothermal OD cooling conditions by tracking the displacement of seeding particles illuminated by strong LED light source and CCD camera. [19].

For studying ON cooling modes, a spontaneous flow can be achieved using a complete hydraulic loop [17]. As shown in [17], \( V_{in} \) develops in time to a parabolic laminar profile. However due to the practical constrains, it can be acceptable in both simulations [12, 14] and experiments [22] to study ON cooling mode by controlling \( V_{in} \). Under ON cooling modes in real transformers, the flow is driven by the thermosyphon force and no external pumps are used. However, a pump is used in the conducted tests to initial the flow. The inlet velocity is set to desired values using a throttling valve. Two reasons are necessitating the usage of the pump. First, the hydraulic pressure in the used experimental setup is large such that no net flow can be created without the external pump. Second, the section of interest to study in the experimental setup is the winding model, with both oil flow and temperature distribution within it, and the rest of the experimental setup is not of interest. Hence, an ON regime is created by setting appropriately the boundary conditions for the winding model regardless of the rest of the experimental setup. Using a pump to simulate experimentally an ON regime gives two main advantages. First, it speeds up the time to reach steady state. Second, it gives flexibility to study ON regime under wider range of operating conditions.

The Richardson number (\( R_i \)) defined as the ratio of Grashof number (Gr) over Reynolds number square (Re\(^2\)), i.e. \( Gr/Re^2 \), indicates whether the flow is ON dominated or OD dominated [14]. Re and Gr are calculated using (1) and (2) [3, 14]. If \( R_i \gg 1 \), then the cooling is considered ON dominated and if \( R_i \ll 1 \), then the cooling is considered OD dominated [14].

\[
Re = \frac{V_{in} \times D_h}{v}
\]

\[
Gr = \frac{9.81 \times \beta_{in} \times D_h^3}{v^2} \times (T_{m} - T_{mo})
\]

where \( D_h \) is the axial duct hydraulic diameter, \( v \) is the oil kinematic viscosity, and \( \beta_{in} \) is the oil volumetric thermal expansion coefficient.

In this paper, an experimental setup is used to study oil flow and temperature distributions using a disc type winding model under ON cooling modes. A PIV system is used to record oil flow rates within radial cooling ducts. Section II gives an overview of the experimental setup. Section III presents test results performed under various transformer operating conditions and winding model geometries. Sections IV and V present discussions and conclusions, respectively.

II. EXPERIMENTAL DESCRIPTIONS

A. Experimental setup

Fig. 1 shows the experimental setup. The setup consists of a hydraulic loop involving a pump, a flow meter, a radiator, an external heating unit, and a disc type winding model. The winding model is made of a polycarbonate material which allows the required optical access to cooling ducts for PIV measurements. The winding model can accommodate up to 55 disc-segments, henceforth plates, which can be arranged in several \( n_d \) as desired. Both inlet side and outlet side axial ducts \( W_d \) are made equal and can be changed to 8 mm, 10 mm, and 12 mm. \( H_d \) can be changed to 4 mm or to 6 mm. Six different geometrical cases, as summarized in Table I, were implemented. Plates have the dimensions of 100 mm \( \times \) 104 mm \( \times \) 10 mm and are made of aluminium blocks. Radial duct length along the flow is 104 mm and its actual depth is 94 mm. Two thermocouples are imbedded 10 mm from both ends of each plate to record the inlet side and outlet side temperature profiles of the winding model. \( T_{in} \) is measured using the average of three thermocouples mounted at the inlet of the winding model lower pass. Winding model outlet oil temperature (\( T_{out} \)) is measured using the average of eight thermocouples mounted along the winding model top pass outlet washer.

<table>
<thead>
<tr>
<th>Case</th>
<th>I</th>
<th>II</th>
<th>III</th>
<th>IV</th>
<th>V</th>
<th>VI</th>
</tr>
</thead>
<tbody>
<tr>
<td>( W_d ) (mm)</td>
<td>10</td>
<td>8</td>
<td>12</td>
<td>10</td>
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<td>( H_d ) (mm)</td>
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<td>4</td>
<td>4</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>( n_d )</td>
<td>10</td>
<td>10</td>
<td>10</td>
<td>15</td>
<td>6</td>
<td>10</td>
</tr>
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<td>3</td>
<td>2</td>
<td>5</td>
<td>3</td>
</tr>
<tr>
<td>Computer to record Temperatures</td>
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<td></td>
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<td>Thermocouple Multi-Channel Meter</td>
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<td></td>
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</tr>
<tr>
<td>Expansion Vessel</td>
<td></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>Main Tank</td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Heat Unit</td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
</tbody>
</table>

Fig. 1. Schematic view of the experimental setup

All thermocouples were calibrated in a controlled temperature water bath and an accuracy of \( \pm 1 \) °C was achieved. Accuracy of the used inlet flow meter is 0.5% of reading. \( V_{in} \) is controlled using a valve while the pump drives
the flow. An external heating unit is used to control $T_{in}$ by heating the oil external to the winding model. Each plate is heated using two resistive cartridge heaters inserted inside each plate. A schematic diagram for the winding model is shown in Fig. 2. Plates are numbered from plate 1 to plate $n_d$ from bottom to top of the top pass of the winding model and radial cooling ducts of the top pass are numbered from duct 1 to duct $n_d + 1$ from bottom to top of the pass. Transformer mineral oil is used with thermal properties listed in [3]. The experimental setup was described in more detail in [20].

B. Oil flow measurement using a PIV system

A PIV system is used to record oil flow rates within radial cooling ducts. A typical PIV system consists of a double head laser source, laser sheet optics, a camera, a synchronizing unit, and computer software. The basic operation of a PIV system is given as follows: firstly, the laser sends a laser pulse which is made using laser sheet optics from a beam to a laser sheet; Secondly, the camera, which is synchronized with the laser source through the synchronizing unit, captures an image called Frame A of the flow with seeding particles where seeding particles reflect part of the laser sheet toward the camera; Thirdly, after a known time difference $\Delta t$, the laser fires a second pulse while the camera captures a second image called Frame B; Finally, By tracking the movements of seeding particles in space between Frame A and Frame B, the velocity field is calculated using the known time difference between the two frames. The maximum duct velocity ($V_{max}$) is extracted from the parabolic laminar velocity profile obtained from the PIV analysis. Only flow rates within the top pass radial cooling ducts are measured. The use of PIV system to measure oil flow rates is discussed in more detail in [21].

III. RESULTS

Tests are split into two main categories. The first involves the study of the influence of transformer operational conditions such as $T_{in}$, $V_{in}$, $P_{loss}$, and $Q_{eddy}$ on oil flow and temperature distributions. The second category involves the investigation of the influence of winding model geometries, such as $W_{de}$, $H_{de}$, and $n_d$, on oil flow and temperature distributions. Results are discussed as follows.

A. Influence of winding operational conditions on oil flow and temperature distributions

Table II summarizes testing conditions for the four operational varied parameters. The range for $V_{in}$ and $P_{loss}$ were selected such that they are within a typical range of power transformer operational conditions under ON cooling modes [2, 23].

![Fig. 2. Geometries of the experimental winding model with PIV measurement arrangement.](image)

![Fig. 3. Oil flow and temperature distributions with $V_{in} = 0.021 \text{ m/s}$, $P_{loss} = 1200 \text{ W/m}^2$ and a range of $T_{in}$.](image)
Experimental Study of Transformer Liquid Flow and Temperature Distribution

Eddy losses are only simulated in the top 5 plates of the top pass and the remaining plates are heated with the resistive-base loss of 1200 W/m². All tests are conducted under fixed winding model geometries of Case II shown in Table I.

1) Effect of $T_{in}$ on oil flow and temperature distributions

At steady state, $T_{in}$ depends on the radiator heat dissipation, heat dissipation through the winding model, and the ambient temperature. $V_{in}$ and $P_{loss}$ are fixed to 0.021 m/s and 1200 W/m², respectively. Four values of $T_{in}$ are tested under uniform losses $Q_{eddy} = 1.0$. Measured oil flow and temperature distributions at steady state are shown in Fig. 3. A summary of test parameters and the $R_i$ number are given in Table IV. The $R_i$ number is much larger than 1 indicating an ON cooling mode. Higher velocities are observed in lower cooling ducts and hence higher temperatures are observed at top pass plates. The temperature of plate 1 is higher than that of plate 2 most likely because hot streaks get sucked into duct 1 reducing the effective heat transfer from plate 1 compared to plate 2 as noticed by [12]. It was found that $T_{in}$ has minor influence on oil flow distributions and the HST rise.

2) Effect of $V_{in}$ on oil flow and temperature distributions

$V_{in}$ was set initially to 0.017 m/s with $T_{in} = 33$ °C, $P_{loss} = 1200$ W/m², and $Q = 1.0$. Test results are shown in Fig. 4 and summarized in Table V. It is observed that higher $V_{in}$ induces a more uniform oil flow and temperature profiles. Oil stagnation was observed under $V_{in} = 0.017$ m/s in duct 10. The increase in $V_{in}$ from 0.021 m/s to 0.025 m/s significantly reduces the HST. The $R_i$ number increases with the reduction of $V_{in}$ indicating the development of a more ON dominating cooling mode.

<table>
<thead>
<tr>
<th>$V_{in}$ [m/s]</th>
<th>$T_{in}$ [°C]</th>
<th>$T_{max}$ [°C]</th>
<th>$v$ [mm/s²]</th>
<th>$HST_{rise}$ [K]</th>
<th>$Re$</th>
<th>$Gr$</th>
<th>$R_i$</th>
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<tr>
<td>0.017</td>
<td>48</td>
<td>72</td>
<td>9.2</td>
<td>60</td>
<td>33</td>
<td>12917</td>
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<tr>
<td>0.021</td>
<td>45</td>
<td>68</td>
<td>9.2</td>
<td>52</td>
<td>41</td>
<td>12126</td>
<td>7</td>
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<tr>
<td>0.025</td>
<td>44</td>
<td>62</td>
<td>9.2</td>
<td>32</td>
<td>49</td>
<td>9490</td>
<td>4</td>
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<tr>
<td>0.033</td>
<td>43</td>
<td>58</td>
<td>9.2</td>
<td>23</td>
<td>65</td>
<td>7645</td>
<td>2</td>
</tr>
</tbody>
</table>

Fig. 4. Oil flow and temperature distributions under varied $V_{in}$. Tests were conducted under $P_{loss} = 1200$ W/m², $T_{in} = 33$ °C, and $Q = 1.0$.

3) Effect of $P_{loss}$ on oil flow and temperature distributions

$V_{in}$ was fixed to 0.025 m/s and $Q = 1.0$. Tests were conducted under different $P_{loss}$. Results are shown in Fig. 5 and Table VI. Higher $P_{loss}$ causes a more distorted oil flow and temperature distributions. As an example, looking at results for $P_{loss} = 2010$ W/m² and for ambient temperature $= 23$ °C and $T_{in} = 32$ °C as mentioned in Table II, $T_{out}$ rise over ambient, can be calculated to be 41 K, is well within the 60 K rise limits specified for ON cooling mode while the HST is above the 78 K rise limit specified in the IEC 60076-2:2011 standard [24]. Hence, results indicate that it might not be reliable if only relying on $T_{out}$ rise as an indication of the HST rise under varying loading conditions. Higher $P_{loss}$ makes $R_i$ number higher. From conducted tests, it is observed that higher $R_i$ indicates a more distorted oil flow and temperature distributions.

<table>
<thead>
<tr>
<th>$P_{loss}$ [W/m²]</th>
<th>$T_{in}$ [°C]</th>
<th>$T_{max}$ [°C]</th>
<th>$v$ [mm/s²]</th>
<th>$HST_{rise}$ [K]</th>
<th>$Re$</th>
<th>$Gr$</th>
<th>$R_i$</th>
</tr>
</thead>
<tbody>
<tr>
<td>800</td>
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<td>9.2</td>
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<td>49</td>
<td>6854</td>
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<td>9.2</td>
<td>62</td>
<td>49</td>
<td>7118</td>
<td>3</td>
</tr>
<tr>
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<td>9.2</td>
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<td>11599</td>
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<tr>
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<td>71</td>
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<td>49</td>
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<tr>
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<td>48</td>
<td>82</td>
<td>9.2</td>
<td>108</td>
<td>49</td>
<td>18190</td>
<td>8</td>
</tr>
</tbody>
</table>

Fig. 5. Oil flow and temperature distributions under varied $P_{loss}$. Tests were conducted under $V_{in} = 0.025$ m/s, $T_{in} = 32$ °C, and $Q = 1.0$.

4) Effect of $Q_{eddy}$ on oil flow and temperature distributions

$V_{in}$ was fixed at 0.021 m/s and $T_{in}$ was fixed at 55 °C. Oil flow and temperature distributions are shown in Fig. 6. Extra losses did not influence the oil flow distribution; nonetheless, eddy losses directly increased the temperature of overheated plates and so increased the HST.
Experimental Study of Transformer Liquid Flow and Temperature Distribution

B. Influence of winding geometries on oil flow and temperature distributions

In this section, three sets of tests are conducted under varied $W_d$, $H_d$, and $n_d$ as defined in Table I. Tests are conducted under fixed $P_{loss} = 1200 \text{ W/m}^2$ and $Q = 1.0$. Results are discussed as follows. For tests conducted under different $W_d$, pass inlet flow rate ($FR_{in}$) is specified instead of $V_{in}$. $FR_{in}$ can be related to $V_{in}$ using $FR_{in} = V_{in} \times W_d \times 6000$.

1) Effect of $W_d$ on oil flow and temperature distributions

Tests were conducted under $FR_{in} = 1.0 \text{ litres/minute (lpm)}$, $FR_{in} = 1.25 \text{ lpm}$, and $FR_{in} = 1.5 \text{ lpm}$. For each $W_d$, $V_{in}$ can be calculated from $FR_{in}$ using (3) where $D_{th}$ is the winding model depth which is fixed to 0.1 m.

$$FR_{in} = V_{in} \times D_{th} \times W_d \times 60000 \quad (3)$$

Tests were conducted under three values of $W_d$ of 8 mm, 10 mm, and 12 mm. Results under $FR_{in} = 1.0 \text{ lpm}$ and $FR_{in} = 1.25 \text{ lpm}$ relay the same observations and so only results under $FR_{in} = 1.0 \text{ lpm}$ are shown in Fig. 7. In Fig. 7, the oil flow and temperature distributions are quite similar for different $W_d$. On the other hand, results under $FR_{in} = 1.5 \text{ lpm}$ are shown in Fig. 8 which indicate that oil flow distributions are comparable but the HST under $W_d = 10 \text{ mm}$ is lower. As mentioned in [11], hot streak dynamics influence the temperature distribution and so they may be the cause of the temperature differences under $FR_{in} = 1.5 \text{ lpm}$. As a conclusion, $W_d$ in the investigated range showed a minor influence on oil flow and temperature distributions.

2) Effect of $H_d$ on temperature and oil flow distributions

Three sets of tests were conducted under two radial duct heights $H_d = 4 \text{ mm}$ and $H_d = 6 \text{ mm}$. Results of tests under three values of $V_{in}$ of 0.017 m/s, 0.021 m/s, and 0.025 m/s are shown in Fig. 9, Fig. 10, and Fig. 11, respectively. Overall $H_d = 6 \text{ mm}$ results in easier occurrence of oil reverse flow compared to $H_d = 4 \text{ mm}$ and HST is usually located on the plate near the ducts with oil stagnation. It is worth noting that oil flow rates in ducts 10 and 11 are almost identical for both $H_d$ tested under 0.017 m/s shown in Fig. 9 but the temperature of plate 10 under $H_d = 6 \text{ mm}$ is almost 15 °C lower than that under $H_d = 4 \text{ mm}$. This is an indirect indicator that the hot streak dynamics have a big impact on the temperature distribution within the transformer winding under ON cooling mode. On the contrary, HST differences for tests conducted under $V_{in} = 0.025 \text{ m/s}$ are negligible despite the clear differences in the shape of both the oil flow and temperature distributions. As a conclusion, $H_d$ has a major influence on oil flow and temperature distributions and it influences the hot streak dynamics within the winding structure.
Experimental Study of Transformer Liquid Flow and Temperature Distribution

Fig. 9. Oil flow and temperature distributions under different $H_d$ with $V_{in} = 0.017$ m/s, $P_{loss} = 1200$ W/m$^2$, $T_{in} = 60^\circ$C and $Q = 1.0$.

Fig. 10. Oil flow and temperature distributions under different $H_d$ with $V_{in} = 0.021$ m/s, $P_{loss} = 1200$ W/m$^2$, $T_{in} = 60^\circ$C and $Q = 1.0$.

Fig. 11. Oil flow and temperature distributions under different $H_d$ with $V_{in} = 0.025$ m/s, $P_{loss} = 1200$ W/m$^2$, $T_{in} = 60^\circ$C and $Q = 1.0$.

3) Effect of $n_d$ on temperature and oil flow distributions

Tests are conducted under several arrangements of plates/pass $n_d$ of 6, 10, and 15. To aid the presentations of results, the plates in the winding model are reassigned with references as plate 1 to plate 30 from bottom to top of the winding model. Tests are conducted under $V_{in} = 0.017$ m/s, $V_{in} = 0.021$ m/s, and $V_{in} = 0.025$ m/s of which the results are shown in Fig. 12, Fig. 13, and Fig. 14, respectively. Under $n_d = 6$, temperature distributions are almost uniform; but this comes with the cost of expected increase in pressure losses in the winding [1]. Therefore, $n_d = 6$ is unlikely a practical case for ON transformers.

Fig. 12. Oil flow and temperature distributions under different $n_d$ with $V_{in} = 0.017$ m/s, $P_{loss} = 1200$ W/m$^2$, $T_{in} = 60^\circ$C and $Q = 1.0$.

Fig. 13. Oil flow and temperature distributions under different $n_d$ with $V_{in} = 0.021$ m/s, $P_{loss} = 1200$ W/m$^2$, $T_{in} = 60^\circ$C and $Q = 1.0$. 

Fig. 14. Oil flow and temperature distributions under different $n_d$ with $V_{in} = 0.025$ m/s, $P_{loss} = 1200$ W/m$^2$, $T_{in} = 60^\circ$C and $Q = 1.0$. 

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The HSTs under \( n = 10 \) and \( n = 15 \) are comparable under \( V = 0.017 \) m/s due to the existence of obvious oil reverse flow and oil stagnation in the ducts near the plate hosting the HST. With the increase of \( V \) to 0.021 m/s, reverse flow is removed under \( n = 10 \) but it is sustained under \( n = 15 \) and so the HST under \( n = 10 \) is lower than the HST under \( n = 15 \). With further increase of \( V \) to 0.025 m/s, oil stagnation in duct 10 is removed under \( n = 10 \) but the oil reverse flow and stagnation are sustained under \( n = 15 \). Hence, a large difference in the HST exists between \( n = 10 \) and \( n = 15 \) when testing under \( V = 0.025 \) m/s. The reduction in the HST with increased \( V \) is minimal under \( n = 15 \) due to the sustained oil stagnation near the location of the HST, i.e., ducts 10 to 13. The HST was reduced under \( n = 10 \) when \( V \) was increased from 0.017 m/s to 0.021 m/s due to the removal of oil reverse flow despite that oil stagnation persisted top of the pass. It can be concluded that the HST increases when oil reverse flow occurs in pass top cooling ducts.

IV. DISCUSSION

ON temperature profile is developed when the buoyancy force is the dominant driving force for the flow within the winding. This situation can be created under combinations of both \( V \) and \( P_{loss} \). Lower \( V \) and higher \( P_{loss} \) result in a more distorted ON profile and higher HST, as reflected by higher \( R \) values. It was observed that increasing of \( V \) helps improve the flow distribution in order to avoid the occurrence of oil stagnation which in turn helps reduce the HST. However, if oil stagnation is sustained in the top winding pass, the increase of \( V \) may not reduce the HST. It became more apparent that hot streak dynamics are a major influential factor on temperature profile within the transformer winding under ON cooling mode. Laser Induced Fluorescence (LIF) may be applicable and can be used in future investigations to track the hot streak dynamics by measuring oil temperature distribution in cooling ducts. The location and value of the HST depend on transformer winding geometries, transformer loading levels, and the hot streak dynamics. To conduct comparative studies under ON cooling mode, the effect of parameter under question on \( V \) must be considered. For example, oil properties, especially viscosity, affects pressure losses within the winding structure and so they influence \( V \). If a transformer is retro-filled with a higher viscosity oil, the reduction of \( V \) should be estimated beforehand in order to compare the performance of the two oils. As such is the effect of different winding model geometries under ON cooling conditions.

V. CONCLUSION

An experimental study was carried out using a lab scale disc type winding model to study ON cooling regimes. It was observed that lower \( V \) or higher \( P_{loss} \) causes oil stagnation to occur in top pass cooling ducts and hence results in higher HST. Eddy losses only influenced temperatures in overheated plate but not the oil flow distribution. \( T \) has minor influence on both oil flow distributions and the HST rise over \( T \). In the investigated range of winding geometries, \( H \) has more prominent effect on flow distribution than \( C \). Higher \( H \) make the occurrence of reserve flow and oil stagnation easier. Finally, higher \( n \) caused sustained oil stagnation in the pass top cooling ducts and hence reduced the effectiveness of higher \( V \) in reducing the HST. Hot streak dynamics play an important role on determining the HST within the transformer winding under ON cooling mode.

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REFERENCES

Experimental Study of Transformer Liquid Flow and Temperature Distribution


Chapter 8 Comparisons of Thermal Performance of Alternative Transformer Oils

In this chapter, an experimental study is carried out to document the influence of emerging alternative transformer insulating liquids, such as gas to liquid oils and ester based oils, on the hot spot temperature and oil flow distribution in disc type winding transformers. Results are documented in a paper format.

*Experimental Study of the Influence of Different Oils on the Transformer Cooling Performance*

*Muhammad Daghrarah, Zhongdong Wang, Member, IEEE, and Qiang Liu, Member, IEEE*
Experimental Study of Transformer Liquid Flow and Temperature Distribution

Muhammad Daghrah, Zhongdong Wang, Member, IEEE, and Qiang Liu, Member, IEEE, Andree Hilker, Attila Gyore, Member, IEEE

Abstract—Mineral oil is traditionally used in liquid immersed transformers to act as a coolant, an information carrier, and as an electrical insulator. Emerging alternative transformer liquids provide advantages, such as improved fire safety and better biodegradability, of which transformer operators would like to utilize. In this paper, an experimental study is conducted to compare the thermal performance of a mineral oil, a gas-to-liquid oil, and a synthetic ester liquid as coolants in a zig-zag disc type winding model. Comparisons are made under liquid directed cooling modes and under liquid natural cooling modes. It was found that under both cooling modes, the mineral oil and the gas-to-liquid oil behaved almost with comparable liquid flow and temperature distributions. Under liquid directed cooling modes, the synthetic ester gave more uniform flow distribution and delayed the occurrence of liquid reverse flow compared to the other oils. Under liquid natural cooling modes and using the zig-zag disc type winding model, synthetic ester gave more distorted liquid flow and temperature distributions due to its higher viscosity which causes lower inlet flow rate to develop under the specific tested retrofilling conditions.

Index Terms—Transformer, Experiment, Particle Image Velocimetry, Mineral Oil, Gas-To-Liquid Oil, Synthetic Ester

I. INTRODUCTION

Transformer life expectancy is affected by the ageing of paper insulation within the winding, which is determined by its experienced temperature. The hot spot temperature (HST) is defined as the hottest temperature in the transformer winding structure which causes the severest paper ageing. In disc type power transformers, liquid is circulated through cooling ducts within the transformer winding to transport heat from within the winding to the external cooling medium. The thermal and physical properties of the liquid, such as density ($\sigma$), kinematic viscosity ($\nu$), and specific heat ($C_p$), influence its cooling performance [1, 2]. Cooling performance is characterized by first the heat transfer coefficient, which is a local cooling factor, and second by liquid flow distribution within the winding, which is a global cooling factor [3].

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Muhammad Daghrah, Zhongdong Wang and Qiang Liu are with the School of Electrical and Electronic Engineering at The University of Manchester, Manchester, M13 9PL, UK (e-mail: zhongdong.wang@manchester.ac.uk).
Andree Hilker is with Shell Global Solutions (Deutschland) – GmbH
Attila Gyore is with M&I Materials, Manchester, M32 0ZD, UK

NOMENCLATURE

$\sigma$ Density (kg/m$^3$)
$\nu$ Kinematic viscosity (m$^2$/s)
$C_p$ Specific heat (J/kg.K)
$Re$ Pass inlet Reynold number $Re = D_h \times V_{in}/\nu$
$H_d$ Winding model radial cooling duct height (m)
$W_d$ Winding model axial cooling duct width (m)
$k_v$ Thermal conductivity of liquid (W/m.K)
$\beta_v$ Volumetric thermal expansion coefficient of liquid (1/K)
$V_{in}$ Winding model inlet velocity (m/s)
$P_\text{loss}$ Pressure loss over winding model (Pa)
$T_b$ Winding model bottom liquid temperature (°C)
$P_{\text{lossiston}}$ Power loss injection in each plate (W)
$V_{\text{max}}$ Maximum velocity in radial cooling duct (m/s)
$V_{av}$ Average velocity in radial cooling duct (m/s)
$D_h$ Hydraulic diameter of axial duct (m)
$A_{sc}$ Inlet cross sectional area of axial cooling duct (m$^2$)
$W_p$ Wet perimeter of axial cooling duct (m)

HST$_{rise}$ Hot spot temperature rise over $T_b$ (K)
$T_{\text{cap}}$ Winding model top liquid temperature (°C)
$ho_{\text{TS}}$ Thermosyphon driving force (Pa)
$g$ Acceleration of gravity (m/s$^2$)
$\Delta H$ Height difference between centres of radiator and winding (m)
$\Delta T$ Average liquid temperature rise between $T_b$ and $T_{\text{cap}}$ (K)

$P_{\text{loss, total}}$ Total losses in the winding model (W)
$I_i$ Winding model inlet mass flow rate (kg/s)
$L_i$ Friction coefficient of component i in the hydraulic loop
$L_e$ Equivalent pipe hydraulic length of component i (m)
$C$ Constant
$R_i$ Richardson number ($R_i = Gr/Re^2$)
$Gr$ Grashof number
$T_w$ Average winding temperature (°C)
$T_m$ Average liquid temperature (°C)

Under Liquid Directed (OD or KD) cooling modes, the flow distribution is affected by pass inlet Reynold number ($Re$), and by the winding geometries such as the radial cooling duct height ($H_d$), the axial cooling duct width ($W_d$), and the number of discs per pass [4, 5]. Under Liquid Natural (ON or KN) cooling modes, the developed liquid flow rate is affected by the thermosyphon force within the winding structure and the total pressure loss within the transformer hydraulic loop [6-8].

Mineral oil is traditionally used as the insulating and cooling media in power transformers. Recently alternative transformer liquids, such as ester based liquids and gas-to-liquid oils, emerged as alternatives in power transformers compared to mineral oils. Ester based liquids are biodegradable and have high flash and fire points. A comparison was made for the thermal performance of a 50
MVA, 141 kV/13 kV transformer that was filled in with mineral oil and natural ester [9]. It was reported that under KNAN cooling mode an increase in the HST by 12 °C was observed in the low voltage winding when filled with natural ester compared to when filled with mineral oil [9]. Hence, it was concluded that retrofitting transformers designed for mineral oil with natural ester requires careful considerations [9]. The impact of mineral oils and ester based liquids on transformer winding temperatures was investigated [10]. It was observed that when using ester the flow velocities within radial cooling ducts were lower, compared to when using mineral oil, which was the reported reason for the increase in the winding temperatures [10]. A 15 MVA, single phase, 154 kV/22.9 kV transformer was filled with mineral oil and then with natural ester [11]. Temperature rise tests were conducted and an increase of near 16 °C in recorded HST was observed in the low voltage winding after retrofitting using natural ester [11].

In this paper, an experimental study is carried out to compare the thermal performance between different transformer liquids in terms of liquid flow and temperature distributions in a zig-zag disc type winding model under directed and natural cooling modes. Used experimental setup is introduced in section II. Comparisons of the thermal performances of tested liquids under OD/KD cooling modes are presented in section III. Comparisons of the thermal performances of tested liquids under ON/KN cooling modes are presented in section IV. Discussion and conclusions are given in sections V and VI, respectively.

II. EXPERIMENTAL DESCRIPTION

A. Experimental setup

Used experimental setup, as shown in Fig. 1, consists of a disc type winding model, a radiator, a pump and a flow meter. The pump is used to force the liquid to circulate through the system. The winding inlet velocity (V_{in}) is measured using a positive displacement flow meter with 0.5% accuracy of reading. A throttling valve is used to set and control V_{in} as desired. The pressure loss over the winding model (P_{z}) is measured under directed cooling mode using a differential pressure instrument with ± 10 Pa accuracy. The liquid is heated within the winding model and circulated through the radiator to be cooled down. To control the winding bottom liquid temperature (T_{b}), an external controlled heater was connected in series with the winding model. The winding model consists of 3 passes, where each pass consists of 10 aluminium plates modelling disc segments [5, 12-14]. Each plate was made with dimensions 100 mm x 104 mm x 10 mm, and can be heated using two resistive cylindrical heaters embedded within the plate. Power losses (P_{loss}) in plates are controlled. Thermocouples were used to record average plate temperatures within the winding model with accuracy ± 1 °C. Each plate was embedded with two thermocouples and their average is used as a representative average plate temperature. The winding model was made of polycarbonate material and was milled to create H_{d} = 4 mm and W_{d} = 10 mm. Particle Image Velocimetry (PIV) system was used to measure liquid flow rates within radial cooling ducts of the third pass in the winding model. The PIV system consists of dual head laser source, laser optics, a camera, a synchronizer, and a software to process captured images [15]. The PIV system records velocities by tracking the movement of seeding particles within the flow. The laser source fires two consecutive pulses with known time difference and the camera captures correspondingly two raw images. Using the time difference between the two laser pulses and the seeding particles displacement in the captured pair of images, liquid velocities within cooling ducts are calculated using the software.

Liquid flow profile within a radial cooling duct is a laminar profile and so it can be characterized by the maximum velocity (V_{max}) of its parabolic shape. The average duct velocity (V_{av}) can be related to V_{max} using V_{av} = V_{max}/1.5. From processed PIV images, V_{max} is extracted instead of V_{av} as V_{max} deemed to be more accurately measured. The experimental setup and the PIV system were presented in more detail in [15, 16].

![Fig. 1.Sketch of experimental setup](image-url)

Fig. 1. Sketch of experimental setup

B. Liquids under investigations

Three different liquids were used, Nytro Gemini X as a mineral oil [17], a gas-to-liquid Shell Diala S4 ZX-I oil [18], and a synthetic ester MIDEL 7131 [19]. Table I summarizes the values of key thermal parameters of the liquids at three different operating temperatures. The system is filled with liquid using a liquid tank connected to the experimental setup. Air can leave the experimental setup through both bleeds above the winding model and the radiator. Before using different liquid, a strict procedure was used to drain the old liquid from the system before testing the next liquid.
### III. COMPARISON UNDER DIRECTED COOLING MODES

Under OD/KD cooling modes, liquid is forced to circulate within the winding using a pump. Washers within the winding structure force the liquid to flow in a zig-zag fashion. According to [4, 20], liquid flow distribution is influenced by dimensionless numbers such as the \( Re \) number and geometrical based dimensionless numbers \( \alpha, \beta, \) and \( \gamma \) [4]. Since the \( Re \) is defined as \( Re = (D_h \times V_{in})/\nu \), only \( \nu \) and \( V_{in} \) affect the \( Re \) for fixed winding geometry. Higher \( \nu \) causes lower \( Re \) resulting in more uniform liquid flow distribution within the winding radial cooling ducts [4, 20]. \( \nu \) is a temperature dependent parameter, which is also related to liquid type.

#### A. Liquid flow and temperature distributions under different \( V_{in} \)

To investigate the influence of different liquids on liquid flow and temperature distributions, experiments were conducted under four different \( V_{in} \) of 0.1 m/s, 0.2 m/s, 0.27 m/s, and 0.3 m/s. Uniform loss distribution in each plate with \( P_{loss} = 50 \text{ W/plate} \), which is equivalent to \( 2010 \text{ W/m}^2 \), was applied. \( T_h \) was fixed to 70 °C. Fig. 2 shows liquid flow and temperature distributions in the third pass within the winding model at \( V_{in} \) of 0.1 m/s. Liquid flow distributions of Gemini X and Diala S4 ZX-I are almost identical whereas MIDE 7131 caused slightly more uniform flow distribution. The calculated \( Re \) of MIDE 7131 is almost half that of the other oils as shown in Table II. Liquid flow distributions for tested liquids are relatively uniform which lead to uniform temperature distributions. Temperature profiles of tested liquids are almost comparable.

Fig. 3 represents liquid flow and temperature distributions at \( V_{in} \) of 0.27 m/s. At this relatively high inlet liquid velocity, liquid reverse flow starts to occur in duct 1 for both Gemini X and Diala S4 ZX-I. The flow distribution using Diala S4 ZX-I is slightly more uniform compared to Gemini X due to its slightly lower \( Re \) number. Due to its higher \( \nu \), MIDE 7131 causes more uniform flow distribution and so offers more resistance to liquid reverse flow phenomenon. Compared to the previous test under \( V_{in} \) of 0.1 m/s, liquid flow distributions are more distorted which directly cause more distorted temperature profiles. The occurrence of liquid reverse flow causes higher \( HST_{rise} \) for both Gemini X and Diala S4 ZX-I compared to MIDE 7131.

#### Table I: Thermal properties of used liquids, units in nomenclature

<table>
<thead>
<tr>
<th>Liquid properties at operating temperatures (°C)</th>
<th>( \sigma )</th>
<th>( \nu )</th>
<th>( k_c )</th>
<th>( C_p )</th>
<th>( \beta_{th} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mineral oil Gemini X</td>
<td>40</td>
<td>858</td>
<td>9.2</td>
<td>0.132</td>
<td>1794</td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>845</td>
<td>5.2</td>
<td>0.130</td>
<td>1877</td>
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<td></td>
<td>80</td>
<td>832</td>
<td>3.4</td>
<td>0.128</td>
<td>1956</td>
</tr>
<tr>
<td>Gas-to-liquid Diala S4 ZX-I</td>
<td>40</td>
<td>793</td>
<td>9.6</td>
<td>0.142</td>
<td>2271</td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>780</td>
<td>5.6</td>
<td>0.140</td>
<td>2365</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>767</td>
<td>3.7</td>
<td>0.138</td>
<td>2458</td>
</tr>
<tr>
<td>Synthetic ester MIDE 7131</td>
<td>40</td>
<td>956</td>
<td>28</td>
<td>0.143</td>
<td>1933</td>
</tr>
<tr>
<td></td>
<td>60</td>
<td>941</td>
<td>14</td>
<td>0.141</td>
<td>1994</td>
</tr>
<tr>
<td></td>
<td>80</td>
<td>926</td>
<td>8</td>
<td>0.139</td>
<td>2023</td>
</tr>
</tbody>
</table>

#### Table II: Tests summary under directed cooling mode with \( T_h = 70 \text{ °C} \)

<table>
<thead>
<tr>
<th>( V_{in} ) (m/s)</th>
<th>Liquid Type</th>
<th>( T_{top} ) [K]</th>
<th>( Re )</th>
<th>( HST_{rise} ) [K]</th>
<th>Location</th>
<th>Reverse flow</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>Gemini X</td>
<td>78.6</td>
<td>443</td>
<td>17.5</td>
<td>Plate 4</td>
<td>No</td>
</tr>
<tr>
<td></td>
<td>Diala S4 ZX-I</td>
<td>78.1</td>
<td>387</td>
<td>17.9</td>
<td>Plate 4</td>
<td>No</td>
</tr>
<tr>
<td></td>
<td>MIDE 7131</td>
<td>76.7</td>
<td>173</td>
<td>18.6</td>
<td>Plate 3</td>
<td>No</td>
</tr>
<tr>
<td>0.2</td>
<td>Gemini X</td>
<td>74.7</td>
<td>887</td>
<td>14.3</td>
<td>Plate 1</td>
<td>No</td>
</tr>
<tr>
<td></td>
<td>Diala S4 ZX-I</td>
<td>74.6</td>
<td>774</td>
<td>15.3</td>
<td>Plate 1</td>
<td>No</td>
</tr>
<tr>
<td></td>
<td>MIDE 7131</td>
<td>71.8</td>
<td>346</td>
<td>13.4</td>
<td>Plate 1</td>
<td>No</td>
</tr>
<tr>
<td>0.27</td>
<td>Gemini X</td>
<td>73.7</td>
<td>1183</td>
<td>17.6</td>
<td>Plate 1</td>
<td>Yes</td>
</tr>
<tr>
<td></td>
<td>Diala S4 ZX-I</td>
<td>73.8</td>
<td>1032</td>
<td>17.2</td>
<td>Plate 1</td>
<td>Yes</td>
</tr>
<tr>
<td></td>
<td>MIDE 7131</td>
<td>74.1</td>
<td>461</td>
<td>12.4</td>
<td>Plate 1</td>
<td>No</td>
</tr>
<tr>
<td>0.3</td>
<td>Gemini X</td>
<td>73.8</td>
<td>1330</td>
<td>15.9</td>
<td>Plate 1</td>
<td>Yes</td>
</tr>
<tr>
<td></td>
<td>Diala S4 ZX-I</td>
<td>73.5</td>
<td>1161</td>
<td>16.6</td>
<td>Plate 1</td>
<td>Yes</td>
</tr>
<tr>
<td></td>
<td>MIDE 7131</td>
<td>73.4</td>
<td>520</td>
<td>12.3</td>
<td>Plate 1</td>
<td>No</td>
</tr>
</tbody>
</table>

*\( Re \) was calculated using \( \nu \) at \( T_h = 70 \text{ °C} \) and \( D_h = 0.018 \text{ m} \).
B. Pressure loss under isothermal conditions with uniform liquid temperature = 70 °C

At a fixed winding geometry and $V_{in}$, higher viscosity liquid would lead to higher pressure losses in the winding. Under isothermal testing conditions with uniform liquid temperature of 70 °C, $P_L$ was measured under a range of $V_{in}$ from 0.05 m/s to 0.35 m/s for the three different liquids as shown in Fig. 4. Gemini X and Diala S4 ZX-I cause considerable pressure loss over the winding model. MIDEL 7131 causes higher pressure loss over the winding model compared to the other two liquids.

To summarize, three main observations are being made. First, with higher $V_{in}$, liquid flow distribution is more distorted for all liquids and this directly results in more distorted temperature profiles. Higher distortion reflects that more liquid flows in top radial cooling ducts compared to bottom cooling ducts which cause higher temperatures in bottom plates. Second, Gemini X and Diala S4 ZX-I showed similar liquid flow and temperature distributions whereas MIDEL 7131 caused less distorted liquid flow and temperature distributions. Hence, MIDEL 7131 offered higher resistance to the occurrence of liquid reverse flow. Third, assuming a retrofitting scenario with the same pump and so fixed pressure loss over the winding of 300 Pa as shown in Fig. 4, MIDEL 7131 would cause lower $V_{in}$ close to 0.22 m/s compared to the two other liquids which would cause $V_{in}$ close to 0.27 m/s. Lower $V_{in}$, and higher $v$, indicate lower $Re$. Lower $Re$ leads to more uniform flow distributions within the winding passes [4, 20]. With lower $Re$, liquid reverse flow is less likely to occur under MIDEL 7131. In case of liquid reverse flow not occurring for the three liquids under both $V_{in}$ of 0.1 m/s and 0.2 m/s, the $HST$ was comparable between the liquids. On the other hand and under $V_{in} = 0.27$ m/s, liquid reverse flow occurred under Gemini X and Diala S4 ZX-I and it did not occur under MIDEL 7131 and so the $HST$ was lower when using MIDEL 7131.

$p_{Th}$ is equal to the total pressure loss in the hydraulic loop [6]. According to [6], $p_{Th}$ can be calculated using (1). $\Delta T$ can be related to the total electric losses in the winding ($P_{loss, total}$) and to the inlet mass flow rate ($\dot{m}$) as presented in (2). $\dot{m}$ is related to $V_{in}$ using $\sigma$ and $A_c$ as shown in (3). $P_L$ can be calculated by adding all pressure drops around the hydraulic loop. Darcy – Weisback equation shown in (4) is usually used to calculate pressure drop within individual sections of the winding model and transformer loop [6, 21, 22].

Assuming that at a specific steady state operating point, the transformer hydraulic loop can be modelled as an equivalent pipe, with an equivalent length $L_{eq}$ and an equivalent hydraulic diameter $D_{h,eq}$ such that the flow is laminar, a reasonable assumption for ON/KN cooling modes, with average liquid flow velocity equals to $V_{in}$ and pressure drop over the pipe equals to $P_L$. In this case, the equivalent friction coefficient $f_{eq}$ can be presented by (5). To relate $V_{in}$ to liquid only dependent properties, first, (3) is substituted in (2) and the resulting (2) is substituted in (1). Then, (5) is substituted in (4). By equating resulting (1) and (4), i.e. $p_{Th} = P_L$, (6) is produced. The first term in (6) is related to the transformer geometries and to the total power losses whereas the second term is related to only liquid properties. In the current experimental setup, (6) can be used approximately to answer the question as what would be the new $V_{in}$ if using a new liquid in the winding model under the same winding model power losses and geometrical conditions compared to the old liquid case to reflect a retrofitting scenario.

$$p_{Th} = g \sigma_o \beta_{Th} \Delta T \Delta H$$

(1)

where $\sigma_o$ is the liquid density at a reference temperature, $\beta_{Th}$ is the liquid thermal expansion coefficient, $g$ is the acceleration of gravity, $\Delta H$ is the height difference between radiator and winding centres, and $\Delta T$ is the average liquid temperature rise from bottom to top of the winding

$$\Delta T = \frac{P_{loss, total}}{\dot{m} \times C_p}$$

(2)

$$\dot{m} = \sigma \times A_c \times V_{in}$$

(3)

$$P_{L,i} = f_i \times \frac{L_i}{D_{h,i}} \times \frac{\sigma \times V_{in}^2}{2}$$

(4)

where $f_i$ is the friction coefficient, $L_i$ is the equivalent duct length, and the $i$ subscript refers to the $i$ component in the hydraulic loop

$$f_{eq} = \frac{C}{Re} = \frac{C \times v}{D_{h,eq} \times V_{in}}$$

(5)

where $C$ is a constant which depends on the flow regime and the pipe shape

$$V_{in}^2 = \left( \frac{2g \times D_{h,eq}^2 \times \Delta H \times P_{loss, total}}{C \times A_c \times L_{eq}} \right) \times \left( \frac{\beta_{Th}}{v \times C_p \times \sigma} \right)$$

(6)

In tests conducted henceforth, $P_{loss}$ is fixed to 30 W/plate which is equivalent to 1200 W/m². $T_B$ is maintained at 60 °C. $V_{in}$ is still controlled by external pump and valve. To check whether the testing conditions can represent natural cooling
regime, Richardson number \( (R_i) \) is calculated for all conducted tests. \( R_i \) is presented in (8) and defined as the ratio of Grashof number \( (G_r) \), shown in (7), to the Re number square. According to [23], if \( R_i < 1 \), then the cooling regime is considered OD/KD dominated regime. If \( R_i > 1 \), then the cooling regime is considered ON/KN dominated regime.

\[
G_r = \frac{g \rho T h D_h^3}{\nu^2} \times (T_{mw} - T_{mo}) \tag{7}
\]

\[
R_i = G_r / Re^2 \tag{8}
\]

where \( T_{mw} \) is the average winding temperature and \( T_{mo} \) is the average liquid temperature.

A. Liquid flow and temperature distributions under different \( V_{in} \)

Tests were conducted with three different \( V_{in} \) of 0.026 m/s, 0.021 m/s, and 0.017 m/s which are within the expected range of ON/KN transformers [7]. Table III represents key test results with the calculated \( R_i \) number for each test. It is observed that lower \( V_{in} \) causes more distorted liquid flow and temperature distributions, i.e. more dominated natural cooling regimes, which are reflected in higher \( R_i \) numbers. In general under ON/KN cooling modes, more liquid flows into pass bottom cooling ducts and less liquid flows into pass top cooling ducts [8]. Individual tests are discussed as follows.

Fig. 5 shows recorded liquid flow and temperature distributions at steady state for tests under \( V_{in} \) of 0.026 m/s. Liquid flow and temperature distributions follow almost an opposite trend to that under OD/KD cooling modes. Gemini X and Diala S4 ZX-I have almost the same liquid flow distributions whereas MIDEAL 7131 has relatively more flow distribution contributing to higher flow rates into pass top cooling ducts. Therefore, MIDEAL 7131 has lower \( HST_{rise} \) compared to the other liquids. Using MIDEAL 7131, it can be observed that the \( Re \) number is almost half of the other liquids and the \( R_i \) number is slightly lower.

**TABLE III**

<table>
<thead>
<tr>
<th>Liquid Type*</th>
<th>( T_{top} ) [°C]</th>
<th>( T_{mw} ) [°C]</th>
<th>( HST_{rise} ) [K]</th>
<th>( Re^* )</th>
<th>( G_r^* )</th>
<th>( R_i )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tests conducted under ( V_{in} = 0.026 ) m/s</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gemini X</td>
<td>80.4</td>
<td>80.3</td>
<td>36.5</td>
<td>177</td>
<td>28436</td>
<td>2.1</td>
</tr>
<tr>
<td>Diala S4 ZX-I</td>
<td>76.6</td>
<td>77.5</td>
<td>33.6</td>
<td>102</td>
<td>26435</td>
<td>2</td>
</tr>
<tr>
<td>MIDEAL 7131</td>
<td>77.3</td>
<td>78.3</td>
<td>25.8</td>
<td>46</td>
<td>3939</td>
<td>1.9</td>
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<tr>
<td>Tests conducted under ( V_{in} = 0.021 ) m/s</td>
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<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Gemini X</td>
<td>83.7</td>
<td>85</td>
<td>49.2</td>
<td>97.7</td>
<td>36968</td>
<td>3.9</td>
</tr>
<tr>
<td>Diala S4 ZX-I</td>
<td>80.6</td>
<td>82.9</td>
<td>47.5</td>
<td>85</td>
<td>26645</td>
<td>3.7</td>
</tr>
<tr>
<td>MIDEAL 7131</td>
<td>81.2</td>
<td>80.4</td>
<td>36.5</td>
<td>38.2</td>
<td>3919</td>
<td>2.7</td>
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<tr>
<td>Tests conducted under ( V_{in} = 0.017 ) m/s</td>
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<td></td>
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<td></td>
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<tr>
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<td>89.8</td>
<td>54</td>
<td>78.2</td>
<td>44584</td>
<td>7.3</td>
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<tr>
<td>Diala S4 ZX-I</td>
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<td>86.1</td>
<td>54</td>
<td>68.2</td>
<td>3259</td>
<td>7</td>
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<tr>
<td>MIDEAL 7131</td>
<td>83.5</td>
<td>85.2</td>
<td>52</td>
<td>30.5</td>
<td>5450</td>
<td>5.8</td>
</tr>
</tbody>
</table>

* \( Re \) and \( G_r \) were calculated using \( V \) at \( T_B = 70^\circ \)C and \( D_h = 0.018 \) m.

Results under \( V_{in} \) of 0.021 m/s are shown in Fig. 6. Liquid stagnation occurs in duct 10 under both Gemini X and Diala S4 ZX-I whereas it did not occur when testing under MIDEAL 7131. Liquid stagnation causes a large increase in temperature of nearby plate; in this case it is plate 9. Liquid flow distribution was almost identical when testing using Gemini X and Diala S4 ZX-I and it was more uniform when testing MIDEAL 7131. As a result of liquid stagnation not occurring under MIDEAL 7131, \( HST_{rise} \) was lower compared to the other liquids. In comparison to tests under \( V_{in} \) of 0.026 m/s, higher \( R_i \) numbers are observed which indicate more dominating ON/KN regimes.

Fig. 5. Liquid flow and temperature distributions under ON/KN cooling mode with \( V_{in} = 0.026 \) m/s and \( T_B = 60^\circ \)C.

Fig. 6. Liquid flow and temperature distributions under ON/KN cooling modes with \( V_{in} = 0.021 \) m/s and \( T_B = 60^\circ \)C.

B. Comparison of Gemini X and MIDEAL 713 under a specific retrofitting scenario

As demonstrated in the previous tests, Gemini X and Diala S4 ZX-I provided almost the same liquid flow distributions and hence their temperature profiles were very close to each other. Under same \( V_{in} \), MIDEAL 7131 caused relatively more uniform liquid flow distributions resulting in more liquid to flow in pass top radial cooling ducts. Nonetheless, when liquid stagnation occurred using MIDEAL 7131 under \( V_{in} = 0.017 \)
m/s, the $HST_{rise}$ was very close for all the liquids. The performances of different liquids were compared under fixed $V_{in}$. However, as demonstrated in (6) the liquid properties, especially $v$, influence the developed $V_{in}$ under ON/KN cooling modes.

It was reported in literature that when retrofitting a power transformer with a new liquid, in point of view of thermal aspects, careful considerations need to be made since the $HST$ may be increased [9-11]. The main reason for the increase in the $HST$ is the reduction of the developed $V_{in}$ when using a different liquid which has higher viscosity. Under the present experimental setup, $V_{in}$ is controlled instead of allowing the system to determine its value similar to what was conducted in [17]. For this reason, (6) can be used, though requires further verifications, to approximately answer the question as of what would be the new $V_{in}$ when using a new liquid under the same winding model geometries and operating conditions, presented in the first term in (6), so that different liquids performances are judged fairly. As an example, a specific case study is discussed in which MIDEL 7131 is compared to Gemini X in a hypothetical retrofitting scenario. According to (6) and the liquids information are shown in Table I at temperature 70 °C, Gemini X would have an inlet velocity ($V_{in, GX}$) of 1.69 times higher than that of MIDEL 7131 ($V_{in, MD}$), i.e. $V_{in, GX} = 1.69 \times V_{in, MD}$. It is very important to emphasize here that this relationship is only valid for the related testing conditions and the studied winding arrangements. Further experimental validations are required to justify the validity of (6) in a broader context.

Tests were conducted under uniform losses of $P_{loss} = 30$ W/plate or equivalently 1200 W/m$^2$. $T_B$ was maintained at 60 °C. The winding model geometries are relatively close to the specified low voltage winding of a 22kV, 250 MVA transformers mentioned in [24]. Table IV provides a summary of key test results in which $V_{in, GX} = 0.028$ m/s and $V_{in, MD} = 0.017$ m/s. Test results are presented in Fig. 8. A difference in the $HST_{rise}$ between Gemini X and MIDEL 7131 of 26 °C was recorded while only 6 °C difference in $T_{top}$ was recorded between the two liquids. Though this case may be the worst case possible of a bad retrofitting choice for ON/KN transformer, it is intended to raise the awareness that appropriate consideration and selection of good transformer candidates is important to fully benefit from the advantages provided by alternative liquids. The increase in the $HST$ under MIDEL 7131, in this studied case, is because liquid stagnation occurs in duct 10 and that liquid flow distribution under Gemini X is much more uniform compared to MIDEL 7131. This can show that the proper selection of the candidate for retrofitting of any transformers is really important.

**TABLE IV**

<table>
<thead>
<tr>
<th>Liquid Type*</th>
<th>$V_{in}$ [m/s]</th>
<th>$T_{top}$ [ºC]</th>
<th>$T_{max}$ [ºC]</th>
<th>$HST_{rise}$ [ºC]</th>
<th>Re*</th>
<th>$G_*$</th>
<th>$R_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gemini X</td>
<td>0.028</td>
<td>77</td>
<td>78</td>
<td>26</td>
<td>125</td>
<td>25544</td>
<td>1.6</td>
</tr>
<tr>
<td>MIDEL 7131</td>
<td>0.017</td>
<td>83.5</td>
<td>85.2</td>
<td>52</td>
<td>30.5</td>
<td>5450</td>
<td>5.8</td>
</tr>
</tbody>
</table>

*Re and $G_*$ were calculated using $v$ at $T_B = 70$ °C and $D_B = 0.018$ m

![Fig. 7. Liquid flow and temperature distributions under ON/KN cooling modes with $V_{in} = 0.017$ m/s and $T_B = 60$ °C.](image)

![Fig. 8. A comparison between the thermal performance of Gemini X and MIDEL 7131 under a specific retrofitting scenario in which according to (6), which required experimental validations, Gemini X develops 1.69 higher flow rate compared to MIDEL 7131.](image)

**V. DISCUSSIONS**

Under directed cooling modes, it was predicted that dimensionless numbers, such as the Re number and the ratio between the radial to the axial cooling ducts (a), influence liquid flow distribution in a pass [4]. The Re number is a function of $V_{in}$ and $v$. Higher $v$, or lower $V_{in}$, leads to a lower Re and hence a more uniform flow distribution. Ester based liquids have higher viscosities compared to mineral and gas-to-liquid oils and so they develop a more uniform flow...
distribution within the winding passes. Higher viscosity liquids provide more resistance to the occurrence of liquid reverse flow in pass bottom cooling ducts. Nonetheless, the ester liquid causes higher pressure loss within the winding and so for the same pump, lower $V_{in}$ develops and so even more uniform liquid flow distribution is produced.

Under natural cooling modes and for the zig-zag disc type winding model, results showed that the difference in the \( HST \) between different liquids is relatively small when liquid reverse flow, or liquid stagnation, occurs under all liquids in pass top cooling ducts. To compare the liquids’ thermal performances fairly, (6) was used to roughly estimate as of what would be the new $V_{in}$ when using a new liquid compared to the old one. The reported data of measured inlet and outlet liquid temperatures under similar transformer loading levels indicate that mineral oils have a near 1.6 higher \( \eta \) compared to natural ester [9, 11] which supports the estimated relationship between $V_{in, GX}$ and $V_{in, MD}$ using (6), though experimental validation of (6) is still required before further expansion of conclusions. Ester based liquid was compared to mineral oil, using (6), in a retrofitting scenario and it was observed that the differences in the $HST_{rise}$ and in $T_{top}$ between the two oils are consistent with reported measurements in the literature under actual retrofitting tests [11] for zig-zag disc type power transformers. Results facilitate that the difference in $T_{top}$ may not be a good indicator of the difference in the $HST_{rise}$. Ester based liquid may increase the $HST$ due to its higher viscosity which causes lower $V_{in}$ to develop. Still, in the industry the measurement of top liquid temperature could be carried out easier.

VI. CONCLUSION

Experimental investigation and comparison of the thermal performance of different transformer liquids were conducted under both OD/KD and ON/KN cooling modes. A disc type zig-zag winding model was used and inlet liquid velocity was controlled and set to desired values. Three liquids were compared which are mineral oil Nynas Gemini X, gas-to-liquid oil Shell Diala S4 ZX-I, and synthetic ester MIDEL 7131. Under both OD/KD and ON/KN cooling modes, Gemini X and Diala S4 ZX-I gave similar liquid flow distributions within the winding model and very close temperature profiles. Under OD/KD cooling modes, MIDEL 7131 showed more resistance to the liquid reverse flow phenomenon and gave more uniform liquid flow distribution. Nonetheless, MIDEL 7131 caused higher pressure losses within the winding model. Under ON/KN cooling modes and in a retrofitting perspective, MIDEL 7131 increased the $HST$ compared to mineral oil Gemini X under the conducted testing conditions and winding model geometries.

ACKNOWLEDGMENT

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REFERENCES


Chapter 9 Conclusions and Future Work

9.1 General Review

In disc type power transformers, the knowledge of the location and value of the hot spot temperature is required for better transformer thermal design and more accurate estimations of the transformer thermal ageing rate. In this PhD research, oil immersed disc type power transformers are focused on. The aim is to conduct experimental studies to enhance the understanding of the influence of contributing factors on oil flow and temperature distributions within the disc type winding model. The study was carried out to cover the following aspects.

Experimental test platform

- A disc type, zig-zag, winding model was designed and constructed from polycarbonate sheets to allow the application of PIV to record oil flow rates.
- Oil flow rates and temperature distribution were measured using a PIV system and K-type thermocouples, respectively.
- The experimental test platform allows conducting tests under different winding model geometries and operational conditions.

Tests under OD cooling conditions

- Experimental validations were provided for the application of dimensional analysis for disc type winding transformer thermal design.
- The influence of operating conditions, such as inlet oil velocity, electric loss distributions and inlet oil temperature, on oil flow, pressure loss, and temperature distributions were studied.
- The influence of axial cooling duct width and radial cooling duct height on oil flow, pressure loss, and temperature distributions, were studied.

Tests under ON cooling conditions

- The influence of operating conditions, such as inlet oil velocity, electric loss distributions and inlet oil temperature, on oil flow and temperature distributions were studied.
The influence of axial cooling duct width, radial cooling duct height, and number of discs per pass on oil flow and temperature distributions were studied.

**Comparison of alternative transformer liquids in zig-zag disc type transformers**

- The thermal performance of transformer mineral oil, Gas-to-Liquid oil, and synthetic ester oil were compared under both OD and ON cooling modes.
- Comparisons of oils were made at fixed inlet oil velocity under OD cooling modes and for retro-filling scenarios under ON cooling modes.

**9.2 Research Conclusions**

**9.2.1 Experimental test platform**

The experimental test platform was designed to allow the study of thermal performance of disc type transformer windings. The winding model provides the ability to test under a wide range of axial cooling duct widths, radial cooling duct height, and different number of discs per pass. It was facilitated that measurement of oil flow rates in radial cooling ducts using a PIV system, under both ON and OD conditions, is feasible and can be achieved with good accuracy. The existences of oil reverse flow and oil stagnation under both ON and OD testing conditions were successfully captured.

**9.2.2 Conclusions for tests under OD cooling conditions**

Experimental validation was conducted for the application of dimensional analysis to understand the dimensionless contributing parameter, i.e. Reynold number and the radial duct height to the axial duct width ratio, on oil flow proportion in each cooling duct and on pressure drop coefficient over the disc-type winding model. First, it was experimentally verified that by matching the dimensionless controlling parameters, the oil flow proportion in each radial cooling duct and the pressure drop coefficient over the winding model are matched irrespective of how the dimensionless controlling parameters are composed of. Second, it was experimentally verified that the matching is valid within a wider range of pass inlet Reynold number and for different winding model geometries.

Under OD non-isothermal testing conditions, operational and geometrical parameters were studied in a parametric sweep fashion and their effects on oil flow and temperature distributions were documented as follows.
Influence of operating conditions

It was observed that higher $V_{in}$ causes a more distorted oil flow distribution and subsequently causes oil stagnation to occur in pass bottom cooling duct and then oil reverse flow occurs with further increase in $V_{in}$. The hot spot temperature is not reduced in a significant manner with further increase in $V_{in}$. Oil flow distribution was not affected by the loading level, if the pass inlet Reynold number is fixed. Higher $P_{loss}$ caused a linear increase in the hot spot temperature. Eddy current losses do not affect oil flow distributions but they directly increase the temperature of overheated plates. Higher $T_{in}$ causes more distorted oil flow distribution which may lead to oil stagnation or oil reverse flow in pass bottom cooling duct. In the case of oil stagnation occurring, obvious increase of the hot spot temperature was usually observed.

Influence of cooling duct dimensions

Tests were conducted under $H_d = 4$ mm and $H_d = 6$ mm. It was observed that higher $H_d$ causes more distorted oil flow distributions and higher hot spot temperatures. Hence, $H_d$ is a major influential parameter on oil flow and temperature distributions. Tests were conducted under three values of $W_d$ of 8 mm, 10 mm, and 12 mm. Higher $W_d$ slightly causes a more uniform oil flow distribution. However, its influence over temperature distribution and on the hot spot temperature can be described as minor. In the tested range of cooling duct dimensions, higher ratios of $H_d/W_d$, from 0.3 to 0.6, caused more distorted oil flow distributions and increased the hot spot temperature, which agrees with previous CFD based simulations [39].

9.2.3 Conclusions for tests under ON cooling conditions

The pump and the external heating unit were used to create boundary conditions to test under ON regime in the winding model. ON cooling is created when the buoyancy forces within the winding dominates the inertia forces by the pump.

Influence of operating conditions

Lower $V_{in}$ caused more distorted oil flow distribution, i.e. more oil flows in bottom pass cooling ducts and less oil flows into top pass cooling ducts. As $V_{in}$ is reduced further, oil stagnation occurs in top pass cooling ducts. With the occurrence of oil stagnation, the hot spot temperature increases significantly. With further reduction in $V_{in}$, oil starts to reverse its direction in top pass cooling ducts. As long as oil stagnation is sustained, the benefit of having higher $V_{in}$ diminishes. Hence, it is
important to avoid situations at which oil stagnation or oil reverse flow occurs. As $P_{\text{loss}}$ is increased, oil flow distribution becomes more distorted and oil stagnation and oil reverse flow occur in pass top cooling ducts resulting in higher hot spot temperatures. It was also observed that non uniform losses do not affect oil flow distribution within the pass. $T_{\text{in}}$ had a weak influence over oil flow distribution within the winding structure.

- **Influence of winding model geometries**

  Higher $H_d$ causes oil reverse flow to occur in lower radial cooling ducts in the pass. The occurrence of oil reverses flow influences the location of the hot spot temperature. It was indirectly indicated that hot streak dynamics are affected by $H_d$. $W_d$ had a weak influence over both oil flow and temperature distributions. The hot spot temperature was higher under higher $n_d$ as oil stagnation is more likely to be sustained. The positive influence of having higher $V_{\text{in}}$ is diminished in reducing the hot spot temperature as long as oil stagnation is sustained in ducts near the location of the hot spot temperature.

9.2.4 **Conclusions of tests under alternative transformer liquids**

Tests were conducted using three transformer liquids including a mineral oil, a gas-to-liquid oil and a synthetic ester.

- **Comparison under OD cooling modes**

  It was observed that both the mineral oil and the gas-to-liquid oil showed similar behaviours with almost identical oil flow and temperature distributions. The synthetic ester oil caused more uniform oil flow distributions and delayed the occurrence of oil reverse flow as it has higher viscosity and so lower Reynold number at the same pass inlet oil velocity. The temperatures between the oils were comparable when no reverse flow occurred in the pass bottom cooling duct. However, when reverse flow occurs in both the mineral oil and the gas-to-liquid oil but not in the synthetic ester liquid, the hot spot temperature was lower in the synthetic ester liquid with the cost of higher pressure loss.

- **Comparison under ON cooling modes**

  At the same $V_{\text{in}}$, it was observed that both the mineral oil and the gas-to-liquid oil have similar oil flow and temperature distributions. The synthetic ester caused more uniform oil flow distribution and hence had lower hot spot temperature. Nonetheless, in practise
for the same transformer design, the synthetic ester would cause lower $V_{in}$ as it has higher viscosity and so testing at the same $V_{in}$ is not a fair way to compare different oils. If assuming retro-filling scenario, it was estimated that the mineral oil would develop 1.7 times higher inlet mass flow rate than the synthetic ester. This estimated ratio was found to match well with calculated ratio from data in published retro-filling literatures [45, 46]. It was observed that the synthetic ester caused a significant increase in the hot spot temperature compared to the mineral oil. Therefore, careful considerations should be taken into account when retro-filling a zig-zag power transformer designed for mineral oil with ester based liquids.

9.3 Future Work

The following items can be pursued further to enhance the understanding of fluid flow and temperature distributions in disc type power transformers.

**Future work related to experimental validation of dimensional analysis**

Validation of other applications of dimensional analysis on hot spot factor can be carried out using both the experimental setup and the PIV system. Composite plates insulated with paper are needed. Also, more work can be conducted to investigate the application of dimensional analysis on ON cooling modes. To do this, comparison between experimental and CFD simulations are needed. Heat dissipation through the winding model need to be accurately estimated and reduced while allowing the PIV system to have optical access to record oil flow rates.

**Future work related to OD experimental testing**

The winding model lacks a certain degree of complexity in the selected plate design. Though it is not expected that composite plates insulated with paper would affect oil flow distribution, they certainly affect the hot spot temperature, average winding temperatures, and the hot spot factor. It is of interest to study the influence of modelling complexities on results. Tests can be conducted with more detailed plates and compared to the available tests presented in this work to understand experimentally the influence of composite insulated plates on reported conclusion here. Also, the range of geometrical parameters can be expanded to establish more firm understanding for their individual effects given that geometrical deviations are controlled. Characterization of the 3D component of the flow rate in radial cooling duct can be documented.
Future work related to ON experimental testing

Would testing under controlled $V_{in}$ be representative of an ON transformer model which have the equivalent $V_{in}$ developed by thermosiphon force spontaneously?

With this investigation, modelling complexity can be defined. This is especially significant for ON simulations as almost all simulations conducted in the reported literature set $V_{in}$ as a controlled parameter. Having a spontaneous system requires immense computational resources and long simulation times using CFD simulations. Also, with this knowledge, a fair comparison between different winding model geometries can be made by testing under equivalent $V_{in}$.

Future work for testing under alternative oils

In order to compare the oils fairly, it is beneficial to have a spontaneous system that would allow the developed thermosiphon force to create the inlet mass flow rate. In the used experimental setup this was not possible as there are large pressure losses in the hydraulic loop and hence a way to expect the new developed flow rate is needed. This knowledge would also benefit simulation modelling. More work is needed to derive a relationship which would allow a fair comparison between the different oils. This can be done by having a complete ON system which has minimum pressure losses in the hydraulic loop and enables the elevation of radiator to adjust the inlet flow rate.
References


Appendix 1 Design of an Experimental Setup to Study Factors Affecting Hot Spot Temperature (HST) in Disc Type Winding Transformer

In this paper, description of the experimental setup and measurement techniques used are provided. Results of initial temperature and oil flow distribution measurements are reported using both a multi-channel temperature instrument and a particle image velocimetry system. Modelling assumptions of the winding model and heating strategies of disc segments are discussed.

DESIGN OF EXPERIMENTAL SETUP TO STUDY FACTORS AFFECTING HOT SPOT TEMPERATURE IN DISC TYPE WINDING TRANSFORMERS

M. Daghrah¹, Z. D. Wang¹*, Q. Liu¹, D. Walker², P. W. R. Smith³, P. Mavrommatis⁴

¹ The University of Manchester, Manchester M13 9PL, UK
² Scottish Power, Blantyre G72 0HT, UK
³ Shell Global Solutions (UK), Manchester M22 0RR, UK
⁴ TJ/H2b Analytical Services Ltd, Capenhurst, Chester CH1 6ES, UK
*Email: zhongdong.wang@manchester.ac.uk

Keywords: Power Transformer, Disc Type Winding, Hot Spot Temperature (HST), Experiment, Isothermal testing, Transformer oil.

Abstract

Oil immersed power transformers have complex structures for which electrical, mechanical and thermal considerations have to be taken into account during design. To produce a reliable transformer, the thermal profile in terms of temperature for windings is of interest because ageing of the transformer and its constituents is mainly affected by its operating temperature. The temperature profile is not uniform within a winding mainly because of the uneven loss distribution and the uneven flow distribution of the coolant within the winding structure. The Hot Spot Temperature (HST) is defined as the hottest temperature of the transformer which strongly affects the ageing of the transformer oil and the winding paper insulation near it. Understanding the HST helps utilize available assets for a utility in the longer term. Parameters affecting the location and the value of the HST can be split into geometrical and operational parameters. Geometrical parameters include horizontal cooling duct height, vertical cooling duct width and the number of discs per pass for a winding. Operational parameters include the inlet oil flow rate, the oils thermal properties and the loading level of the transformer. In this paper, the design and implementation of an experimental setup is presented which is used to study the effects of these parameters on the HST.

1 Introduction

Studying the temperature distribution within a disc-type winding has been the focus of many researchers. Understanding how temperature is distributed within a winding could give advantages in estimating the ageing rate of the operating transformer. Also, it is important to understand the dynamic behaviour of the HST and the influence of different load profiles on the thermal behaviour of the transformer especially with the emergence of smart grid technologies. Fundamentally, such an understanding could considerably improve future transformer design, especially when different types of oil are used.

Both operational and structural conditions affect a transformer’s ageing rate. It is generally accepted that a nominal ageing rate of transformer paper insulation occurs at temperatures up to 98 °C and that ageing rate doubles each 6 °C increase over 98 °C [1-2]. Average winding temperature rise over ambient temperature are specified by the IEC 60076-2 [2] for Oil Natural (ON) cooling mode as 65 K and for Oil Directed (OD) cooling mode as 70 K.

The distribution of transformer losses contributes to the uneven temperature profile as eddy current losses are not evenly distributed but concentrated at the two ends of the winding. Also, the distribution of flowing oil among horizontal cooling ducts is not uniform. These two effects were combined together by the IEC 60076-2 standard [2] through the definition of the hot spot factor $H = QxS$ where the Q factor represents the loss distribution and the S factor represents cooling efficiency of the winding geometry. The hot spot factor $H$ is used to estimate the HST from temperature rise tests. Wrong estimation of $H$ could result in operating the transformer either at higher or at lower operating temperature than the predicted one. In [3] effective $H$ factors of 35 transformers were estimated and concluded that the median value of effective $H$ was significantly higher than the one originally assumed. To better understand HST, studies were performed on parameters affecting the HST using small scale winding models either through simulations [4] or through experiments [5]. The designed experimental setup presented in this paper allows testing the effects of various parameters on the thermal behaviour of a typical disc type winding model. Existing experiment related literatures studying HST on a small scale winding model are reviewed, and the results of initial tests conducted on the designed system are presented.

2 Literature Review

Different modelling techniques have been applied to transformer thermal modelling. In the literature, these modelling techniques can be classified into two main categories. The first category includes models which are based on an electrical-thermal analogy using combined circuit components [6-7]. The second category includes small scale
winding models which represent enough details of the flow and temperature distributions within a winding [4-5]. The small scale winding model is a circumferential segment of a transformer winding. The transformer winding consists of adjacent segments between two sets of spacers and the segments are hydraulically in parallel as shown in Figure 1. Within each winding segment, the oil flows in a zig-zag pattern and this is controlled by diverting washers. A set of disc sections between two diverting washers is called a pass. Several passes make up a winding segment and these passes are hydraulically in series. Thermal modelling of a transformer winding, and hence the HST location and value, can be performed by studying both oil flow distribution and loss distribution within a small scale winding model either via simulation or by experiment.

Figure 1: Representation of a typical transformer winding

HST study through simulation can be fitted into two main streams; the first stream is the Network Hydraulic Thermal Modelling (NHTM). In the NHTM, the small scale winding model is represented as a network of parallel paths and the conservation of mass, energy and momentum are applied at each node and path in the network to generate a set of equations which are then solved numerically [4]. The second stream is the use of both Computational Fluid Dynamics (CFD) and Finite Element Method (FEM) techniques through commercial software to find both the flow and the temperature distribution within the winding model. CFD simulations are computationally more demanding; on the other hand they are more accurate than NHTM models. In [8] oil flow distributions predicted by both NHTM and CFD modelling were presented. The heat transfer process within a cooling duct was studied in [9] and analytical results were compared with CFD simulation results.

HST study through experimental work involves building a small scale winding model and running thermal tests under different testing conditions to observe and record both the temperature profile and the oil flow distribution within the winding model. Experimental work is governed by the difficulty of measuring both the temperature and the flow rates within the small winding model without affecting the flow pattern.

Very few experimental attempts have been reported within the literature due to the difficulty of measuring the flow rates within horizontal cooling ducts. [5] conducted an isothermal investigation to measure oil flow rates within horizontal ducts of a disc type winding model. The winding model was built to accommodate up to 20 disc sections. Only one pass was implemented. The flow distribution was measured and recorded using Hot Wire Anemometry (HWA) under different inlet flow rates and under different geometrical parameters. It was observed during the experiments that oil stagnation occurred at the middle ducts of the tested pass and oil stagnation occurred at more ducts when the number of disc section per pass was increased.

[10] investigated the cooling performance of an SF6 gas cooled power transformer under isothermal conditions. A small winding model was used and water was used instead of SF6 but under the same Reynold number. A total of 28 disc sections were used. Parameters of interests were the ratio between horizontal cooling ducts height to vertical cooling duct width and the number of disc section per pass. Both the flow distribution within the horizontal ducts and the pressure distribution within the vertical duct were measured. Velocity was measured by tracking the movement of seeding particles using a CCD camera. It was concluded that lower number of discs per pass would lead to better flow distribution but higher pressure loss. The existence of reverse flow under certain conditions was also reported which ultimately affects cooling performance.

Thermal testing on a small scale winding model was conducted [11]. The winding model consisted of two winding segments each consisted of 6 heated blocks. Both the flow distribution and the temperature profile were recorded. Laser Doppler Velocimetry (LDV) was used to record the flow rates within the winding model. Water was used as the coolant instead of transformer oil. The main purpose of this experiment was to verify the proposed numerical model which was based on CFD analysis. It was concluded that CFD simulations can give detailed temperature profile and that smaller grid size improves simulation accuracy.

3 Experimental Setup

The designed experimental setup aims to provide the ability to test the effects of several geometrical and operational parameters on the developed thermal profile inside a small scale winding model. Table 1 lists parameters to be tested with possible ranges for each parameter.

<table>
<thead>
<tr>
<th>Geometrical parameters</th>
<th>Operational Parameters</th>
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<tbody>
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<td>Horizontal duct (mm)</td>
<td>Vertical ducts (mm)</td>
</tr>
<tr>
<td>Discs/pass</td>
<td>Oil Used</td>
</tr>
<tr>
<td>Loading Level</td>
<td>Inlet flow rate</td>
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<tr>
<td>(litres/minute)</td>
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</table>

Table 1: Parameters which may affect the HST

<table>
<thead>
<tr>
<th>Geometrical parameters</th>
<th>Operational Parameters</th>
</tr>
</thead>
<tbody>
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<td>Horizontal duct (mm)</td>
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Experimental Study of Transformer Liquid Flow and Temperature Distribution

Figure 2 illustrates the designed experimental setup. The system consists of the following components which are discussed in sections 3.1) Winding model 3.2) Radiator 3.3) pump and 3.4) Flow meter. The instrumentation are discussed in sections 3.5) for temperature measurement and 3.6) for PIV.

3.1 Winding Model

The winding model is made to replicate a typical winding segment of disc type transformer. Three simplifications were adopted to facilitate the construction of the set-up. First, it was assumed that disc sections within winding segment can be modelled as rectangular plates instead of sections of discs. Second, instead of using composite plates constructed of copper conductors wrapped with paper; solid aluminium plates were used to model disc sections. Third, washers were modelled with the same thickness as the plate which is not the case in real transformers.

Figure 3 shows the designed winding model. It can host up to 60 plates. Each plate is a 100 mm x 104 mm x 10 mm aluminium plate. Two resistive heating elements were inserted in each plate, 30 mm away from the plate edges as detailed in Figure 3. The designed plate differs from the actual disc section in that: 1) a solid rectangular shaped plate is used. 2) The plate was not wrapped with insulating paper. 3) Heat density is not uniformly distributed within the plate due to the location and number of the heating elements used.

The temperature profile of the plate was tested using thermal imaging at different injected heating powers and under natural air cooling conditions. Figure 4 shows a thermal image of one of the plates supplied with a total of 144 watts and the temperature profile along the blue line at the centre of the plate. It can be observed that the temperature profile is fairly symmetrical on the y-axes. The plates are inserted into the winding model such that the oil flows either from right to left or from left to right. Two thermocouples, Th1 and Th2, were embedded in the plate 10 mm from each side as shown in Figure 4. It was observed that the temperature difference between Th1 and Th2 was less than 1 °C during the test.

Lexan® 9030 polycarbonate sheets were used to construct the winding model walls. Lexan® 9030 provides good optical passage so that the winding model can be seen from outside. Lexan® 9030 should withstand the different tests oil and temperature within expected testing range of HST of 110 °C.

3.2 Radiator

Radiator are essential to cool oil down and to dissipate losses to the ambient. Equation 1 is the main governing equation for radiator selection [12] where \( m_r \) is the mass flow rate in kg/s, \( c_p \) is the heat capacity of oil in J/(kg.°C), \( \Delta T = T_{in} - T_{out} \) is the temperature difference between radiator hot inlet oil and radiator cooled outlet oil in °C. \( P \) is the power to be dissipated by the radiator in W. The performance of the radiator depends on the temperature difference between mean oil temperature within the radiator and the ambient temperature. A domestic radiator was chosen to dissipate 4 kW under ambient temperature of 20 °C and mean oil temperature rise over ambient of 50 °C.

\[
P = m_r c_p \Delta T
\]  

(1)

3.3 Pump and AC Drive

A pump is required to create the driving force to circulate the oil in the system at different desired flow rates. Hydraulic
losses within the loop should be estimated and known before acquiring the pump. The energy equation shown by equation 2 defines the energy state of the fluid in a volume controlled system at two different positions 1 and 2 [13]. In equation 2, \( p \) is the static pressure at either positions in Pa, \( V \) is the average velocity at either positions in m/s, \( z \) is the relative height of either positions from a predefined reference in m, \( \gamma \) is the fluid specific weight in N/m\(^3\), \( g \) is the gravitational acceleration constant, \( \alpha \) is a constant which depends on flow type and it is equal to 2 for laminar flows. Both \( h_p \) and \( h_t \) are the pump head and the hydraulic head losses in the system in m.

In a closed system such as the one shown in Figure 5, the pump supplies hydraulic head equivalent to the hydraulic losses and equation 2 is reduced to equation 3 where \( h_{\text{WM}} \) is the hydraulic loss of the winding model, \( h_{\text{FM}} \) is the hydraulic loss of the Flow Meter, \( h_{\text{R}} \) is the hydraulic loss of the Radiator and \( h_{\text{pipe}} \) consists of hydraulic losses related to the pipe work. The hydraulic losses can be divided into major losses and minor losses. Major losses are governed by Darcy-Weisbach equation [13] shown in equation 4 where \( f \) is the friction coefficient of the duct surface, \( L \) is the duct length, \( D \) is the duct hydraulic diameter, \( V \) is the average velocity within the duct and \( g \) is the gravitational acceleration constant. Minor hydraulic losses are usually estimated experimentally and they represent joint, bend, expansion or contraction losses.

\[
\frac{p_1}{\gamma} + \frac{\alpha_1 (V_1^2)}{2g} + z_1 + h_p = \frac{p_2}{\gamma} + \frac{\alpha_2 (V_2^2)}{2g} + z_2 + h_1
\]  
\[h_p = h_t = h_{\text{WM}} + h_{\text{FM}} + h_{\text{R}} + h_{\text{pipe}} \]  
\[h_{\text{Major}} = f \frac{L}{D} \frac{V^2}{2g} \]  

Figure 5: Components of hydraulic losses and pump selection

Two types of pumps can be used; the first is the centrifugal pump and the second is the positive displacement pump. The differences between their pump curves are shown in Figure 5. Centrifugal pumps are more suitable for applications where speed control is required and so a centrifugal pump was chosen. The flow rate can be controlled either by throttling or speed control is required and so a centrifugal pump was chosen. A PD flow meter works by allowing the fluid to pass in predetermined volumes. PD flowmeters are almost insensitive to oil viscosity variations. An Oval Gear flow meter was chosen with an accuracy of 0.5% of reading.

### 3.5 Multi-channel Temperature System

It is desired to measure the temperature in each plate at two locations; the first is the plate temperature near the duct inlet and the second is the plate temperature near the duct outlet. If 30 plates are used then at least 60 temperature sensors are needed. Thermocouples were used because they are small in size, easy to install and inexpensive in price. K-type thermocouples were used. Data acquisition system manufactured by Measurement Computing was used. The system consists of a data acquisition card PCI-DAS08 which is programmable and an external multiplexer CIO-EXP32 which has two 16 by 1 multiplexers on board. The CIO-EXP32 is designed for thermocouples and so it provides a Cold Junction Compensation (CJC) and open thermocouple detection. The PCI-DAS08 allows up to four CIO-EXP32 multiplexers to be used which gives the ability to implement up to 128 channels. LABVIEW was used to acquire and store temperature data. The system was calibrated using a water bath from temperatures ranging from 0 °C to 100 °C in 5 °C steps. Accuracy within +/- 0.5 °C was achieved with repeatability within +/- 0.2 °C.

### 3.6 Particle Image Velocimetry (PIV)

Particle Image Velocimetry (PIV) is a non-intrusive measurement method used extensively in aerospace and fluid dynamic research to record and measure flow distribution. PIV system works by tracking the movement of small tracer particles within the fluid flow. To track the tracer particles, a double head laser source is used to send two consecutive laser pulses which are spread into laser sheets through optical lenses. The laser sheets are usually 1 mm thick and they are aligned in the expected direction of the flow. A flow is captured using a CCD camera placed orthogonal to the laser sheet during each pulse. Two frames, frame A and frame B, synchronized with the two laser pulses, pulse A and pulse B, contain the displacement information of the tracer particles. By knowing the time between the two pulses and the displacement distance of each tracer particle from the two frames, the velocity field can be calculated. A synchronizing unit is needed to synchronize several PIV components. Figure 6 shows basic components of PIV system.

Figure 6: PIV system main components and basic operation

### 4 Preliminary Test Results

The designed experimental setup is meant to be used extensively to study thermal behaviour of a typical disc type
winding. To achieve this purpose, both isothermal and thermal tests are needed. Isothermal tests are needed to record the flow distribution within the horizontal cooling ducts under various geometrical and operational conditions. Thermal tests are needed to study the location of the HST. In the following, initial results of both isothermal and thermal tests are presented for a fixed winding model geometry summarised in Table 2. Transformer insulating oil Nytro Gemini X was used in both tests presented here. Tests with different oil are ongoing and results will be provided in further publications.

Table 2: Geometry of winding model under test

<table>
<thead>
<tr>
<th>Geometrical parameters</th>
<th>Horizontal Ducts (mm)</th>
<th>Vertical Ducts (mm)</th>
<th>Number of Plates/Pass</th>
<th>Number of Passes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Height</td>
<td>4</td>
<td>100</td>
<td>10,10</td>
<td>10</td>
</tr>
<tr>
<td>Length</td>
<td>100</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Depth</td>
<td>100</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

4.1 Isothermal Test

Flow distribution of oil within the third pass was measured using Particle Image Velocimetry (PIV). Inlet flow rate recorded by the inlet flow meter was fluctuating between 15.19 to 15.41 litres/minute during the test. Room temperature was recorded during the test and it was maintained between 24 °C and 26 °C and so the oil temperature during the test was assumed to be 25 °C. Under these test conditions, Figure 7 shows the recorded average velocities in each of the horizontal cooling ducts numbered from 1 to 11 from bottom to top except for the bottom cooling duct because it was not optically accessible for PIV at the time of the conducted test. Table 3 shows the numerical values of the test.

The recorded results using PIV can be checked against the inlet flow meter by summing all the flow rates within the horizontal cooling ducts which should be equal to the inlet flow meter recording. Using data from Table 3, the total flow rates without duct 1 is 14.43 litres/minute and if duct 1 velocity is considered similar to duct 2 then the total flow rate would be 15.33 litres/minute.

Table 3: Third pass average velocity profile

<table>
<thead>
<tr>
<th>Duct</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
<th>11</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity (mm/s)</td>
<td>NA</td>
<td>38</td>
<td>36</td>
<td>43</td>
<td>46</td>
<td>51</td>
<td>59</td>
<td>64</td>
<td>78</td>
<td>89</td>
<td>97</td>
</tr>
</tbody>
</table>

Flow within the horizontal ducts is believed to be laminar based on fluid dynamic theory that if the Reynolds’ number (Re) is less than 2000, then the flow should be laminar [13]. Re is given by equation 5 where V is the mean flow velocity in m/s, D is the hydraulic diameter in m and v is the kinematic viscosity in m²/s. For the fluid used and the horizontal duct geometry, the maximum velocity which would maintain a laminar flow can be calculated as V ≈ 4 m/s. This value is much higher than the recorded velocities shown in Table 3 and so the flow should be laminar. A laminar flow has a parabolic velocity profile [13]. One of the PIV images shown in Figure 8 for the velocity profile, illustrates the characteristics possessed by laminar flow.

$$Re = \frac{VD}{v}$$

4.2 Thermal Test

A thermal test was conducted under an inlet flow rate of 15.0 litres/minute. Only the last pass of the winding model was heated. The total power injection in the pass was 440 W evenly distributed between the 10 plates of the pass. Room ambient temperature at steady state was recorded to be 23 °C. Temperatures were recorded at each plate at two locations either close to the inlet or close to the outlet of the respective horizontal duct, see Figure 4. The winding model inlet temperature was recorded at steady state to be 29.7 °C and the outlet temperature was recorded to be 31.1 °C. During the test, the temperature was logged each second.

Figure 9 shows the temperature of plate 1 versus time until steady state was reached. At steady state, the average temperature of each plate over 1 minute was taken as a representative value.

Figure 10 shows the recorded temperature profile at steady state. It can be observed that the hottest plate is the first plate. The temperature profile corresponds well with the recorded flow distribution. It can be observed that the velocity at duct 11 is highest which correspond to lowest plate temperature. A
trivial conclusion from heat convection, higher flow rate leads to a better cooling.

Figure 10: Third pass temperature profile

5 Conclusions

An experimental setup was implemented to conduct studies to aid understanding of thermal behaviour of disc type winding transformers. Initial isothermal and thermal tests were conducted to verify the functionality of the designed test system. Particle Image Velocimetry (PIV) was used to measure the flow rate within winding horizontal ducts, and it was proven that PIV is a technique which allows velocity measurement within confined and small cooling ducts. Initial thermal test was conducted assuming uniform loss distribution within the winding model, and the temperature profile recorded by thermocouples was found to be consistent with the velocity distribution within the pass which was recorded using PIV under isothermal testing conditions.

The designed system will be used in the future to conduct extensive thermal and isothermal tests. Experimental tests and CFD simulations are planned to be performed. Also, implementation of non-uniform loss distribution within the winding model will be considered. The plate design will be improved further to represent the winding disc sectors more accurately.

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References

Appendix 2 Effect of Installation Positions of Fibre Optics Temperature Sensors (FOTS) on Transformer Hot-Spot Temperature Measurements

In this paper, a small scale experimental setup is designed to investigate the influence of installation positions of fibre optics temperature sensors within spacers on their ability to record accurately the hot spot temperature. Several installation positions are defined by changing the groove angle, the groove width, and the groove length. Kraft paper is wrapped above and below each spacer. Tests were conducted under both oil directed and oil natural cooling conditions.

EFFECTS OF INSTALLATION POSITION OF FIBRE OPTICS TEMPERATURE SENSORS (FOTS) ON TRANSFORMER HOT-SPOT TEMPERATURE MEASUREMENTS

M. Daghrarah\textsuperscript{1}, Z. D. Wang\textsuperscript{1*}, Q. Liu\textsuperscript{1}, Jean-Noël Bérubé\textsuperscript{2}

\textsuperscript{1}The University of Manchester, Manchester M13 9PL, UK
\textsuperscript{2}Neoptix Canada LP, Canada
*Email: zhongdong.wang@manchester.ac.uk

Keywords: Power Transformer, Hot Spot Temperature (HST), Experiment, Fibre Optics Temperature Sensors (FOTS).

Abstract

Fibre Optics Temperature Sensors (FOTS) have been used to measure the Hot-spot Temperature (HST) in power transformers for at least 30 years. The advantage that FOTS have is that they are immune to electromagnetic interference and so unaffected by transformer operation. FOTS are used as a method to accurately measure the HST if its location is known. In this paper, behaviours of FOTS under different installation conditions are to be studied experimentally. A test system was designed and it consists of a small scale winding model with geometries that replicate a typical disc type winding transformer. The winding model is designed to host up to 8 disc sectors. The test system also includes a radiator to cool heated oil down, a flow meter and a pump to control the inlet flow rate. Temperature recordings from both FOTS and thermocouples are compared. More specifically, effects of cooling mode on FOTS reading are studied. Also, the dynamic response of FOTS under simulated loading cycle is studied and finally effects of installation position of FOTS within spacer on reading accuracy are investigated. It is expected that FOTS recording would be very close to the conductor temperature at steady state but not necessary during transient periods.

1 Introduction

An electrical power transformer has a non-uniform temperature profile within its winding mainly because first the uneven loss distribution and second the uneven flow distribution within the winding cooling ducts. This leads to the definition of the Hot Spot Temperature (HST) as the hottest point in the transformer. Ageing of the transformer is mainly dependent on the ageing of its paper insulation which is hugely affected by the HST. In general, it is accepted that the nominal transformer ageing rate occurs at a HST around 98 °C for normal Kraft paper and 110 °C for thermally upgraded Kraft paper. It is also recognized that the ageing rate doubles each 6 °C rise over the rated HST [1-2].

Because it is important to know the actual value of the HST, several methods were proposed to estimate the HST either by calculations or by direct measurements. Both the IEC 60076-2 [2] and the IEEE loading guide [3] provided methods to calculate the HST based on data obtained from heat run tests; these methods provide a quick way to estimate the HST but on the account of lower accuracy. Other methods of estimating the HST are the Network Hydraulic Thermal Modelling (NHTM) initially proposed by J. Oliver [4] and then constantly being developed by researchers. With the emergence of fast processing machines, Computation Fluid Dynamics (CFD) and Finite Element Methods (FEM) are becoming more popular in researching power transformer thermal modelling. They give more accurate and precise prediction of the HST. All these techniques are simulation based techniques to identify the expected location and value of the HST.

Winding Temperature Indicators (WTIs) were used traditionally to measure the winding HST indirectly by heating an oil pocket in proportion of the transformer load current. Calibration is needed of WTI based on simulation results at rated conditions. However, accuracy of WTI at different loading conditions is not verified as it is usually calibrated at only the rated condition based on simulation results.

Fibre Optics Temperature Sensors (FOTS) are considered the most accurate mean to measure directly the HST in live transformers. FOTS are immune to electromagnetic interference as they are not affected by the transformer operation. They can provide valuable information to utilities about the HST to make better decision in loading, utilizing or replacing their assets. The emergence of FOTS can be traced back to 1984 when [5] reported one of the first practical testing and installation of FOTS under typical live transformer environment. Practical field experience for the usage of FOTS in live transformer was reported by [6]. The field experience was conducted on a 240 MVA autotransformer with a total of six FOTS probes used and they were installed on top of tap, common and series windings. FOTS recordings were compared with WTI measurements and [6] reported that the response time of FOTS was faster than WTI but there was a reasonable agreement between WTI and FOTS recordings. FOTS technology has matured significantly since its early ages and more utilities and transformer manufacturers are relying on it to better understand thermal behaviour of power transformers.
Because of recent FOTS developments, the IEC 60076-2 [2] and the IEEE loading guide [3] specified separate sections on recommended number of FOTS probes to use and recommendations on good installation practices were stated.

To better locate the HST, understanding of both oil flow distribution and loss distribution within the winding are needed [7]. The location of HST has to be determined in order to measure it correctly using FOTS. FOTS probes are usually inserted within spacers during winding manufacturing in the location where the HST is expected [8]. It is of interest to understand whether the installation conditions of FOTS probes within spacers would have any effect on measured HST. In this paper, different installation conditions of FOTS probes within spacers are tested on small scale winding model under various operating conditions.

2 Experimental Setup

Figure 1 shows a schematic diagram for the designed experimental setup. It consists of three main components 1) a winding model 2) a pump and 3) a radiator. The winding model permits up to 8 disc segments, plates, to be inserted. An external fan is used to blow air to the radiator to control cooling of oil. The flow rate is measured using a positive displacement flow meter which has an accuracy of 0.5% of reading. The flow rate is controlled by throttling using a ball valve in front of the pump.

Figure 2 shows a more detailed schematic of the winding model and the naming for plates, spacers and used FOTS probes and thermocouples. Plates are named as P1 to P4 from bottom to top. A total of 5 cooling ducts are available and they are named as D1 to D5 from bottom to top. In each cooling duct, a spacer is placed and spacers are named as S1 to S5 corresponding to cooling ducts D1 to D5 respectively. A total of 7 FOTS probes are used and named as F1 to F7. F1 and F2 are installed in S2. F3 and F4 are installed in S3 and F5 and F6 are installed in S4. F7 was used to measure ambient temperature during tests and left placed outside winding model.

A total of 9 thermocouples are used to measure plate temperatures, assumed to be the HST, and oil inlet temperature. T1 and T2 are used to measure P1 temperature, T3 and T4 are used to measure P2 temperature, T5 and T6 are used to measure P3 temperature and T7 and T8 are used to measure P4 temperature. T9 is used to measure inlet oil temperature. In all tests, T1 was placed in contact of P1 surface exactly underneath F1. The same installation approach was followed for T2 and F2, T3 and F3, T4 and F4, T5 and F5 and finally T6 and F6. T7 and T8 are fixed on P4 in the same manner as the rest of thermocouples. This positioning creates two groups of sensors. The first group are all sensors closer to horizontal ducts inlet. The first group is referred to as Inlet Side Group (ISG) which consists of T1, T3, T5 and T7. The second group are all sensors closer to horizontal ducts outlets. The second group is referred to as Outlet Side Group (OSG) which consists of T2, T4, T6, T8 and T9.

All four plates were made from aluminium blocks with dimension of 100 mm x 204 mm x 10 mm. Each plate was heated with 6 cartridge heating elements in order to create a uniform and acceptable plate temperature profiles. Figure 3 shows plate temperature profile under natural air cooling conditions with injected power of 107 W per plate.
Thermocouples were fixed on plates using high temperature, low thermal resistivity tape. Four layers of transformer paper insulation with total thickness of 0.2 mm were wrapped around the plate underneath each spacer. Spacers used were made from typical transformer pressboards with thickness of 3.2 mm. The dimensions of each used spacer are 100 mm x 40 mm x 3.2 mm. Grooves in spacers were cut precisely using laser. All used spacers and papers were first placed in heated oven at 105 °C for two days to remove all moisture content and then spacers and papers were impregnated in oil and placed in heated vacuum oven at 105 °C for 2 days to remove all gas content and to impregnate them with oil. FOTS probes were installed in spacers and then spacers were fixed on plates along plates central lines using screws from both sides. FOTS probe installation is summarized in Figure 4. The assembled experimental setup is shown in Figure 5.

![Figure 4: Thermocouple and FOTS installations on plates](image)

![Figure 5: Assembled experimental setup](image)

3 Investigation Methodology

The purpose of this paper is to investigate whether installation positions of FOTS probes within spacers would create any measurement errors of recorded HST under wide range of operating conditions. Three installation parameters are defined as follows:

1) **Groove size:** It is recommended by leading FOTS manufacture [8] that grooves should have size such as FOTS probes can be inserted in push-fit conditions. A groove size bigger than FOTS probes is to be tested to see if it does influence recorded HST under varying cooling conditions.

2) **Groove Angle:** The effect of groove angle within spacer of FOTS reading is to be investigated.

3) **Head Space:** The effect of space between FOTS probe head and end of groove is to be investigated.

![Figure 6: Parameters of FOTS installation within spacers](image)

Spacers have two grooves and the distance between the FOTS probes head in each spacer is fixed to be 50 mm in all studied cases. Three cases are to be studied as follows:

1) **Case 1:** A reference case in which both ISG and OSG grooves are made such that groove size is 3.1 mm, groove angle is 45° and no head space.

2) **Case 2:** For all ISG grooves, the groove size is the only parameter that was changed from case 1 and it was made to be 5 mm. For OSG, groove angle is the only parameter that was changed from case 1 and it was made to be 30°.

3) **Case 3:** For ISG grooves, the head space is the only parameter that changed from case 1 and it was made to be 15 mm. For all OSG, the groove angle is the only parameter that changed and it was made to be 90°.

![Figure 7: Geometry of different FOTS installation studied](image)
The following FOTS installation rules recommended by [8] were respected and followed:

1) FOTS bending radius should not be less than 1 cm.
2) The last 1 cm of FOTS probe should be free standing.
3) Avoid bending or pressing last 1 cm of FOTS probes
4) FOTS probes are to be inserted in push fit fashion.

In all studied cases, the location of each FOTS probe head within the spacer is the same. Since the tests are conducted on small scale winding model; it is important to define the HST in the context of this lab experiment. In real situations, the probes are installed at the centre of spacers [8] as it is believed where the HST is located. However, in this lab experiment two FOTS were inserted in each spacer as shown in Figure 4 with thermocouples underneath and above each FOTS probe head. The FOTS probes reading are compared to both thermocouples above and below it. The location of FOTS probes within spacers is arbitrary and does not indicate that the HST is located there within actual transformer cooling ducts. Two FOTS probes in each spacer were used to obtain result repeatability and to reduce number of times needed to assemble the winding model. Tests conducted are classified into three categories as follows:

1) **Cooling mode effect:** for spacers of Case 1, Figure 7, tests are conducted under Oil Natural (ON) and Oil Direct (OD) cooling modes. The differences between the two cooling modes are highlighted and respectively the reading of FOTS under each cooling mode is discussed.

2) **Dynamic response of FOTS:** A cooling cycle is simulated in which the starting cooling mode is Oil Natural Air Natural (ONAN) cooling mode under 50 % loading level; then when top oil temperature rises above a threshold value a fan is turned on and the cooling mode becomes Oil Natural Air Forced (ONAF) cooling mode. Anticipating a rise in loading level to 100 %, a pump is switched on if top oil temperature rises above a threshold value shifting the cooling mode to Oil Directed Air Forced (ODAF) cooling mode. Response time of FOTS is evaluated and compared to HST, bottom oil temperature and to top oil temperature response times recorded by thermocouples.

3) **Effect of installation conditions:** Under same operating conditions, recording of FOTS are to be compared to identify which installation condition would create measurement errors. As demonstrated in Figure 7, groove size, groove angle and head space are the three parameters which identify installation conditions.

4 **Results**

All used FOTS and thermocouples were checked for accuracy. Used FOTS probes has an accuracy of +/- 1 °C and used thermocouples has an accuracy of +/- 1 °C. Transformer insulating oil Nytro Gemini X was used as the coolant in all tests.

4.1 **Effect of cooling mode on FOTS reading**

In power transformers, two main factors affect the developed temperature profile inside the transformer and so the location of the HST. The first factor is the loss distribution. The second factor is the flow distribution. It is well known that eddy current losses are higher both at the top and bottom of transformer winding. Similarly, the flow distribution is not uniform within transformer’s cooling ducts.

For ON cooling mode, the flow is driven by the thermosiphon force and the flow is higher at lower cooling ducts [9]. This leads to higher temperatures at top discs of each pass. On the contrary, for OD cooling mode, the flow is driven by inlet pumps with high flow rates such that the flow is higher at the top cooling ducts [7]. Since FOTS are usually installed between two discs in spacers [8], it is expected that FOTS would read a temperature in between the two adjacent discs.

Figure 8 and Figure 9 shows recorded temperature of OSG sensors T7, F5 and T5 under ON and OD cooling modes. Time constant is higher, 8.25 minutes, for ON cooling compared to OD cooling, 5.0 minutes. It can be seen that F7 reading is in between T7 and T5 for both cooling modes. However, the temperature difference between F7 and either T7 or T5 is determined by the overall temperature profile.

![Figure 8: FOTS recordings under ON cooling mode](image-url)

![Figure 9: FOTS recording under OD cooling mode](image-url)
Figure 10 shows steady state developed temperature profile under ON cooling mode. It can be seen that the temperature differences between FOTS probes and respective thermocouples are dependent on operating conditions. In ON cooling mode, flow distribution is mainly driven by thermosiphon force and so more flow passes through lower ducts given lower temperatures at lower plates. From the temperature profile, it can be inferred that the flow distribution is not uniform with large distortion. This creates larger temperature differences between adjacent plates than in OD cooling mode and so larger temperature differences between FOTS and thermocouples.

Figure 11 shows steady state developed temperature profile under OD cooling mode. It can be observed that temperature profile is nearly uniform for the first 3 plates while there is around 4 °C temperature difference between plate 3 and plate 4. The main reason for this relatively large temperature difference is that oil flows in higher rates at the last two cooling ducts surrounding plate 4. Also, the washer above plate 4 is not heated.

As a main conclusion of this test, FOTS reads the HST correctly in both ON and OD modes. Since FOTS are installed in spacers, the temperature recorded by FOTS would be somewhere between temperatures of both discs above and below the spacer. The higher the temperature difference between the two discs, the higher the temperature difference between FOTS recording and discs temperatures.

4.2 Dynamic response of FOTS
Time response of FOTS is to be documented in simulated load cycle starting from ONAN mode at 50 % loading level to ODAF mode at 100 % load. Figure 12 demonstrates the loading cycle which consists of 4 stages as follows:

1) Stage 1: Power supply was switched on as the starting point delivering total power of 550 W. Radiator fan was switched off during this stage. The fan was switched on when the top oil temperature was slightly higher than 60 °C making the transition from ONAN to ONAF. A delay time of around 4.1 minutes was recorded from the time the fan was switched on till the time both FOTS and thermocouples responded. This delay time is related to the system thermal time constant. The FOTS F5 lags the response of thermocouple T5 by 1.3 minutes. This is expected due to the time constant induced by the paper and spacer hosting the FOTS probes.

2) Stage 2: System temperature reduced due to a more effective radiator cooling. A temperature reduction by around 10 °C was achieved. A time constant of around 5.5 minutes was needed for temperature to decay from peak value to 37% of steady state value.

3) Stage 3: An increase in loading level was simulated by increasing the power to 1100 W. Thermocouples responded instantly and then 1 minute later FOTS F5 responded.

4) Stage 4: The pump speed was increased from 1 lpm to 5 lpm when the top oil temperature reached 68 °C. A response time of 1 minute was recorded for FOTS.

It can be observed from this test that FOTS response is within 1 minute to internal changes such as power or flow rate. However, for external fans, it was observed that FOTS had 5 minutes time delay until a response started.

4.3 Effect of installation positions on FOTS accuracy
Three parameters are to be investigated under same operating conditions to check whether certain installation conditions would create inaccuracies in recorded HST. The differences between FOTS sensors and adjacent thermocouples are calculated for different cases. All tests were conducted under same operating conditions of total injected power Pinj = 1070 W and Inlet flow rate Q = 5 lpm.
a) Effect of Groove size on FOTS recording:

Table 1 summarizes tests results for the ISG sensors which have bigger groove size from case 1 to case 2, Figure 7. It can be concluded by comparing both the absolute temperature and the temperature differences between the two cases that there is no observable effect of groove size on FOTS recording. FOTS in case 2 recorded HST in a similar fashion as in case 1. This can be explained by the fact that the oil within the grooves of FOTS is stagnated and does not flow. Nytro Gemini X has close thermal conductivity to that of paper and pressboards.

Table 1: Effect of Groove size on FOTS HST measurement

<table>
<thead>
<tr>
<th>Case</th>
<th>T1</th>
<th>T2</th>
<th>T3</th>
<th>T4</th>
<th>T5</th>
<th>T6</th>
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<th>F2</th>
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<th>F4</th>
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<td>84.32</td>
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<td>81.91</td>
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<td>87.5</td>
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<tr>
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<tr>
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<td>F4-T4</td>
<td>F6-T6</td>
<td>F8-T6</td>
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</table>

b) Effect of Groove angle on FOTS recording:

Table 2 summarizes tests results for the OSG sensors which have different installation angle from 45° in case 1 to 30° in case 2, Figure 7. No observable effects were noticed and all temperature differences are within the 2°C accuracy limit of both FOTS and thermocouples.

Table 2: Effect of Groove angle on FOTS HST measurement

<table>
<thead>
<tr>
<th>Case</th>
<th>T1</th>
<th>T2</th>
<th>T3</th>
<th>T4</th>
<th>T5</th>
<th>T6</th>
<th>T7</th>
<th>F1</th>
<th>F2</th>
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<th>F4</th>
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<td>88.0</td>
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<td>F5-T5</td>
<td>F7-T7</td>
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<tr>
<td>(°C)</td>
<td>F1-T1</td>
<td>F3-T3</td>
<td>F5-T5</td>
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</tbody>
</table>

c) Effect of head space on FOTS recording:

Table 3 shows the results of head space, Figure 7, on FOTS reading accuracy. It can be seen that the head space has no observable effect on recorded HST. It is believed that the oil in the head space is in stagnation and so does not influence recorded HST.

Table 3: Effect of head space on FOTS HST measurement

<table>
<thead>
<tr>
<th>Case</th>
<th>T1</th>
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<th>T3</th>
<th>T4</th>
<th>T5</th>
<th>T6</th>
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<tbody>
<tr>
<td>(°C)</td>
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<td>84.80</td>
<td>84.32</td>
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<td>81.91</td>
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<td>85.6</td>
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<td>86.67</td>
<td>85.0</td>
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</tr>
<tr>
<td>F1-T2</td>
<td>F2-T4</td>
<td>F4-T6</td>
<td>F6-T8</td>
<td>F8-T6</td>
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<tr>
<td>Case 3</td>
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<td>F6-T6</td>
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</tbody>
</table>

5 Conclusions

Under different installation and operational conditions, it was found that FOTS record the HST accurately. It was found that temperature differences between FOTS and HST depend on the overall temperature profile within the winding. Higher

Experimental Study of Transformer Liquid Flow and Temperature Distribution

5 Conclusions

Under different installation and operational conditions, it was found that FOTS record the HST accurately. It was found that temperature differences between FOTS and HST depend on the overall temperature profile within the winding. Higher temperature differences between FOTS and HST were recorded for ON cooling mode. This is due to the fact that the temperature profile is more distorted than in the OD case studied. Accordingly, it is recommended to install FOTS sensors around winding discs which are believed to have the HST.

Under dynamic load profile, FOTS showed fast response to variation such as loading level and pump speed. However, a time lag of about 1 minute between FOTS and thermocouples was recorded and this is believed to be due to the thermal time constant of spacer hosting the FOTS probe.

No observable effect of any of the studied installation parameters, groove width, groove angle and head space was observed. This is mainly because oil is believed to be in stagnation within the FOTS groove. This leads to the main conclusion that FOTS are capable of recording the HST accurately if its location was known through proper design and simulation. It is important to know the temperature distribution within the winding as FOTS probes might read more than 3°C lower than the actual HST.

References


Appendix 3 Experimental Investigation of Hot Spot Factor for Assessing Hot Spot Temperature in Transformers

In this paper, the hot spot factor as defined in the standard is examined. The aim is to split the hot spot factor into its two main components the S factor and the Q factor and to check whether these components are coupled or not. This process is done by first applying uniform losses, hence $Q = 1$, and measuring the hot spot factor from the known hot spot temperature. The flow distribution using particle image velocity is also measured. The second step is to apply non-uniform losses, and hence $Q$ is calculated. Then using the measured hot spot temperature, the hot spot factor and the S factor are calculated.

Experimental Investigation of Hot Spot Factor for Assessing Hot Spot Temperature in Transformers

M. Daghrah¹, Z.D. Wang¹*, Q. Liu¹, D. Walker², Ch. Krause³, G. Wilson⁴

¹The University of Manchester, Manchester, M13 9PL, UK
²Scottish Power, Blantyre, G72 0HT, UK
³Weidmann Electrical Technology AG, Rapperswil, CH 8640, Switzerland
⁴National Grid, Warwick, CV34 6DA, UK

*zhongdong.wang@manchester.ac.uk

Abstract—Ageing of paper insulation within oil immersed power transformers is the determining factor for the transformer expected life. A thermal diagram as shown in the IEC 60076-2:2011 standard specified a hot spot factor method to estimate the hot spot temperature based on heat run test data. The hot spot factor is split into a Q factor, which depends on loss distribution within winding, and an S factor, which depends on oil flow distribution within winding cooling ducts. In this paper, an experiment is conducted on a disc type winding model to study the hot spot factor under fixed winding geometries subjected to wide range of inlet flow rates and uniform or non-uniform power loss injections. The hot spot factor is calculated from measured temperatures within the winding model using thermocouples. The hot spot factor was decoupled by applying first uniform losses to calculate the S factor and then nonuniform losses to calculate the Q factor. It was observed that the hot spot factor is lower for more uniform temperature profiles. It was observed that the S and Q factors are interdependent and cannot be decoupled.

Keywords—Hot Spot; Thermal modelling; Experiment; Transformer

I. INTRODUCTION

In-service ageing of a power transformer is mainly determined by its Hot Spot Temperature (HST), defined as the hottest temperature within the transformer winding. The HST is affected by transformer winding geometries, total winding flow rate and loss distribution within the winding. It is widely accepted that ageing rate of paper insulation doubles each 6 K rise above the rated HST [1-2]. Prediction of the HST requires detailed knowledge of transformer winding geometries, flow distribution within the winding cooling ducts and loss distribution within the winding. An empirical method is given in the IEC 60076-2:2011 to estimate HST using heat run test data and a predefined Hot Spot Factor (H).

During a heat run test, the ambient temperature $\theta_a$, the bottom oil temperature rise $\Delta \theta_o$, the top oil temperature rise $\Delta \theta_t$, and the mean winding temperature rise $\Delta \theta_w$ are recorded. Measured temperature rises during a heat run test are presented on a thermal diagram as shown in Fig. 1 in which oil and winding temperature rises are assumed linear along winding height [2]. The gradient $g$ is the temperature difference between the mean oil temperature rise $\Delta \theta_{om}$ and $\Delta \theta_w$. The HST rise $\Delta \theta_h$ is estimated using equation (1) where $H$ is the Hot Spot Factor.

Presenting the $H$ with an analytical formula was considered by [3] and it was concluded that it is not reasonable to recommend any analytical formula for it. From a collection of heat run tests which included measurements of the HST using fibre optics, it was found that among the transformers studied the $H$ ranged from 0.51 to 2.06 [3]. It was observed in [3] that the $H$ does not depend on cooling mode or transformer power level. The $H$ was calculated and presented in [4] from a set of thermal tests on Oil Natural Air Natural (ONAN), Oil Natural Air Forced (ONAF) and Oil Directed Air Forced (ODAF) power transformers. It was found that for studied transformer types, the average value of the calculated $H$ is 1.3. The effective $H$ was deduced based on paper samples obtained from 35 scrapped transformers [5] and it was found that on average the effective $H$ value is 2.95, a much higher value than the assumed 1.3. It was specified in the IEC standard [2] that the $H$ can be presented as multiplication of two separate factors $Q$ and $S$. The $Q$ factor represents the loss distribution whereas the $S$ factor represents the nonuniformity of flow distribution within cooling ducts. A detailed thermal-hydraulic network model was used by [6] to study a 100-MVA power transformer and the results were used to calculate the $H$ and both $Q$ and $S$ factors. It was concluded that the $S$ and $Q$ factors are not decoupled. It was also observed that the $S$ factor depends on loading level and loss distribution as this affect oil viscosity which influences flow distribution. It was concluded that the concept of the $S$ and $Q$ factors as defined in the IEC 60076-2:2011 might not be particularly meaningful [6].

$$\Delta \theta_h = \Delta \theta_o + g \times H$$  \hspace{1cm} (1)

Fig. 1. Thermal diagram according to IEC 60076-2:2011 [2]
In this paper, an experiment is conducted on a disc type winding model to study the $H$, $S$ and $Q$ factors subjected to uniform and nonuniform losses under OD cooling mode. The experimental setup is presented in Section II. The effect of inlet flow rate on the $H$ is captured. Thermal tests were conducted and all required data to calculate the $H$ were measured and presented in Section III. Finally, Section IV provides a discussion of the results.

II. EXPERIMENTAL SETUP

The experimental setup consists of a disc type winding model, a radiator, a pump and a flow meter as shown in Fig. 2. The experimental setup was described in more details in [7]. An external heating unit is added outside of the winding model to control oil inlet temperature. Brief descriptions of the system components are given as follows:

A. Disc type winding model

The winding model consists of three passes and each pass hosts 10 disc segments, henceforth plates. Each plate was heated using two cartridge heaters located within each plate as shown in Fig. 3. Plates were made of aluminum blocks with 100 mm $\times$ 104 mm $\times$ 10 mm dimensions. Paper was not wrapped over plates due to the induced difficulty in assembling. This would cause lower than if plates were wrapped and hence affect the calculated $H$, which will be discussed in section IV. Baffles were made from acrylic sheets with the same thickness as the plates. Two thermocouples were fitted in each plate 10 mm from each end as shown in Fig. 3. Two thermocouples were used to measure winding model inlet temperature and 1 thermocouple was used to measure winding model outlet temperature. The height of each Horizontal Duct (HD) is 4 mm. Both the Vertical Ducts (VD) have the same width of 12 mm.

B. Instrumentation used

A multi-channel temperature acquisition system was implemented to provide 64 thermocouple channels. The system was calibrated in a water bath and an accuracy of $\pm$ 0.5% of reading. Multiple variable autotransformers, variac, were used to control power injection to simulate uniform or nonuniform loss distributions. The current and voltage were measured to record the power injection per plate.

III. TEST RESULTS

In order to decouple the $H$ into the $S$ and the $Q$ factors, two tests are conducted under uniform and nonuniform loss distributions as shown in Table I. Assuming that a plate models a power transformer copper disc segment with a current density of 4 A/mm$^2$ [1]; 50 W per plate, or equivalently near 2100 W/m$^2$ power flux density, was assumed which would accommodate for DC copper losses, eddy current losses and stray losses within the winding lower discs. Extra power was injected in top top 5 plates to simulate eddy current losses.

A. Hot spot factor under uniform losses

Inlet flow rate (FR) was changed from 3 litres/minute (lpm) to 18 lpm, equivalent average winding inlet velocities are shown on Fig. 4. The total power injection was maintained constant at 50 W per plate. Fig. 4 shows the average winding model temperature profiles for each plate under different flow rates. Table II shows the temperature rises and the calculated $H$. The $H$ is calculated using equation (1). Since losses are uniform, the $Q$ factor was assumed 1.0 [2,6]. The $S$ factor, by definition, was calculated using $H = S \times Q$. It can be observed that the temperature profile is more distorted at higher flow rates due to nonuniform oil flow distribution within horizontal cooling ducts. From Table II, the calculated $H$ is higher for higher flow rates even though the HST is lower.

<table>
<thead>
<tr>
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<th>P26</th>
<th>P27</th>
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<tr>
<td>Non-uniform Power (W)</td>
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<td>60</td>
<td>76</td>
<td>95</td>
<td>111</td>
<td>132</td>
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</table>
Experimental Study of Transformer Liquid Flow and Temperature Distribution

TABLE II TEMPERATURE RISES TO CALCULATE H UNDER UNIFORM LOSSES

<table>
<thead>
<tr>
<th>FR (lpm)</th>
<th>$\theta_a$ (°C)</th>
<th>$\Delta \theta_a$ (°C)</th>
<th>$\Delta \theta_b$ (°C)</th>
<th>$\Delta \theta_{in}$ (°C)</th>
<th>$\Delta \theta_{out}$ (°C)</th>
<th>g (°C)</th>
<th>HSF</th>
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</tbody>
</table>

TABLE III TEMPERATURE RISES TO CALCULATE H UNDER NONUNIFORM LOSSES

<table>
<thead>
<tr>
<th>FR (lpm)</th>
<th>$\theta_a$ (°C)</th>
<th>$\Delta \theta_a$ (°C)</th>
<th>$\Delta \theta_b$ (°C)</th>
<th>$\Delta \theta_{in}$ (°C)</th>
<th>$\Delta \theta_{out}$ (°C)</th>
<th>g (°C)</th>
<th>HSF</th>
<th>S</th>
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</table>

Fig. 4 Average winding model temperature profiles under uniform losses

Nonuniform losses were applied and fixed to a total of 1724 W, as shown in Table I and Fig. 5, under same range of inlet flow rates. Winding model temperature profiles are shown in Fig. 6 and the measured temperature rises are summarized in Table III. The $S$ factor was assumed equal to that found in the previous uniform loss case, shown in Table II, since the winding model geometries and the tested range of inlet flow rates are the same.

In both uniform and nonuniform cases, the bottom oil temperatures are almost similar, within 4 °C differences, so oil viscosities at winding inlet can be considered similar for each flow rate between the two cases. It can be observed that the temperature profile is more distorted at lower flow rates. On the contrary of the uniform loss case, the $H$ is lower at higher flow rates since the temperature profile is less distorted and so the HST rise is closer to $\Delta \theta_{out}$. Despite the fact that nonuniform losses are fixed for all flow rates, the $Q$ factor decreased with the increase of inlet flow rate.

Fig. 6 Average winding model temperature profiles under nonuniform losses

IV. DISCUSSION

The HST is affected by three contributors which are the oil flow rates within horizontal cooling ducts, the oil flow distribution within each pass and the axial and radial loss distributions within the winding. The $H$ is used to empirically compensate for the lack of knowledge of the three contributors by using accumulated experience and the data from heat run tests. The $H$ is split into the $Q$ and $S$ factors. The $S$ factor is affected by uniformity of flow distribution within the winding. The uniformity of oil distribution depends greatly on winding geometries and winding inlet oil flow rates. In the uniform loss tests, it was observed that higher inlet flow rates created more distorted temperature profiles. The $Q$ factor was fixed and set to 1.0 since all the plates had the same power injection. The $H$ was calculated from tests data shown in Table II and the $S$ factor was then derived from the $H$ and the known $Q$ factor. It was observed that higher flow rates created higher $H$ despite the fact that the HST is lower as can be seen by comparing temperature rises for FR = 3 lpm and FR = 18 lpm. This indicates that higher $H$ does not necessarily correspond to higher HST as can be seen in the uniform loss case.
For the nonuniform loss case, similar flow rates and inlet oil temperatures to those of the uniform loss case were reached. This justifies the assumption that the $S$ factor from the uniform loss case can be used in the $H$ calculation in the nonuniform loss case. The nonuniform losses were fixed for all inlet flow rates and it was observed that higher flow rates lead to lower $H$ and lower HST. Even though the loss distribution is fixed, it was observed that the $Q$ factor is lower for higher inlet flow rates; indicating that the $S$ and $Q$ factors are interdependent.

Inlet flow rate affects the uniformity of the developed temperature profile. Under uniform loss distribution, it can be observed that higher flow rates create more distorted temperature profiles and lower the HST, comparing FR = 3 lpm and FR = 18 lpm in Table II. On the contrary, for nonuniform loss distribution higher flow rates created more uniform temperature profiles and lower HST as well as lower $H$. From this observation, it can be concluded that the $H$ is affected by the uniformity of winding temperature profile which is shaped by both the oil flow distribution and the loss distribution.

Disc segments were modelled in this work using rectangular aluminium plates without paper. This simplification affects both the average winding temperature and the HST. For wrapped composite discs, both the HST and the average winding temperature would be higher; this would make the calculated $H$ slightly lower, since $g$ would be higher.

V. CONCLUSION

In this paper, an experimental setup was used to conduct artificial heat run tests on a disc type winding model. All temperature rises required to calculate the Hot Spot Factor ($H$) were recorded using thermocouples. The inlet flow rate was changed from 3 lpm to 18 lpm and both uniform and nonuniform power injection were applied in order to decouple the $H$ into its two defined $S$ and $Q$ components by the IEC 60076-2-3:2011 standard. It was demonstrated that the $S$ and $Q$ factors are interdependent and cannot be decoupled. It was observed that the $H$ is affected by inlet flow rate. Under uniform losses, the $H$ factor increased with increasing flow rate though the HST is lower at higher flow rates. On the other hand, it was observed that the $H$ factor decreased with increasing flow rates for nonuniform losses and so was the HST.

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Appendix 4 Influence of Winding Geometrical Deviations on the Hot Spot Temperature in Disc Type Power Transformers

In this paper, experience accumulated during the assembly of a disc type winding model indicated that disc segments inside the winding if misaligned, within 1 to 2 mm range, they would influence both temperature and oil flow distributions. If the misalignment happens to exist near the hot spot temperature location, the hot spot temperature may either be increased or reduced. On top of that, the existence of the geometrical deviations may mask the influence of higher vertical duct width or higher number of disc segments per pass. The paper does raise awareness of the influence of these geometrical deviations so that future researchers and perhaps transformer manufactures would pay more attention when assembling winding models or winding structures especially near the expected location of the hot spot temperature.

INFLUENCE OF WINDING GEOMETRICAL DEVIATIONS ON THE HOT SPOT TEMPERATURE IN A DISC TYPE POWER TRANSFORMER

M. Daghrarah¹, Z.D. Wang¹*, Q. Liu¹, Ch. Krause², P. Dyer³ and P. Jarman⁴
¹The University of Manchester, M13 9PL, UK
²Weidmann Electrical Technology AG, Rapperswil, CH 8640
³UK Power Networks, Crawley, RH10 0FL, UK
⁴National Grid, Warwick, CV34 6DA, UK
*Email: zhongdong.wang@manchester.ac.uk

Abstract: Loss distribution and oil flow distribution within a transformer winding influence the location and value of the hot spot temperature. Oil flow distribution is affected by the winding geometries, and deviations in the cooling duct dimensions influence the hot spot temperature. Deviations can be created during transformer assembly by discs protruding into axial cooling duct or by insulating paper purging and so narrowing the radial cooling duct. In this work, an experimental investigation was carried out to highlight the influence of geometrical deviations on the hot spot temperature near the diverting washers which could adversely affect the hot spot temperature. A disc type winding model was used to create three cases of geometrical deviations. The temperature within the winding model was measured using thermocouples while oil flow rate in cooling ducts was measured using particle image velocimetry. Case 1 highlights the influence of geometrical deviations on oil flow and temperature distributions in a 15 disc segments per pass arrangement in which protrusions of winding discs into the axial cooling duct influenced the oil flow distribution and hence affected the hot spot temperature. Case 2 shows a 5-pass arrangement, 6-disc segments per pass, in which the influence of geometrical deviations on the temperature profiles of pass 3 and pass 4 is highlighted which caused the hot spot temperature in pass 4 to be around 5 °C higher than that in pass 3. Finally, case 3 shows a comparison between two winding model assemblies which have the same winding geometries but different geometrical deviations. Overall it is shown that geometrical deviations of discs, or washers, into axial cooling ducts may increase the hot spot temperature by several degrees. Such geometrical deviations should be minimised during the transformer manufacturing process.

1. INTRODUCTION

The hot spot temperature (HST) defined as the hottest point in a transformer winding is of interest to study since the transformer paper ageing rate depends on it [1-3]. Disc-type power transformers are used in transmission systems and are essential for the integrity of the electricity grid. Unexpected failure of a major power transformer could have serious economic and security consequences. Hence it is essential to estimate expected remaining life of power transformers. Thermal modelling is used to predict the HST of a transformer under a specific design and loading conditions. Under oil directed and cooled (OD) disc type power transformers, the oil is pumped and guided inside the winding structure in a zig-zag fashion to enhance the cooling performance. Different modelling approaches were presented in literature such as the network models [4-6] and the computational fluid dynamic based models [7, 8], to estimate the HST, [6, 9]. The major geometries that influence the flow distribution within the winding structure are the radial cooling duct height, the axial cooling duct width [10], and the number of discs per pass. It was reported in [10, 11] that lower ratio between the axial cooling duct height to the axial cooling duct width would result in a more uniform flow distribution. It was also reported that higher discs per pass would result in a more distorted flow distribution and oil reverse flow in the pass bottom cooling ducts are more likely to occur [11].

Experimental verifications of the simulation models are required to enhance model understanding. A lab-scale winding model was used in [12] to record oil flow rates within cooling ducts using hot wire anemometry under isothermal conditions. Laser doppler velocimetry was used in [13] to record oil flow rates within cooling ducts of a small scale winding model and to compare the measurement with simulation predictions. In the 2D modelling, the winding geometries used are precise as given by the transformer design specifications. In practice, however, this level of precision cannot be guaranteed during transformer assembly due to manufacturing tolerance, henceforth called geometrical deviations. It is of importance to check whether these geometrical deviations have an impact on the HST. The geometrical deviations are considered a local effect which can either increase or reduce the disc temperatures near the deviations.
In this work, an experimental setup is used to show the influence of certain geometrical deviations on the HST and oil flow distribution in a disc type winding model. Particle image velocimetry (PIV) is used to record oil flow rates within the winding model while temperature is recorded using thermocouples. Measurements and results repeatability are first presented for both flow and temperature measurements under fixed geometry. Three cases, each with different geometrical deviations, are then defined and results are discussed to highlight the influence of geometrical deviations on both the oil flow and temperature distributions.

2. EXPERIMENT SETUP

The experimental setup is designed to enable an extensive study to be conducted under OD cooling mode. The setup consists of a disc type winding model, a pump, a flow meter external to the winding model, and a radiator to cool the oil down. The inlet flow meter is a positive displacement type with specified accuracy of 0.5% of reading. The winding model is imbedded with aluminium plates to model the disc segments. Each aluminium plate is heated using two resistive heaters. Variable transformers are used to apply the desired voltage which allows an accurate power injection in each plate with ± 1 W accuracy at testing conditions. K type thermocouples are used to record each plate average temperature, bottom oil temperature ($T_{\text{bottom}}$), and top oil temperature ($T_{\text{top}}$). All used thermocouples were calibrated and temperature measurement accuracy is within ± 1 °C. A PIV system is used to record oil flow rates within radial cooling ducts.

The PIV system consists of a camera, a laser source, laser sheet optics, a synchronizer and a software. The experimental setup and the PIV system were introduced in detail in [14, 15]. A schematic diagram of the experimental setup is shown in Figure 1.

2.1 WINDING MODEL ASSEMBLY

The winding model is designed to provide flexibility when changing geometrical parameters of winding structure, such as axial cooling duct width and the number of plates per pass. Polycarbonate material is used to construct walls and guides of the winding model. As shown in Figure 2, side walls are milled and grooves are made to host the washers and the plates. Two reciprocal side walls are placed and held with a supporting structure. Washers and plates are then inserted in the winding model. The resistive heaters are then inserted into the plates, through holes in one side wall, along with the thermocouples. At this stage, plates should be aligned manually before any sealing. Silicone sealant is used to seal the holes in the winding model for both the heating element wires and for the thermocouples. Once the sealant is cured, the plates are fixed in their position. Geometrical deviations occur in the winding model if a plate or a washer is slightly misaligned from its desired position. Silicone sealant is used to seal the washers near the blocking end. Adding extra sealant would cause the washer to protrude into the winding model axial cooling duct which creates a geometrical deviation. The winding model is made such that the radial cooling duct is 4 mm and the axial cooling duct is 10 mm. The total number of plates in the winding model is 30 divided into passes as desired.

2.2 MEASUREMENT REPEATABILITY

To separate the effect of geometrical deviations from measurement uncertainty, a test was conducted three times on three different days. The test was conducted under fixed winding model geometries of 4 mm radial cooling duct height, 10 mm axial cooling ducts width, and 10 plates per pass. The power injection in each plate ($P_{\text{loss}}$) is 50 W/plate, equivalent to 2010 W/m², and inlet oil flow velocity ($V_{\text{in}}$) set to 0.2 m/s. $T_{\text{bottom}}$ was 44 °C at steady state. Figure 3 shows both measured temperature and oil flow distributions in the third pass. Third pass ducts are numbered from duct 1 to duct 11 from bottom to top of the pass as shown in Figure 2. Likewise, third pass plates are numbered from plate 1 to plate 10 from bottom to top of the pass. Repeatability of temperature and oil flow rates is discussed as follows.
Temperature measurement repeatability: Temperature measurement uncertainty is caused by the thermocouple measurement uncertainty which is within ± 1 °C and by the uncertainty of power injection in each plate which is within ± 1 W. From the temperature profile shown in Figure 3, the temperature measurement repeatability is identified to be within ± 1 °C.

Oil flow rate measurement repeatability: The PIV measurement resolution is within 1 mm/s; nonetheless, the PIV measurement repeatability depends on the accurate positioning of both the laser source and the camera field of view and on the good quality of captured PIV raw images. Error caused by image calibration was investigated and it is within ± 1% of maximum duct velocity. From Figure 3, the maximum deviation occurs in duct 11 with 16 mm/s discrepancies. Measurements in duct 1, duct 2, and duct 8 have maximum discrepancies of 13 mm/s while the rest of the ducts have measurement discrepancies within 8 mm/s. A contributing factor to PIV measurement uncertainty is the internal circulation of oil in most top and most bottom cooling ducts at higher operating $V_{in}$ and $T_{bottom}$ as described in [15].

Figure 3: Measurement repeatability defined with a test repeated three times under $V_{in} = 0.2$ m/s and $P_{loss} = 2010$ W/m². Temperature measurement accuracy is +/- 1 °C whereas PIV measurement accuracy can be assumed 13 mm/s.

3. RESULTS

Three cases are conducted to show the influence of geometrical deviations on measured temperature and oil flow distributions within the winding model as discussed individually in the following.

3.1 CASE 1

Case 1 was conducted under winding model geometries of 15 plates per pass and a total of 2 passes in the winding model. Radial cooling duct height was 4 mm and axial cooling duct width was 10 mm. $T_{bottom}$ was maintained at 70 °C while $V_{in}$ was set at 0.3 m/s. $P_{loss}$ was fixed to 2010 W/m² in each plate. The geometrical deviations of pass 2, the top pass, demonstrated in Figure 4 where Plate 3 is protruding by 2 mm toward the axial cooling duct.

The temperature and the oil flow distributions are shown in Figure 5. It is expected under OD cooling mode that the temperature profile is monotonically decreasing under uniform loss injection whereas the oil flow distribution is monotonically increasing from bottom to top of the pass [10]. It can be observed on the oil flow distribution that because Plate 3 protrudes towards the axial cooling duct, duct 4 suffers and the oil is forced to flow more into the first three ducts, duct 1 to duct 3. Higher oil
flow rates into the first three ducts enhance the cooling performance of the first three plates. This is more apparent on the temperature distribution in which the temperatures of Plate 2 and Plate 3 are near 4°C lower than that of Plate 1.

Figure 4: Case 1 showing geometrical deviations near Plate 3, highlighted by zoomed area where plate 3 protrudes towards the axial cooling duct.

Figure 5: Temperature and flow distribution profiles related to case 1 in which Plate 3 protrudes towards the axial cooling duct which caused more oil to flow in first three ducts and hence shaping the temperature profile.

3.2 CASE 2

Case 2 was conducted under winding model geometries of 6 plates per pass and a total of 5 passes. Radial cooling duct height is 4 mm and axial cooling ducts widths are 10 mm each. Case 2 was conducted under $T_{\text{bottom}} = 70 \, ^\circ\text{C}$, $V_{\text{in}} = 0.3 \, \text{m/s}$, and $P_{\text{loss}} = 2010 \, \text{W/m}^2$.

By looking at Pass 3 on Figure 6, it can be noted that Washer 3 is aligned properly while Plate 13 protrudes toward the axial cooling duct. This position allows more oil to flow into duct 1 of Pass 3 and hence enhances the cooling of Plate 13, which host the pass HST, as can be seen by the temperature profile in Figure 7. This is a similar condition as Plate 3 in case 1. In contrast, looking at Washer 4 in Pass 4 on Figure 6, Washer 4 protrudes towards the axial cooling duct causing duct 1 of Pass 4 to suffer less oil flow. This in effect increases Plate 19 temperature. In theory, the difference between Plate 19 and Plate 13 should not exceed 2°C, based on $V_{\text{in}}$ and $P_{\text{loss}}$ as shown by the dashed line on Figure 7. However, the temperature difference between Plate 19 and Plate 13 is almost 5°C due to the geometrical deviations near the washers.

Figure 6: Case 2 geometrical deviations where in pass 3, Plate 13 protrudes toward the axial cooling duct causing more oil to flow in duct 1 of the pass. On the contrary, Washer 4 in pass 4 protrudes toward the axial cooling duct causing less oil to flow in duct 1 of the pass.

3.3 CASE 3

Case 3 is conducted under 10 plates per pass, Figure 8, and cooling ducts dimensions are the same as in the previous cases. Case 3 was conducted under $T_{\text{bottom}} = 70 \, ^\circ\text{C}$, $V_{\text{in}} = 0.3 \, \text{m/s}$ and $P_{\text{loss}} = 2010 \, \text{W/m}^2$. Two separate assemblies were performed of the winding model under the same dimensions in which Assembly A had a different geometrical deviation than Assembly B. Assembly A had the third pass bottom washer protruding towards the axial cooling duct while Assembly B had the same washer aligned with the plates as shown in Figure 8.
Experimental Study of Transformer Liquid Flow and Temperature Distribution

Figure 7: Temperature profile related to case 2 testing conditions. Temperatures of Pass 3 and Pass 4 were affected by geometrical deviations near corresponding pass bottom washers.

Both temperature and oil flow distribution were measured in the third pass and are shown in Figure 9. In Assembly A, the geometrical deviations near the bottom washer caused the oil flow distribution in duct 1 to suffer and hence oil reverse flow occurred in the duct. Under oil reverse flow, the relatively hotter oil in duct 2 flows back into duct 1 which in effect raises the temperature of plate 1. Under Assembly B, the bottom washer was aligned with the plates and no reverse flow occurred though the oil flow velocity in duct 1 is relatively low. The HST difference between Assembly A and Assembly B is near 3 °C.

Figure 8: Assembly B had the pass bottom washer aligned properly whereas the washer protrudes towards the axial cooling duct in Assembly A causing oil reverse flow to occur in duct 1.

Figure 9: Case 3 temperature and oil flow distribution in which Assembly A had the third pass bottom washer protruding into the axial duct causing reverse flow to occur and hence increasing the pass HST

4. DISCUSSION

Under OD cooling modes, an emphasis is placed on the proper selection of cooling duct dimensions to optimize the thermal performance of a power transformer. The oil flow distribution can be classified as a global factor which influences the HST. The HST, however, is a local phenomenon which is affected by the losses in discs and the local heat transfer. During transformer winding assembly, geometrical deviations should be controlled as they may either increase or reduce the HST. Geometrical deviations may influence the oil flow distribution within a pass. It was observed in conducted tests that if the pass bottom washer protrudes towards the axial cooling duct, less oil flows into the first cooling duct. Under case 3, the protrusion of the bottom washer caused the oil to reverse its direction. The protrusion of the pass bottom washer would have stronger influence on both temperature and oil flow distribution for narrower axial cooling ducts. Thus, it is recommended that particular attention is paid during transformer winding assembly when reaching the top passes of the winding where the HST is expected to be.
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5. CONCLUSIONS

In this work, an experimental setup was used to study the influence of winding geometrical deviations on the HST in a disc-type transformer winding under oil directed cooling mode. The experimental setup consisted of a winding model, a pump, a radiator and an inlet flow meter. Particle image velocimetry was used to measure oil flow rates within the cooling ducts while thermocouples were used to measure the temperature within the winding structure. Three cases were presented to highlight the possible influence of geometrical deviations on the oil flow distributions, the temperature distributions, and the HST. It was observed that when a washer or a plate, which represents a disc, protrudes slightly towards the axial cooling duct, the duct above the washer, or the plate, suffers less oil flows into it and so the plate temperature near that duct increases.

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