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A case study on thrust bearing failures at the SÃO SIMÃO hydroelectric power plant

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ABSTRACT

After twenty years without any apparent problems on their combined guide and thrust bearings, the six 280 MW hydrogenerators of the São Simão Hydroelectric Power Plant were failing. The source of the failure was the melting of the thrust pad babbitt lining. The machines began showing performance failures, leading to a sudden interruption in their operation. This caused considerable losses with high direct and indirect costs. The solution proposed by the bearing manufacturer was an improvement in the bearing design and the installation of new water–oil heat exchangers. The direct cost of their solution was estimated to be US $2,400,000.00. In a search for a less expensive alternative, CEMIG started a parallel study focused on the heat exchangers. A methodology based on heat transfer was applied, indicating that an increase in the heat exchange surface area could solve the problem. A third heat exchanger was added in one machine that already possessed two. The results fulfilled the preliminary predictions, eliminating the risk of additional babbitt lining failures. As a consequence of this success modeling, heat exchangers were replaced by stainless steel plate ones in all machines. This alternative solution had a total direct cost of US $600,000.00.

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1. Introduction

The problem first appeared in the Unit 1 of the São Simão Hydroelectric Power Plant was due to melting of the babbitt pad linings and led to an emergency interruption on the machine operation, which was kept out of service until the pads were repaired. Failures were associated with high temperatures in the pads, up to 84 °C on summer time, when the temperature of cooling water can reach 30.8 °C. A similar problem had occurred in the other five machines of the plant [1]. CEMIG contacted the bearing manufacturer to evaluate and propose solutions for the issue. After lengthy testings, they recommended an improvement on bearing design and replacement of water–oil heat exchangers for a greater exchange capacity ones. But the high cost involved an average of US $400,000.00 (four hundred thousand dollars) per machine, led CEMIG to search for a less expensive alternative.

In an attempt to avoid new failures and subsequent interruptions in the machines' operation, the company was forced to limit its generators output in the summer until the problem was solved.

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Nomenclature

A	surface area, m^2
c	specific heat capacity, kJ $°C^{-1}$ kg^{-1}
$LMTD$	log mean temperature difference, °C
p	pressure, kPa
$\dot{Q}$	heat transfer, kW
ρ	density, kg m^{-3}
$\overline{T}$	average temperature, °C
T	temperature, °C
U	overall coefficient of heat transfer, kW $°C^{-1}$ m^{-2}
U*	Modified overall coefficient of heat transfer (UA), kW $°C^{-1}$
$\dot{V}$	flux, m^3 s^{-1}
ν	viscosity, m^2 s^{-1}
$\dot{W}$	power, kW

Subscripts

1,2,3,4 and 5	see Figs. 2 and 3
b	bearing
e	heat exchanger
o	oil
p	pump
s	thrust pads
w	water

1.1. A new point of view

While searching for a simpler and cheaper alternative for the problem at hand and one that could be implemented in the shortest amount of time, the research targeted the water–oil heat exchangers as the best solution. An increase in the exchange surface area would decrease the temperature of the oil flowing out of the heat exchangers, providing the necessary decrease in the thrust pad temperature. Making use of an existent spare heat exchanger, it was considered adding it to Unit 1, with the oil circuit in series and water in parallel. Fig. 1 shows the original configuration of the oil cooling system, in which a third heat exchanger was placed.

1.2. Model

To determine the consequences of installing a third heat exchanger, a semi-empirical model was developed, based on values measured in Unit no. 1 with the original configuration set (two heat exchangers), as can be seen in Table 1. Pipes, pumps and heat exchangers external surfaces were considered adiabatic.

Due to the low temperature of water during testing the thrust pad average temperature reached only 77.3 °C, with a highest value of 78.5 °C in pad no. 10.

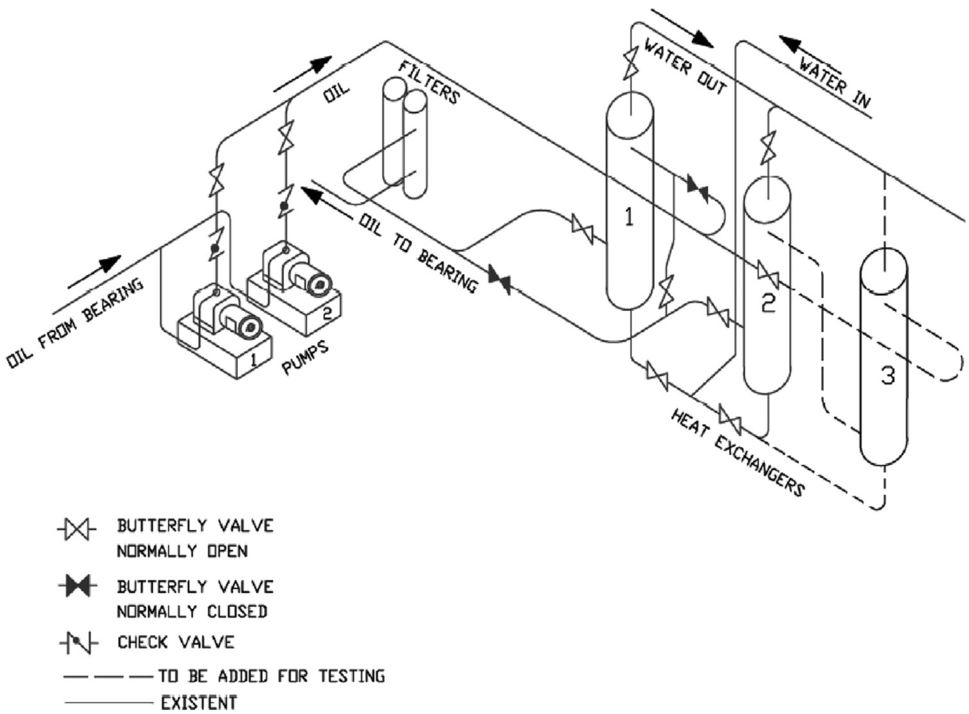

Fig. 1. Combined bearing oil refrigeration system.

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