Empirical Model of Bearing Temperature Saturation at Asahan II Hydro-Turbine-Generator

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Abstrak--*Kejenuhan suhu bantalan adalah peristiwa umum yang diamati selama pengujian tanpa-beban dan beban-penuh pada turbin-generator-hidro PLTA Asahan II. Walaupun melibatkan mekanisme perpindahan kalor yang kompleks, sebuah model sederhana dapat dibangun untuk meramal perubahan suhu bantalan terhadap waktu. Tingkat kepercayaan atas bentuk umum dari model tersebut cukup tinggi karena divalidasi dengan data yang melimpah yang dikumpulkan selama 35 tahun ke belakang. Investigasi lanjutan juga dilakukan untuk membandingkan kejenuhan suhu pada berbagai beban, laju rotasi, dan kinerja pendingin yang berbeda.*

Kata kunci: *Bantalan, kejenuhan suhu, perpindahan kalor, pemodelan*

Abstrac--*Bearing temperature saturation is a common phenomenon observed during no-load run and full-load tests of Asahan II hydro-turbine-generators. Despite complex heat transfer mechanism involved behind it, simple model can be constructed to predict bearing temperature change with time. Confidence on the general form of such model is rather high because it is validated using abundant data collected during the past 35 years. Further investigations are also made to compare temperature saturations at different load, rotational speed, and performance of cooling coil.*

Keywords: *Bearing, temperature saturation, heat transfer, modeling*

1. INTRODUCTION

Asahan II hydroelectric power plant consists of two power stations: Siguragura (4×79.4MVA) and Tangga (4×88MVA). Each generating unit on both power stations uses vertical shaft Francis turbine and vertical shaft semi-umbrella synchronous generator. To permit smooth rotary motion of vertical shafts, four bearings are employed: upper guide bearing, lower guide bearing, turbine guide bearing, and thrust bearing—the latter also acts to support axial load of vertical shafts. All bearings are self-oillubricated, Babbitt-lined, segmental structure, except upper guide bearing at Siguragura which is one ring structure.

Upper and turbine guide bearings are located in their own oil reservoir, while thrust and lower guide bearings are located together in one oil reservoir. Heat from the bearings is effectively removed by open-circulation, water-cooled cooling coils provided in each oil reservoir. Bearing temperature monitoring is available through RTDs and dial thermometers installed on several segments or locations of each bearing.

Bearing temperature shall not exceed 65°C under 30°C cooling water [1]. However, bearing temperature may rise beyond this value and the causes are traditionally associated with declining performance of cooling coil, shaft-misalignment, or axial load unbalance. Arising from this fact, bearing temperature recording has become standardized test performed in maintenance of each hydro-turbine-generator to confirm the performance of cooling coil, shaft-alignment, and axial load balance. Based mainly on Japanese standards, test of bearing temperature is carried in two approaches: no-load run and full-load, with full-load test is done twice, before and after maintenance work. Full-load test is similar to duty type S1 of IEC 60034-1 [2].

A typical test of bearing temperature may last from two until five hours, depending on how fast temperature saturation is. Bearing temperature at rated operating conditions is said to be saturated whenever its rate of change had decreased to practically 0.5°C per hour. This requirement is stricter compared to other international standards, which is maximum 1°C per 30 minutes in [3] or less than 2K per hour in [2]. With the main interest is to know the final saturated bearing temperature, lengthy test duration is unavoidable.

This paper aims to develop model to predict final saturated bearing temperature as well as to understand its significance, thus providing faster and deeper analysis of bearing system performance.

2. THEORETICAL REVIEW AND RESEARCH METHODOLOGY

Heat is generated by dynamic friction between lubricated contacts of bearing and shaft. This heat is then distributed mainly to metal bodies (bearing, shaft, and metal support) via conduction, lubricating oil via forced convection caused by centrifugal pumping action of the shaft [1] and later to watercooled cooling coil, also via forced convection.

To know temperature distribution in space and temperature rate of change of a system, heat transfer analysis shall be performed. Regardless of the method used, difficulties in bearing heat transfer analysis arise from many factors, including discontinuities of bearing structure [4], material inhomogeneity, variation of thermal properties of material with temperature, etc. Despite such difficulties, under some proper assumptions, classical method of heat transfer analysis, i.e. by finding analytical solution of a system of differential equations, could yield a very good insight on actual heat transfer phenomenon.

Let us begin by formulating heat balance of bearing system,

$$Q_f - Q_b - Q_l = 0 \quad (1)$$

where Q_f, Q_b, and Q_l are heat generated by friction, heat conducted to bearing, and heat carried away by lubricating oil, respectively. Heat conducted to other metal parts other than bearing is considered negligible. In dealing with Q_b, it will greatly simplify the analysis by noticing that lumped-capacity method is applicable to our situation, in which the bearing is assumed to have almost uniform internal temperature, T_b [5]. Meanwhile, Newton's Law of Cooling will be used when dealing with Q_l. Thus, the former equation (1) can be expanded as follow

$$Q_f - m_b c_b \frac{d(T_b - T_i)}{dt} - hA(T_b - T_c) = 0 \quad (2)$$

where m_b is bearing mass, c_b is bearing specific heat capacity, T_i is bearing initial temperature, h is average film coefficient over the lubricated surface of bearing, A is bearing surface area which depends on whether the bearing is fully immersed or half-immersed in lubricating oil, and T_c is average temperature of lubricating oil in contact with cooling coil.

The solution to equation (2) is general formula for bearing temperature variation with time as follow

$$T_b(t) = \left(\frac{Q_f}{hA} + T_c - T_i\right)\left[1 - \exp\left(-\frac{hA}{m_b c_b} t\right)\right] + T_i \quad (3)$$

According to [6], this so-called zero-order model is very good and seems to capture the behavior of the empirical temperature data very well considering the simplicity of the model. Other more precise models are available as in [7, 8, 9] which include additional parameter to describe heat loss previously neglected in our zero-order model. However, this obviously will lead to more complicated calculation. To keep calculation to minimum, while in the same time maintaining adequate accuracy, zero-order model will be used.

To make our next analysis easier, equation (3) need to be rewritten in a more compact form. This can be done by introducing a time constant,

$$\tau = \frac{m_b c_b}{hA} \quad (4)$$

and also by noticing that bearing temperature reaches equilibrium state, T_e, as t becomes very large,

$$T_e = T_b(\infty) = \frac{Q_f}{hA} + T_c \quad (5)$$

Finally, equation (3) becomes

$$T_b(t) = (T_e - T_i)\left[1 - \exp\left(-\frac{t}{\tau}\right)\right] + T_i \quad (6)$$

Equation (6) above has two unknown quantities. The first one is time constant τ. Referring to equation (4), it contains h, which is not readily available and very hard to determine [5]. Next unknown quantity is equilibrium temperature T_e, which, referring to equation (5), contains Q_f.

Despite this disadvantage, we can still obtain a numerical description of bearing temperature formula. The strategy is to switch from pure theoretical investigation to numerical calculation using empirical data by nonlinear least squares fitting. Generally, this procedure aims to find the best-fitting curve to a given set of points by evaluating the sum of the squares of the offsets of the points from the curve, known as R^2 [10]. In our case, given paired dataset of time and bearing temperature,

$$t_j = \{t_1, t_2, \cdots, t_n\}$$
$$T_{bj} = \{T_{b1}, T_{b2}, \cdots, T_{bn}\}$$

the objective is to find τ and T_b such that R^2 becomes minimum, where

$$R^2 = \sum_{j=1}^{n}\left\{T_{bj} - T_{b1} - (T_e - T_{b1})\left[1 - \exp\left(-\frac{t_j}{\tau}\right)\right]\right\}^2 \quad (7)$$

In this paper, Solver add-in of Microsoft Excel ® will be used to solve the above optimization problem of equation (7) for data gathered since 1983 until 2018, with the general procedure is described in [11]. Statistical correlation between given data and calculated model is evaluated using Pearson correlation coefficient and justified by visual examination.

However, the above procedure will be meaningless if either τ or Te turned out to be non-constant. Fortunately, this is not the case because of following reasons,

a. h is independent of temperature difference in forced convection, as it is influenced by flow dynamics and thermal conductivity of fluid in use [5].
b. Although dynamic friction between lubricated contacts of ok2bearing and shaft is highly nonlinear and depends on many parameters [12], it is sufficient to assume that Q_f is constant under steady rotational speed and friction force.

3. RESULTS AND DISCUSSION

3.1 Statistical correlation between given data and calculated model

Calculated model as of equation (6) for all types of bearing has Pearson correlation coefficient as high as 0.991 and 0.989 on average to given data for Siguragura Power Station and Tangga Power

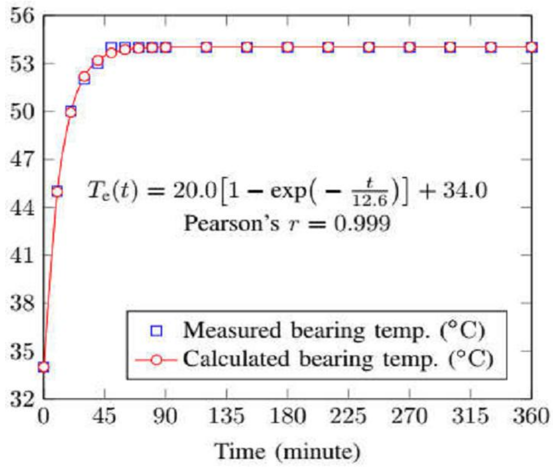

Figure 1. Verification of model for lower bearing at Tangga Power Station no. 1 (1983)

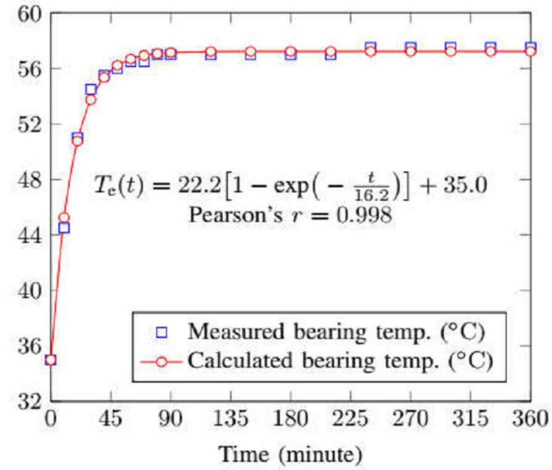

Figure 2. Verification of model for thrust bearing at Tangga Power Station no. 2 (1983)

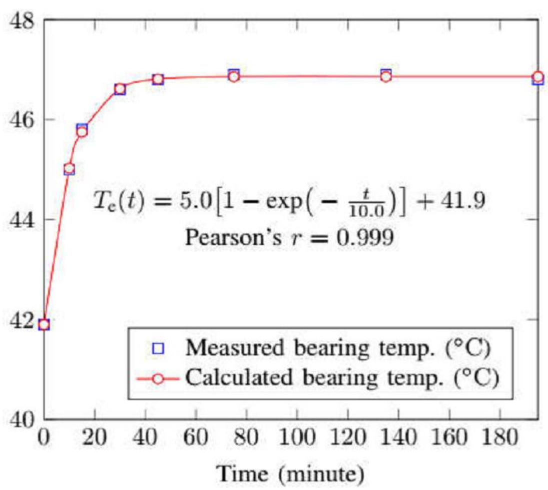

Figure 3. Verification of model for upper earing at Siguragura Power Station no. 1 (2016)

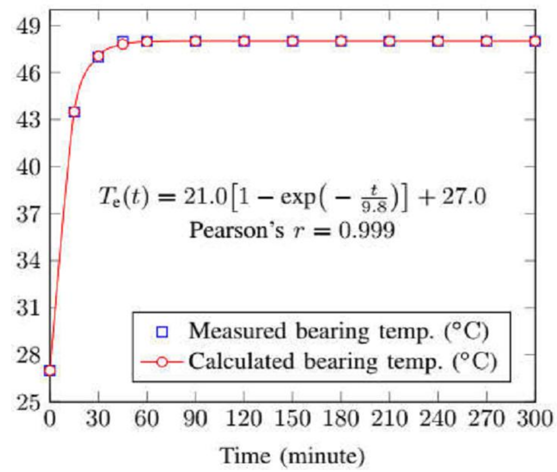

Figure 4. Verification of model for turbine bearing at Siguragura Power Station no. 2 (1982)

Station, respectively. Figure 1 until figure 4 show visual examination of sample models.

High statistical correlation between given data and calculated model suggests that equation (6) indeed offers excellent model for bearing temperature variation with time.

3.2 T_e and τ from data with different sensor

Equation (4) and (5) suggest that equation (6) should have fixed coefficients if Q_f, h, and T_c are fixed. On the other hand, different sensors would not show same readings of bearing temperature. Despite those sensors are attached to same equipment and expected to experience same condition for Q_f, h, and T_c, they will not produce same model for bearing temperature variation. Table 1 shows such differences for the case of rated-load, rated speed condition of equipment under test.

Following factors may contribute to observed discrepancies among models from different sensors:

a. As explained earlier in section 1, each sensor is installed on different location of bearing segment. Non-uniform average gap between that bearing-segment and corresponding shaft may have influence on Q_f.
b. Different measurement principle between fluid-filled dial thermometer and temperature dependent resistance of RTD sensor may have influence on apparent value of h.

3.3 T_e and τ at different load condition

It is very natural to suspect that loading of hydro-turbine-generator has influence on Q_f, hence also T_e. Particularly, higher load will manifest in higher hydraulic thrust, which is proportional to frictional force on thrust bearing. Table 2 shows that rated-load values of T_e for thrust bearing are almost always greater than the no-load ones. This is in accordance with the fact that thrust bearing acts to support axial load of vertical shafts. Similar pattern seemingly does not occur for other bearings. However, referring to equation (4),

Table 1. Calculated T_e and τ based on data from dial thermometer and RTD

Unit ID[a] [Year]	Upper Bearing		Lower Bearing		Thrust Bearing		Turbine Bearing	
	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ
TNP3 [1983]	51.5[b]	12.7	49.2	17.7	63.9	15.5	55.4	12.1
	(52.0)[c]	(10.4)	(53.0)	(13.6)	(60.9)	(13.7)	(52.7)	(23.7)
TNP4 [1983]	50.6	23.1	50.5	20.5	59.0	15.4	53.5	14.5
	(49.8)	(23.1)	(54.5)	(14.0)	(56.0)	(14.4)	(53.2)	(14.7)
SGP3 [1982]	49.0	26.1	53.9	8.8	59.8	6.3	50.0	19.7
	(51.7)	(11.9)	(52.1)	(14.0)	(62.1)	(6.8)	(50.6)	(15.3)
SGP2 [1982]	53.0	9.0	54.9	18.9	59.1	27.5	53.0	27.6
	(55.9)	(25.9)	(55.1)	(24.0)	(60.0)	(12.1)	(54.1)	(22.1)

[a] Unit ID uses the following format: TNPx for Tangga Power Station and SGPx for Siguragura Power Station, with "x" represents unit number.

[b] Based on data from RTD.

[c] Based on data from dial thermometer.

Table 2. Calculated T_e and τ at no-load and rated-load

Unit ID[a] [Year]	Upper Bearing		Lower Bearing		Thrust Bearing		Turbine Bearing	
	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ
TNP1 [1983]	50.9[b]	23.2	56.0	16.2	56.4	17.7	54.4	13.4
	(50.1)[c]	(16.3)	(54.0)	(12.6)	(58.0)	(13.0)	(52.9)	(12.3)
TNP2 [2018]	43.7	19.8	48.2	7.4	54.0	7.6	45.3	10.9
	(44.9)	(18.4)	(50.7)	(18.3)	(58.6)	(10.3)	(45.2)	(12.8)
SGP1 [1982]	54.9	14.3	50.0	4.0	56.5	13.5	49.3	12.0
	(55.4)	(11.2)	(50.0)	(10.6)	(56.5)	(7.6)	(47.9)	(10.2)
SGP2 [2016]	50.1	13.1	52.0	12.7	57.0	9.9	45.9[d]	14.7[d]
	(50.6)	(5.5)	(52.5)	(3.4)	(58.0)	(3.7)	(46.0)[d]	(4.9)[d]

[a] see note (a) in Table 1.

[c] Rated-load, rated speed condition case.

[d] Based on data from RTD. All other values of T_e and τ are based on data from dial thermometer.

there should be no obvious direct explanation regarding effect of loading on τ, contrary to result in table 2. Further investigation shall be made on this case.

3.4 T_e and τ at different rotational speed

As demonstrated in table 3 below, rotational speed affects Q_f, in which higher rotational speed is followed with higher T_e. Rotational speed may also affect flow dynamics of cooling fluid; hence also affect h and τ. However, there is no general patterns observed in table 3b regarding effect of rotational speed on τ. Further investigation shall be made on this case.

Table 3. Calculated T_e and τ at various rotational speed (no-load)

(a) SGP4[a] [1982]

Speed [RPM]	Upper Bearing		Lower Bearing		Thrust Bearing		Turbine Bearing	
	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ
200	53.0[b]	10.2	51.8	8.6	55.1	7.8	49.7	9.7
	(52.1)[c]	(8.2)	(48.6)	(9.3)	(55.3)	(6.9)	(49.6)	(9.7)
333	55.0	20.4	55.1	8.3	59.0	13.1	52.5	15.5
	(53.8)	(18.7)	(52.7)	(14.9)	(59.1)	(27.4)	(51.9)	(24.4)

(b) TNP3[a] [1983]

Speed [RPM]	Upper Bearing		Lower Bearing		Thrust Bearing		Turbine Bearing	
	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ	T_e (°C)	τ
90	32.3[b]	28.1	33.5	13.0	36.1	7.7	34.3	22.2
	(32.9)[c]	(15.7)	(33.7)	(6.9)	(36.8)	(10.7)	(32.4)	(16.2)
180	38.5	24.0	39.3	16.7	43.4	13.8	42.4	11.8
	(40.8)	(25.2)	(41.2)	(12.3)	(44.1)	(18.8)	(42.1)	(14.2)
260	45.4	29.2	44.4	11.2	49.5	14.5	48.3	15.0
	(44.6)	(19.0)	(47.3)	(10.7)	(48.1)	(8.5)	(46.7)	(11.3)
333	49.4	22.3	48.8	18.0	56.5	35.6	52.4	18.2
	(50.4)	(18.5)	(52.4)	(9.3)	(54.9)	(44.3)	(51.5)	(15.0)

[a] see note (a) in Table 1

[b] see note (b) in Table 1

[c] see note (c) in Table 1

Table 4. Calculated T_e and τ at difference performance of cooling coil, TNP1[a] [2015]

Lower Bearing		Thrust Bearing	
Before	After	Before	After
53.0	51.9	60.8	59.5
(58.0)	(54.5)	(60.5)	(58.5)

[a] see note (a) in Table 1.

[b] Before and after means before and after replacement of lower cooling coil, respectively.

3.5 T_e and τ at different performance of cooling coil

Performance of cooling coil has influence on T_c, hence on T_e. Table 4 shows decrease in T_e as cooling coil replaced with the new, scale-free one. Other factors that might affect T_e had been eliminated as much as possible.

3.6 Generality of proposed model

Equation (6) has been shown to be an excellent model for bearing temperature variation with time. However, there is no single numerical version of the model capable of reproducing the same accuracy in two consecutive tests, even if same particular condition seems to be satisfied, e.g. same unit, bearing, sensor, rotational speed, load, and same cooler performance. Close look upon this phenomenon reveals that what seems to be identical condition between two tests turns out to be different in some aspects.

The calculated numerical model is unique to particular assembly or installation. A slight difference on shaft run-out/alignment, axial load balance, or sensor installation may result in noticeable variation of coefficients in calculated numerical model. Thus, generality of proposed model should be understood in its basic form of equation (6) only, without assigning any numerical coefficient.

If the main interest is to forecast bearing temperature change with time, one should apply following procedures:

a. Collect early data of bearing temperature.
b. Calculate numerical coefficient of T_e and τ using nonlinear least squares fitting.
c. Insert those two coefficients back to equation (6).
d. Forecast bearing temperature change with time using obtained empirical model.

4. CONCLUSION

General model of bearing temperature variation with time has been developed based on simple lumped-capacity method and Newton's Law of Cooling. Nonlinear least squares fitting is used to obtain numerical version of the model with high statistical correlation to empirical data. Comparisons of calculated equilibrium-state bearing temperature and time constant on different sensors, load condition, rotational speed, and performance of cooling coil have been made. Although basic form of proposed model is generally applicable to all situations, its numerical version is unique to particular assembly or installation from which empirical data of bearing temperature is gathered.

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