

AN EXPERIENCE IN THERMODYNAMIC TEST OF LOW HEAD TURBINE IN THAILAND THE CASE OF MAE NGAT HYDRO POWER PLANT

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ABSTRACT

The aim of this paper is to show an experience in efficiency measurement of hydraulic turbine with Thermodynamic method at Mae Ngat hydro power plant which is the first project that takes place for the low head turbine in Thailand. Mae Ngat hydro power plant owned by Electricity Generating Authority of Thailand (EGAT) includes of two (2) Units of Kaplan turbine and its rated head is 40 m, rated output is 4.7 MW and rated discharge is 13 m³/s. According to IEC41-1991 with temperature sampling probe designed and made by EGAT, the test result indicated that the turbine head lower than 100 m can be measured with Thermodynamic method by using high accuracy and precision thermometer with the condition of the penstock that buried underground.

The test results for both Units show that the efficiency curves quite conform with and lower than the design curve which Unit 2 is slightly lower efficiency than Unit 1. From this Thermodynamic testing, both Units have its own equation of the relation between Discharge and Differential Pressure in order to determine the absolute turbine efficiency for the next testing with differential pressure measurement but without discharge measurement.

Moreover, it is very useful in applying the results for calibration of Ultrasonic Flow Meter since the regular calibration cost charged by calibration company is quite expensive and also have many restrictions on the size that can be calibrated.

KEYWORDS: *Hydraulic Turbine, Performance Test, Thermodynamic, Efficiency*

1. INTRODUCTION

The Thermodynamic Test is one of method accepted by IEC60041-1991 “Field Acceptance Tests to Determine The Hydraulic Performance of Hydraulic Turbines, Storage Pumps and Pump-Turbines” to determine the efficiency of hydraulic turbine by detecting heat loss dissipate through the water passing the turbine. This method some time called heat loss method. It is an advantage of Thermodynamic method to find the absolute efficiency without measuring the discharge that is a high uncertainty variable.

The “absolute efficiency” is the actual efficiency that can be used to compare to the guaranteed value directly. While the another way commonly used that also no need to measure turbine discharge such as Index Test that measure “relative efficiency” can compare only before and after turbine replacement (or any turbine modification) and show that the curves conform with the guaranteed curves or not.

The only one problem of Thermodynamic method is the heat loss that detected (temperature difference) is very small until the equipment cannot measure if its resolution is not high enough. And also this tiny loss can be disturbed by ambient that fluctuated. These are the cause of the measurement error and the test failed. That is the reason why IEC recommends testing at the net head above 100 m, which water temperature generally will increase about 0.01 °C for the test performing at net head of 100 m.

However, this project Mae Ngat hydro power plant has the penstock that buried underground, which means the temperature of water inside the penstock may not be disturbed by the environment. And the recent instrument has higher accuracy and resolution than before. The digital oceanographic thermometer “Sea-Bird” has high accuracy and resolution better than 0.001 °C and 0.00025 °C respectively, which means that the very low temperature difference (heat loss) can then be detected below net head of 100 meters which is the height of mostly Thailand’s dam. It is the reason for this experiment.

2. BASIC PRINCIPLE OF THERMODYNAMIC METHOD FOR MEASURING TURBINE EFFICIENCY

2.1 Determination of Turbine Efficiency

The turbine efficiency calculation of Thermodynamic method start with the general expression of turbine efficiency that defined the boundary by IEC:

$$\eta = \frac{P_{out}}{P_{in}} \quad (1)$$

$$\text{Thus } \eta = \frac{P}{P_h} \quad (2)$$

Where η [-] is turbine efficiency, P_{out} [W] (or P [W]) is turbine output (or mechanical power of machine) which is generator input located as Figure 1, and P_{in} [W] (or P_h [W]) is turbine input (or hydraulic power). The Thermodynamic method consider mechanical power transmitted through the coupling of the runner and shaft (P_m) as an output for hydraulic efficiency (η_h), so the turbine efficiency equation becomes the separated efficiency equation:

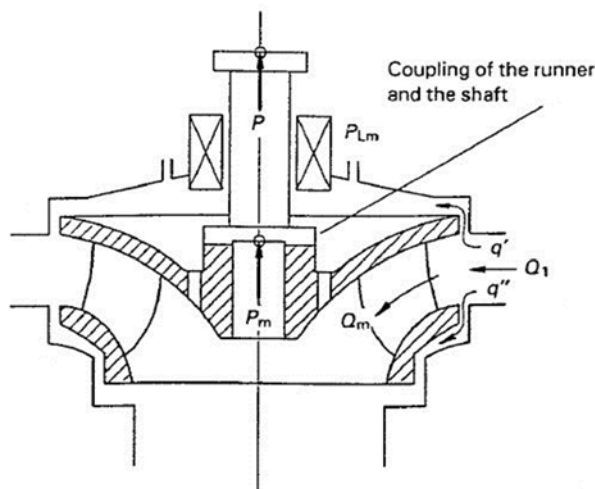
$$\eta = \frac{P}{P_m} \left(\frac{P_m}{P_h} \right) \quad (3)$$

$$\text{Thus } \eta = (\eta_m)(\eta_h) \quad (4)$$

Where P_m [W] is mechanical power of runner which is located as Figure 1, η_m [-] ($= P/P_m$) is mechanical efficiency which is a result of losses in the turbine bearing, and η_h [-] ($= P_m/P_h$) is hydraulic efficiency which is a result of losses in the turbine runner transferred to the water resulting in a temperature increase that detectable by “Sea-Bird” thermometer.

The turbine output (generator input) can be determined by dividing the generator output by generator efficiency ($P = P_{gen}/\eta_{gen}$), which the assumption is non-degradation on generator efficiency. Then the generator efficiency can be found by using the generator efficiency curve of model test or acceptance test.

Figure 1: Flux diagram for power and discharge



Source: IEC60041-1991

Note: $P_m = P + P_{Lm}$ where P_{Lm} [W] is mechanical power loss dissipated in turbine guide bearing and thrust bearing, which is weighted by only turbine proportion.

As mentioned, the Thermodynamic method interests in hydraulic efficiency that tells how efficient the turbine transforms hydraulic power to rotating mechanical power. And we know the general relation between power (P) and energy (E) is $P = \rho QE$ where ρ [kg/m³] is water density, and Q [m³/s] is turbine discharge. So the hydraulic efficiency becomes:

$$\eta_h = \frac{E_m}{E} \quad (5)$$

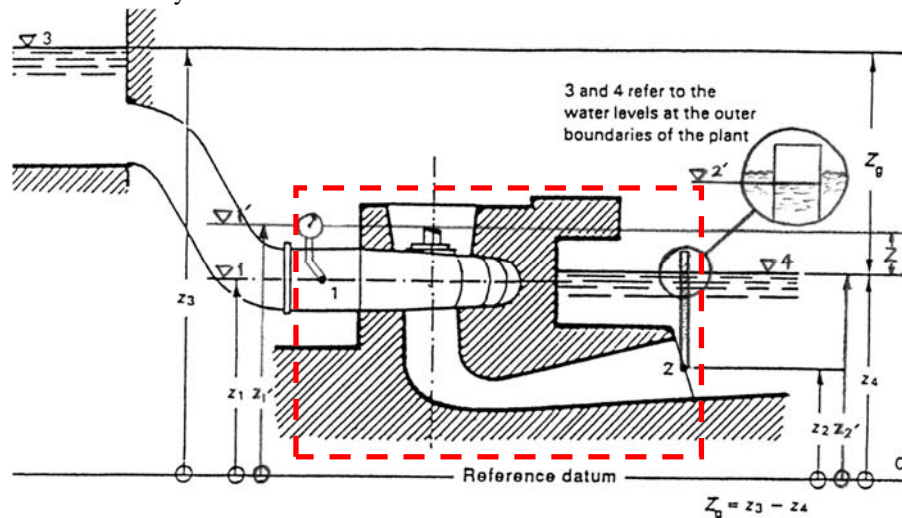
Where E_m [J/kg] ($= P_m/\rho Q$) is specific mechanical energy which is specific energy of mechanical power transmitted through the coupling of the runner and shaft, and E [J/kg] ($= P/\rho Q$) is specific hydraulic energy which is specific energy of water available between the high and low pressure reference sections of the machine.

The specific hydraulic energy can be determined by measurement of entering and leaving the turbine performance variables such as pressure, velocity and level of water, while an additional measurement for determining the specific mechanical energy is temperature of water, which this different temperature reflect the heat loss in turbine. So the hydraulic efficiency Equation (5) can be expressed as:

$$\eta_h = \frac{a(p_{abs1} - p_{abs2}) + c_p(T_1 - T_2) + \frac{v_1^2 - v_2^2}{2} + g(z_1 - z_2)}{\frac{(p_{abs1} - p_{abs2})}{\rho} + \frac{v_1^2 - v_2^2}{2} + g(z_1 - z_2)} \quad (6)$$

Where the performance variables: p_{abs} [Pa] is absolute pressure, T [°C] is temperature, v [m/s] is velocity, and z [m] is level; the Thermodynamic properties: a [m³/kg] is isothermal factor, c_p [J/kg·°C] is specific heat capacity, and ρ [kg/m³] is density ($1/\rho$ or v are also called isentropic factor); and the subscripts: “1” and “2” are high pressure and low pressure section which are referring to the centerlines of inlet and outlet respectively. See Figure 2 test boundary.

Figure 2: Test Boundary



Source: Adapted from IEC60041-1991

Note: The red dashed line is the test boundary

In practical, the “Sea-Bird” temperature sensor cannot be inserted directly into main flow at centerline inside of penstock. This led to the use of special equipment that called “sampling probe” which is designed and made by EGAT to measure “the temperature and its corresponding pressure at measuring vessel of sampling probe” instead of “the temperature and pressure at centerline of penstock”, which these two measuring position are the same enthalpy because of the adiabatic processes are assumed. To account for this influence subscripts “₁₁” and “₂₁” will be used to replace “1” and “2” of Equation (6), respectively. The enthalpy drop process across the turbine can be shown by the enthalpy-entropy diagram as shown in Figure 3.

Figure 3: Enthalpy-Entropy Diagram

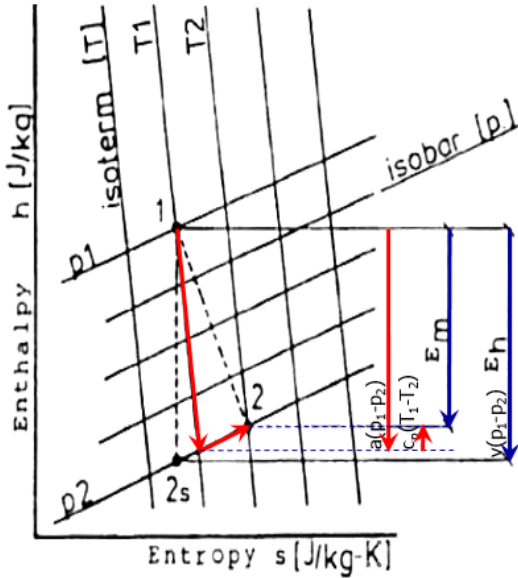


Figure 3a: Theoretical Enthalpy Drop

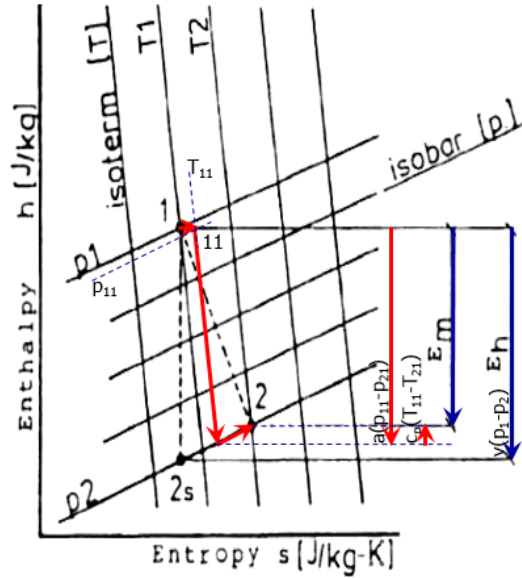


Figure 3b: Practical Enthalpy Drop

- Note:**
- 1) Figure 3a show the theoretical enthalpy drop for ideal process (isentropic or entropy is constant) which is form “1” to “2s” and real process (entropy increase) which is from “1” to “2”
 - 2) Figure 3b show the practical enthalpy drop for ideal process (isentropic or entropy is constant) which is form “1” to “2s” and real process (entropy increase) which is form “11” to “21”
 - 3) Point “2” and point “21” are at the same position in case of outlet parameter can be measured directly by submerging the “Sea-Bird” into the tailrace.
 - 4) The specific mechanic energy (E_m) of Figure 2a are equal to Figure 2b although the path is different.

Moreover, due to the fact that water will take time to travel between two sections and combine with the temperature may be fluctuated by surroundings before flowing through the turbine. To account for this influences a corrective term (dE_m) will be added to the Equation (6). Then the final equation for hydraulic efficiency becomes:

$$\eta_h = \frac{a(p_{abs11} - p_{abs21}) + c_p(T_{11} - T_{21}) + \frac{v_{11}^2 - v_{21}^2}{2} + g(z_{11} - z_{21}) + dE_m}{\frac{(p_{abs1} - p_{abs2})}{\rho} + \frac{v_1^2 - v_2^2}{2} + g(z_1 - z_2)} \quad (7)$$

Where dE_m [J/kg] is a corrective term which is calculated by multiplying the temperature gradient [°C/sec] with the time traveling [sec] and its specific heat capacity [J/kg-°C], and the subscripts: “₁₁” and “₂₁” are high pressure and low pressure point which are referring to the exact measuring point of inlet and outlet respectively.

2.2 Determination of Turbine Discharge

After the turbine efficiency has been calculated and subsequently calculate the turbine discharge (flow rate).

According to Equation (2), we know turbine output is equal to generator input which is calculated as the generator output divided by generator efficiency ($P = P_{gen}/\eta_{gen}$). And we know hydraulic power is equal to the product of water density, local gravity, turbine discharge, and turbine net head ($P_h = \rho gQH$). So we can find the turbine discharge by derived into the equation:

$$\eta = (P_{gen}/\eta_{gen})/(\rho gQH) \quad (8)$$

$$\text{Thus } Q = (P_{gen}/\eta_{gen})/(\rho gH\eta) \quad (9)$$

Where P_{gen} [W] is generator output, η_{gen} [-] is generator efficiency, g [m/s^2] is local gravity and H [m] is turbine net head which is equal to specific hydraulic energy divided by local gravity ($H = E/g$).

It is very useful in implementing the results to apply for the calibration of ultrasonic flow meter if there is appropriate measuring point of ultrasonic flow meter.

And for the next testing, it is not necessary to measure the discharge again if there is the measurement of differential pressure in this Thermodynamic test because the calculated discharge and the measured differential pressure of all test load can be created to the relation that can be applied to the next testing.

$$Q = k(\Delta p)^n \quad (10)$$

Where k is constant, n is exponent, and Δp is differential pressure which is located at suitably taps such as the taps on turbine spiral case (inner tap - outer tap).

2.3 Correction to Specified Net Head

From the entire steps of calculation, we will get the group of performance parameters such as η , P , Q and H for each load testing which suggest for six loads at least as IEC requirement. In actually, the turbine net head (H) will vary in each load testing depending on the different loss in the system. But the turbine performance curve is usually established for each specified net head. Thus the correction of each testing load to the specified net head are necessary. The correction formula are:

$$P_{sp} = P(H_{sp}/H)^{3/2} \quad (11)$$

$$Q_{sp} = Q(H_{sp}/H)^{1/2} \quad (12)$$

$$\eta_{sp} = \eta \quad (13)$$

Where H_{sp} [m] is specified net head, P_{sp} [W] is mechanical power at specified net head, Q_{sp} [m^3/s] is discharge at specified net head, and η_{sp} [-] is turbine efficiency at specified net head. According to IEC, the allowed range of these correction formula is approximately 2% ($0.98 < H_{sp}/H < 1.02$). In case the range over these allowance, the different efficiency taking form hill chart shall be applied to as tested efficiency to get the turbine efficiency at specified net head ($\eta_{sp} = \eta + \Delta\eta$), which is not described in this article.

3. AN EXPERIENCE AT MAE NGAT HYDRO POWER PLANT

The purpose of this Thermodynamic test is to prove that the test can still be achieved under a net head of 100 m, although it is lower than the level recommended by IEC. The test took place on Mae Ngat hydro power plant located in Mae Taeng district of Chiang Mai which is the first project that performed by Electricity Generating Authority of Thailand (EGAT) to use the Thermodynamic method on the hydraulic turbine performance test in Thailand. This plant is owned by EGAT includes of 2 Units of Kaplan turbine and its rated head is 40 m, rated output is 4.7 MW, and rated discharge is 13 m^3/s with buried penstock.

The test was performed respectively on the load 100, 95, 90, 85, 80, 75, 70, 60, 50, 40, 30, and 20% of 4.5 MW generator output which is kept nearly unity power factor ($PF = 1$) for each unit. Two runs averaged were reported for each load testing. During a test run, the fluctuation and variations were controlled within the limit: $\pm 1.5\%$ of generator power, $\pm 1.0\%$ of specific hydraulic energy, $\pm 0.5\%$ of rotational speed, and ± 0.005 °C/min of water temperature, in order to meet the IEC requirements.

The main apparatus using in this Thermodynamic test consists of three “Sea Bird” thermometers measuring the water temperature at sampling probe (T_{11}) and at sampling frame (T_{21}), which the first thermometer located at the turbine inlet (penstock) is inserted to the sampling probe (measuring section “11”) that led the water to flow out of penstock, and the left two thermometers located at the turbine outlet (draft tube) are inserted to the sampling frame (measuring section “21”) that guide the thermometers submerged into the tailrace water: The sampling probe and the sampling frame are designed and made by EGAT which follow the IEC recommendation.

The other test instruments and redundancy consist of one power meter measuring generator output (P_a), four pressure transmitters used to find absolute pressure of turbine inlet static pressure (p_i) and sampling probe pressure (p_{11}), two differential pressure transmitters measuring spiral case differential pressure (Δp), one level transmitter measuring tailrace water level used to find the absolute pressure of turbine outlet pressure (p_{21}), one barometric pressure transmitter measuring ambient pressure (p_{amb}), one displacement transmitter measuring servomotor stroke of guide vane opening (a), one rotameter measuring sampling probe discharge (q_{11}), and one measuring tape with plate gauge measuring tailrace water level used to find the absolute pressure turbine outlet pressure (p_{21}).

These following Figure 4 to Figure 13 are all the measuring device used in the test and Figure 14 show the diagram of all measuring point.

Figure 4: The Installation of Sampling Probe at the Penstock nearby the Inlet Valve



Figure 5: EGAT’s Sampling Probe (Left), Sea-Bird Thermometer (Middle) and Rotameter (Right)



Figure 6: Pressure Transmitter (2 Left for Sampling Probe, 2 Right for Static Taps)

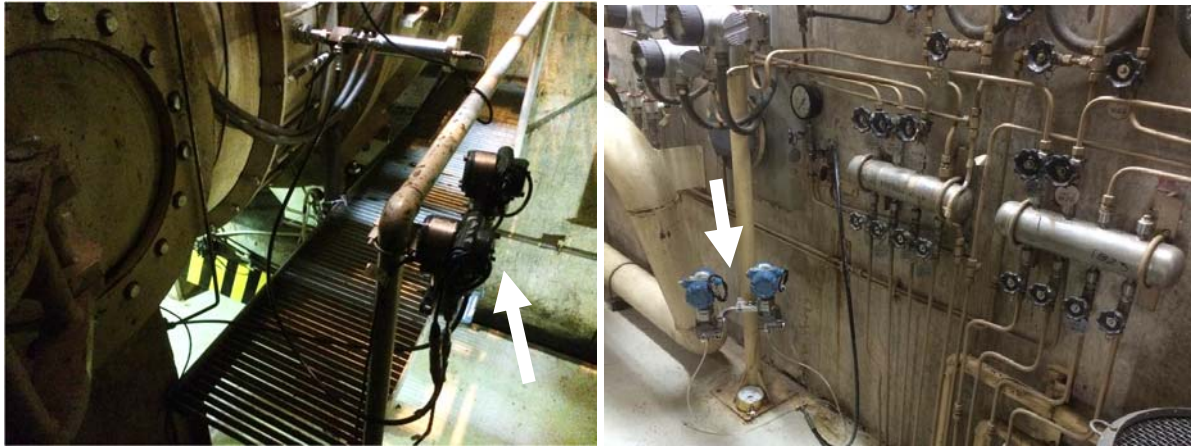


Figure 7: The Installation of Sampling Frame at Draft Tube



Figure 8: Layout for 2 Sea-Bird Thermometers (Middle), and Level Transmitter (Right)



Figure 9: Measuring Tailrace Water Level by Measuring Tape with Plate Gauge



Figure 10: Differential Pressure Transmitter Located at the Taps on Turbine Spiral Case

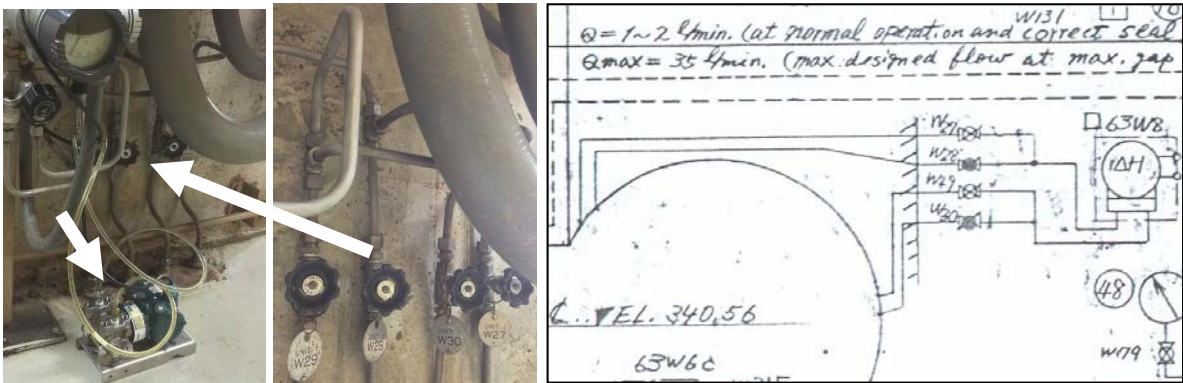


Figure 11: The Connecting Point of Power Meter

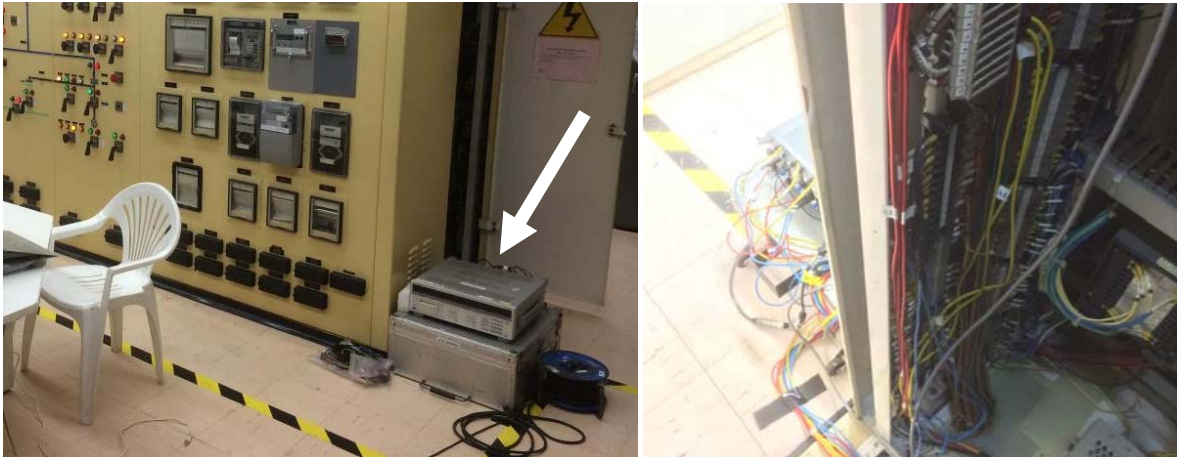


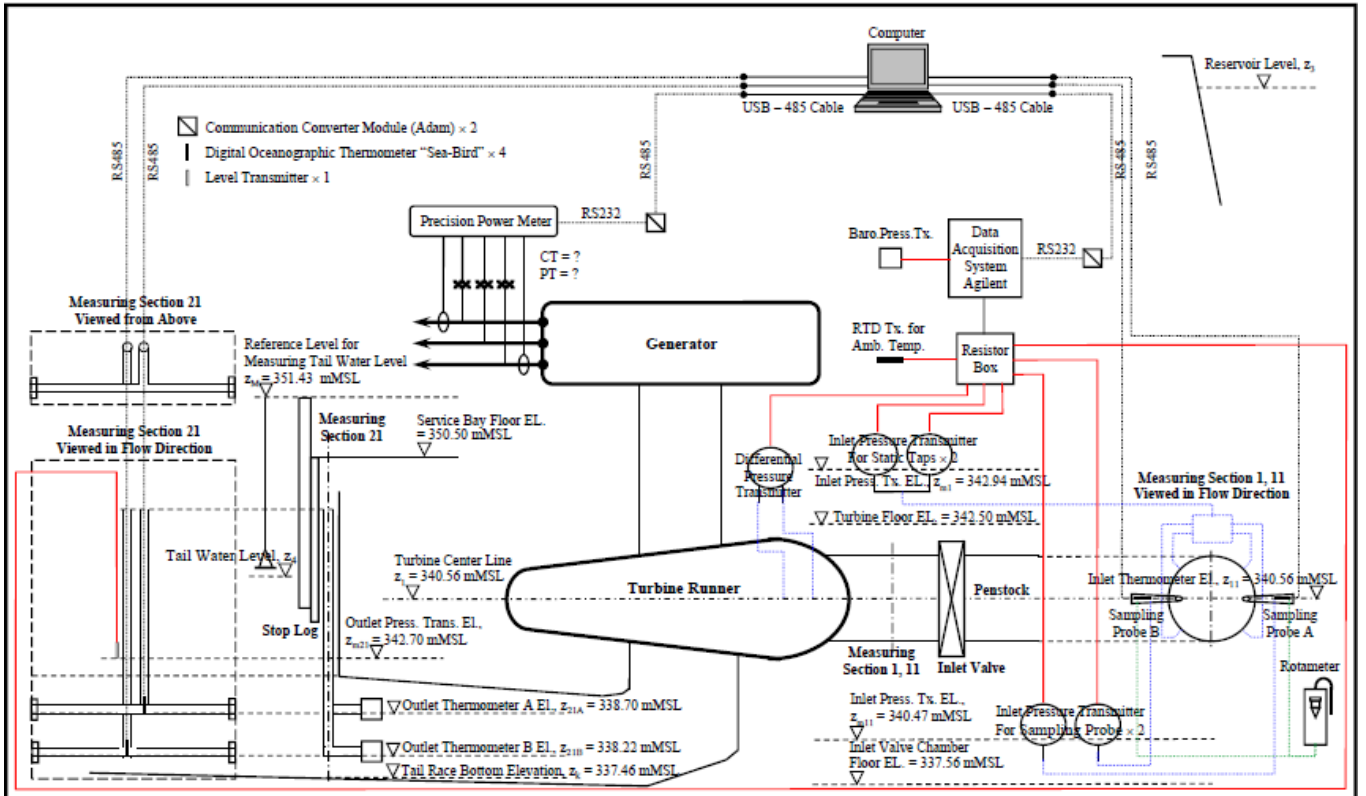
Figure 12: Displacement Transmitter Measuring Servomotor Stroke of Guide Vane Opening



Figure 13: Data Logging with Data Acquisition System



Figure 14: Measuring Point Diagram of All Instrument Used in the Test



Note: Only the sampling probe B was used during the test because sampling probe A was error.

During the test, the calculated turbine net head is around 49.48 to 51.75 meters. So the test results will be corrected to the specified head that is 52.5 meters according to maximum characteristic of turbine (design).

The Figure 15 and 16 show the test results at net head 52.5 m for both Units in form of performance curves compared with design turbine show that the efficiency curves quite conform with and lower than the design curve which Unit 2 is slightly lower efficiency than Unit 1. The best efficiency of Unit 1 is 85.1% which is 6.9% lower than design at operating point of mechanical power 4.5 MW (generator output 3.8 MW), while Unit 2, the best efficiency is 83.8% which is 8.2% lower than design at the same operating point.

Figure 15: Unit 1 Test Results - Performance Curves Comparison with Design

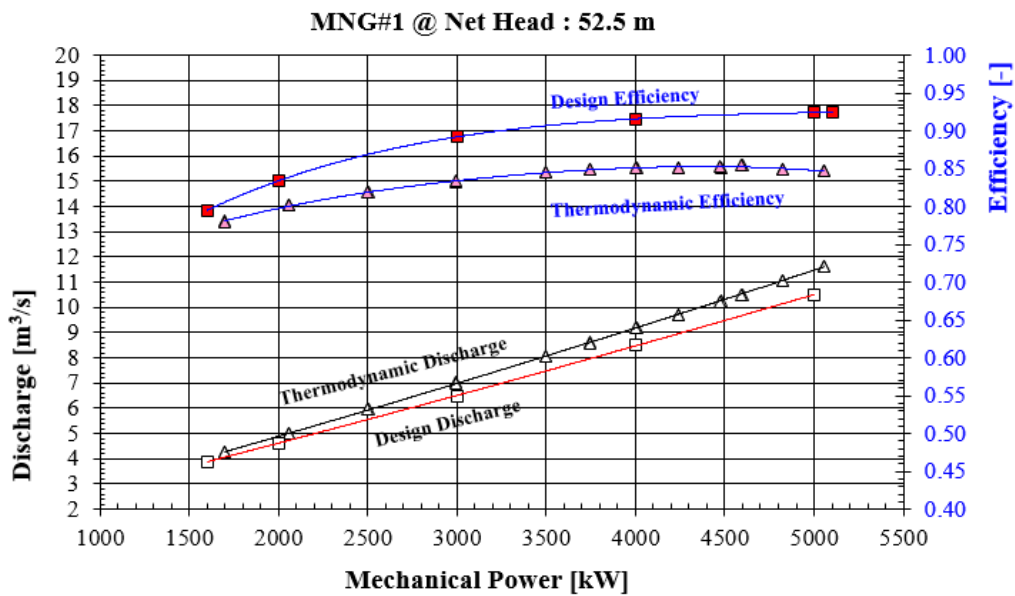
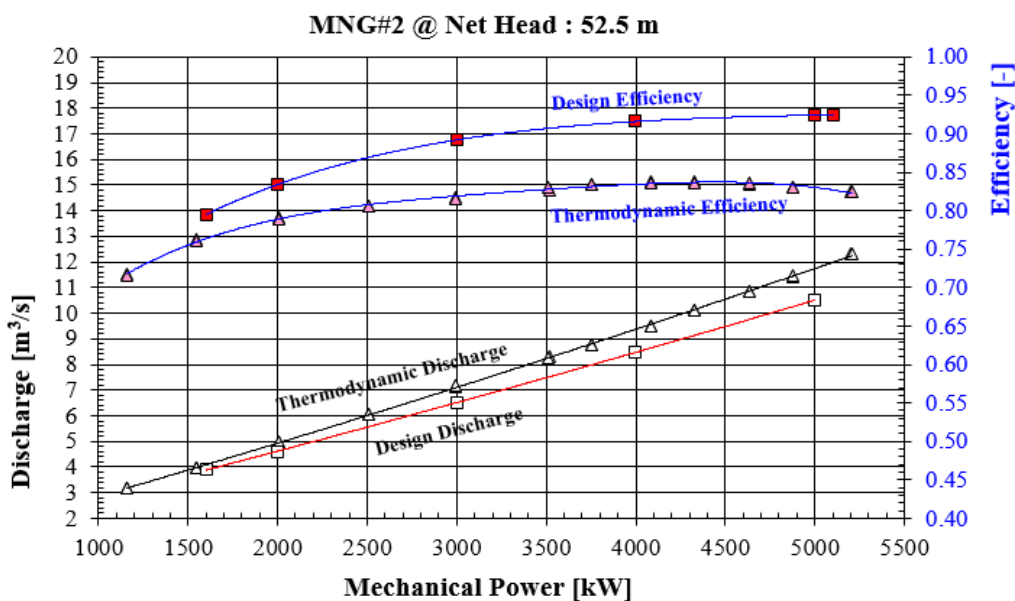


Figure 16: Unit 2 Test Results - Performance Curves Comparison with Design



After we get the absolute efficiency from this Thermodynamic testing and the discharge is consequently calculated together with the measurement of differential pressure, so the relation between discharge and differential pressure can be established in order to create the polynomial equation of this relation. Then in the next testing, it is not necessary to use the Thermodynamic method because the created polynomial equation can be used to get the discharge by measuring only differential pressure. But the important constraint is the differential pressure taps must be in the same condition which is no any obstacle inside the taps.

Both Units have its own equation of the relation between discharge (Q) and differential pressure (Δp) which are $Q = 1.4537(\Delta p)^{0.5} - 0.4329$ for Unit 1 and $Q = 1.6164(\Delta p)^{0.5} - 1.4510$ for Unit 2 as shown in the Figure 17 and 18 below.

Figure 17: Unit 1 – The Relation between Discharge and Differential Pressure

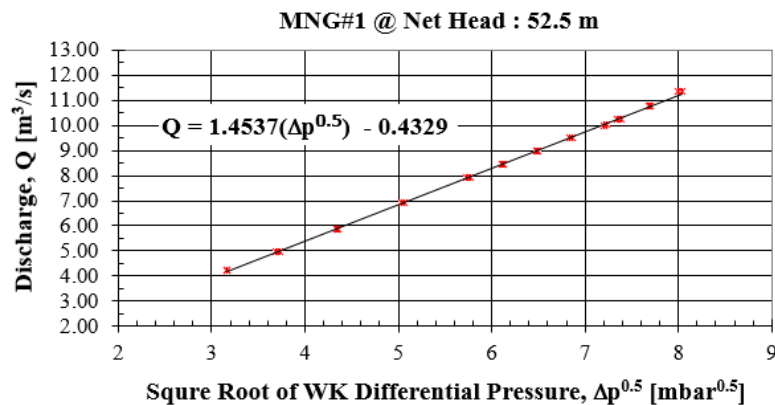
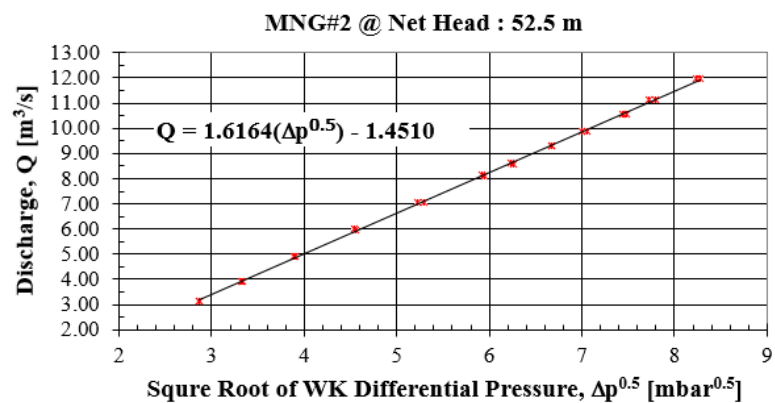


Figure 18: Unit 2 – The Relation between Discharge and Differential Pressure



4. CONCLUSION

Although the net head is lower than the level recommended by IEC, the testing efficiency curves that quite conform with the design curve indicate that the test results of net head lower than 100 m can be achieved with Thermodynamic method by using high accuracy and precision thermometer with the condition of the penstock buried underground which water temperature inside may not be disturbed by the environment.

The added benefit of the Thermodynamic test is the results that can be applied to calibrate the ultrasonic flow meter since the regular calibration cost charged by calibration company in Thailand is quite expensive (approximately 1,000 USD) and also have many restrictions on the size that can be calibrated because the calibration company cannot calibrate the meter at the large scale of pipe size (not

over 8 inches or inside diameter of 212.9 mm) and high flow rate (not over 54 m³/h). While the penstock is larger than 2,000 mm and the discharge is larger than 40,000 m³/h in this Mae Ngat hydro power plant.

However the discharge calculated from this Thermodynamic test cannot be compared with ultrasonic flow meter and plant monitoring. Because there is no suitable measuring point of the discharge in this power plant and the discharge displayed on plant monitor calculated from the design since commissioning is not accurate enough to be used to compare with the test results of this performance test.

For the next testing, the other low head project whether it is a turbine or pump with a suitable measuring point of discharge will be chosen to be the research project in order to compare between the discharge from Thermodynamic test and the discharge from ultrasonic flow meter. This will ensure that the Thermodynamic test results is highly accurate.

5. REFERENCES

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