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Estimation of performance of steam turbines using a simple predictive tool

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Mechanical drive steam turbines are major prime movers for compressor, blower, and pump applications. Steam turbines are available for a wide range of steam conditions, power ratings and speeds. In this work, a simple predictive tool, which is easier than existing approaches, less complicated with fewer computations, is presented for rapid prediction of steam rate, turbine efficiency, and the inlet and exhaust nozzle diameters to determine the actual steam rate (ASR) and total steam requirements for both multi-stage and single-stage turbines. The proposed method predicts the above mentioned parameters for inlet steam pressures up to 12,000 kPa, turbine ratings up to 10,000 kW as well as the exhaust air over inlet air ratios of up to 0.55. The predictions from the proposed predictive tool have been compared with reported data and found good agreement with average absolute deviation hovering around 1.4%.

1. Introduction

A steam turbine is a module to convert heat energy into mechanical energy. The principal task in operating a steam turbine is to convert the energy of hot steam into rotational energy [1]. The design of high efficiency turbo-machinery is a promising and challenging task in view of their important function and complex internal flow field [2]. Any malfunction occurring inside steam turbines will increase the overall heat losses [3]. Steam turbines used as process drivers are usually required to operate over a range of speeds in contrast to a turbine used to drive an electric generator which runs at nearly constant speed. The energy available in each kilogram of steam which flows through the turbine is a function of overall turbine pressure ratio (inlet pressure/exhaust pressure) and inlet temperature. Condensing turbines are those whose exhaust pressures are below atmospheric. They offer the highest overall turbine pressure ratio for a given set of inlet conditions and therefore require the lowest steam flow to produce a given power. A cooling medium is required to totally condense the steam. Non-condensing or back-pressure turbines exhaust steam at pressures above atmospheric and are usually applied when the exhaust steam can be utilized elsewhere [3].

In a single-stage turbine, steam is accelerated through one cascade of stationary nozzles and guided into the rotating blades or buckets on the turbine wheel to produce power. A Rateau design has one row of buckets per stage. A Curtis design has two rows of buckets per stage and requires a set of turning vanes between the first and second row of buckets to redirect the steam flow. A multi-stage turbine utilizes either a Curtis or Rateau first stage followed by one or more Rateau stages. Single-stage turbines are usually limited to about 2000 kW although special designs are available for larger units. Below 2000 kW the choice between a single and a multi-stage turbine is usually on economic grounds. For a given shaft power, a single stage turbine will have a lower capital cost but will require more steam than a multi-stage turbine because of the lower efficiency of the single-stage turbine [4].

The objective of the steam turbine is to maximize the use of the available steam energy where the available steam energy is defined as the difference between the inlet and exhaust energies (enthalpies) for a 100% efficient constant entropy (i.e. isentropic) process [4]. There are numerous loss mechanisms which reduce the efficiency from isentropic process such as throttling losses, steam leakage, friction between the steam and the nozzles/buckets, bearing losses, etc. Efficiencies can range from a low of 40% for a low power single-stage turbine to a high approaching 90% for a large multistage, multi-valve turbine [4].

Most equipment driven by steam turbines are centrifugal machines where power varies as the cube of speed. Part load efficiency varies as a function of speed, flow, and the number of stages. By assuming power to vary as the cube of speed the turbine part load efficiency can be approximated as a percentage of the design efficiency [5].

Current methods are more complicated and need longer computations for rigorous investigation of the design and performance characteristics of hybrid system configurations consisting of gas turbine, and steam turbine for power applications [5]. Moreover, when the mechanical equipments such as steam turbines are put into operation, their performance will degenerate with the increases in

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operation time. The levels of performance degeneration may vary with the working environment, mission, working character and maintenance of the equipment [6]. In addition, the high cost that companies incur in these days for energy consumption makes it necessary to develop a practical, reliable and easy-to-use method for power generating industries in terms of assessing operational parameters [7].

In light of the above mentioned issues faced by the power generating industries, there is an essential need to develop a practical, reliable and easy-to-use method for practice engineers for the estimation of steam rate, turbine efficiency, and the inlet and exhaust nozzle diameters to determine the actual steam rate (ASR) and total steam requirements for both multi-stage and single-stage steam turbines. The present study discusses the formulation of a simple method which can be of significant importance for engineers dealing with steam turbines. The present approach is of practical significance for power generating industries in terms of assessing operational issues. In particular the proposed predictive tool gives an advance indication of key parameters which could potentially enable practice engineers to take appropriate measures so as to avoid operational problems with steam turbines in power generating industries.

2. Methodology to develop new predictive tool

The required data to develop this predictive tool includes the reported data [4] for steam rate, turbine efficiency, the inlet and exhaust nozzle diameters to determine the actual steam rate (ASR) and total steam requirements for both multi-stage and single-stage turbines. The following methodology [8–12] using MATLAB software [13] has been applied to develop a predictive tool for a typical parameter such as part load efficiency correction factor vs percent power multi-valve steam turbines (W) and number of stages (N).

Firstly, part load efficiency correction factor is correlated as a function of Percent Power Multi-Valve Steam Turbines for different number of stages. Then, the calculated coefficients for these polynomials are correlated as a function of number of stages. The derived polynomials are applied to calculate new coefficients for equation (1) to predict the part load efficiency correction factor. Table 1 shows the tuned coefficients for equations (2)–(5).

In brief, the following steps are repeated to tune the predictive tool's coefficients.

1. Correlate the part load efficiency correction factor as a function of percent power multi-valve steam turbines for a given number of stages (N).
2. Repeat step 1 for other number of stages (N).
3. Correlate corresponding polynomial coefficients, which were obtained for different percent power multi-valve steam turbines versus number of stages, $a = f(N)$, $b = f(N)$, $c = f(N)$, $d = f(N)$ [see equations (2)–(5)].

Table 1
Tuned coefficients used in equations (2)–(5) for part load efficiency correction factor vs percent power multi-valve steam turbines.

Coefficient	Part load efficiency correction factor vs percent power multi-valve steam turbines
A_1	$-9.67554468989956 \times 10^{-2}$
B_1	$-3.864271558704358 \times 10^{-1}$
C_1	$1.044184496346528 \times 10^{-1}$
D_1	$-8.181357318296115 \times 10^{-3}$
A_2	$-1.150178844252596 \times 10^{-2}$
B_2	$1.528358655177229 \times 10^{-2}$
C_2	$-3.893637295630038 \times 10^{-3}$
D_2	$3.169814158242376 \times 10^{-4}$
A_3	$2.44354207528106 \times 10^{-4}$
B_3	$-1.89257100557497 \times 10^{-4}$
C_3	$4.747677297164477 \times 10^{-5}$
D_3	$-4.00379604790941 \times 10^{-6}$
A_4	$-1.19638058919324 \times 10^{-6}$
B_4	$7.50620062251362 \times 10^{-7}$
C_4	$-1.899443544787457 \times 10^{-7}$
D_4	$1.653733711580176 \times 10^{-8}$

Table 2
Tuned coefficients used in equations (7)–(10) for basic efficiency of multi-valve, multi-Stage condensing and non-condensing steam turbines.

Coefficient	Basic efficiency of multi-valve, multi-stage condensing turbines	Basic efficiency of multi-valve, multi-stage non-condensing turbines
A_1	4.37877350555386	4.3579522488849
B_1	$-5.886246936292 \times 10^{-6}$	$1.442866040264652 \times 10^{-5}$
C_1	$-4.37824615985392 \times 10^{-10}$	$-4.49835638024881 \times 10^{-9}$
D_1	$4.9052617628984 \times 10^{-14}$	$2.79985661346127 \times 10^{-13}$
A_2	$-1.387851243025345 \times 10^{2}$	$-3.45898953966307 \times 10^{1}$
B_2	$-6.589550191385539 \times 10^{-2}$	$-1.58632573100788 \times 10^{-1}$
C_2	$9.729602056575966 \times 10^{-6}$	$3.29243029645988 \times 10^{-5}$
D_2	$-5.27789047337105 \times 10^{-10}$	$-2.12357823149299 \times 10^{-9}$
A_3	$3.98069674568167 \times 10^{4}$	$-3.7896243395658 \times 10^{4}$
B_3	$2.980006167118549 \times 10^{1}$	$1.11304136766883 \times 10^{2}$
C_3	$-5.19648879302579 \times 10^{-3}$	$-2.86203302643892 \times 10^{-2}$
D_3	$3.2495031828337 \times 10^{-7}$	$2.033476179976455 \times 10^{-6}$
A_4	$-4.522126353628195 \times 10^{6}$	$1.6118197770998 \times 10^{7}$
B_4	$-5.198744177935268 \times 10^{3}$	$-3.03480606125437 \times 10^{4}$
C_4	1.06667544357578	8.7404857491443
D_4	$-7.3315284072605 \times 10^{-5}$	$-6.61654062984245 \times 10^{-4}$

Equation (1) represents the proposed governing equation in which four coefficients are used to correlate part load efficiency correction factor as a function of percent power multi-valve steam turbines (W) and number of stages (N) and for different number of stages (N) where the relevant coefficients are reported in Table 1.

$$\ln(\eta_F) = a + bW + cW^2 + dW^3 \tag{1}$$

where:

$$a = A_1 + B_1 N + C_1 N^2 + D_1 N^3 \tag{2}$$

$$b = A_2 + B_2 N + C_2 N^2 + D_2 N^3 \tag{3}$$

$$c = A_3 + B_3 N + C_3 N^2 + D_3 N^3 \tag{4}$$

$$d = A_4 + B_4 N + C_4 N^2 + D_4 N^3 \tag{5}$$

Equations (6)–(10) predict the basic efficiency of multi-valve, multi-stage condensing turbines and non-condensing turbines as

Table 3
Tuned coefficients used in equations (12)–(15) (speed efficiency correction factor for condensing and non-condensing turbines).

Coefficient	Speed Efficiency Correction Factor for Condensing and Non-Condensing Turbines
A_1	$7.139838347762626 \times 10^{-1}$
B_1	$-1.259929507384354 \times 10^{-4}$
C_1	$2.0812766931417 \times 10^{-8}$
D_1	$-1.149692664292995 \times 10^{-12}$
A_2	5.41269669960742
B_2	$7.4668488807785 \times 10^{-4}$
C_2	$-1.3976426645952 \times 10^{-7}$
D_2	$8.321824042644553 \times 10^{-12}$
A_3	$-2.98007219398855 \times 10^{1}$
B_3	$-2.582366410572227 \times 10^{-4}$
C_3	$1.542107843250927 \times 10^{-7}$
D_3	$-1.269395300867692 \times 10^{-11}$
A_4	$5.124041889735471 \times 10^{1}$
B_4	$-3.074217078802187 \times 10^{-3}$
C_4	$3.110231585517153 \times 10^{-7}$
D_4	$-9.70009445567164 \times 10^{-12}$