UNIVERSITY OF TURKISH AERONAUTICAL ASSOCIATION
INSTITUE OF SCIENCE AND TECHNOLOGY

INVESTIGATING THE PERFORMANCE OF THE STEAM TURBINE BY SIMULATION MODELING

MASTER THESIS

Mothana Bdaiwi AHMED

MECHANICAL AND AERONAUTICAL ENGINEERING

MASTER THESIS PROGRAM

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Supervisor: Assist. Prof. Dr. Ibrahim MAHARIQ

Mothana Bdaiwi AHMED, having student number 1406080014 and enrolled in the Master Program at the Institute of Science and Technology at the University of Turkish Aeronautical Association, after meeting all of the required conditions contained in the related regulations, has successfully accomplished, in front of the jury, the presentation of the thesis prepared with the title of: "Investigating The Performance OF The Steam Turbine By Simulation Modeling"

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Thesis Defense Date: 28.12.2017

THE UNIVERSITY OF TURKISH AERONAUTICAL ASSOCIATION
INSTITUTE OF SCIENCE AND TECHNOLOGY

I hereby declare that all the information in this study I presented as my Master's Thesis, called "Investigating The Performance OF The Steam Turbine By Simulation Modeling" has been presented in accordance with the academic rules and ethical conduct. I also declare and certify on my honor that I have fully cited and referenced all the sources I made use of in this present study.

28. 10. 2017

Mothana Bdaiwi AHMED

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December 2017 Mothana AHMED

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NOMENCLATURE

Symbol		Description	Units
A	:	Constant	–
B	:	Constant	–
C	:	Constant	–
c_f	:	steam speed	m/s
C_P	:	Specific heat	J/kg. °C
D	:	Constant	–
E	:	Constant	–
Ex	:	Exergy	kJ/kg
G	:	Inlet mass flow rate	kg/s
h	:	Specific enthalpy	kJ/kg
H_v	:	Lower heating value	kJ/kg
I	:	Exergy losses	kJ/kg
k_o	:	Constant value for linear equation	–
k_a	:	Constant value for linear equation	–
M	:	Constant in equation	–
m	:	Mass flow rate	kg/s
P	:	Power	MW
P	:	Pressure	MPa
$P(S)$	:	Saturation pressure	MPa
R	:	Constant	–
S	:	Entropy	kJ/kg.K
T	:	Temperature	°C
$T(S)$	:	Saturation temperature	K
u	:	Blade speed	m/s
V	:	Volume	m^3
v	:	Specific volume	m^3/kg

Greek Symbols

Symbol		Description	Units
Δ	:	Change of conditions	–
τ	:	Time constant	(Sec)
α	:	Steam quality	–
ρ	:	Density	kg/m^3
ρ_s	:	Density of steam	kg/m^3
δ	:	Rotor angle	(rad)

Subscripts

Symbol		Meaning
B	:	Boiler
C	:	Condenser

cfwh	:	Closed feed water heater
e	:	Electrical
ex.	:	Exergy
ext.	:	Extraction
f	:	liquid phase
fuel	:	Fuel
g	:	Vapor phase
in	:	input
m	:	Mechanical
o	:	Ambient
ofwh	:	open feed water heater
out	:	out
rh.	:	Reheater
s	:	Steam
w	:	Water

Superscript

Symbol		Meaning
ctrl	:	Controller

Abbreviation

Symbol		Description
HPC	:	High pressure turbine
HPV	:	High pressure valves
HTGR	:	High temperature gas reactor
IPC	:	Intermediate pressure cylinder
LPC	:	Low pressure cylinder
MPC	:	Model predictive control
RHV	:	Reheater valves

ABSTRACT

INVESTIGATING THE PERFORMANCE OF THE STEAM TURBINE BY SIMULATION MODELING

AHMED, Mothana Bdaiwi

Master, Department of Mechanical & Aeronautical Engineering

Thesis Supervisor: Assist. Prof. Dr. Ibrahim MAHARIQ

December 2017, 75 pages

This thesis describes a possible way to build a simulation model of the most important parts of power plant Al – Dura (K– 160–13.34–0.0068). Matlab/Simulink software was used to simulate the behavior of Steam turbine with high-pressure (HP), intermediate pressure (IP), and low – pressure (LP) steam, with a load response in a stable circumstance over a range within 50% to 100%. The model is based on "Stodol's law" and simulates the pressure and enthalpy alongside the dissimilar turbine phases and the vapor and water extraction. The effect of the vapor and water extraction on the turbine is also elucidated. Areas of essential energy loss and exergy decimation will be resolved. The impact of changing the power plant load on the exergy analysis will likewise be researched. A non – linear function is developed to evaluate thermodynamic properties at above phases of turbine. The response of suggested purposes to estimate these properties of vapor is compared with standard data and showed high accuracy (the modeling error is less than 0.01%). In this thesis analysis of temperature and quantity balance of thermal power plant for Al – Dura (K– 160–13.34–0.0068) station was firstly studied and used for references objectives in addition to justification aims of applied form. The Cycle – Tempo and

Matlab/Simulink package are used to model the energetic and exergetic, analysis of the power plant.

Keywords: Exergy, energy, extraction steam, reheated steam.

ÖZET

SİMÜLASYON MODELLEME İLE BUHAR TÜRBİNİNİN PERFORMANSINI ARAŞTIRMA

AHMED,Mothana Bdaiwi
Yüksek lisans, Mekanik and Havacılık Mühendisliği
Tez Danışmanı: Yardımcı Prof. Dr. Ibrahim MAHARIQ
Aralık 2017, 62 sayfa

Bu tez, Al-Dura enerji santralinin (K-160-13.34-0.0068) en önemli parçalarının simülasyon modelini oluşturmak için muhtemel bir yolu tanıtmaktadır. Matlab/simulink yazılımı, %50'den %100 e kadar değişen stabil durumdaki yük yanıtı ile yüksek basınçlı (HP),orta basınçlı (IP) ve düşük basınçlı (LP) buharla çalışan buhar türbininin buhar basıncının davranımı benzetimini yapmak için kullanılmıştır.Bu model Stodol's yasası üzerine kurulu olup,basınç,entalpi bunun yanında benzemeyen türbin fazları, buhar ve su çıkarmanın taklidini yapmaktadır. Türbindeki buhar ve su çıkarımının etkisi açıklanmıştır. Esas enerji kaybı, enerji örnek seyreltme bölgeleri ortadan kaldırılacaktır. Ekserji analizindeki değişen enerji santralinin etkisi de aynı şekilde araştırılacaktır. Türbin fazları üzerindeki termodinamik özellikleri değerlendirmek için doğrusal olmayan fonksiyon geliştirilmiştir. Buharlaşmanın bu özelliklerini öngörmek için önerilen amaçlar standart data ve gösterilen yüksek doğruluk (modelleme hatası %0.01'den azdır) ile karşılaştırılmıştır. Bu tezde Al-Dura termal enerji santralinin sıcaklık ve denge miktarı analizi ilk olarak incelenip araştırılmış ve kaynak hedefle, başvurulmuş formun doğrulama amaçları için kullanılmıştır. Cycle-tempo ve Matlab/Simulink

paketi enerjetik ve ekserjetiği modellendirmek ve enerji santralinin analizi için kullanılmıştır.

Anahtar Kelimeler: Ekserji, enerji, ekstraksiyon buhar, yeniden ısıtılmış buha

CHAPTER ONE

INTRODUCTION

1.1 Background

Because of the costs and efficiencies of steam turbines, they are utilized in power generating. A new level is offered for steam turbines structure with regards to application, capacity and required accomplishment. There is a complex characteristic in the steam turbines. The multistage steam expansion is included in these turbines for raising the thermal efficiency. Because of the turbine structure complexity, it is not easy to discover the influences of the estimated control system. Studying turbine dynamics leads to progress the nonlinear analytical models. These models are utilized for accomplishing real-time simulations, controlling system design synthesis and monitoring the required states [1].

By using identification techniques in the applications of the power plant, the mathematical models are advanced. The advanced models include complexities which characterize the system in a good way in specific operating conditions [2]. In normal operation without disruption or any foreign excitation, the system identification is considered as an optimal goal. Using operating data is demanded to identify the external excitation and also to identify the limitations of faces [3].

In the supposed case of parametric models availability, it would be advantageous to use soft computing methods.

1.2 Steam Turbine Power Plant

A thermal power plant is usually defined by fuel type, and it is used for the purpose of heating water and creating steam. In order to provide the steam which is needed in operating a thermal plant, Coal, oil, solar and nuclear powers can be utilized. Most of the thermal plants operate on a partially closed loop, utilizing a Rankine cycle [4].

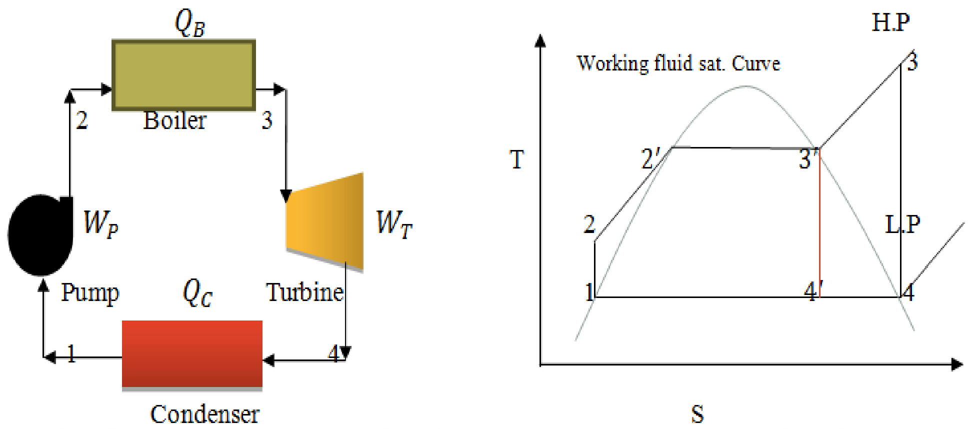

Figure 1.1: Simple steam turbine power plant.

There is a basic working principle in the simple steam turbine. This principle is utilized for the generation of electric power as in figure 1. 1; entering of saturated water, where the, pump, is compressed it, to, a high, pressure. After compressing the water, it is heated to reach the state of superheated steam in the steam generator where the heat is converted from gases to water. From the steam generator, the superheated steam goes out and enters the turbine at high pressure and temperature. The electricity is produced through utilizing work output of the rotating turbine shaft. Therefore, part of the heat which is converted to water is utilized as electrical energy. The wet steam which goes out the turbine is condensed in a condenser to the state of saturated water. In the cooling towers, the cooling water going out the condenser at high temperature is cooled.

1.3 Advance Steam Power Plant

By raising the super-heated steam degree, pressure and by multistage expansion with reheating, the Rankine cycle performance can be advanced. The Rankine cycle efficiency limitation is performed through the high heat of working fluid vaporization. The entry temperatures of a steam turbine are about $565\,C^o$ and steam condenser temperatures are about $30\,C^o$ when the temperature and pressure reach the supercritical levels in the steam boiler. A hypothetical greatest Carnot efficiency for the steam turbine of 63% which is contrasted and a real total thermal efficiency of up to 42% for a coal-fired power station is introduced.

By consolidating the feed water heaters (FWHS), another progress of thermal efficiency is achieved. Regeneration enhances aids in controlling the greater rate of the volume flow of the steam and the cycle efficiency [5]. There are types of F.W.H, and these types are as follow:

1. Direct – contact or (open-sort) feed water heater or open type.

In this sort, to item saturated water at the extraction steam pressure, the extraction steam is straightforwardly blended with the approaching sub cooled feed water.

2. Close type feedwater heater.

In this type, the steam does not blend with feed water, yet rather heat is transported from the extraction steam when it condenses on the external part of tubes when the supplied water streams in them. The supplied water and steam may be at very various pressure in the closed heater.

An ideal arrangement of the basic components in a real power plant is clarified in figure 1.2.

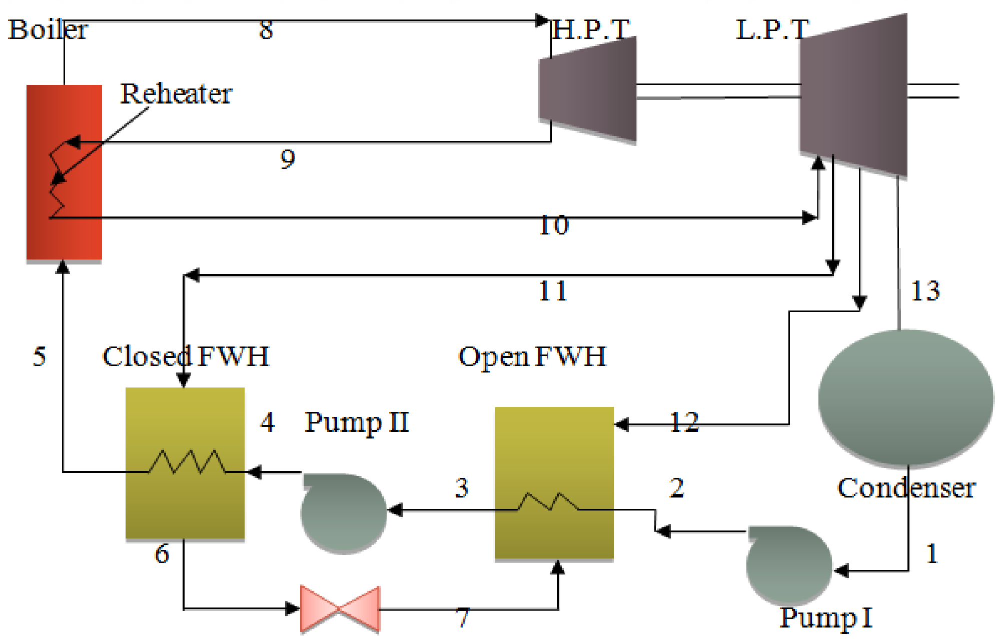

Figure 1.2: Flow diagram of the advanced steam turbine power plant.

1.4 Simulation of Power Plants

High-performance simulators can be developed for analysis and training on complex systems when there are strong and perfect processors, progressive numerical methods and adaptable software. Power plants are complex systems groups with remarkable influences on the industries processes. For analysis, the simulation was designed by applying Simulink Mat lab and Cycle tempo. Mat lab and Cycle tempo can be utilized by the simulation which is designed for purposes of analysis. These programs survey the long-lasting control frameworks conduct and the aggregate plant achievement following framework unsettling influence.

The steam turbine changes over saved vitality of high pressure and high-temperature steam into turning vitality, and this rotating vitality is changed over into electrical vitality by methods for the generator. In every section of the turbine, there are moving blades which are relevant to rotor and stationary blades. Through outing throttle valves (wellbeing close – off valves), fresh steam is funneled to the turbine channel and it is run by free on – off water powered servomotor with bounce back springs. They are intended to work completely open under typical operation and they have the ability to close rapidly. After that, in the high-pressure turbine lodging, the steam goes throughout control valves and it is run by autonomous corresponding hydraulic servomotors.

From the control valves and through a group of nozzles, the steam enters the turbine. The steam is heated again after the high-pressure part. This reheated steam is channeled to the valve chambers, where security close – off halfway pressure control valves associated with the pole can keep running by means for a solitary servomotor described by the earlier thought to be water driven control framework [6].

There are five main parts of a classic steam power plant. These parts are Condenser, Boiler, Feedwater system, Turbine and Generator.

These parts will be briefly described as follow:

1. Boiler:

For the purposes of heating and power, the boiler is used. It can be defined as an enclosed vessel in which water is heated and circulated as hot water or steam. This water goes inside the boiler or steam generator. In the boiler, water takes energy by the chemical reaction of burning fuel in the furnace. The saturated water goes from the economizer to the steam drum. As soon as the water goes inside the steam drum, it passes over the down comers to the lower inlet water wall headers. The water rises through the water walls from the headers and it is converted steam. As the water is converted steam/vapor, the steam/vapor goes inside the steam drum once more. The steam/vapor went along a series of steam and water separators and then dryers in the steam drum. The water droplets are taken off from the cycle and the steam by the steam separators and dryers [7].

A plain simulation, which uses SIMULINK, frames the thermal/control behavior of a domestic gas boiler. In a research programmer, simulation can be considered as the initial stage in investigating the efficiency of control. A summarized review of boiler control gives a rough idea of research developments and current practice [8]. The mathematical methods are put in an application for electrical power generating plants and they are developed by Benson type boiler to describe the basic dynamic behavior of the boiler subsystems. An optimization approach is performed depending on genetic algorithm (GA) for the purpose of estimating the model parameters and also for the purpose of fitting the model's reaction on the actual system dynamic. Simulink is a computer simulation package that was devised by Math works and runs with MATLAB. The package is an icon-driven simulator, it uses standard control system functions to illustrate integrators, delays, scopes and signal sources [9].

2. Turbine:

It extracts thermal energy of pressurized steam for utilizing it in mechanical work on a rotating output shaft . It generates rotary motion , so it is suitable for driving an electrical generator. The steam turbine can be considered as a heat engine. This engine can draw its progress in thermodynamic efficiency by using multiple steps in the steam expansion. The lumped models for every turbine subsections substitute the dynamic system. An optimization approach which is relied on genetic algorithm is applied to estimate the unknown parameters of dynamic models that have more complex structure depended on empirical data. For training process, optimization parameters and suitable fitness function are selected. The operation of models training is accomplished through MATLAB Genetic Algorithm Toolbox and MATLAB Simulink. This operation enables the model training to be based on recorded data or to be performed on the line in simulation space.

3. Condenser:

It makes the exhaust steam communicate with a cool medium (often cold water) for taking the heat out and condensing it back to water, and this process is known as condensate. The cooled surface condenser is utilized by Steam electric power plants. In the turbine, the used steam is exhausted into the condenser. So, when the steam touches the cool tubes which are full of circulating water, it is condensed and then it is taken from the condenser. The cooling towers may be utilized for rejecting the waste heat to the atmosphere when there is a good source of cooling water [10]. The condenser simulation models were advanced through utilizing MATLAB/SIMULINK. This model can calculate condenser pressure, cooling water temperature and hot good water temperature. The simulation conclusions were contrasted with design parameters, and the relative errors were between -0.2% and $+0.2\%$, which means that the simulation model is sensible. The developed model of the condenser is a significant base for the simulation of the power plant [11].

4. Feed water System:

The feed water is often saved, preheated and conditioned in a feed water tank at thermal power stations. Then it is provided to the boiler through a pump. The surface condenser removes the latent heat of vaporization when a conventional steam electric power plant utilizing a drum boiler . Then, the condensate pump pumps the condensate water through a feed water heater. Preheating the feed water progresses the thermodynamic efficiency of the system by decreasing the irreversibility engaged in steam generation. That leads to decrease the plant operating costs and also leads to avoid the thermal shock.

By performing analysis utilizing probabilistic approach, the model of Simulation for Feed Water System and Condensate of a thermal plant are done. Ash handling system, feed water system, coal handling system, water treatment system, condensate, steam generating system, feed water system and air distribution system are included in the plant. By depending on Markov birth-death process, the mathematical formulation is performed. The availability simulation model helps in performing the evaluation of the system by using the rates of failure and repair which are taken from sheets of maintenance history. By applying software package

MATLAB 7.0.4., the analysis is performed. The different availability matrices are developed useful for progressing the existing maintenance schedule [12].

The feed water system is constructed of the following components [8]:

a) Closed FWH.

b) Open (Direct – contact) FWH or deaerator.

c) Pumps.

d) An economizer and valves.

Closed feedwater heaters are shell and tube heat exchangers. The feed water is heated through turbine extraction steam and it is streams throughout the tubes. For removing dissolved gases such as oxygen from the feed water to steam generating boilers, a deaerator or open type FWH is used. The water is pumped back to the boiler by the means of feed pump. For doing this, pressure should be raised to at least boiler pressure. For regulating, directing or controlling the fluid flow (gases, liquids, fluidized solids, or slurries), the valve is used for opening, closing, or partially obstructing different passageways.

5. Generator (Electromechanical Energy Transfer):

The mechanical energy is converted into electrical energy, and the steam turbine operates a generator to perform this process. The efficiency of energy transformation of the elevated capacity generators can be 98% or 99% for a huge machine. Figure 1.3, shows the electricity generation.

The simulations of the industrial turbine were advanced with methodology, and they are used for analysis and training. The simulated turbine – generator system is controlled by using a Programmable Logic Controller (PLC), which is connected to the Digital Control System (DCS) that supplies operator screens and supervisory control. Through utilizing graphic user interface (GUI) tools and general – purpose simulation software, the simulator was constructed. The system of simulator reached a networked three – PC platform which should be with a special keyboard and touch – screens to emulate the actual DCS keyboard. The mathematical models of the systems of generator, auxiliary and turbine are included in the resulting simulation [13].

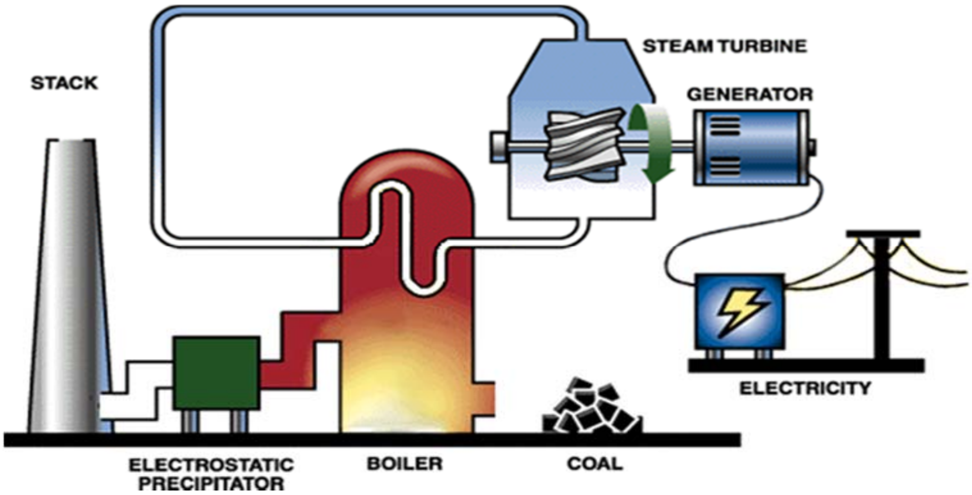

Figure 1.3: Depicted of the electricity generation.

1.5 The Analysis of Energy and Exergy

This analysis to power generation systems has an importance for the economical utilization of the resources of energy. So, it draws the attention of system designers and scientists in our time. Some of these system designers and scientists studied the exergy analyses [14, 15] and the improvement of efficiency [16]. Efficiency is one in every of the foremost oftentimes used terms in thermodynamics, and it indicates however well an energy conversion or method is accomplished. Efficiency is additionally one in every of the foremost frequently victimized terms in thermodynamics and is commonly a supply of bewilderment. This is as a result of efficiency is commonly used while not being properly outlined initial [17]. Efficiency traditionally has been primarily outlined supported the first law (i.e., energy). Exergy analysis is a great tool within the energy systems style, assessment, optimization, and improvement [18, 19]. A thermodynamical analysis system includes energy and exergy analyses that leads to get an entire image of system behavior.

1.6 The Aims of the Thesis

The objectives of this study were to accomplish:

Creating the theoretical model in order to simulate steam turbine power station. In order to reach the best state of parameters which is used by a steam turbine, the two programs Mat lap and Cycle tempo are used.

The mathematical structure of the mentioned techniques is analyzed by using Cycle tempo and Mat lab (V2014a) to simulate steam turbine power station.

Comparing simulated cycle with practical data that taken from the actual plant. As well as to reach the best state of parameters which is used by a steam turbine. The exergy destruction sources are defined and classified that leads to make workable recommendations.

1.7 Layout of Thesis

This thesis is included six chapters, it presents as follows:

Chapter 1: Introduction

Chapter 2: Literature Survey

Chapter 3: Theoretical analysis and, simulations models

The Steam Plant Energy and Exergy analysis,

Chapter 4: The Results and Discussion:

Energy and Exergy Analysis Results

Chapter 5: Conclusions and Suggestions

CHAPTER TWO

LITERATURE SURVEY

2.1 Introduction

The literature review is concluded by focusing on the steam turbine predicting the performance under different environmental factors and operating conditions. The literature review contained prediction methods. These methods are of experimental nature section 2.2 shows the mathematical and analytical steam turbine power plant formation genetic algorithms ways. Part 2.3 displays the analysis of plant exergy. Part 2.4 presents the essential idea of the study. This literature review deals with simulation and mathematical model, genetic algorithms and exergic analytical of the steam turbine.

2.2 The plant mathematical and simulation model of steam turbine

The mathematical modeling is performed by**.Ghaffariet.al, 2007 [20],** advanced variable parametric models for the steam power generating plant turbine and for a boiler subsystem relying on the thermodynamic and physical laws. The GA was accomplished in order to set the model parameter by using experimented data. These models were used for the purpose of super heaters part. The simulation conclusions revealed that the progressive way flexibility and influence was less deviation and more accurate. This way was developed for startup and shutdown modes and such strange conditions. They took into consideration the combustion process and heat exchange in the furnace. Different models were improved for these conditions.

Ali Chaibakhsh and Ali Ghaffari, 2008 [6], took into consideration a 440 MW plant steam turbine and Benson boiler for the approach of formation. These researchers described the dynamic steam turbine, through progressing the nonlinear mathematical way. The associated variables of advanced ways were either adjusted through applying Genetic Algorithm (GA) or they were determined through empirical relations. Comparing the actual system response validated the progressive method accuracy with the response of the turbine generates a model in steady state.

Philip S. Bartels and Christine K. Kovarch, 2002 [8], improved a simulation model for industrial turbine which is used for analysis and training. The simulation was ruled by a Programmable Logic Controller (PLC), that is linked to a Digital Control System (DCS) and operator screens. Also, it is utilized for assessing system design relied on the actual system data.

Tomasz Barszcz and Piotr Czop, 2009 [10], improved a plant method of 225MW coal-fired unit. VPP is a Virtual Plant depended on a method appeared in Mat lab / Simulink environment. Depending on the Stodola equation which was founded in order to extract the transient conditions, the static features presented relationships between the quantities of output and input of an elementary turbine section.

Through facilitated data-driven model, the dynamics model of station turbine was approximated and is expressed from the principle of mass conservation. For the delay of inlet and steam chest, the time of delay (τ) has been used in (0.3 - 0.35) sec. arrangement, steam extraction pipeline in (0.35) sec. arrangement and crossover delay in the (0.4- 0.6) sec. arrangement.

HataitepWongsuwarn, 2010 [11], used the controller of industrial process and thermodynamic properties of substances in numerical simulation. The subtractive clustering and Neuro-fuzzy system (NFs) were utilized for counting energy properties in an experiential steam power plant. From the subsystem of the properties of thermodynamic like superheat steams or saturated water, Neuro-fuzzy models are constructed. The comparison between experiential conclusions of the nonlinear Neuro-fuzzy model and many back propagation neural networks (BNNs), improved that the NFs modeling was closer to thermodynamic properties.

Omar R. I M, 2012 [12], researched, by mathematical modeling, simulation and identification, the dynamic responses of SC (Supercritical) power plants. In the operation of modeling improvement Genetic Algorithms were utilized for model response optimization and parameter identification. The model was emphasized for confirmed process states with various groups of data gained from 600MW SC power plant. A more dependable and plainer identification technique than classical optimization techniques is presented by a Genetic Algorithm. In this study, three schedules were examined. The first schedule was made for small load changes about symbolical situations whereas the others were used in order to manage high load changes.

M S Jamelet. al, 2013 [13], simulated a 200 MW gas-fuelled steam turbine plant in Basra, in Iraq. The simulation system rated the thermodynamic parameters of this plant. For this purpose, he applied the program of Cycle Tempo. The results of the simulation were affirmed by data gained from the log sheet information equipping by the station.

Mircea Dulau and Dorin Bica, 2013 [21], shown the numerical displaying of the steam turbine unit, relied on the progression condition. This method identified the recreation scheme with IP, HP and LP segment. The use of Simulink/Mat lab reproduced the conduct of the pole torque, relying on the control valves opening, with uncommon parameters of the procedure, at that point the progression reaction of the steam turbine, with relative and load control calculation. A first-order transfer function can be used as clarified by Simulation conclusions, in spite of the difference of the steam turbine parameters. In general, there are two factors effect the dynamic response of a steam turbine: the storage action in the reheater and entrained steam into high-pressure turbine section. Characterizing the dynamic response of a steam turbine with regard to changes in the opening of steam valves (RHV position and HPV position) can be performed.

2.3 Exergy Analysis

Eric Conklin, 2006 [22], presented the analysis of natural gas-fired steam boiler and the analysis of second law thermodynamic of a cogeneration plant at Rice University. The analysis consists of a lot of plant components such as valves, piping, boilers, turbines. This relentless state plant working conditions were identified relying upon obtainable slanting information and the finishing up conditions were illustrative of the heft of the plant's working hours. The exergy losses from every plant competent were ascertained relied on these working states to identify the correct framework misfortunes.

A. Rashad and A. El Maihy, 2009 [23], displayed the analysis of exergy and energy of Shobra El-Khima power plant in the city of Cairo. The accomplishment of the plant was characterized through independent component method. The condensers had extra energy losses for the Maximum load (404.653 MW) and at 50% load (278.849 MW) is lost to the environment. In the turbine system, the ratio of the exergy losses to the overall exergy losses was maximized (42%) at Max load and (46.1%) at 50% load, followed by the condenser (28%) at Maximum load and (20.3%) at 75% load.

C. Mborahand E.K. Gbadam, 2010 [24], researched exergy and energy analysis and their use to determine the magnitude and locations of losses to raise the fulfillment of a 500 MW open system steam power plant. The wanted outputs (heat, work and irreversibility) of the different components were identified and assessed utilizing mass, exergy and energy balance equations. The conclusions appeared that 50 % of heat energy was destroyed. As a result, extra improvement in the combustor will increase the plant accomplishment.

S.C. Kaushiket.al, 2011 [25], presented a review of various researches on thermal power plants. Exergitic analyses which are displayed in this study helped in understanding the accomplishments of combined cycle thermal power plants and in determining design probable advancements of efficiency. For advancing the power generation opportunities in the power plants, it presented a perfect solution. The progressing of the efficiency of some components was proposed through raising their size. This concerned raises the cost and therefore there was a limiting size would not be economically due to the increasing size**.**

Vosough Amir, **2012 [26],** explained the valuable idea of exergy and vitality use, as utilized for the evaporator framework. The exergy and vitality efficiencies had been set for the heater. The exergy and vitality efficiencies were 89.21% and 45.48%, separately. The combustion chamber, as it had been discovered, was the starting donor for exergy pulverization that took after by warm converter of a kettle framework. Moreover, in order to raise gas-fired steam power plant efficiency, the modifications were investigated by decreasing irreversibilities in the steam generator, through diminishing the abundance burning air portion, or/and the stack gas temperature. The efficiencies of aggregate plant exergy and vitality raised by 0.19% and 0.37%, separately when the part of overabundance ignition air diminished from

0.4 to 0.15, and by 0.84%, 2.3% when reducing the stack-gas temperature from 137°C to 90°C.

M. Pandy and T.kGogoi, 2013 [9], demonstrated the exergy and energy examination of a reheat recovery vapor control cycle. Exergy effectiveness, energy productivity, and the irreversibility comes about picked up from the reproduction had been displayed as charts. The temperature differed from 500 to 800 K for every supercritical pressure and the supercritical pressure was changed from 250 to 400 bar. The parametric search clarified that the exergy effectiveness and cycle effectiveness raise with raising the temperature and pressure, as it is clarified by the parametric search. This was because of reducing exergetic and energetic losses at increased temperature and pressure.

M. M. Rashidiet. al, 2014 [27], looked into usage an exergy investigation of a steam cycle with dual reheat and turbine extraction. Six radiators were utilized, three of these heaters were at high pressure and the other three heaters were at low pressure with deaerator. The irreversibility and exergy examinations of every part were recognized. Impacts of heater outlet steam temperature, turbine input pressure, and condenser back weight on the first and second law efficiencies were identified. The conclusions identified that the biggest exergy misfortune happened in the heater took after by the turbine. The conclusions additionally uncovered that the aggregate warm proficiency and the second law productivity lessened as the condenser weight raises for any settled outlet heater temperature. Also the better estimations of the extricated pressure of (HP, IP, and LP) turbine that presented the greatest first law efficiencies were gained relied on the wanted heat stack comparing to every leave boiler temperature.

2.4 Contribution

Most of the searches were contained steam plants simulations over computer program and contrasted the conclusions with the workable information of the actual power plant. As well, prior searches were achieved on the one steam plant and either Cycle– Tempo or Mat lap. There are essential points in this study, and these points are:

This research used Cycle –Tempo and MATLAP computer programs and it used one steam power plants (160MW).

The power plant simulation that compared with practical data for the one real plant and the results gave good agreement between them.

The thermodynamic properties are modeled by usage of Mat lap program and they calculate the energy properties within the practical steam power plant.

Exergy and energy analysis are used to analyze the accomplishment of thermal systems.

In this study included the power station with a low power cycle, because they are available in Iraq.Exergy results importance has been taken in this study and the presence of the results of Simulation in Cycle Tempo program.

CHAPTER THREE

THEORETICAL ANALYSIS AND SIMULATION MODEL

3.1 Introduction

Power plants utilize steam turbines to transfer heat energy to mechanical energy and then to electric energy. High-pressure steam and high temperature steam increase by the turbine. The variation between the outlet and the inlet steam of the concluding enthalpy is transferred into rotational energy on the shaft that rounds a generator leading to produce electrical power.

There are two parts in this chapter; the first one involves plants exergy and energy analysis, while the second part mathematical simulation model with computer programs. The steam turbine plant modeling is established by two techniques, the first one is MATLAB version (V2014a) with m – files Simulink and the second one is Cycle – Tempo (Release 5) program Simulink to characterize the mass, heat balances and thermodynamics for the components, in and steady case. The power station of the AL – Dura in Baghdad type (K–160–13.34–0.0068) is investigated figure 3.1.

The simulation model was used for the purpose of solving this station for steady state condition through software MATLAB and Cycle– Tempo programs and for different load capacity.

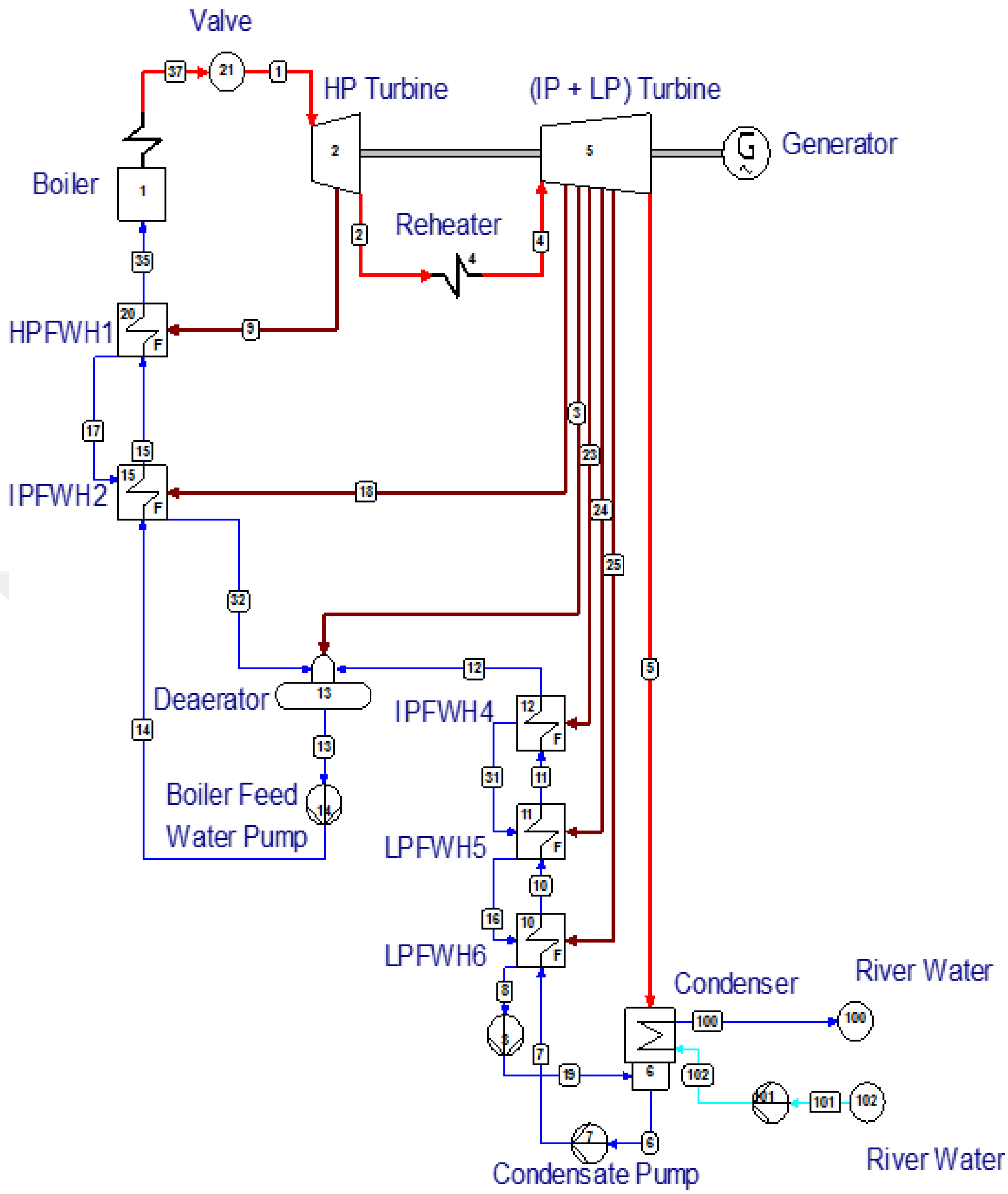

Figure 3.1: Steam turbine plant heat cycle (K–160–13.34–0.0068)

3.2 The Steam Plant Energy and Exergy Analysis

3.2.1 System Configuration

The type (K–160–13.34–0.0068) of the steam turbine is examined and it is contrasted with the modeling approach. IP, LP and HP turbines are included in this type. Additionally, the feed water heaters, bleeding extractions, and other auxiliary system are comprised of the system. The arrangement of turbine and steam are clarified in figure 3.2. There is four LP closed feedwater heaters (CFWHs) in the system, one is open feed heater/deaerator and the other three are high pressure closed feedwater heaters. The feed water pump is turned by a separate 32 MW turbo driver. The feed water pump pressurizes the water to 29.8 MPa, despite the pressure at the HP turbine inlet falls to 13.34 MPa. The output power control can meet the demand by the feed water pump speed control together with the flow control valve.

The superheated steam at $560C^{\circ}$ and 13.34 MPa pressure comes in the HPT section. The inlet steam pressure falls about 1.2 MPa through passing over the turbine chest system. After that, the steam extends in the HPT and it went into the heater. The outlet pressure and temperature of the HP section at the full load processes are $300C^{\circ}$, 3.51 MPa, respectively. At $540C^{\circ}$ and 3.42 MPa pressure, the reheated steam goes into IP section. Exhaust steam from IP turbine is supplied to LP turbine at $231.24C^{\circ}$ and 0.292 MPa. The inlet pressure and temperature of the condenser are $26.27C^{\circ}$ and 0.0034MPa. The taken – out steam from the latter LP and IP extractions point is used for heating the feed water in LP heaters train.

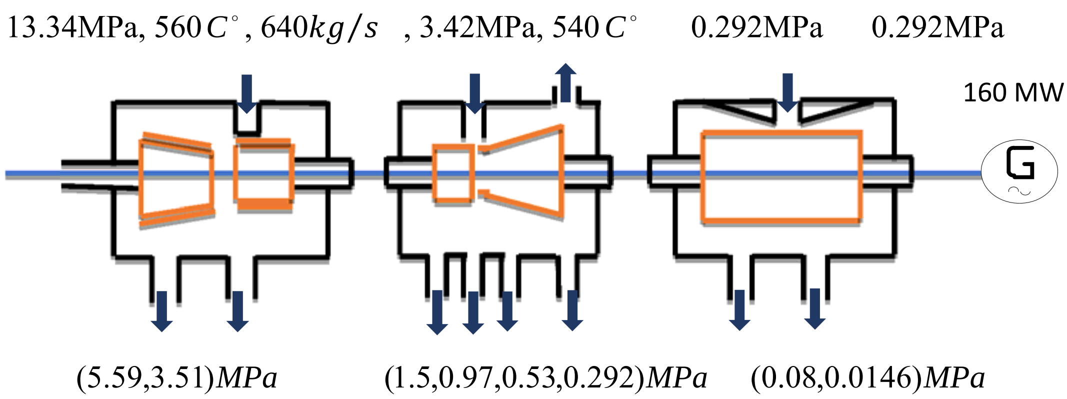

Figure 3.2: Steam turbine section and (K–160–13.34–0.0068) type of extraction

3.2.2 Assumptions

For this model, the following assumptions are taken into consideration:

1. The reference environment temperature (T_0) is $25C^{\circ}$ and pressure (P_0) is 1.013 bar.
2. In each fully opened governing valve, steam mass flow rate is the same
3. $\approx$ 4% of inlet pressure is the pressure losses in fully opened governing valve.
4. The speed ratios ($^{u}/_{c_f}$) is supposed be similar in every turbine phase,
5. The pump's efficiency is equal to 85%.
6. Terminal temperature difference (TTD) is $4C^{\circ}$.
7. When going into the combustor (T_{og}) = 25 C°, the temperature of flue gases is equal to an ambient air temperature [28].

3.2.3 Thermodynamic Properties of Steam and Water

In mathematical controller and simulations of industrial operation, these properties are needed. Controller and Simulation need them to calculate energy and exergy in the system. Fast calculation of these properties is necessary for measuring many process variables [11]. The thermodynamic properties are through utilizing the program of Mat lab that is constructed for every thermodynamic properties subsystem.

1. Saturation Properties

Three main equations from the triple to the critical point can represent the different thermodynamic properties.

The main equations which cover the nine properties are clarified in (Appendix A) [29];

$$T(s) = A + \frac{B}{[\log P(s)] + C} \tag{3.1}$$

$$LogP(s) = \sum_{N=0}^{9} A(N)T(S)^{N} + \frac{A(10)}{T(s) - A(11)} \tag{3.2}$$

$$Y(S) = A + BT(C)^{1/3} + CT(C)^{5/6} + DT(C)^{7/8} + \sum_{N=1}^{5} E(N)T(C)^N \tag{3.3}$$

Where: Y(S) represents a property of saturation liquid. The first four terms of equation (3.3) refer to the variation of properties in the critical region, while equation (3.1) and (3.2) are modified forms. In equation (3.3), T(C) = [T (CR) – T(S)]/T (CR) where T (CR) is the critical temperature (Appendix A).

2. Steam properties, vapor and liquid phase

The following functions are applied to certain properties of steam and liquid phases relied on steam pressure where $P \approx P(S)$ are as following [29];

Presenting equation (3.3) to find Y(S) in this case.

And for liquid phase;

$$h_f = Y(s)h(fCR) \tag{3.4}$$

$$s_f = Y(s)h(fCR) \tag{3.5}$$

And for vapor phase:

$$h_g = Y(s)h(gCR) \tag{3.6}$$

$$s_g = Y(s)s(gCR) \tag{3.7}$$

The specified enthalpy in two phases is presented by;

$$h_{fg} = Y(s)h(fgTP) \tag{3.8}$$

Where: h(fCR), h(gCR), h(fgTP), s(fCR), s(gCR) all constant value defined in (Appendix A)

3. Superheat Properties

The equations are more complex in the superheat region. The pressure (P_{in}) and temperature (T_{in}) should enter as independent variables in this region. These equations are as follows (Appendix A) [29];

$$v(PT) = \frac{RT_{in}}{P_{in}} - B(1)\exp[-B(2)T_{in}] + \frac{1}{10P_{in}}\{B(3) - \exp[\textstyle\sum_{n=0}^{2} A(N)T(S)^N]\}\exp\left[\frac{T(S) - T_{in}}{M}\right] \tag{3.9}$$

$$h(PT) = \exp\sum_{n=0}^{2} A(N)T_{in}^{N} - A(3)\exp\left[\frac{T(s)-T_{in}}{M}\right] \tag{3.10}$$

$$s(PT) = \sum_{N=0}^{4} A(N)T_{in}^{N} + B(1)\log[10P_{in} + B(2)] - \sum_{N=0}^{4} C(N)T(S)^{N}\left\{\exp\left[\frac{T(S)-T_{in}}{M}\right]\right\} \tag{3.11}$$

4. Compressed liquid

The following equations are utilized in the liquid region for compressed water [13];

$$v = \sum_{i=0}^{6}[c_{i,0}(\frac{T_{in}}{100})^{i} + \frac{P_{in}}{10}\sum_{i=0}^{6}[c_{i,1}(\frac{T_{in}}{100})^{i}] + \left[\frac{1}{(\frac{P_{in}}{10})+2} - 0.227 + 0.018\frac{P_{in}}{10}\right] [\sum_{i=0}^{6}[c_{i,2}(\frac{T_{in}}{100})^{i}] + 4.55.10^{-12}.(\frac{T_{in}}{100})^{i}] \tag{3.12}$$

Where, (c) is polynomial coefficient (Appendix B).

$$h = \sum_{i=0}^{6}[c_{i,3}(\frac{T_{in}}{100})^{i} + \sum_{i=0}^{6}[c_{i,4}(\frac{T_{in}}{100})^{i}\frac{P_{in}}{10}] + \left[\frac{1}{(\frac{P_{in}}{10})+2} - 0.24927 + 0.021\frac{P_{in}}{10}\right] [\sum_{i=0}^{6}[c_{i,5}(\frac{T_{in}}{100})^{i}] + 1.682.10^{-9}.(\frac{T_{in}}{100})^{16}]]1000 \tag{3.13}$$

3.2.4 Calculations of the Characteristics Preliminary

The objective determining the basic points of the inlet and exit of steam from a steam turbine. The pressure estimation relies on the pressure losses as the following [28];

$$\Delta P_{0'} = (0.03 - 0.05)P_{0'} \tag{3.14}$$

Where, ($P_{0'}$) refers to the pressure which comes from the boiler to the valve.

Therefore,

$$P_1 = P_{0'} - \Delta P_{0'} \tag{3.15}$$

The point corresponding to the steam state can be found from the certain values of (P_1&T_1). The pressure value of steam exit from turbine with high pressure is clarified as equation [37];

$$P_3 = (0.17 - 0.3)P_1 \tag{3.16}$$

The steam pressure value can be identified as the following formula;

$$P_4 = P_3 - (\Delta P_4)P_3 \tag{3.17}$$

The pressure which losses range in the reheater (ΔP_4) is≈ (1 – 2 %), and the formula [28] can determine the steam pressure at the low-pressure turbine inlet;

$$P_{24} = (0.085)P_4 \tag{3.18}$$

3.2.5 Calculations of the Steam Flow Rate

The rate of steam flow to the first stage of turbine (G) can be identified by choosing the nominal electrical power (N_{el}) as the following formula;

$$G = \frac{N_{el}}{\Delta h_{act}.\eta_m \eta eg} \tag{3.19}$$

Where; the mechanical efficiency of a turbine that equals to 0.996 is represented by(η_m), while (η_{eg}) represents the electric generator efficiency that can be selected as 0.987 [30]. The following formulas clarify the reduction of the actual heat through the steam turbine (Δh_{act}) is determined by the formula;

$$\Delta h_{act} = \eta_{ir}(Q_B + Q_{rh}) \tag{3.20}$$

Where; (η_{ir}) Represents the plant thermal (inlet relative) efficiency without regenerative heating and it is identified as in the following formula;

$$\eta_{ir} = \frac{(\Delta h_s \eta_s)^{HPT} + (\Delta h_s \eta_s)^{IPT+LPT}}{(\Delta h_s \eta_s)^{HPT} + (h_4 - h_{5s})} \tag{3.21}$$

Where; (η_s) is the entropic efficiency of the supposed turbine to be; $(\eta_s^{HPT}) = 0.88$ and $(\eta_s^{IPT+LPT}) = 0.86$ [30]. At every turbine section, the isentropic reduction in enthalpy (Δh_s) can be determined, where (h_4) represents the enthalpy at reheater and (h_{5s}) represents the enthalpy of steam that leaving LP turbine. In the case of finishing expansion in the turbine in the superheated steam region, it can be determined as follows [31];

$$\Delta h_s = \frac{K}{K-1} P_a v_a [1 - (\frac{P_b}{P_a})^{\frac{K-1}{K}} \quad (3.22)$$

Where; At outlet and inlet of the turbine phase (P_a, P_b) represent the steam pressure, (v_a) refers to the certain steam inlet value, and (k) refers to the isentropic index k =1.135 for wet steam and k = 1.3 for dry superheated steam. $(Q_B and Q_{rh})$ refer to the boiler and reheater heat respectively.

$$Q_B = (h_{37} - h_{35}) \text{ at boiler} \quad (3.23)$$

$$Q_{rh} = (h_4 - h_3) \text{ at reheater} \quad (3.24)$$

The values of enthalpies which enter the HP turbine, the IP+LP turbine and the enthalpy of feed water which enter the boiler can be determined. The value which develops the relative efficiency of the turbine is determined below;

$$\xi_r^{rh\infty} = \xi_r^{\infty} \left(\frac{\xi_r^{rh}}{\xi_r^{\infty}} \right) \quad (3.25)$$

Where: $\left(\frac{\xi_r^{rh}}{\xi_r^{\infty}} \right)$ the coefficient of improvement of the efficiency with extractions and reheating is mentioned in figure 3.5 [31].

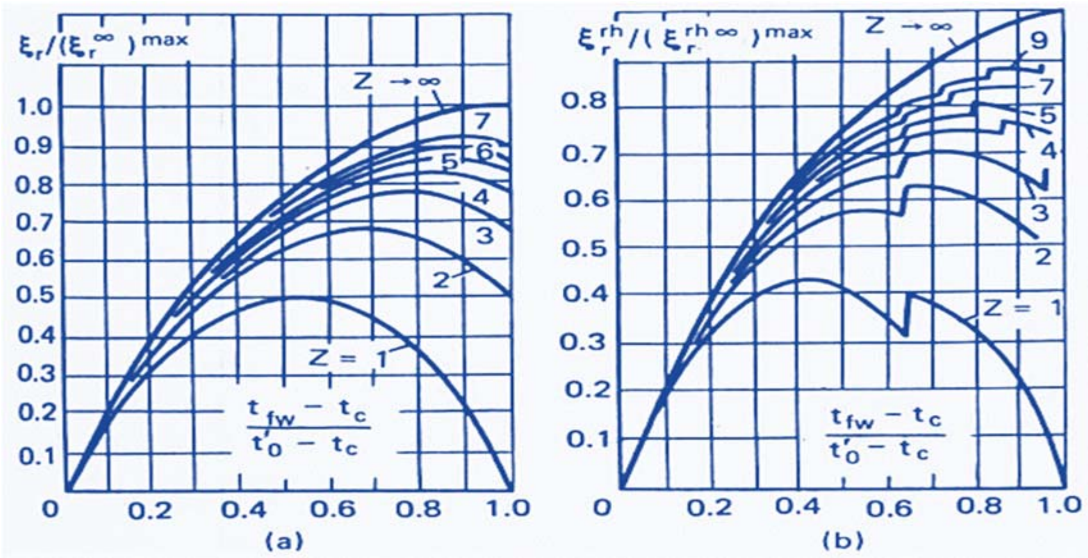

Figure 3.5: Relative gain in particular heat consumption for plants (a) without steam reheating and (b) with steam reheating [39]

For an infinite number of regenerative water heaters, the gain in efficiency (ξ_r^{∞}) is clarified as below;

$$\xi_r^{\infty} = 1 - \frac{1 - \dfrac{T_5(S_4 - S_{5s})}{(h_0 - h_{1s}) + (h_4 - h_{5s})}}{1 - \dfrac{T_s(S_4 - S_{35})}{(h_0 - h_{1s}) + (h_4 - h_{35})}} \tag{3.26}$$

Where; $(h_4 \,\&\, S_4)$ represent the enthalpy and entropy at reheater,

(S_{5s}) represents the entropy of steam which leaves L.P.T,

(T_5) represents the temperature at condenser,

(h_{35}) and (S_{35}) represent the feed water enthalpy and entropy which enter the boiler,

(h_0) represents the enthalpy of basic steam

and (h_{1s}) represents the steam enthalpy upon isentropic expansion in the turbine with high pressure.

The internal relative efficiency of the plant with regenerative and reheat $(\eta_{ri})regen$ of a turbine is determined below;

$$(\eta_{ri})regen = \frac{\eta_{ir}}{1 - \zeta_r^{rh\infty}} \tag{3.27}$$

Equation (3.19) determines the rate of mass flow (G) which goes through the basic steam value to the turbine, and the rate of steam flow to the condenser is determined in the equation [32];

$$G_C = \frac{N_{el}}{(h_s - h_{5s})\eta_m \eta_{eg}} \left(\frac{1}{(\eta_{ri})_{regen}} - 1 \right) \tag{3.28}$$

The efficiency of every turbine section is determined as below:

The equation [32]evaluates the efficiency of the governing phase;

For single – row phase;

$$\eta^{gst} = \frac{K_u}{C_f} \left(0.83 - \frac{2 \times 10^{-4}}{G} \sqrt{\frac{P_{0'}}{V_{0'}}} \right) \tag{3.29}$$

Where: $(\frac{K_u}{C_f})$ represents a coefficient that is calculated from the curve in figure 3.6. ($v_{0'}$) *and* ($P_{0'}$) refer to the specific volume and pressure before the governing phase nozzles and (G) refers to the rate of steam flow.

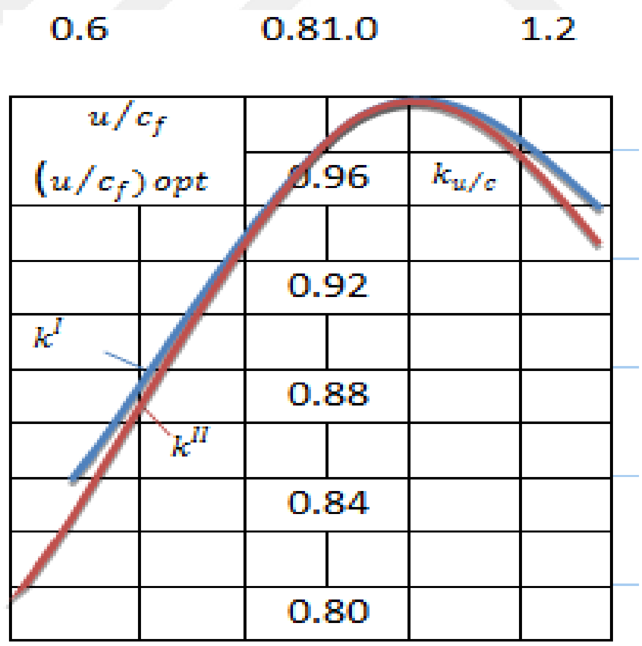

Figure 3.6: Rectification of efficiency of a governing phase to examine the deviation of velocity ratio u/c_f from the ideal value

k^{I}_{u/c_f}– For single – row phases (0.88 – 0.98) and k^{II}_{u/c_f} – for double – row phases (0.82 – 0.98) [33].

Calculating the first section efficiency of non-controlled phases of turbine is as below;

$$\eta^{HP.I}=\left(0.925-\frac{0.5}{G_{av}V_{av}}\right)\left(1+\frac{\Delta h_0^{gr}-600}{20000}\right)\left(1-\xi_{ev}\right)$$

(3.30)

Where; $G_{av}=\sqrt{G_1 G_2}$ represents the rate of average steam flow to the set of phases.

$v_{av}=\sqrt{v_1 v_2}$ Represents the average certain volume of steam.

$(G_1,G_2) and (v_1,v_2)$ represent the rates of flow and specify the volumes of steam before and after the phases set.

(Δh_0^{gr}) represents the obtainable heat fall of the set,

(kJ/kg) and (ξ_{ev}) represent the exit velocity coefficient that is equal to0.01 [32].

The second, third section and intermediate pressure turbine efficiency $(\eta^{HP.II}, \eta^{HP.III} and, \eta^{IPT}$ is determined similarly by equation (3.30) In the similar way efficiency (η^{LPT}) of stages of low pressure turbine is determined [32].

3.2.6 Mass and Heats Balances of Feed Waters Heaters

From figure 3.1 the sample heat balance of FWH are:

1. The heat balances for the heaters, No.20;

$$m_{16}h_{16}+h_{34}=h_{35}+m_{16}h_{36} \tag{3.31}$$

2. For heater (Deaerator);

$$m_{20}h_{20}+\left(1-\sum_{1}^{n}m_n\right)h_{12}+(m_{16}+m_{17}+m_{18})h_{32}=h_{13} \tag{3.32}$$

3. The heat balances of the heaters, No.11;

$$m_{24}h_{24}+m_{23}h_{31}+(1-\sum\nolimits_1^n m_n)h_{10}=(m_{24}+m_{23})h_{30}+(1-\sum\nolimits_1^n m_n)h_{11} \tag{3.33}$$

3.2.7 Exergy Analysis of the Steams Plants

When the system is in a dead case, exergy is mainly the maximum work which is gained from it. A clear datum which should be utilized in many of the calculations is the ambient (surrounding) states. When the system is in mechanical and thermal equilibrium, it approaches this state. The seconds laws in thermodynamic is a generic terms for concept that identify the maximum of potential works for the systems. The analysis of energy relies on the 1^{st} laws in thermodynamic, which has related to the energy conservations. The analysis of second law utilized the laws of energy conservation and mass with the entropy's. The exergitic analysis merges the 1^{st} and 2^{nd} thermodynamics laws. Generally, the overall exergy of the systems is the totality of K.E exergy's, P.E exergy, physical exergy and chemical exergy. The 2nd law efficiency presents a correct measure of energy system performance from the thermos dynamic view point [9]. Presenting the analysis of components division exergy of the thermals power plants systems is below:

a. Boilers

The equation of energy balances for the gaining: $m_g C_{pg}$ in figure 3.7 be;
Heat fed to the steams = Heat refused from flues gases, So:

$$(G_{37} - m_{16} - m_{17}) * (h_4 - h_3) + G_{37}(h_{37} - h_{35}) = (T_{g1} - T_{g2}) * m_g C_{pg} \tag{3.34}$$

The reduction unavailability across the components is equals to the exergy loss (irreversibility) in the components. Boiler irreversibility is: $I_{boi} = (Ex_{in} - Ex_{out}.)$

$$I_{boi} = \left[(T_{g1} - T_{g2}) m_g C_{Pg} - \left(In \frac{T_{g1}}{T_{g2}} \right) m_g C_{Pg} T_0 \right] - \begin{bmatrix} G_{37}((h_{37} - h_{35}) + ((h_4 - h_3) * (1 - m_{16} - m_{17})) - T_0 (s_{37} - s_{35}) + \\ (s_4 - s_3)(1 - m_{16} - m_{17}) T_0) \end{bmatrix} \tag{3.35}$$

b. Steams turbines

In the turbine, the irreversibility is presented as follows;

$$I_{tur} = (I_{I.P} + I_{H.P} + I_{L.P}) \tag{3.36}$$

$$I_{H.P} = [(1 - m_{16})(s_{17} - s_{16}) + (s_{16} - s_1)] * G_{37} T_0 \tag{3.37}$$

$$I_{I.P} = G_{37}T_0 * [(1 - m_{16} - m_{17} - m_{18})(s_{19} - s_{18}) + (1 - m_{16} - m_{17})(s_{18} - s_4) + (1 - \sum_1^5 m_n)(s_{24} - s_{23}) + (1 - \sum_1^4 m_n))(s_{23} - s_{19})] \quad (3.38)$$

$$I_{L.P} = G_{37}T_0 * \left[(s_{25} - s_{24})(1 - \sum_1^6 m_n) + (s_{26} - s_{25})(1 - \sum_{i=1}^7 m_n) + (s_5 - s_{26})(1 - \sum_{i=1}^8 m_n) \right] \quad (3.39)$$

c. The Condensers

The cooling water mass which is circulated to condenser m_w is identified as follows;

$$G_5 * (h_6 - h_5) = (T_{w102} - T_{w100}) * m_w C_{pw} \quad (3.40)$$

Where: C_{pw}= 4.1868 kJ.K. (kg)$^{-1}$

The Irreversibility in a condenser: $\mathrm{I_{cond}} = (Ex_w - Ex_s)$

$$I_{cond} = \left[m_w C_{pw} In \frac{T_{w100}}{T_{w102}} \right] * T_0 - [G_5 T_0 (s_6 - s_5)] \quad (3.41)$$

a. Pump

In every pump the irreversibility is;

$$I_{pump} = m_w T_0 (s_{out} - s_{in}) \quad (3.42)$$

And the irreversibilities in (n) pumps are;

$$I_{pump} = \sum_{i=1}^{n} I_{pump} \quad (3.43)$$

b. The Feed water heater

1. In an open type (deaerator), the exergy losses are:

$$I_{ofwh} = (m_{13}s_{13} - m_{20}s_{20} - m_{32}s_{32} - m_{12}s_{12}) * T_0 \quad (3.44)$$

2. The irreversibility (exergy losses) for every closed heater is;

$$I_{cfwh} = mT_0 (s_{out} - s_{in}) \quad (3.45)$$

In the closed feedwater heaters, the overall irreversibilities are;

$$I_{tcfwh} = \sum_{i=1}^{n} I_{cfwh} \quad (3.46)$$

The overall Irreversibility (I_{total}) for plant components is;

$$\sum I = I_{boi} + I_{tur} + I_{con} + I_{pump} + I_{ofwh} + I_{cfwh} \tag{3.47}$$

The input of exergy to the system is;

$$E_{in} = \left[m_g C_{pg} (T_{g1} - T_{g2}) - m_g C_{pg} T_0 \left(In \frac{T_{g1}}{T_{g2}} \right) \right] \tag{3.48}$$

The rate of irreversibility of the singular components to the overall irreversibility of the cycle represent the fractional exergy destruction of the component. The loss of fractional exergy for every component is clarified as follows;

$$Ratio = \left(\frac{I_{component}}{I_{total}} \right) \times 100\% \tag{3.49}$$

The exergy efficiency is clarified below;

$$\text{Exergy efficiency} = \left[\frac{E_{x.in} - \sum I}{E_{x.in}} \right] \times 100\% \tag{3.50}$$

Table 3.1 presents the efficiencies of functional exergy for the apparatuses [27].

Table 3.1: The efficiencies of functional exergy for the apparatuses.

Type	*Name*	η_{ex}
1 4	Boiler Reheater	$\frac{Ex_{out} - Ex_{in}}{Ex_{fuel}}$
2 5	Turbine	$\frac{P_m}{Ex_{in} - Ex_{out}}$
4 8,10,11,12,15,16,20	Condenser Flashed heater	$\frac{Ex_{w\,out} - Ex_{w\,in}}{Ex_{s\,in} - Ex_{s\,out}}$
13	Deaerator	$\frac{m_w Ex_{w\,out} - Ex_{w\,in}}{Ex_{s\,in} - m_s Ex_{s\,out}}$
7,19,14	Pump	$\frac{Ex_{out} - Ex_{in}}{P_m}$

The formula [34] identifies the Universal Exergy efficiency (Univ. Exergy eff.);

$$\text{Universal Exergy Efficiencies} = \frac{E_{Xout}}{E_{Xin}} \times 100\% \tag{3.51}$$

The following formula determines the Relative Exergy loss (Rel. Ex. Loss) is:

$$\text{Relative Exergy loss} = \frac{\text{the exergy losses}}{\text{the input of total exergy}} * 100\% \tag{3.52}$$

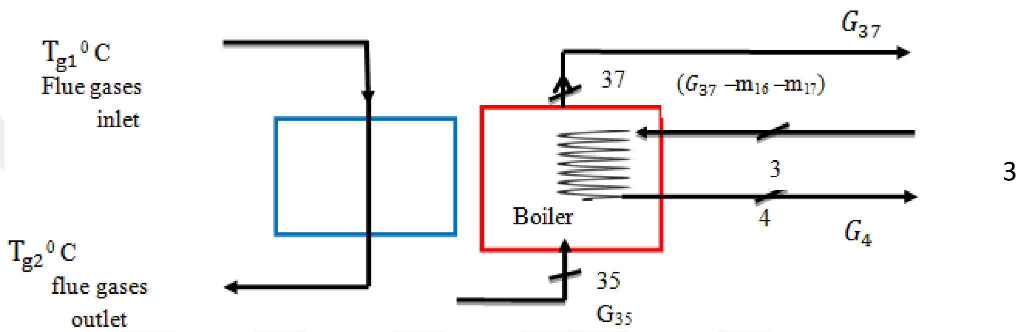

Figure 3.7: Steam generation mass balance

3.3 Simulation Model of Steam Turbine

3.3.1 Turbine Performance

The model carries out primary stable state analysis, relied on input specifications which are identified by users such as power output or primary steam and condensing pressures, and many parameters concerning either boiler and turbine efficiency or power block. The conclusions are summed up by the heat and mass balance of steam generator and power block .

During operating, a turbine is probably operated for a remarkable long time with the changeable rate of steam flow in startup and shut down regimes. The estimated conditions are probably confused due to salt deposition in the steam path [34].

The inlet pressure and extraction pressure to every section of the turbine are determined by using the law of Stodola's Cone (Stodola – 1922) as appear below [35]:

$$\frac{G}{G_0}=[\frac{P_a}{P_{a,0}}*\sqrt{\frac{1-(\frac{P_b}{P_a})^{\frac{n+1}{n}}}{1-(\frac{P_{b,0}}{P_{a,0}})^{\frac{n+1}{n}}}}*\sqrt{\frac{P_{a,0}v_{a,0}}{P_a v_a}}] \tag{3.53}$$

Where (*P*) refers to the pressures

(v) refers to the specific volume.

The sub – index (a) represents the inlet value ,

(b) represents the outlet value

(0) represents the design values,

and n represents wet steam. The polytrophic exponent calculation can be explained as follows (Traupal, 2001);

$$n=\frac{k}{[\frac{kP*(v_{steam}-v_{liquid})}{h_{fg}}*(1-\eta_T)]+1} \tag{3.54}$$

Where *K*: refers to the isentropic exponent, η_T refers to the total turbine efficiency.

3.3.2 The Modeling of Steam Turbine

It is possible to construct the simulation model in terms of energy and mass equation, the equation of state and semi – empirical equation. The dynamic models for every component are uncomplicated empirical relations which connect the variables of the system with a restricted number of parameters. The models training operation is achieved by following Cycle – Tempo and MATLAB Simulink.

The power model contains a model of steam turbine, a generator, and control system. To formulate models of components, changeable equations of energy and mass and conservation have been utilized. Some models were contracted in simplified and advanced versions to perform the model in Simulink and to reach the simulation time within the available time. This model improvement is addressed to the appropriate representation of these variables relevant for modeling. The model, in this case, can clarify the turbine behavior in the MAT LAB/SIMULINK code . In order to merge this model into MATLAB/SIMULINK, it should present variables as enthalpy drop through the turbine, the pressure drop through the turbine , power output , pressure at the extraction lines, etc.…. [35]. The model s components are linked by ports propagating and enabling current steam parameters (pressure, temperature, etc.) or/and rates of mass/energy flow figure 3.8. Figure 3.9 [36]. Clarify the block diagram of steam turbine representation.

Performing the power output control is done by the control valve position which controls the flow of steam to the turbines. Usually, the first order filter models the delay between the various steam path sections. In the different turbines, specific total power fractions are removed, and this is modeled through(F_{HP},F_{IP} andF_{LP}). (τ_{HP}) is the time constant of the delay between the high-pressure turbine and the control valve. (τ_{RH}) Represents the time constant at reheated system and ($\tau_{I,L.P}$) represents the time constant of the delay between low and intermediate pressure turbine, (s) represents Laplace factor.

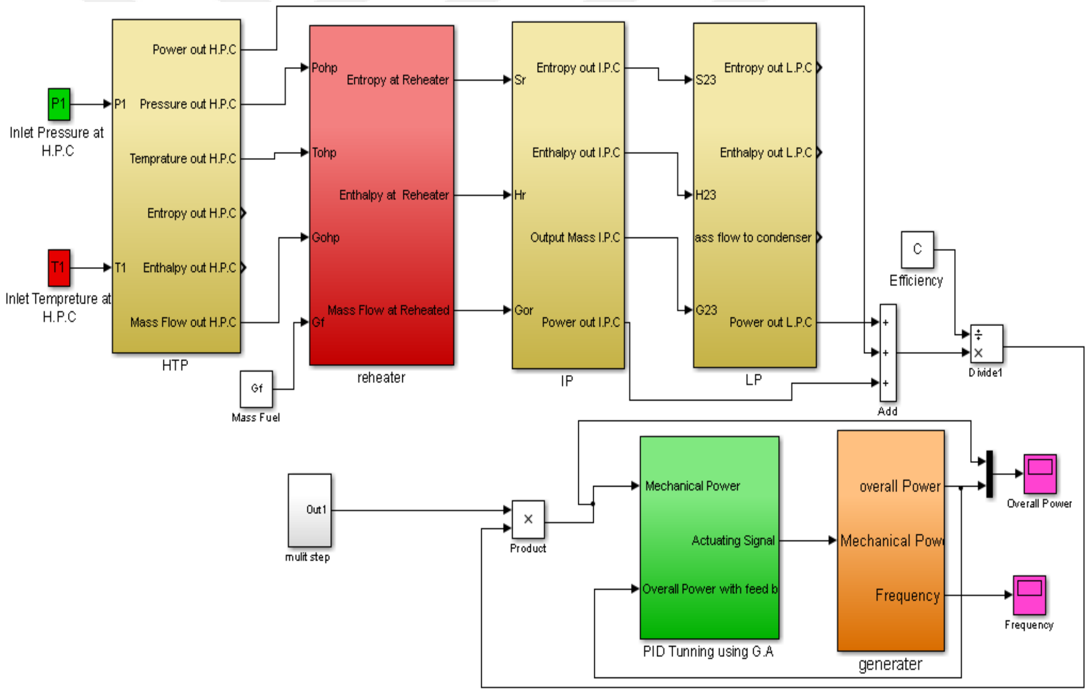

Figure 3.8: Overall turbine and generator models for the power station of AL-Dura (K– 60–13.34–0.0068)

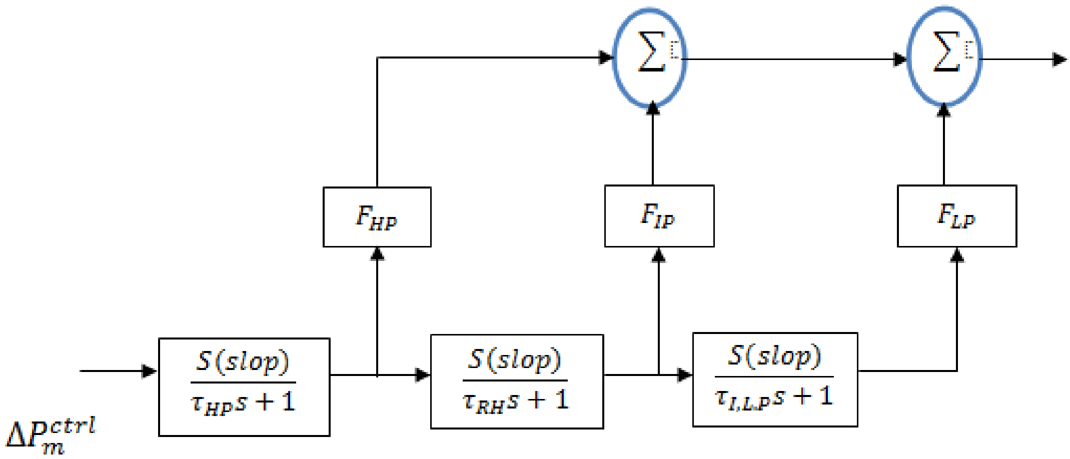

Figure 3.9: The block diagram of steam turbine

3.3.2.1 High pressure turbine

The mass flow should be constant during the entire set of phases, making the application of equation (3.53) possible only in turbine sections with the similar mass flow. So, several interconnected sections will be necessary to describe the behavior of the total turbine.

Thus, the HP turbine three stages. Figure 3.10 clarifies the model for the high-pressure turbine.

The rate of mass flow which is involved with the pressure fall across the first phase of HP turbine is presented by Stodola (1922) [36];

$$G = K_1 \lambda \tag{3.55}$$

Where K_1: represents a constant gained from the information that taken of the responses of turbine,

and (λ) is clarified as follows;

$$\lambda = \sqrt{\frac{(P_1^2 - P_{16}^2)}{T_1}} \tag{3.56}$$

By plotting (λ) verseinlet mass flow gained from the information (presented of Cycle–Tempo program), the linear fitting slop is represented as $K_1 = 714.37$, as in figure 4.22. This represents the constant accuracy.

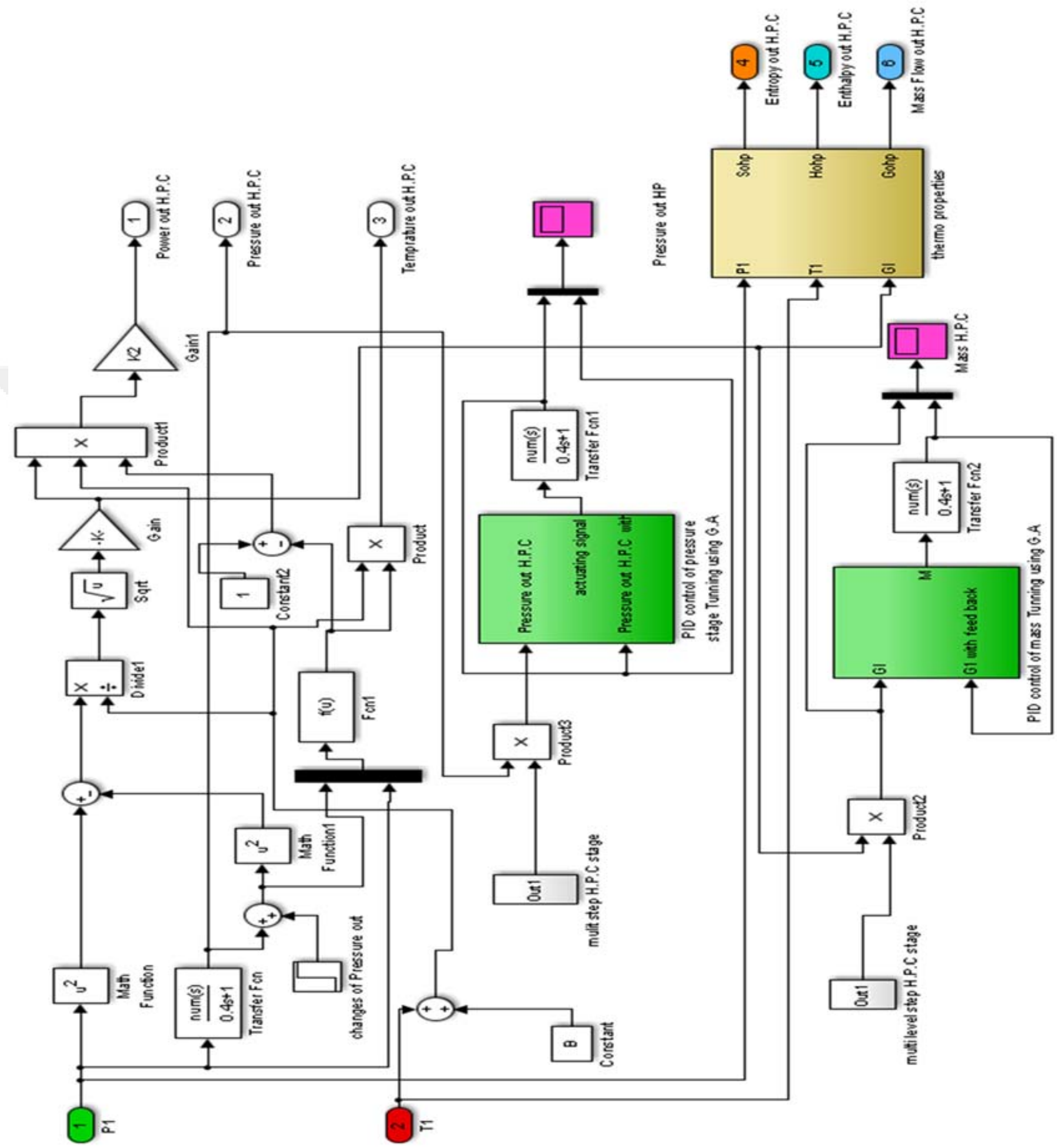

Figure 3.10: High-pressure turbine model for AL-Dura power station.

Performance at variable power is equal to ≈ 2.009 is the ideal value for particular heat capacity(C_P), and 1.2536 is the polytropic index (n).

$$K_2 = \frac{C_p \eta^{HP.1}}{1000} \tag{3.57}$$

The transfer function of existing and entering pressure is;

$$(\frac{P_{16}}{P_1}) = \frac{(slop)S}{\tau_{HPT}s+1} \tag{3.58}$$

The output and input pressure relationship for turbine with high pressure relied on practical information (of Cycle – Tempo program), is clarified in figure 4.24. The higher linear and slope of S (slop) = 0.26101 relation is explained. The first order filter usually models the delay between the various steam path parts. (τ) represents the time constant that is clarified in [37];

$$\tau = \frac{Steam\ mass\ inside\ turbine}{Mass\ flow\ throu\ HP\ turbine} \tag{3.59}$$

Where the Steam mass inside turbine = $Density * volume$.

Normally, for the high-pressure turbine, the time constant is between 0.1s and 0.4s [36]. The pressure, rate of mass flow and temperature of steam at output and input of every part of high pressure turbine is demanded. The relation between the output and input and the rate of steam were identified in preceding section. The steam temperature at the exit of first stage turbine (T_{16}) canbe identified in the terms of inlet steam pressure and temperature. In HP turbine, estimating the expansion of steam is an isentropic expansion, at discharge of HP turbine, the temperature of steam can be evaluated through utilizing the relation between isentropic thermodynamic of perfect as below;

$$\frac{T_{16}}{T_1} = \left(\frac{P_{16}}{P_1}\right)^{\left(\frac{n-1}{n}\right)} \tag{3.60}$$

From first the phase of HP turbine, the out temperature (T_{16}) is;

$$T_{16} = T_1\left(\frac{P_{16}}{P_1}\right)^{\left(\frac{n-1}{n}\right)} \tag{3.61}$$

The power output ($\mathbf{P}_{HP.I}$) which is advanced in first stage at HP turbine, is clarified below;

$$P_{HP.I} = \eta^{HP.I} G(h_1 - h_{16}) = \eta^{HP.I} C_P G(T_1 - T_{16}) \tag{3.62}$$

Then,

$$P_{HP.I} = \eta^{HP.I} C_P G(T_1 - T_1\left((\frac{P_{16}}{P_1})^{\left(\frac{n-1}{n}\right)} \right)$$
$$= \eta^{HP.I} C_P G(T_1 + 273.15)\left(1 - (\frac{P_{16}}{P_1})^{\left(\frac{n-1}{n}\right)} \right) \quad (3.63)$$

Then, the entropy and enthalpy in each phase at HP turbine are found, through utilizing the thermodynamic property equations for steam as clarified in section (3.2.3). The rate of mass flow at each phase in HP turbine is possible to be found through utilizing the balance of mass and heat in section (3.2.6).

3.3.2.2 Intermediate and low-pressure turbine

The low – pressure sections and intermediate turbines are very complex in structure. Across the turbine stages, the loss of heat of the steam leads to condensation effect that reduces the quality of steam and influences the accomplishment of the turbine. The mathematical models enable estimating the energy which is released from steam expands in the phases of the turbine. The IP turbine model is clarified in figure 3.11.

The relationship between the rate of the mass flow at all extraction that relied on actual information and rate of input mass flow from reheater to IP turbine is clarified in figure 4.25 and 26). Extraction.no.3 and extraction.no.5 clarify a high linearity relation with the slope. The conversion function of the output and input mass is as follows;

$$\frac{m_{est}}{G_{IP}} = \frac{S(slop)}{\tau_{I,LPT} S + 1} \quad (3.64)$$

From equation (3.59), the time constant of LP & IP turbine are calculated. The delay between the LP and IP turbines is represented as time constant in (0.3−0.6) order [36]. The following formula determines the power ($\mathbf{P}_{IP}$) for IP turbine:

$$P_{IP} = \eta^{IPT}.[G_{IP}(h_{IP} - h_{ext.18}) + (G_{IP} - m_{est.18}).(h_{est.18} - h_{est.19}) +$$
$$(h_{ext.19} - h_{ext.23}).(G_{IP} - m_{ext.18} - m_{est.19}) + \quad (3.65)$$
$$(h_{ext.23} - h_{ext.24}).(G_{IP} - m_{ext.18} - m_{ext.19} - m_{ext.23})]$$

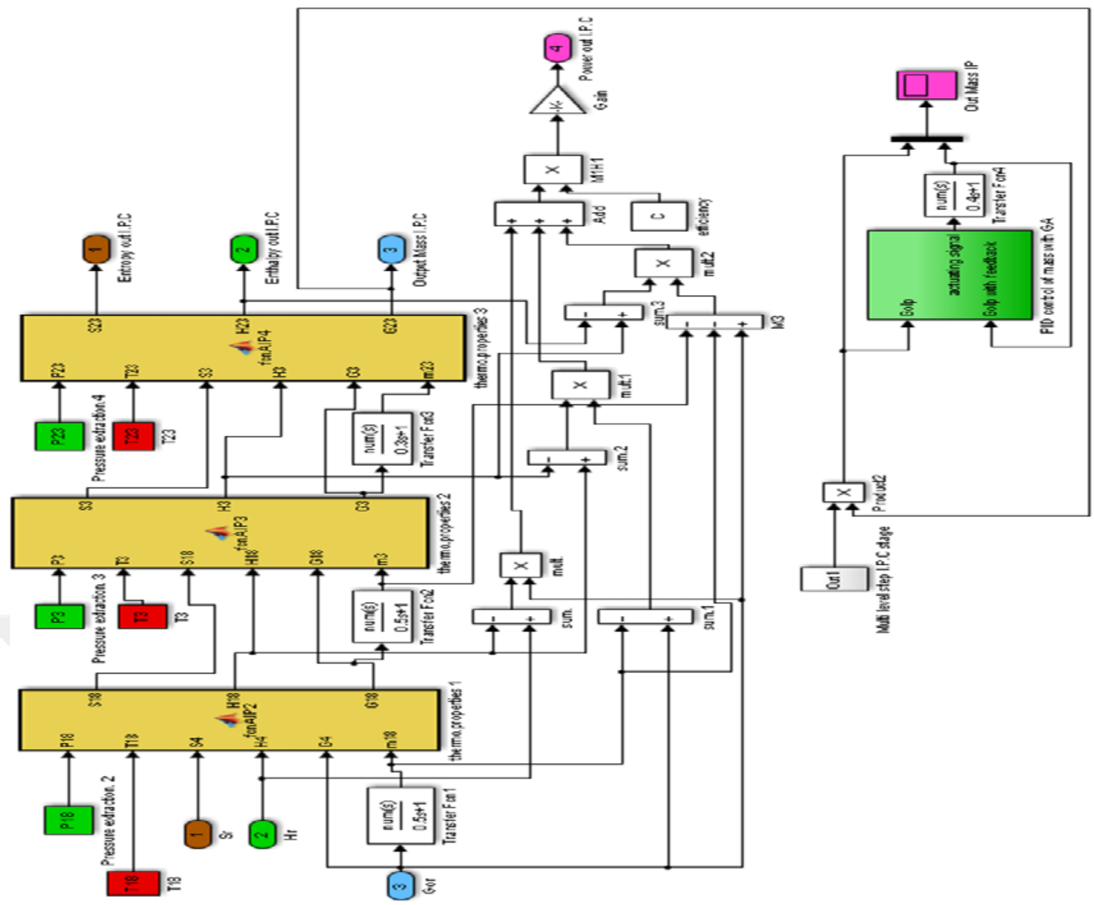

Figure 3.11: Intermediate pressure turbine model for power station of AL-Dura (K–160–13.34–0.0068).

Through utilizing the water and steam thermodynamic properties, LP turbine model is the same, as in figure 3.12.

Figure 4.27 and figure 4.28 clarify the relationship between the rate of mass flow at all extraction relied on actual information (by the program of Cycle – Tempo) and the rate of input mass flow IPT to LPT, and in these figures, a linear relation with the slope at (extraction.no.7 and extraction.no.8) is shown. The conversion function of the output and input mass is explained in equation (3.64)

Now, there are two extraction points in the turbine with low pressure. The power (**P_{LP}**) in LP turbine is identified blow;

$$P_{LP} = \eta^{LPT}[G_{LP}(h_{LP} - h_{ext.25}) + (G_{LP} - m_{est.25})(h_{est.25} - h_{est.26}) + (G_{LP} - m_{ext.25} - m_{est.26})(h_{ext.26} - h_5)] \quad (3.66)$$

At intermediate & low-pressure turbine phases, the rate of finally mass flow through utilizing the balance of mass and heat balance as clarified in (3.2.6).

The following formula determines the overall mechanical power generated ($\mathbf{P}_m$);

$$P_m = P_{HP} + P_{IP} + P_{LP} \tag{3.67}$$

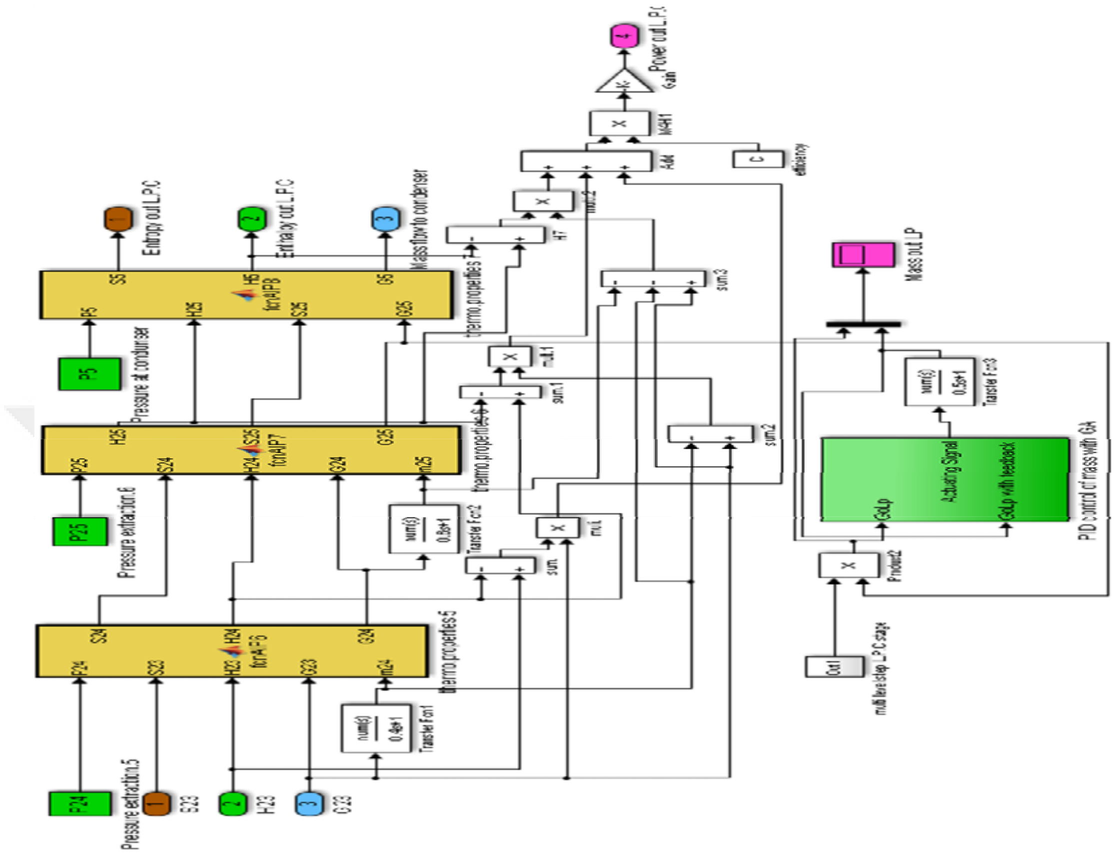

Figure 3.12: Low-pressure turbine model for power station of AL-Dura (K–160–13.34–0.0068).

3.3.3 Reheater Model

One of the components of steam generating unit is reheater. It influences the economy advancement by the following means;

1. The reheater raises the plant capacity.
2. Eliminates the steam turbine corrosion.
3. Minimizes the steam consumption of the steam turbine.

For the reheater section, models are advanced by applying conservation of energy and mass principles [38].

Accurate mathematical models for boiler component and heat balance have been advanced. The following equation clarify the model of superheated temperature [39];

$$\frac{dT_4}{dt} = R_2[R_1 G_{Fuel} + D_2 + G_2.(T_2 - T_4 + D_1)] \tag{3.68}$$

Where: $R_1 = C_p * H_v$, $R_2 = \frac{1}{\rho_s V_s}$, $D_1 = \frac{k_a}{C_p}$, $D_2 = k_0 \rho_s V_s$.

The lower heating value (H_v) of the fuel which is constant for each fuel type, (T_2)wich refers to the temperature output of the high – pressure turbine, (T_4) which represents the temperature output from the reheater and (G_2)which refers to the mass flow input to reheater, the heat flow can be captured. The reheated dynamics raises the nonlinearity and turbine time of delay and must be considered as turbine model. The fuel and water /steam specifications and this model parameters are captured from construction information. For the superheated temperature model, the equation (3.68) can be used.

The function of transfer rate of fuel flow and quality of steam is shown below [40];

$$\frac{\alpha}{G_{fuel}} = \frac{S(slop)}{\tau_{RH} s + 1} \tag{3.69}$$

The reheating system has time constant a which is varied between 10 to 20s [6, 39, 41]. Figure 3.13 clarifies an adjusted model of the temperature model for the reheater and figure 3.14 represents the subsystem model. Appendix C presents the modified parameters (R_1, R_2, D_1, D_2) of the advanced models.

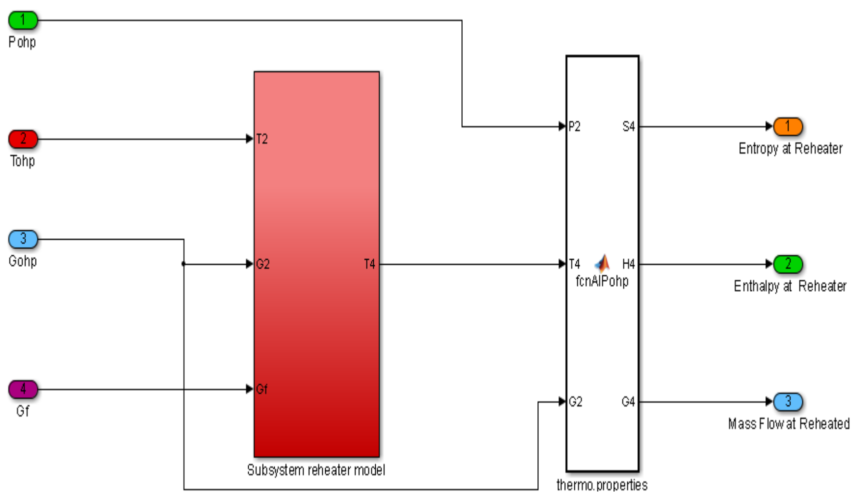

Figure 3.13: Reheater model for AL – Dura power station.

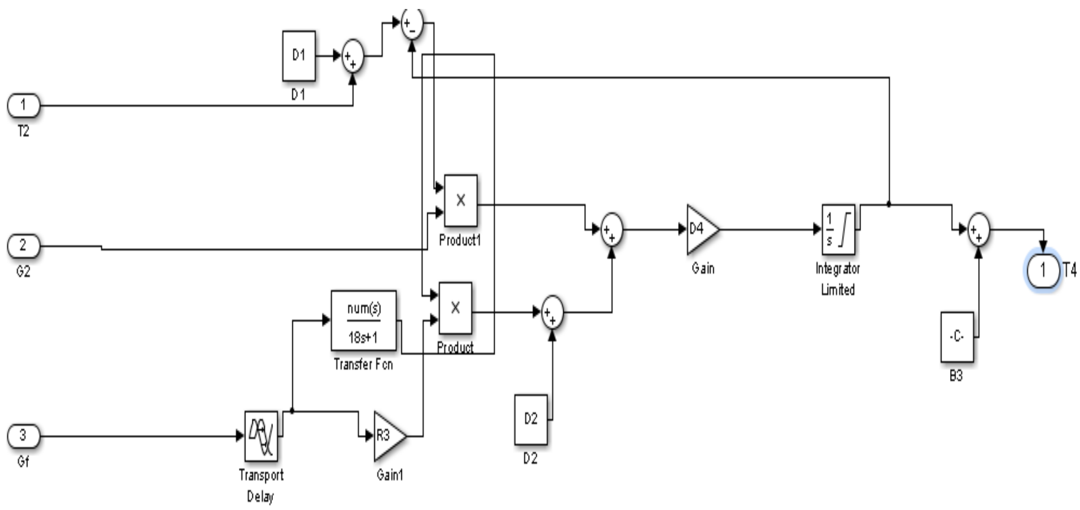

Figure 3.14: Subsystem reheater model for AL–Dura power station (K–160–13.34–0.0068)

3.3.4 Generator Model

The generator block diagram is explained as in figure 3.15 [42].

Where; the change in mechanical power output is presented as ($\Delta \boldsymbol{P}_m$), the change in electrical power is presented as ($\Delta \boldsymbol{P}_e$) and the frequency deviation is represented as(ΔF).

The rotating shaft of the steam turbine is connected to the generator rotor. The speed of the shaft is associated directly with the frequency of the electrical power as it is tightly controlled [43].

Putting the swing formula of asynchronous machine in application to small perturbation, there is [42];

$$M \frac{d^2\delta}{dt^2} = P_a = P_m - P_e \tag{3.70}$$

Where; the angular momentum of rotor is represented as (M) and the

$$P_{\max} = \frac{EV}{X} \text{ Steady state stability limit (S.S.S Limit)} \tag{3.71}$$

$$\text{Then, } P_e = P_{\max} \sin \delta = \frac{EV}{X} \sin \delta \tag{3.72}$$

The identification of the electrical power is performed in regard with terminal voltage (V), machine excitation voltage (E) and direct axis synchronous reactance (X).

$$M = \frac{SH}{\pi f} \tag{3.73}$$

Where; (H) represents the inertia constant (MJ/MVA).

Then equation (3.71) is written again for the system operating frequency electric $f\,(H)$;

$$\frac{SH}{\pi f}\frac{d^2\delta}{dt^2} = P_a = P_m - P_e \tag{3.74}$$

After splitting formula (3.74) by (S), the rating of machine (base) in MVA results in a formula as follows;

$$\frac{H}{\pi f}\frac{d^2\delta}{dt^2} = P_a = P_m - P_e \tag{3.75}$$

Figure 3.16 clarifies an adjusted model of the power electric model for the generator.

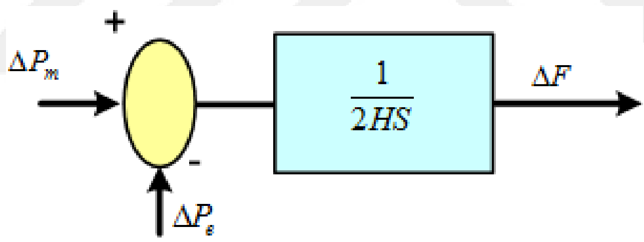

Figure 3.15: Generator block diagram.

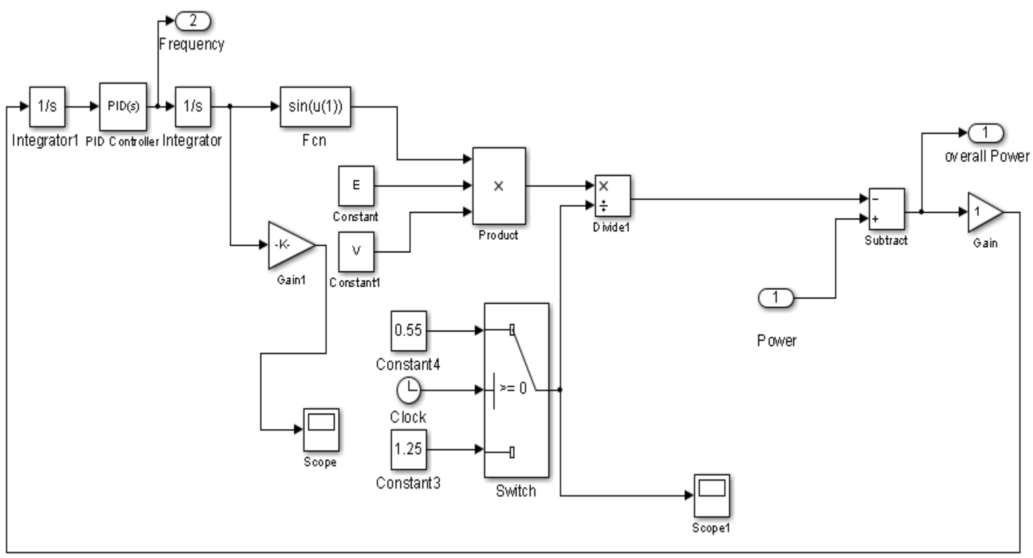

Figure 3.16: Generator model for AL–Dura power station (K–160–13.34–0.0068).

3.5 Computer Program

Using Mat lab software and comparing it with the **Cycle–Tempo** software characterizes the performance of each study. The program makes it possible to get all properties at every node by the thermodynamic cycle through utilizing the proper thermodynamic relations. The chosen study is power station AL–Dura type (K–160–13.34–0.0068) steam power plant in Baghdad, Iraq. It is possible to change the different parameters to construct and simulate all cases and also Cycle– Tempo" that are utilized for this purpose. The flow charts of the computer programs as given in figure 3.17 and figure 3.18 clarifies the simulation and the most important parameters.

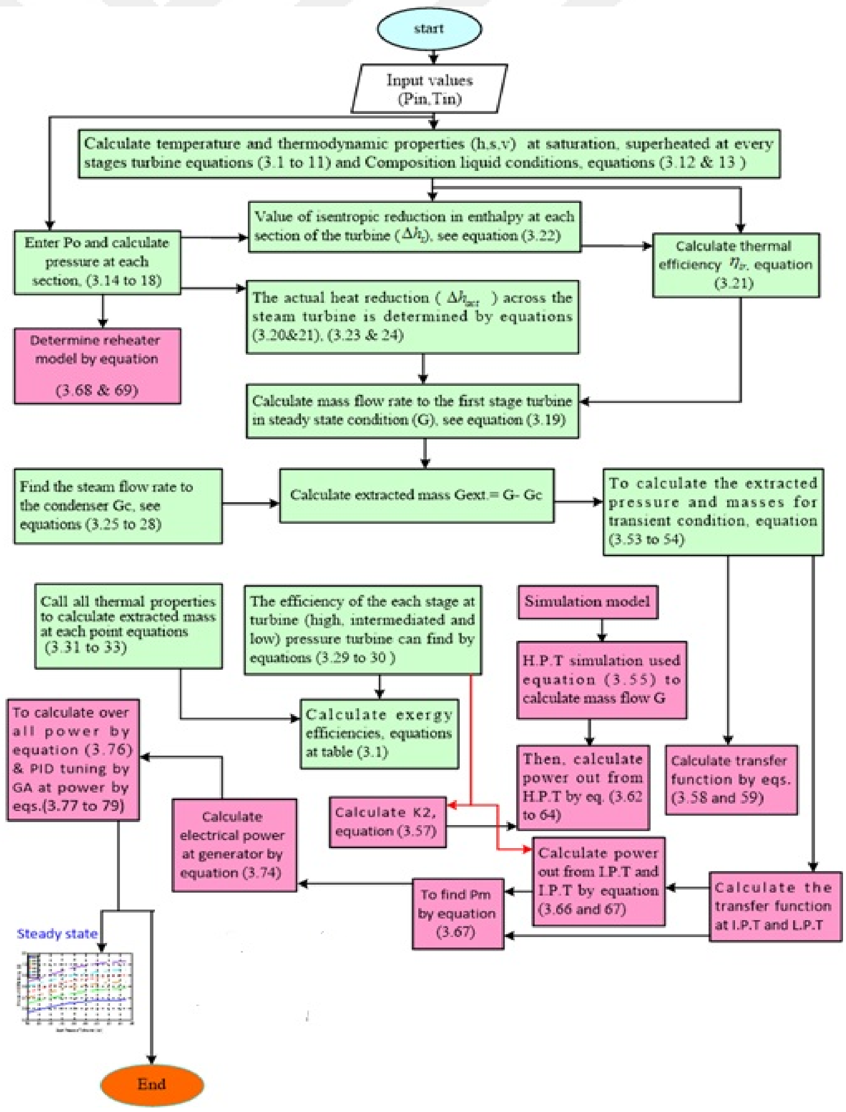

Figure 3.17: Flow chart of the computer program MATLAB

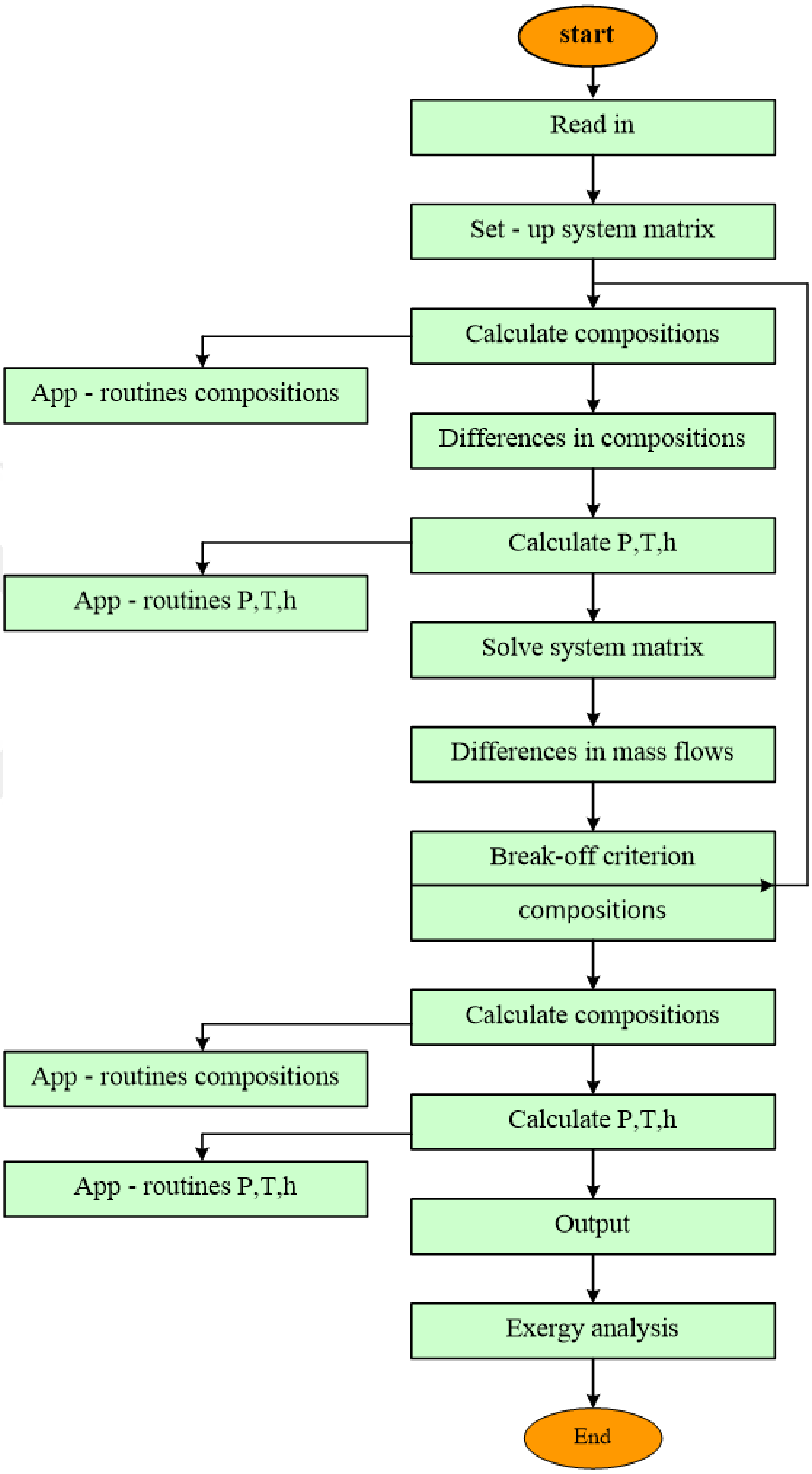

Figure 3.18: Flowchart of the computer program cycle– tempo

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Introduction

There are two parts of conclusions, first one contains the analysis of exergy and energy of AL–Dura power plant type. The second one contains the mathematical simulation model conclusions. The simulation of the steam turbine power plant uses two techniques (Programs) and it has components such as boiler, turbine, generator, and reheater. The first one of these two techniques use MATLAB (V2014a) with m–files Simulink and the other one applies Cycle – Tempo (Release 5).

4.2 Energy and Exergy Analysis Results

They are firstly compared with standard data to compute the accuracy of the thermodynamic properties like entropy and enthalpy of water steam. The response of proposed function for specified enthalpy and entropy are applied for this case, and they are applied at two extractions point (extraction no.1 and extraction no.3) with various temperatures and pressures which are clarified respectively in figures (4.1) and (4.2). The error functions in (Appendix D) are; minimum bound error Min(|e|), maximum bound error Max(|e|), average absolute deviation AAD (e), mean absolute error MAE and correlation coefficient R^2 (e). The comparison reveals a high accuracy and shows that the modeling error is less than 0.01%.

In figures (4.1), (4.2) and (4.3), more definition of thermal analysis of this **station** (K–160–13.34–0.0068) was listed.

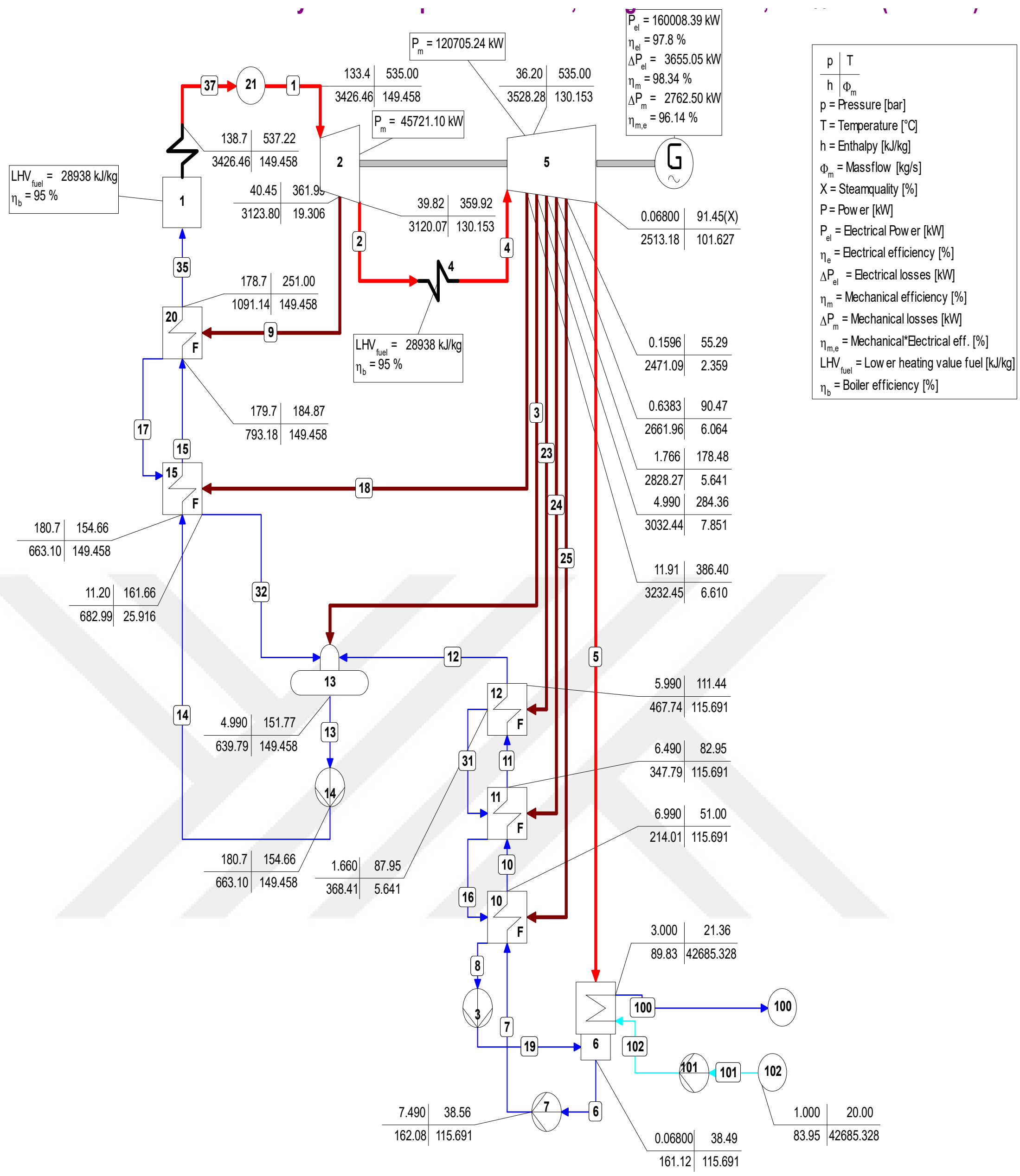

Figure 4.1: Design calculation at 100% load and power 160 MW for water– steam cycle at AL–Dura power station

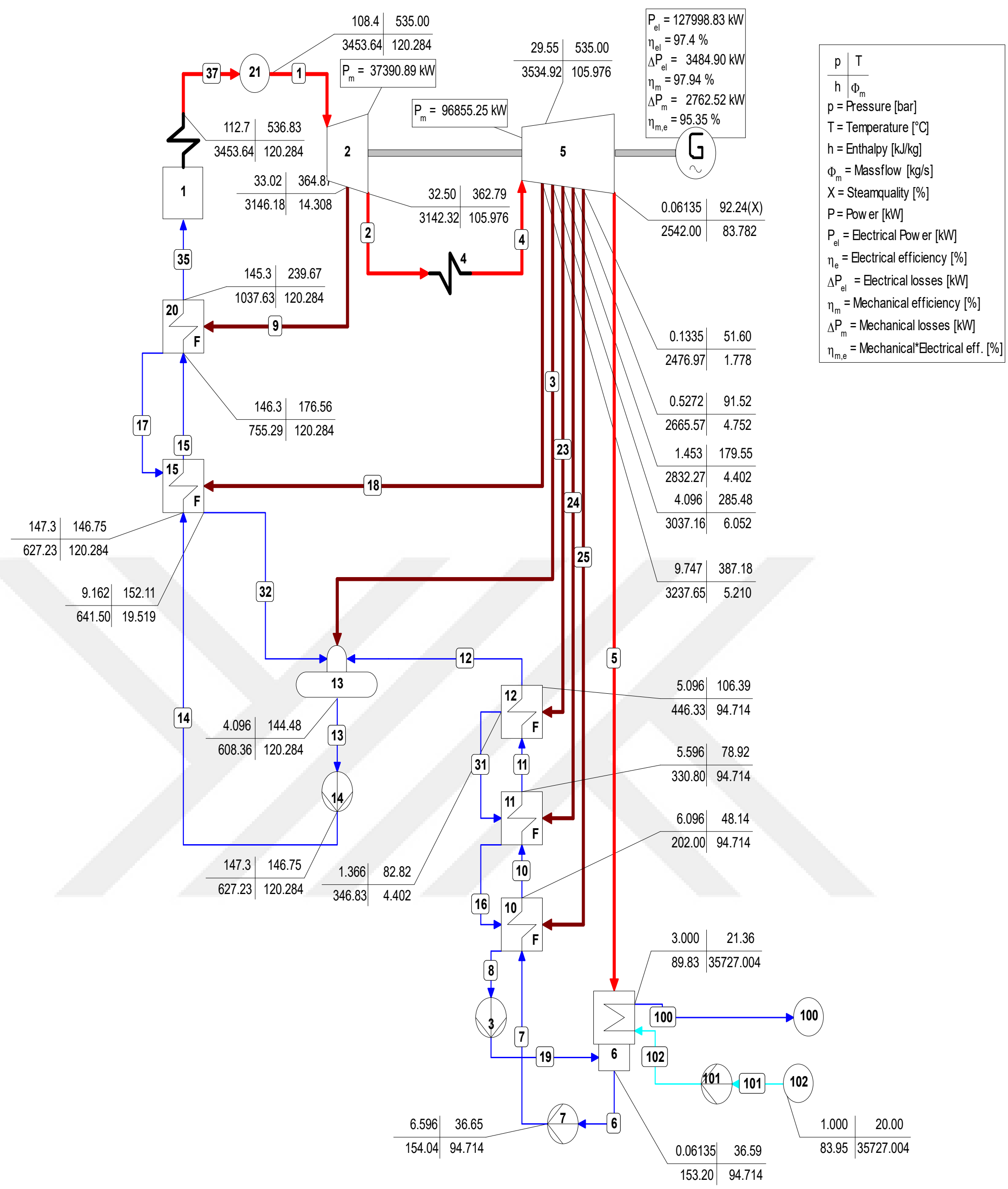

Figure 4.2: Design calculation at 75% load and power 128 MW for water– steam cycle at AL–Dura power station

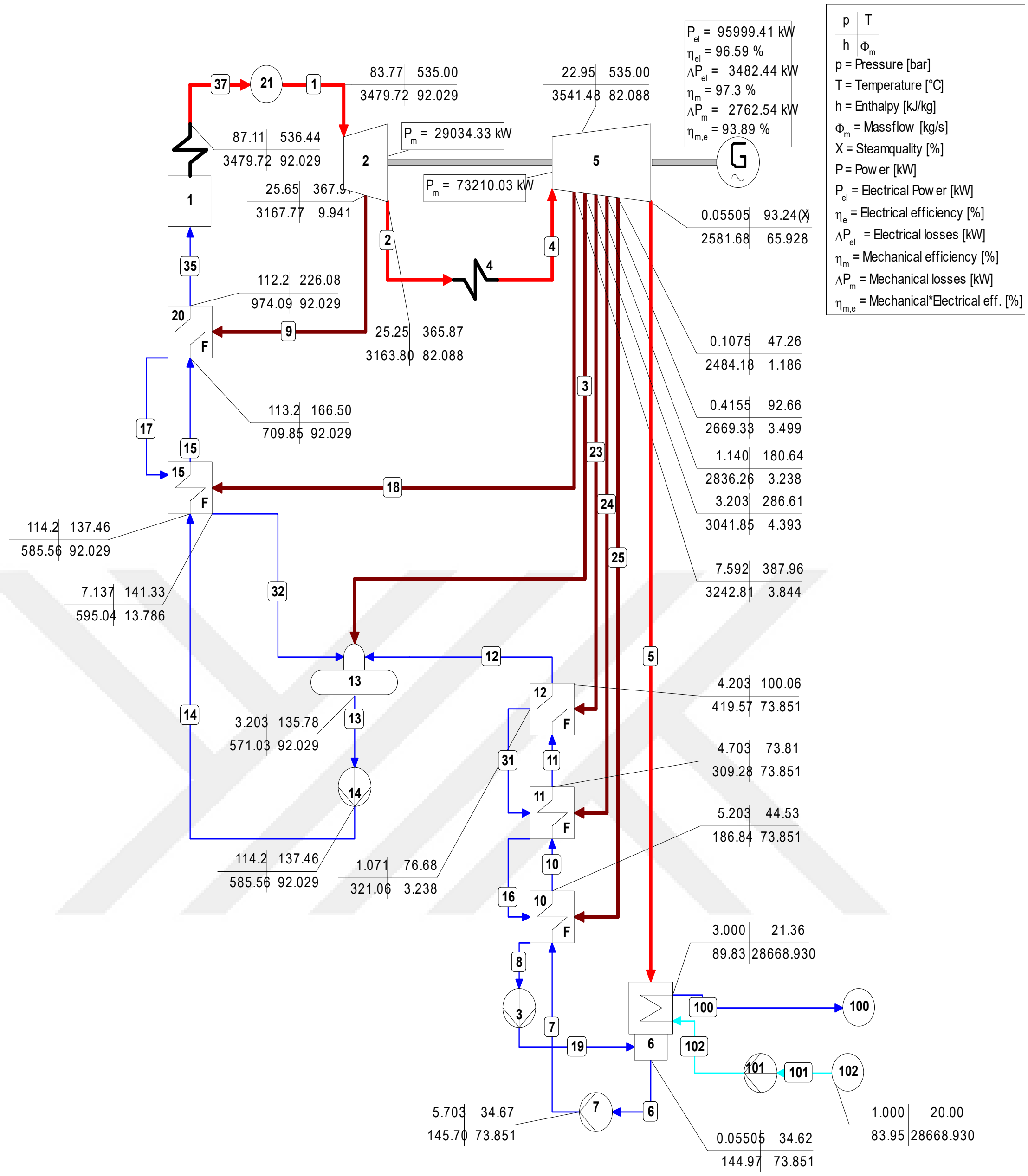

Figure 4.3: Design calculation at 50% load and power 96 MW for water – steam cycle at AL–Dura power station

For full load, these figures present thermodynamic properties, mass, and energy balance at each section in details. Conventional water– steam cycle for the station (K–160–13.34–0.0068) is described by figures 4.1, 4.2 and 4.3. The calculations of design are performed for three loads and power 100%; 160 MW, 75%; 128 MW and 50%; 96 MW. These calculations are applied by utilizing cycle – Tempo program.

From tables 4.1 and 4.2, there is an, agreements between the calculated values of thermodynamics properties with the conclusions gained from (Cycle– Tempo & Mat lab). The two mentioned tables display the conclusions gained through utilizing two programs, MATLAB and Cycle– Tempo, every station at full loads. This reveals an, excellent agreements between the conclusions gained through utilizing these programs. The results for AL-Dura powers station with specifications 75% loads and 50% loads are shown respectively in tables 4.3 and 4.4.

Table 4.1: Results of 160 MW and 100 % load for cycle – temp program (AL–Dura Station)

Data from program Cycle – Temp at 160 MW and 100% load (AL-Dura Station)						
Parameter	P (MPa)	T $(°C)$	Mass (k_g/s)	h (k_J/k_g)	S $(k_J/k_g.k)$	Exe. (k_J/k_g)
Input for the first stage of HP turbine	13.34	535	149.458	3426.46	6.5437	1542.50
Extraction (1) at HPC	4.045	361.99	19.306	3123.80	6.6279	1215.58
at the reheat	3.620	535	130.153	3528.28	7.2411	1443.35
Extraction (2) at IPC	1.191	386.40	6.610	3232.45	7.3389	1119.30
Extraction (3) at IPC	0.499	284.36	7.851	3032.44	7.4007	914.83
Extraction (4) at LPC	0.176	178.46	5.641	2828.27	7.4725	676.60
Extraction (5) at LPC	0.063	90.47	6.064	2661.96	7.5275	494.44
Extraction (6) at LPC	0.015	55.29	2.359	2471.04	7.5906	285.39
Output from the LPC to condenser	0.0068	38.49	101.627	2513.18	8.0983	180.96

Table 4.2: Results of 160 MW and 100 % load for mat lab program (AL – Dura Station).

Data from program Matlab at 160 MW 100% load (AL – Dura Station)					
Parameter	P (MPa)	T $(°C)$	Mass (k_g/s)	h (k_J/k_g)	S $(k_J/k_g.k)$
Input for the first stage of HP turbine	13.34	535	149.518	3428.4	6.5474
Extraction (1) at HPC	4.055	363.53	19.360	3144.4	6.6233
at the reheat	3.620	535	130.60	3518.5	7.2230
Extraction (2) at IPC	1.191	384.28	6.625	3217.2	7.3283
Extraction (3) at IPC	0.53	289.19	7.835	3022.4	7.3992
Extraction (4) at LPC	0.176	172.66	5.655	2818.7	7.4699
Extraction (5) at LPC	0.063	88.75	6.079	2663.7	7.5280
Extraction (6) at LPC	0.015	55.39	1.559	2481.0	7.6911
Output from the LPC to condenser	0.0068	38.50	101.8	2464.1	7.9823

Table 4.3: Results of 128 MW and 75 % load for cycle – temp program (AL-Dura Station).

Data from program Cycle – Temp at 128 MW and 75% load (AL-Dura Station)						
Parameter	P (MPa)	T ($°C$)	Mass (k_g/s)	h (k_J/k_g)	S ($k_J/k_g.k$)	Exe. (k_J/k_g)
Input for the first stage of HP turbine	10.84	535	120.284	3453.64	6.6650	1534.71
Extraction (1) at HPC	3.302	364.87	14.308	3146.18	6.7507	1202.56
at the reheat	2.955	535	105.976	3534.92	7.3409	1421.23
Extraction (2) at IPC	0.9747	387.18	5.210	3237.65	7.4381	1095.96
Extraction (3) at IPC	0.4096	285.48	6.052	3037.16	7.5036	876.60
Extraction (4) at LPC	0.1453	179.55	4.402	2832.27	7.5705	652.42
Extraction (5) at LPC	0.05272	91.52	4.752	2665.57	7.6250	470.03
Extraction (6) at LPC	0.01335	51.60	1.778	2476.97	7.6866	263.67
Output from the LPC to condenser	0.006135	36.59	83.782	2542.00	8.2386	169.63

Table 4.4: Results of 96 MW and 50 % load for cycle – temp program (AL – Dura Station).

Data from program cycle – Temp at 96 MW and 50% load (AL – Dura Station)						
Parameter	P (MPa)	T ($°C$)	Mass (k_g/s)	h (k_J/k_g)	S ($k_J/k_g.k$)	Exe. (k_J/k_g)
Input for the first stage of HP turbine	8.377	535	92.029	3479.72	6.8086	1519.42
Extraction (1) at HPC	2.565	367.97	9.941	3167.77	6.8953	1182.48
at the reheat	2.295	535	82.088	3541.48	7.4636	1392.44
Extraction (2) at IPC	0.7592	387.96	3.844	3242.81	7.5597	1066.06
Extraction (3) at IPC	0.3203	286.61	4.393	3041.85	7.6244	846.48
Extraction (4) at LPC	0.1140	180.64	3.238	2836.26	7.6905	621.83
Extraction (5) at LPC	0.04155	92.66	3.499	2669.33	7.7442	439.42
Extraction (6) at LPC	0.01075	47.26	1.186	2484.18	7.8038	237.12
Output from the LPC to condenser	0.005505	34.62	65.928	2581.68	8.4162	158.15

The steam pressures for AL – Dura station is shown in figure 4.4. The conclusions clarify that the pressure fall throughout the turbines parts are linear and can be identified by the first function of order transfer. This figure exhibits that the extractions pressures raises with an inlet pressure of IP, HP and LP turbine [[. The extraction pressures for taps no.1 is larger than at no. (2s, 3s, 4, 5 and 6).

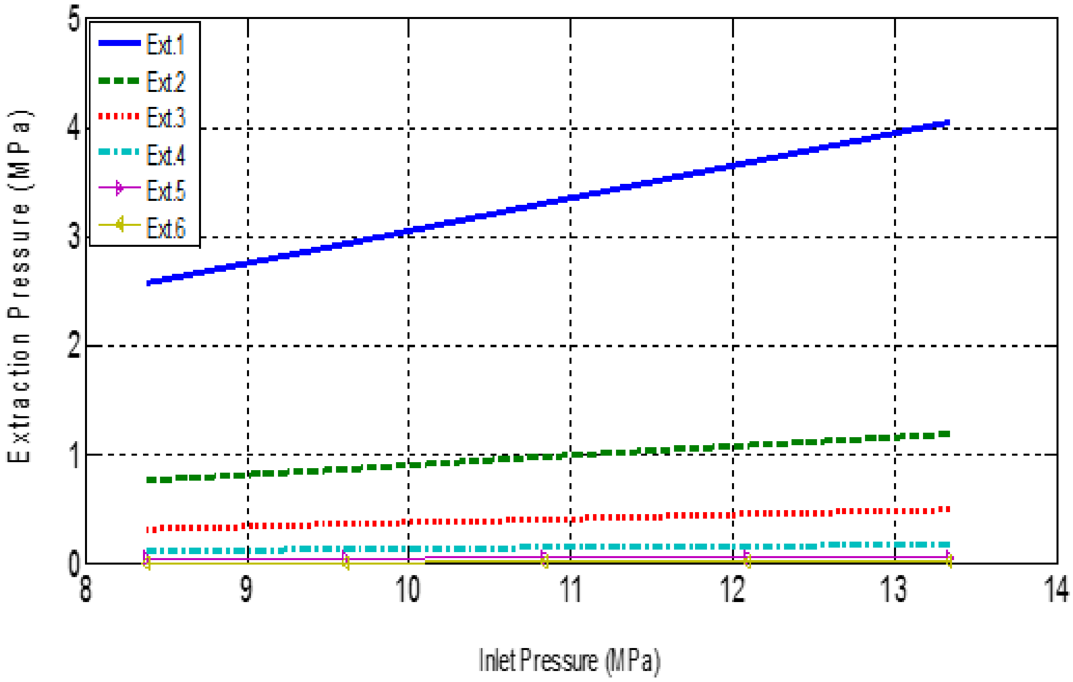

Figure 4.4: The varies of extractions pressure with different of inlet basic pressure for AL-Dura power station.

Figure (4.5), indicated comparison of extraction masses with power for the real data and (Cycle – Tempo program) data for AL – Dura station. It becomes clear that there is an agreement between them and the error is less than 2%.

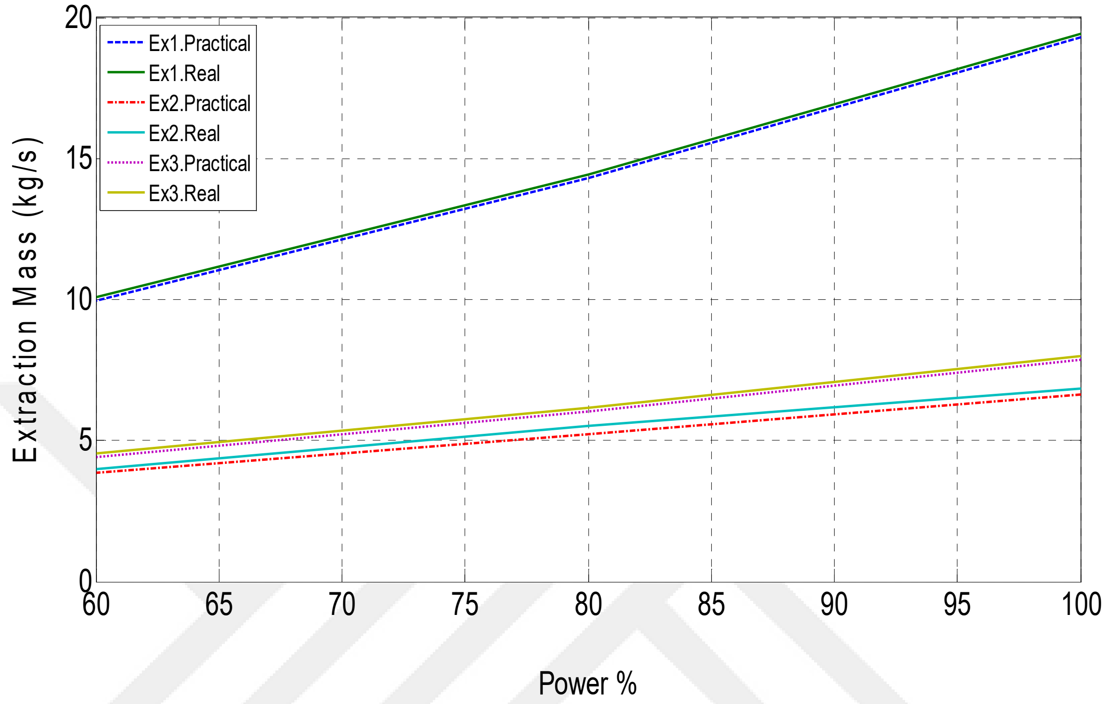

Figure 4.5: Compared between the real – time data and (Cycle – Tempo program) data for AL – Dura power station.

In high, intermediate and lower pressures turbines phases for AL–Dura power station, the extraction mass flow rate is illustrated in figure (4.6). The rate of extracted steam raises with inlet mass of high, intermediated and low turbines. The extraction mass at taps no.1 is greater than all other extractions masses for the reason of reheating and extraction to gather. In intermediated pressure turbine the extraction mass at taps no.3 is greater because it bleeding to the deaerator.

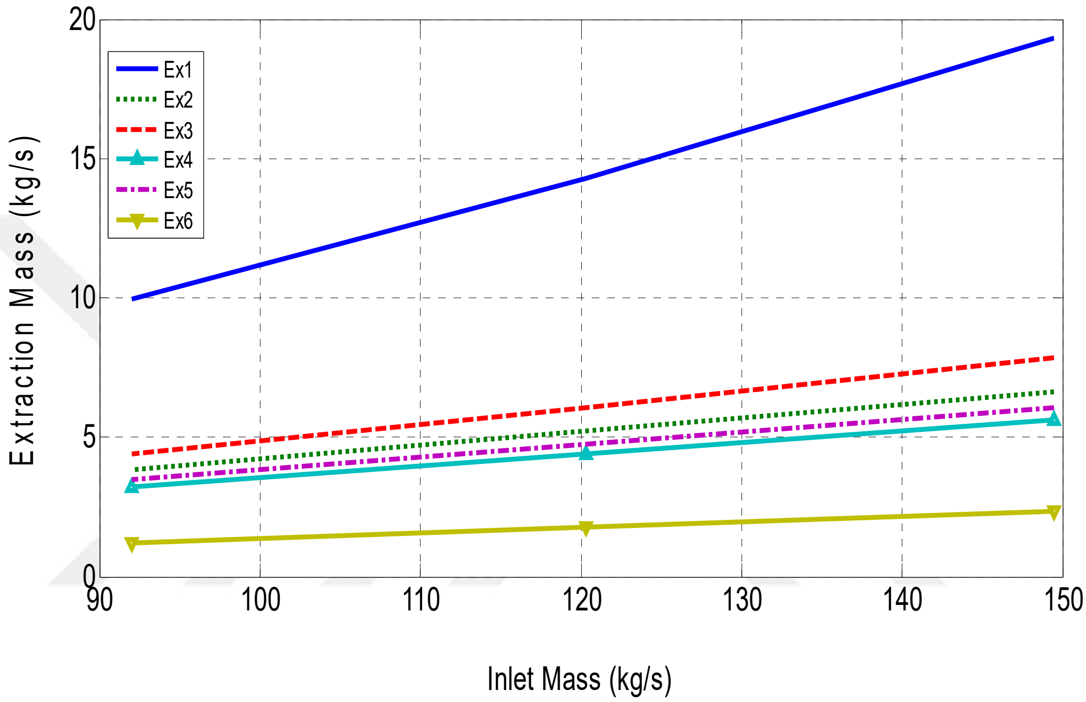

Figure 4.6: The relationship between inlet main mass and extractions mass for AL-Dura power station

The relationships between the rate of mass flow and total powers for AL – Dura powers stations is illustrated in figure (4.7).

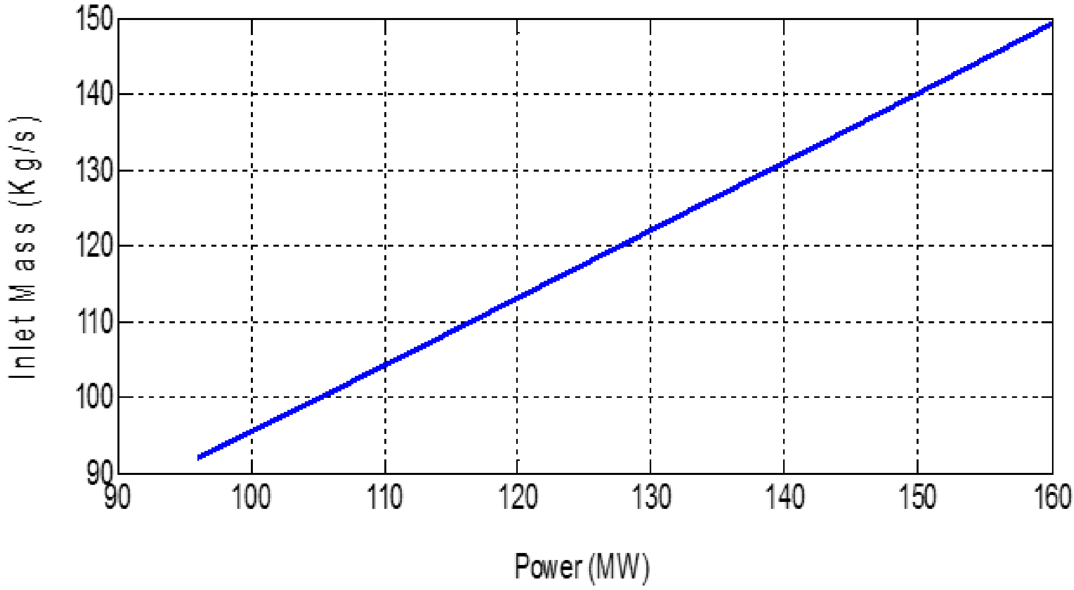

Figure 4.7: The varies of the total power with different inlet mass for AL-Dura power station

Figure 4.8, refers to the relation between the inlet pressures and total powers for AL–Dura powers stations which clarifies a linear relation between the inlet pressure and power.

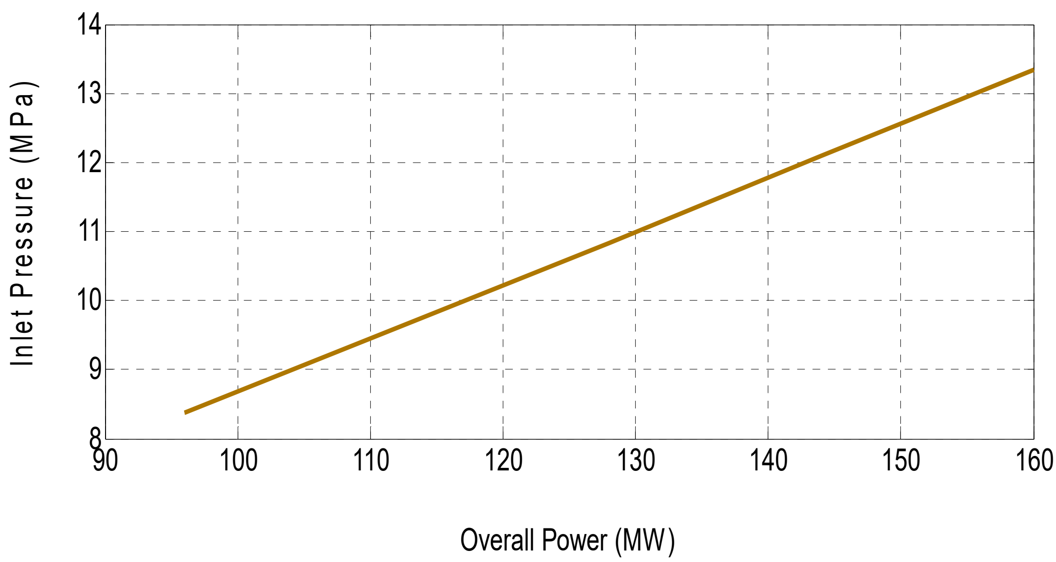

Figure 4.8: Relation between the inlet pressure and total power for AL-Dura power station.

The change ratio in inlet pressures of high-pressure turbine is related to the initial pressure, for the existing steams power plant (AL–Dura power station), as illustrated in figure 4.9. The changing ratio of initial mass to inlet mass for the same conditions is clarified in figure 4.10.

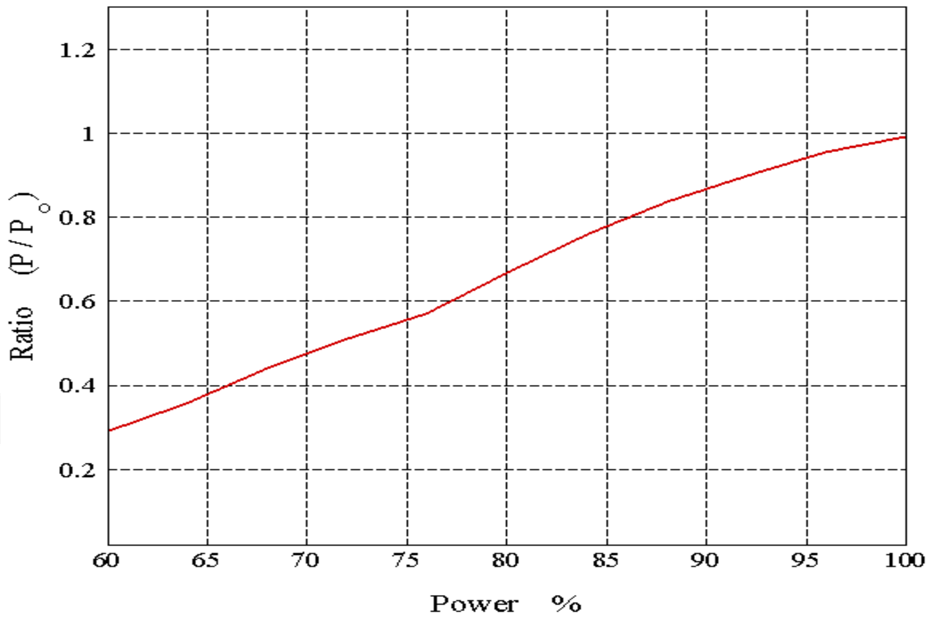

Figure 4.9: The varies inlet pressure ratio at high-pressure turbine with total power for AL-dura power station.

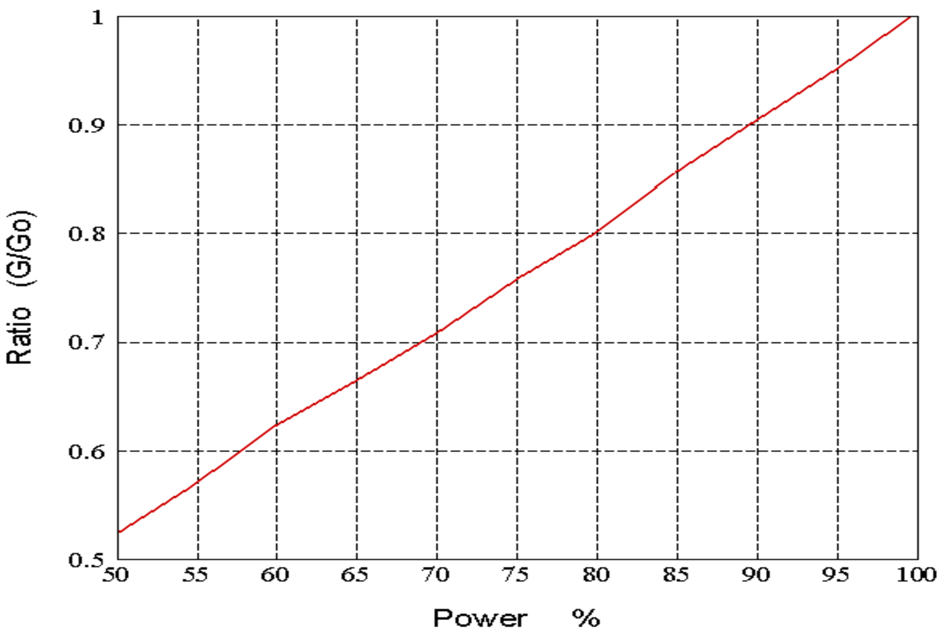

Figure 4.10: Relation between the ratio of inlet mass flow rate at high-pressure turbine and total power for AL-Dura power station

Figure 4.11, shows that the relation between the ratio of overall mass flow rate and the total power for AL – Dura powers station and this relation was also linear.

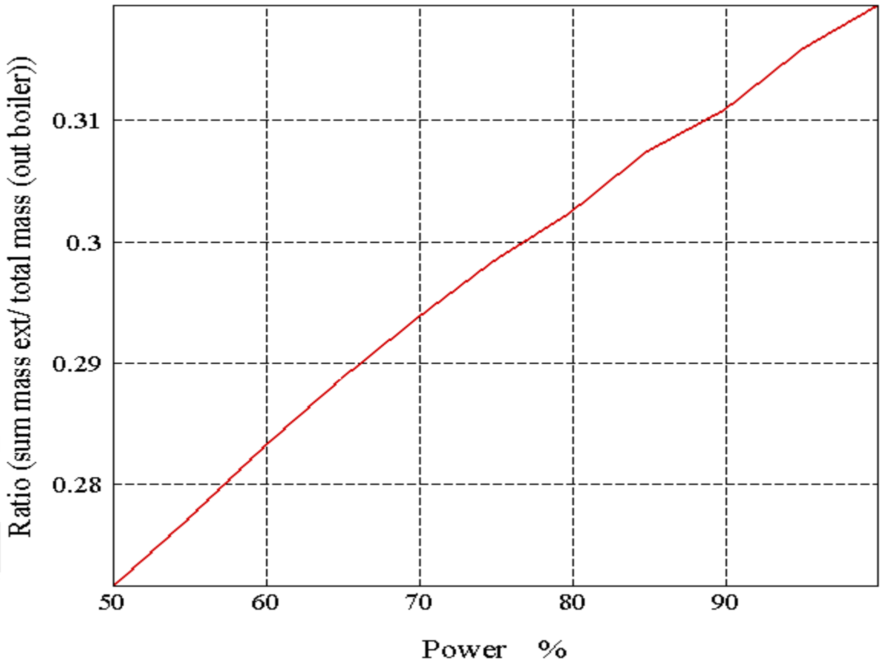

Figure 4.11: Relation between ratio of the overall mass flow rate and the total power for AL – Dura power station.

The efficiency of internal exergy was calculated as in [table 4.1 and figure 4.12] to understand the results from the cycle calculation. Some necessary data for calculating these efficiencies are evaluated because of the unavailability of the original system calculations. The equation 4.1 is for calculating the internals exergy efficiency ($\eta_{ex.intern.}$) of the cycles. The steam cycle thermal efficiency is clarified below;

$$\eta_{th.,cycle} = \eta_{ex.intern.}\left[1-\frac{T_C}{T_H}\right] \tag{4.1}$$

All systems use the same condenser temperature, so (T_C) refers to the constant. By only the value for (T_H) ($\eta_{th.,cycle}$) is determined, the mean temperature of heat addition to the reheater and boiler, and ($\eta_{ex.intern.}$). From equation 4.2, the temperature (T_H) is calculated. After that, the entropies and enthalpies at the outlet and inlet of the reheater and boiler should be considered as the ratios of the mass flows to reheated and boiler. These values of T_H are identified by;

$$T_H = \frac{\sum (h_{out} - h_{in}) * m}{\sum (s_{out} - s_{in}) * m} \qquad (4.2)$$

From the calculations with the Cycles – Tempo models, the ($\eta_{th.cycle}$) values are available. At last, the ($\eta_{ex.intern.}$) values which are shown in table (4.5) are calculated.

Table 4.5: Results of cycles calculations (singles reheat steam cycle) for (Ks–160–13.34s–0.0068).

Results of cycle calculations (single reheat steam cycle) 160 **MW**						
P_{steam}	T_{steam}(°C)	T_H (K)	T_C (K)	η_{rev}	$\eta_{th,cycle}$	$\eta_{ex,intern}$
180	530	645.56	295.52	0.5422	0.4571	0.8430
	560	650.99	295.52	0.5460	0.4652	0.8520
	580	654.71	295.52	0.5480	0.4703	0.8582
	600	658.50	295.52	0.5512	0.4750	0.8617
	620	662.38	295.52	0.5538	0.4802	0.8671
	650	668.34	295.52	0.5578	0.4843	0.8682
250	530	662.26	295.52	0.5537	0.4650	0.8398
	560	670.23	295.52	0.5590	0.4735	0.8470
	580	676.26	295.52	0.5630	0.4792	0.8511
	600	682.33	295.52	0.5668	0.4847	0.8551
	620	688.43	295.52	0.5707	0.4896	0.8578
	650	696.69	295.52	0.5758	0.4960	0.8621
300	530	669.86	295.52	0.5588	0.4676	0.8367
	560	679.41	295.52	0.5650	0.4770	0.8442
	580	688.25	295.52	0.5706	0.4828	0.8461
	600	691.07	295.52	0.5723	0.4882	0.8530
	620	699.51	295.52	0.5775	0.4933	0.8541
	650	708.12	295.52	0.5826	0.5005	0.8590
350	530	673.52	295.52	0.5612	0.4679	0.8337
	560	689.15	295.52	0.5711	0.4783	0.8375
	580	691.63	295.52	0.5727	0.4845	0.8459
	600	698.28	295.52	0.5767	0.4902	0.8500
	620	704.64	295.52	0.5806	0.4957	0.8537
	650	714.62	295.52	0.5864	0.5030	0.8577

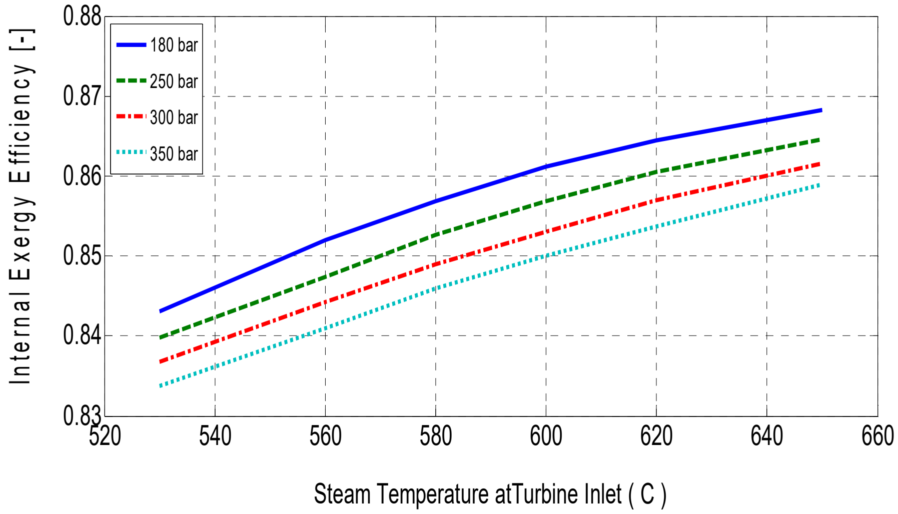

Figure 4.12: The internal exergy efficiency with different steam temperature (Single reheat) for AL – Dura power station.

The efficiencies of internal exergy are initially identified through the of the steam turbine isentropic efficiencies as illustrated in figure 4.13. Generally, as the rate of steam volume flow is higher, this isentropic efficiency is higher. In this case, the internal efficiency is in full agreement with this rule. The influences on the external exergy efficiency and the thermal efficiency of the reversible cycles (the Carnot efficiency) are considered separately. As shown in results, the reversible cycle thermal efficiencies raise linearly with the temperature of steam, and this increase is per Kelvin.

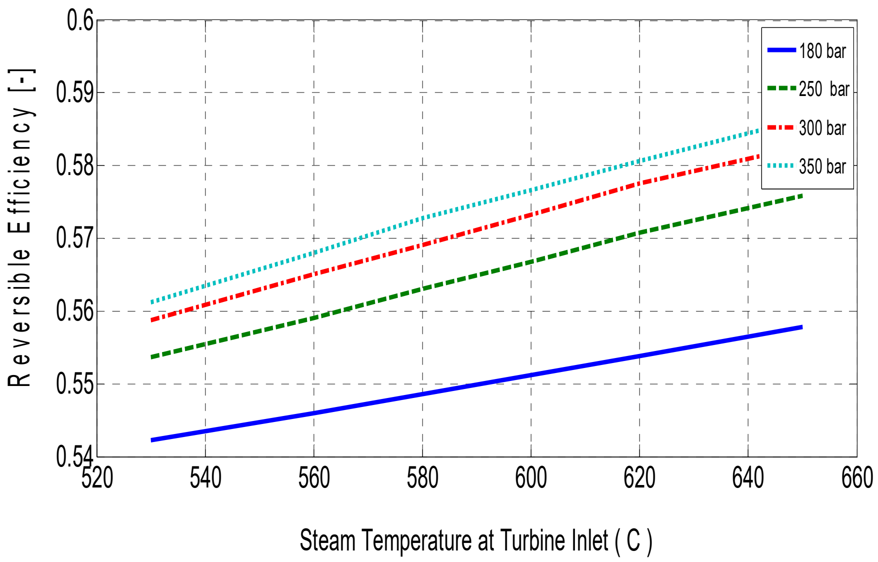

Figure 4.13: The reversible efficiency with different steam temperature at turbine inlet for AL – Dura power station.

The change of thermal efficiency with inlet steam pressure for various temperatures is clarified in figure 4.14. This figure illustrates that for a constants boilers pressures, (180s bar) the efficiency raises from (s46.8%) to (s47.5%) in the case of differing the inlet turbine temperature from [(560°C) - (600°C)]. Figure 4.15, shows that at a constants inlet, turbines, temperatures, (650°C), the efficiency raises from [(s48.4%) - (s50.3%)] as the boiler pressure increases from (s180 bar) to (s350 bar). Clearly, the increase of the boilers pressures and turbines temperature raises the steams enthalpy at turbines inlet and condenser pressures stays constant at (0.0034) bar.

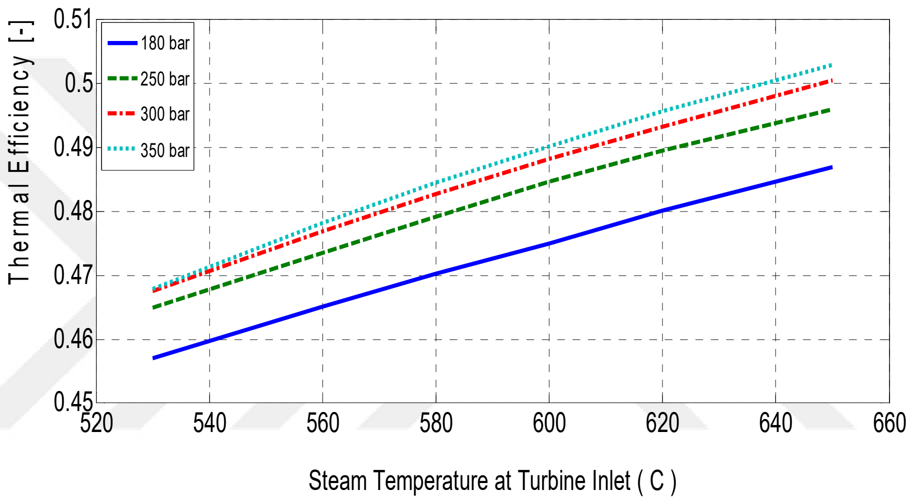

Figure 4.14: Thermal efficiency versus steam temperature at turbine inlet temperature, for AL – Dura power station.

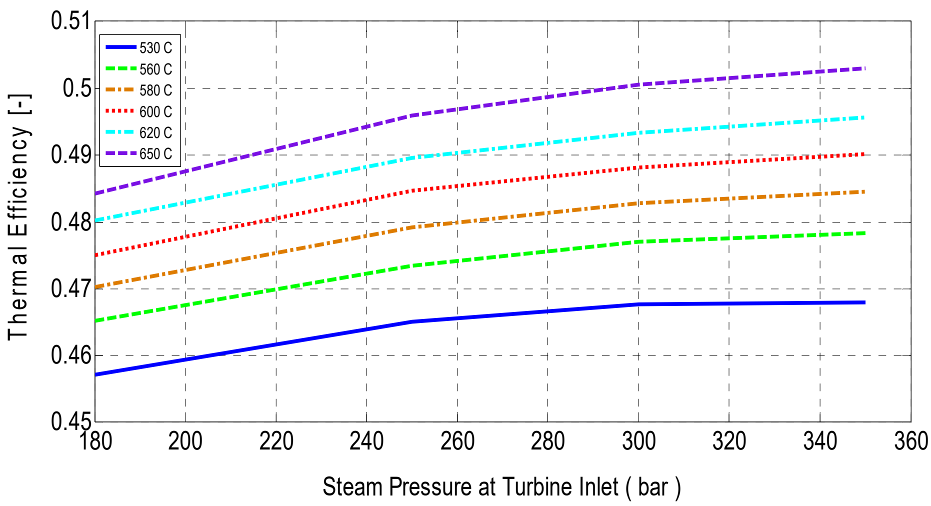

Figure 4.15: Thermal efficiency with different steam pressures at turbine inlet the temperature for AL – Dura power station.

Figure 4.16, illustrates the internals exergy efficiency of the power cycles. The efficiency of internal exergy is initially identified through the isentropic efficiency of the steam turbine.

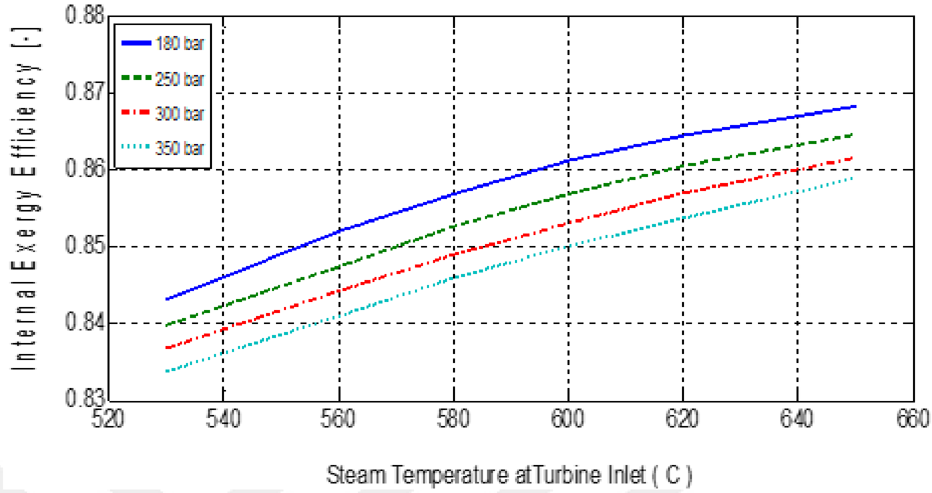

Figure 4.16: The internal exergy efficiency versus steam temperature (Single reheat) for AL – Dura power station.

The inlets and extractions exergy for AL-Dura power station are clarified in figure 4.17. The extracted exergy raises with raising the inlets exergy for the highs, intermediate and low-pressures turbines with a linear, relations between the inlet and extraction exergy. The enthalpy at no.1 is bigger than the no. (2, 3, 4 and 5) and 6 because of the high pressure and temperature, so the extraction exergy value at no.1 is more than it at no. (2, 3, 4 and 5) and 6.

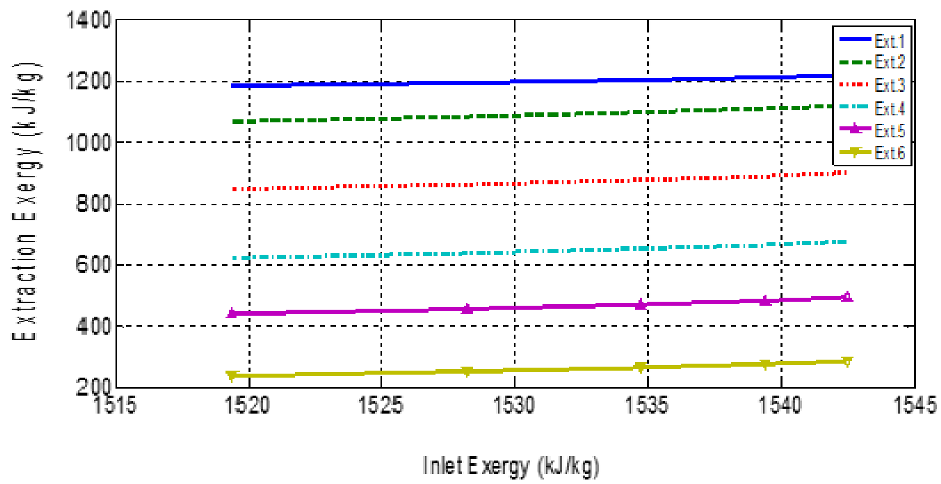

Figure 4.17: Relation between inlet and extraction exergy for AL-Dura power station.

The exergys at extraction no. (s1, 2, s3, 4, 5 and 6) in high, intermediated and low-pressures turbines for AL-Dura power station is shown in figure 4.18. With rising power, the extracted exergy raises as well as the linear relationship between them.

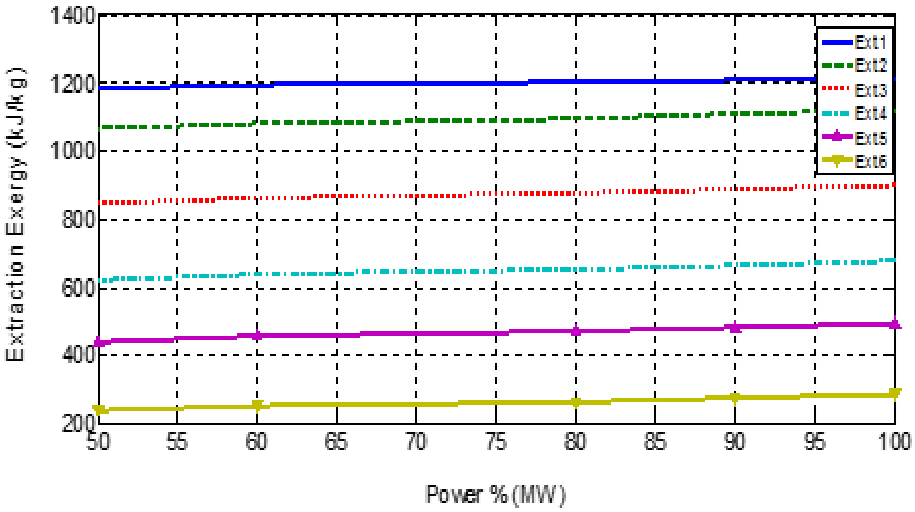

Figure 4.18: The relationship between extractions exergy and power % for AL-Dura power station

The values of exergys, relatives, function exergy's losses and efficiencies of universals exergys for this station at changeable load capacity are clarified in tables (4.6). The value of relative exergy decreases for plant components, and it decreases with a decreased capacity of power for the boiler system components only.

These tables also clarify that the universal and functional exergy's efficiencies have the greatest values in the steam turbines in this station, and the power capacity does not affect the universal exergy. By the exergy efficiencies definitions, these cases can be understood as illustrated in table (3.1) and equations [(3.51) and (3.52)].

Table 4.6: The values of exergy in the system of AL-Dura power station 160MW, design calculation at 100% load

No.	Name	Type	Exergy transmitted from system [kW]			Rel. Ex. Loss	Func. Exergy eff.	Univ. Exergy eff.
			Total	Power/Heat	Losses	[%]	[%]	[%]
1	Boiler	1	202075.55	0.00	202075.55	44.25	48.15	53.37
4	Reheater	2	27554.77	0.00	27554.77	6.03	53.55	87.30
2	Turbine	3	48827.22	45721.10	3106.12	0.68	93.64	98.65
5	Turbine	3	147510.27	120705.24	26805.02	5.87	81.83	85.73

Table 4.6 (Continued): The values of exergy, in the systems of AL-Dura powers station 160MW, design calculations at 100% loads

No.	Name	Type	Exergy transmitted from system [kW]			Rel. Ex. Loss	Func. Exergy eff.	Univ. Exergy eff.
			Total	Power/Heat	Losses	[%]	[%]	[%]
6	Condenser	4	13402.92	0.00	13402.92	2.93	25.69	61.34
10	Flash. Heater	5	150.59	0.00	150.59	0.03	78.71	88.42
11	Flash. Heater	5	655.10	0.00	655.10	0.14	78.21	84.44
12	Flash. Heater	5	513.14	0.00	513.14	0.11	85.68	92.87
15	Flash. Heater	5	699.42	0.00	699.42	0.15	90.65	97.62
20	Flash. Heater	5	1799.88	0.00	1799.88	0.39	91.08	96.33
13	Deaerator	7	874.52	0.00	874.52	0.19	86.75	94.75
3	Pump	8	2.57	3.66	-1.10	0.00	70.10	101.50
14	Pump	8	-3061.10	-3656.71	595.62	0.13	83.71	96.93
7	Pump	8	-88.31	-125.98	37.67	0.01	70.10	93.20
101	Pump	8	-8599.74	-11876.42	3276.68	0.72	72.41	83.18
21	Sink/Source	10	0.00	0.00	0.00	0.00		100.00
1	Pipe		550.60		550.60	0.12		
4	Pipe		1607.48		1607.48	0.35		
12	Pipe		9.13		9.13	0.00		
16	Pipe		0.73		0.73	0.00		
17	Pipe		43.10		43.10	0.01		
18	Pipe		53.48		53.48	0.01		
23	Pipe		45.91		45.91	0.01		
24	Pipe		49.36		49.36	0.01		
25	Pipe		18.27		18.27	0.00		
31	Pipe		0.60		0.60	0.00		
32	Pipe		20.58		20.58	0.00		
35	Pipe		405.44		405.44	0.09		
37	Pipe		705.99		705.99	0.15		
100	Pipe		4184.15		4184.15	0.92		
	Medium to/from env.							
1	Boiler	1	-389735.62	-389735.62	0.00	0.00		
4	Reheater	2	-59325.26	-59325.26	0.00	0.00		
100	Sink/Source	10	16649.94	0.00	16649.94	3.65		
102	Sink/Source	10	-7600.66	0.00	-7600.66	-1.66		
	Total:		0.01	-298289.99	298290.01	65.32		

CHAPTER FIVE

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

The following conclusions were included of the present work for analysis of exergy and energy and simulation modeling:

1. At the abnormal condition, such as turbine over speed or load rejection, the turbine and generator model were advanced. For this reason, the model is utilized for design synthesis the control system thus give actual–time simulations.
2. The cycle thermal efficiency rises with raising the temperature and pressure of steam (0.4571% at 180 bar, 530°C to 0.5030 % at 350 bar, 650°C). Raising the inlet steam temperature (0.5422 % to 0.5864%) results in raising the reversible efficiency at the same range of temperature and pressure.
3. The efficiencies of internal exergy are raised with temperature and they are reduced with pressure.
4. The calculation revealed that the highest losses of exergy in the (K – 160 –13.34 –0.0068) power plant had been presented at different loads occurred in boiler, which accounted for (76.97%) at full load, (77.41%) at 75% load and (77.76%) at 50% of irreversibilities, followed by turbine which accounted for (10.02 at full load), (10.22%) at 75% load and (10.52%) at 50% of irreversibilities. Then condenser (4.48% at 100% load), (4.04%) at 75% load and (3.59%) at 50% of irreversibilities.
5. By exergitic analysis of the plant , a logical procedure to improve the power production of thermal power plant is presented

6. The mathematical model is processed with the built MATLAB code its results compared with those obtained from the Cycle- Tempo program. They give reasonable agreement.

5.2 Recommendations

The following recommendations are supposed to progress plant performance.

1. Study the simulation model for gas turbine plant existing in IRAQ.
2. Using control methods such as (Fuzzy, Neural…) in GA solution.
3. Study the exergetic and energetic analysis of all steam turbine plants in IRAQ.
4. Studying the effect of the atmospheric condition on the steam power plant performance.

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APPENDIX

Appendix-A: Thermodynamic Property Equations

Thermodynamic Property Equations for Steam (Saturated):

Saturation Properties:

Saturation Temperature T(S)	
$T(S)=A+\frac{B}{[\log P(S)]+C}$	
Range I	Range II
$0.000611\leq P(S)<12.33\ MPA$	$12.33\leq P(S)\leq 22.1\ MPA$
$273.16\leq T(S)<600\ \ K$	$600\leq T(S)\leq 647.3\ \ K$
A= 0.426776E2	A= − 0.387592E3
B= −0.389270E4	B= − 0.125875E5
C= − 0.948654E1	C= − 0.152578E2

Saturation Pressure P(S)	
$\text{Log } P(S)=\sum_{N=0}^{9} A(N)T(S)^N+\frac{A(10)}{T(S)-A(11)}$	
Range I	
$273.16\leq T(S)\leq 647.3\ K$	
A(0)= 0.1045592E2	A(6)= 0.903668E−15
A(1)= −0.404897E−2	A(7)= − 0.199690E−17
A(2)= −0.417520E−4	A(8)= 0.779287E−21
A(3)= 0.368510E−6	A(9)= 0.191482E−24
A(4)= − 0.101520E−8	A(10)= −0.396806E4
A(5)= 0.865310E-12	A(11)= 0.395735E2

Constant for Saturation Properties:

$$Y(S) = A+ BT(C)^{1/3}+CT(C)^{5/6}+D(C)^{7/8}+\sum_{N=1}^{7}E(N)T(C)^N$$

T(C) = [T(CR) – T (S)]/T(CR) (dimensionless), where: T(CR) =647.3 K

Specific Volume			
V(f)= Y(S) V(fCR)		V(g)=Y(S)P(CR)V(gCR)/P(S)	
Range I		**Range II**	
273.16≤T(S)≤647.3K		273.16≤T(S)≤647.3K	
A=1.0	B= −1.9153882	A= 1.0	B= 1.6351057
C= 1.2015186E1	D= −7.8464025	C= 5.2584599E1	D= −4.4694653E1
E(1)= −3.888614	E(2)= 2.0582238	E(1)= −8.9751114	E(2)= −4.3845530E−1
E(3)= −2.0829991	E(4)= 8.2180004E−1	E(3)= −1.9179576E1	E(4)= 3.6765319E1
E(5)= 4.7549742E−1	E(6)= 0.0	E(5)= −1.9462437E1	E(6)= 0.0
E(7)= 0.0		E(7)= 0.0	
V(fCR)= 3.155E−3		V(fCR)= 3.155E−3	
		P(CR)=2.2089E1	

Enthalpy	
H(f)=Y(S)H(fCR)	H(f)=Y(S)H(fCR)
Range I	Range II
273.16≤ $T(S) \leq 300\ K$	300≤ $T(S) \leq 600\ \ K$
A=0.0	A=8.839230108E−1
B=0.0	B=0.0
C=0.0	C=0.0
D=0.0	D=0.0
E(1)= 6.24698837E2	E(1)= −2.67172935
E(2)= −2.34385369E3	E(2)= 6.22640035
E(3)= −9.50812101E3	E(3)= −1.31789573E1
E(4)= 7.16287928E4	E(4)= −1.91322436
E(5)= −1.63535221E5	E(5)= 6.87937653E1
E(6)= 1.66531093E5	E(6)= −1.24819906E2
E(7)= −6.47854585E4	E(7)= 7.21435404E1
H(fCR)= 2.0993E3	H(fCR)= 2.0993E3

H(f)=Y(S)H(fCR)	H(fG)=Y(S)H(fgTP)
Range VI	Range III
600≤ $T(S) \leq 647.3\ \ K$	273.16≤ $T(S) \leq 647.3\ \ K$
A=1.0	A= 0.0
B= −4.41057805E−1	B= 7.79221E−1
C= −5.52255517	C= 4.62668
D= 6.42994847	D= −1.07931
E(1)= −1.64578795	E(1)= −3.87446
E(2)= −1.30574143	E(2)= 2.94553
E(3)= 0.0	E(3)= −8.06395
E(4)= 0.0	E(4)= 1.15633E1
E(5)= 0.0	E(5)= −6.02884
E(6)= 0.0	E(6)= 0.0
E(7)= 0.0	E(7)= 0.0
H(fCR)= 2.0993E3	H(fgTP)= 2.5009E3

H(g)=Y(S)H(gCR)
Range V
273.16≤ $T(S) \leq 647.3\ \ K$
A= 1.0
B= 4.57874342E−1
C= 5.08441288
D= −1.48513244
E(1)= −4.81351884
E(2)= 2.69411792
E(3)= −7.39064542
E(4)= 1.04961689E1
E(5)= −5.46840036
E(6)= 0.0
E(7)= 0.0
H(gCR)= 2.0993E3

Entropy			
S(f)=Y(S)S(fCR)	S(f)=Y(S)S(fCR)	S(f)=Y(S)S(fCR)	S(g)=Y(S)S(gCR)
Range I	Range II	Range III	Range VI
$273.16 \leq T(S) \leq 300\ K$	$300 \leq T(S) \leq 600\ \ K$	$600 \leq T(S) \leq 647.3\ \ K$	$273.16 \leq T(S) \leq 647.3\ \ K$
A= 0.0	A= 9.12762917E−1	A= 1.0	A= 1.0
B= 0.0	B= 0.0	B= −3.24817650E−1	B= 3.77391E−1
C= 0.0	C= 0.0	C= −2.990556709	C= −2.78368
D= 0.0	D= 0.0	D= 3.2341900	D= 6.93135
E(1)= −1.83692956E3	E(1)= −1.75702956	E(1)= −6.7806759E−1	E(1)= −4.34839
E(2)= 1.47066352E4	E(2)= 1.68754095	E(2)= −1.91910364	E(2)= 1.34672
E(3)= −4.314660	E(3)= 5.82215341	E(3)= 0.0	E(3)= 1.75261
E(4)= 4.86066733E4	E(4)= −6.33354786E1	E(4)= 0.0	E(4)= −6.22295
E(5)= 7.9975096E3	E(5)= 1.88076546E2	E(5)= 0.0	E(5)= 0.0
E(6)= −5.83339887E4	E(6)= −2.52344531E2	E(6)= 0.0	E(6)= 0.0
E(7)= 3.31400718E4	E(7)= 1.28058531E2	E(7)= 0.0	E(7)= 0.0
S(fCR)= 4.4289	S(fCR)= 4.4289	S(fCR)= 4.4289	S(gCR)= 4.4289

Thermodynamic Property Equations for Steam (Superheated):

Superheat Properties:

Specific Volume V(PT)

V(PT)=$\frac{RT_{in}}{P_{in}} - B(1)\exp[-B(2)T_{in}] + \frac{1}{10P_{in}}\{B(3) - \exp[\sum_{N=0}^{2} A(N)T(S)^N]\}\exp\left[\frac{T(S)-T}{M}\right]$

R= 4.61631E−4	M= 4.0E1
B(1)= 5.27993E−2	A(0)= −3.741378
B(2)= 3.75928E−3	A(1)= −4.7838281E−3
B(3)= 2.2E−2	A(2)= 1.5923434E−5

Enthalpy H(PT):

H(PT)=$\sum_{N=0}^{2} A(N)T_{in}^N - A(3)\exp\left[\frac{T(S)-T_{in}}{M}\right]$

A(0)= B(11)+B(12)$P_{in} + B(13)P_{in}^2$	
A(1)= B(21)+B(22) $P_{in} + B(23)P_{in}^2$	
A(2)= B(31)+B(32)$P_{in} + B(33)P_{in}^2$	
A(3)= B(41)+B(42)$T(S) + B(43)T(S)^2 + B(44)T(S)^3 + B(45)T(S)^4$	
B(11)= 2.04121E3	B(21)= 1.610693
B(12)= −4.040021E1	B(22)= 5.472051E−2
B(13)= −4.8095E−1	B(23)= 7.517537E−4
B(31)= 3.383117E−4	B(42)= −1.699419E1
B(32)= −1.975736E−5	B(43)= 6.2746295E−2
B(33)= −2.87409E−7	B(44)= −1.0284259E−4
B(41)= 1.70782E3	B(45)= 6.4561298E−8
M= 4.5E1	

Entropy S(PT)

S(PT)=$\sum_{N=0}^{4} A(N)T_{in}^N + B(1)\log[10P_{in} + B(2)] - \sum_{N=0}^{4} C(N)T(S)^N\left\{\exp\left[\frac{T(S)-T_{in}}{M}\right]\right\}$

A(0)= 4.6162961	B(2)= 1.0E−3
A(1)= 1.039008E−2	C(0)= 1.777804
A(2)= −9.873085E−6	C(1)= −1.802468E−2
A(3)= 5.43411E-9	C(2)= 6.854459E−5
A(4)= V1.170465E−12	C(3)= −1.184424E−7
B(1)= −4.650306E−1	C(4)= 8.142201E−11
M= 8.5E1	

Appendix-B: Compressed Liquid

$$
c := \begin{pmatrix}
9.9942695\cdot10^{-4} & -4.609152\cdot10^{-6} & 3\cdot10^{-6} & 6.438\cdot10^{-4} & 9.6859913\cdot10^{-3} & -2.88\cdot10^{-3} \\
-3.4631890\cdot10^{-6} & 3.504430\cdot10^{-6} & 7\cdot10^{-7} & 4.1982594\cdot10^{-1} & -3.0725260\cdot10^{-3} & 3.2\cdot10^{-3} \\
6.5359352\cdot10^{-5} & -6.688444\cdot10^{-6} & 0 & -6.521815\cdot10^{-3} & 1.6722356\cdot10^{-3} & 0 \\
-2.6236907\cdot10^{-5} & 4.868174\cdot10^{-6} & 0 & 6.365696\cdot10^{-3} & -8.069604\cdot10^{-4} & 0 \\
7.856555\cdot10^{-6} & -1.835593\cdot10^{-6} & 8\cdot10^{-7} & -1.850814\cdot10^{-3} & 1.209302\cdot10^{-4} & 0 \\
-5.000042\cdot10^{-7} & 2.128961\cdot10^{-7} & 0 & 4.519971\cdot10^{-4} & -2.200941\cdot10^{-5} & 0 \\
0 & 0 & 0 & 0 & 0 & 5.44\cdot10^{-5}
\end{pmatrix}
$$

Appendix-C: Parameters of Reheater Model

Table C.Parameters for superheated temperature model:

D1	D2	R1	R2
0.1797	7.285	0.00245	0.1013
0.1798	7.289	0.002461	0.102
0.1799	7.294	0.002472	0.1028
0.1801	7.298	0.002483	0.1035
0.1802	7.303	0.002494	0.1043
0.1803	7.307	0.002506	0.105
0.1804	7.312	0.002517	0.1058
0.1806	7.316	0.002528	0.1065
0.1807	7.321	0.002539	0.1073
0.1808	7.325	0.00255	0.108

Appendix-D: Thermodynamic Property Error Functions

Thermodynamic property error functions for power station AL – Dura (K – 160 – 13.34 – 0.0068)

Table D.1: Turbine modeling error of enthalpy at extraction no.1 of high pressure turbine, entropy at extraction no.3 of intermediate pressure cylinder.

	Max(e)	Min(e)	MAE	AAD(e)	R^2 (e)
Enthalpy for Ext.1	0.8221	$2.321e^{-4}$	0.4521	$1.0905e^{-4}$	0.9987
Entropy for Ext.3	0.0097	$1.101e^{-4}$	0.0045	$5.4632\ e^{-4}$	0.9975

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