ECONOMICS OF USING BIOMASS ELECTRIC POWER FOR ENERGY APPLICATION OF LIGHT COMMERCIAL BUILDING

A Thesis for Masters of Science Degree University of Tennessee at Chattanooga

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May 2010
I dedicate my thesis to my mother, Syamala Nellore, Brother, Vamsi Nellore, and Sister, Lavanya Nellore for their love, financial and mental support. It is also dedicated to my advisor Dr. Prakash Dhamshala who taught me the best kind of knowledge and without whom this work would never been accomplished.
ACKNOWLEDGEMENT

This thesis work would not have been possible without the support of many people. I am especially thankful to my advisor, Dr. Prakash Dhamshala, for his fruitful guidance, support and for keeping up my confidence all the time. I also express my gratitude to my graduate thesis committee members, Dr. Joseph Owino, Dr. Sagar Kapadia for serving on the committee and for their valuable suggestions.

I express my sincere gratitude to my family, for being supportive during the completion of this work. To all my friends and family members whose support gave me motivation to complete this thesis.
ABSTRACT

Due to increased concern on global warming coupled with dwindling supply of conventional fuels, attention has been focused recently on use of renewable energy resources. The energy derived from biomass has a great potential in meeting this challenge by use of the state of art technology in gasifiers.

Commercial units recently placed in the market have a potential of using the energy available in the biomass in producing electric power as well as in meeting the other building loads most efficiently, with negligible environmental effect. Woody mass, switch grass, hazzle nut shells and other biomass wastes can be utilized in a gasifier to meet the electrical, heating and cooling loads of a building. Based on current economic data, characteristics of a gasifier and that of the building, computer simulations have been performed using hourly weather data, to evaluate the economic feasibility of using a commercially available gasifier unit for a light commercial building located in Chattanooga, TN. Results obtained from the computer simulation show that gasifier units are best suited and extremely attractive for areas with abundance in biomass sources enabling a very short payback periods.

The key variables that affect the payback period for gasifier plant are the capital cost, the utility purchase factor (UPF), the total equivalent subsidies, local electric power and gas costs, fraction of rated electrical power capacity generated. The other factors that affect the payback period are the COP of the absorption cycle and the cost of the feedstock.
The excess heat energy recovered after meeting the heating and cooling loads of the building, when utilized for other applications or marketing for local sales can significantly reduce the payback period.

The gaseous emissions from the use of producer gas is very low compared to the emissions from the use of coal or natural gas resources and therefore use of the biomass resources through the investigated gasifier technology is extremely environment friendly, especially when one considers the carbon neutral aspect of the resources.
# TABLE OF CONTENTS

Acknowledgement iii  
Abstract iv  
List of Tables ix  
List of Figures xi  
Abbreviations and Acronyms xiii  

## CHAPTER I  
INTRODUCTION 1  
Global Warming 2  
Breakdown of Energy Production by Renewable Energy 5  
Zero Energy Building 8  
Biomass for ZEB 8  

## CHAPTER II  
BIOMASS  
Overview 10  
Types of Feedstock 12  
  Forestry 13  
  Agriculture 13  
  Energy Crops 13  
  Other forms of Biomass 14  
Biomass Conversion Process 14  
Different Types of Conversions 16  
  Combustion 16
Anaerobic Digestion 16
Thermo-Chemical Conversion 17
Carbon Neutral 18
Variety of Fuels 19
Demerits 19
Most Suitable Technology 20

CHAPTER III
GASIFICATION
Overview 21
Gasifier Types 22
Fixed Bed Gasifier 22
Fluidized Bed Gasifier 24
Entrained Flow Gasifier 25
Downdraft Gasifier 27
Working and Different Zones 28
Factors Effecting the Biomass Gasification 30
  Effect of the Temperature 31
  Effect of Pressure 31
  Equivalence Ratio 32
  Moisture Content in Feedstock 32
BioMax Gasifier Power Module 33
  Gasifier 34
  Gas Cooling 35
  Gas Cleaning 36
  Waste Heat Recovery 37

CHAPTER IV
CHEMICAL REACTIONS AND CALCULATIONS 39
Combustion of Producer Gas 40
  Air- Fuel Ratio and Dew Point Temperature of Producer Gas 40
Combustion Reaction Calculation 41
Adiabatic Flame Temperature 46
Emission from Combustion of Fossil Fuels and Producer Gas 48
   Combustion of Fossil Fuels 48
   Combustion of Natural Gas 50
   Combustion of Producer Gas 51

CHAPTER V
DEVELOPMENT OF MODEL FOR GASIFIER POWER MODULE 54
Principles of Reverse Engineering 56
Specifications of Light Commercial Building 57
Double Effect Absorption Chiller 64
Building Peak-Loads and Part-Loads Condition 66
Economic Analysis 67

CHAPTER VI
RESULTS 70

CHAPTER VII
CONCLUSIONS 82

LIST OF REFERENCES 84

APPENDIX 1 86
Combine Cycle Power Plant Example

APPENDIX 2 90
Transfer Function Method

APPENDIX 3 111
Double-Effect Absorption Chiller
## LIST OF TABLES

<table>
<thead>
<tr>
<th>Table No.</th>
<th>Title</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Enthalpy at Different Temperatures</td>
<td>43</td>
</tr>
<tr>
<td>2</td>
<td>Variation of Simple Payback Period with COP of Absorption Chiller and Capital Cost</td>
<td>72</td>
</tr>
<tr>
<td>3</td>
<td>Variation of Simple Payback Period with Utility Purchase factor (UPF) and Capital Cost</td>
<td>72</td>
</tr>
<tr>
<td>4</td>
<td>Variation of Simple Payback Period with Fraction of Electric Power Capacity cost of feedstock and Capital Cost</td>
<td>73</td>
</tr>
<tr>
<td>5</td>
<td>Variation of Simple Payback Period with Fraction of Electric Power Capacity cost of feedstock and Capital Cost</td>
<td>73</td>
</tr>
<tr>
<td>6</td>
<td>The Variation of Simple Payback Period with Electric Power &amp; Gas Cost and Capital Cost</td>
<td>74</td>
</tr>
<tr>
<td>7</td>
<td>The Variation of Simple Payback Period with Electric Power &amp; Gas Cost and Capital Cost</td>
<td>74</td>
</tr>
<tr>
<td>8</td>
<td>The Variation of Simple Payback Period with Federal Subsidies and Capital Cost</td>
<td>75</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page No.</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Increase of the Globe Temperature and Concentration of CO₂</td>
<td>3</td>
</tr>
<tr>
<td>2</td>
<td>Projected Growth of Worlds Energy Consumption</td>
<td>4</td>
</tr>
<tr>
<td>3</td>
<td>Renewable Energy Projected Growth</td>
<td>5</td>
</tr>
<tr>
<td>4</td>
<td>Annual Energy Consumption of Different Energy Sources</td>
<td>7</td>
</tr>
<tr>
<td>5</td>
<td>Distribution of Feed Stock</td>
<td>11</td>
</tr>
<tr>
<td>6</td>
<td>Various Biomass Feedstock Available Across US</td>
<td>12</td>
</tr>
<tr>
<td>7</td>
<td>Biomass Resources and Conversion Processes</td>
<td>15</td>
</tr>
<tr>
<td>8</td>
<td>Anaerobic Digester Plant</td>
<td>17</td>
</tr>
<tr>
<td>9</td>
<td>Carbon Neutral Accountability</td>
<td>19</td>
</tr>
<tr>
<td>Page</td>
<td>Section</td>
<td>Page</td>
</tr>
<tr>
<td>------</td>
<td>-------------------------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>10</td>
<td>Fluidized Bed Gasifier</td>
<td>24</td>
</tr>
<tr>
<td>11</td>
<td>Taxaco Entrained Flow Gasifier</td>
<td>26</td>
</tr>
<tr>
<td>12</td>
<td>Downdraft Gasifier</td>
<td>29</td>
</tr>
<tr>
<td>13</td>
<td>Effect of Equivalence Ratio and Required Temperature of Gasifier</td>
<td>33</td>
</tr>
<tr>
<td>14</td>
<td>Components of BioMax Power Module</td>
<td>34</td>
</tr>
<tr>
<td>15</td>
<td>Downdraft Gasifier in BioMax</td>
<td>34</td>
</tr>
<tr>
<td>16</td>
<td>Cooling System of BioMax</td>
<td>36</td>
</tr>
<tr>
<td>17</td>
<td>Gas Cleaning in BioMax</td>
<td>37</td>
</tr>
<tr>
<td>18</td>
<td>Waste Heat Recovery in BioMax</td>
<td>38</td>
</tr>
<tr>
<td>19</td>
<td>Carbon Dioxide Emissions by Different Energy Sources</td>
<td>52</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>---</td>
<td>---</td>
<td>---</td>
</tr>
<tr>
<td>20</td>
<td>Schematic of BioMax Power Module</td>
<td>55</td>
</tr>
<tr>
<td>21</td>
<td>Distinction Between the Mechanical Vapor Compression Cycle and Absorption Refrigeration Cycle</td>
<td>64</td>
</tr>
<tr>
<td>22</td>
<td>Variation of Cooling Load, Heat Extraction Rates on Peak Cooling Day</td>
<td>66</td>
</tr>
<tr>
<td>23</td>
<td>Flow Chart for Computer Simulations</td>
<td>69</td>
</tr>
<tr>
<td>24</td>
<td>The Input / Output Screen of the TABLET Computer Code</td>
<td>71</td>
</tr>
<tr>
<td>25</td>
<td>Variation of Payback Period with COP</td>
<td>76</td>
</tr>
<tr>
<td>26</td>
<td>Variation of Payback Period with Capital Cost and UPF</td>
<td>77</td>
</tr>
<tr>
<td>27</td>
<td>Variation of Payback Period with Cost of Feed stock</td>
<td>78</td>
</tr>
<tr>
<td>28</td>
<td>Variation of Payback Period with Capital Cost and Electric Power and Gas Cost</td>
<td>79</td>
</tr>
<tr>
<td>29</td>
<td>Variation of Payback Period with Capital Cost and Federal Subsidies</td>
<td>79</td>
</tr>
</tbody>
</table>
30  Variation of Payback Period with Capital Cost and Rated Electrical Power Capacity  80

31  Variation of Annual Operating Costs and Payback Period with Operation Mode  81
### ABBREVIATIONS AND ACRONYMS

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Annuity of Capital Investment</td>
</tr>
<tr>
<td>ACH</td>
<td>Air change equivalent to the volume of the building per hour, ft$^3$/hr (m$^3$/s)</td>
</tr>
<tr>
<td>$C_e$</td>
<td>Cost of electric energy consumption (cents/kWh)</td>
</tr>
<tr>
<td>$C_{ep}$</td>
<td>Cost of electric power produced (cents/kWh)</td>
</tr>
<tr>
<td>$C_{ed}$</td>
<td>Cost of excess electric power demand ($/yr)</td>
</tr>
<tr>
<td>$C_{htg}$</td>
<td>Annual cost of space heating ($/yr)</td>
</tr>
<tr>
<td>$C_{opr}$</td>
<td>Total annual cost of building operations ($/yr)</td>
</tr>
<tr>
<td>COP</td>
<td>Coefficient of Performance</td>
</tr>
<tr>
<td>CTF</td>
<td>Conduction Transfer Function</td>
</tr>
<tr>
<td>$E_{dL}$</td>
<td>Electrical power demand limit (kW)</td>
</tr>
<tr>
<td>$i$</td>
<td>Blended interest rate per annum</td>
</tr>
<tr>
<td>$E_{ts}$</td>
<td>Equivalent total subsidies (%)</td>
</tr>
<tr>
<td>HVAC</td>
<td>Heating, ventilating and air-conditioning system</td>
</tr>
<tr>
<td>$P_{ed}$</td>
<td>Peak electric demand for a given month (kW)</td>
</tr>
<tr>
<td>PB</td>
<td>Simple payback period (yrs)</td>
</tr>
<tr>
<td>$q_{heat}$</td>
<td>Heat capacity output of the heat pump, Btu/hr (kW)</td>
</tr>
</tbody>
</table>
\( q_{\text{lat}} \)  Latent cooling capacity of the heat pump, Btu/hr (kW)

\( q_{s} \)  Sensible cooling capacity of the heat pump, Btu/hr (kW)

\( q_{\text{tot}} \)  Total cooling capacity of the heat pump, Btu/hr (kW)

\( t_{i} \)  Moist air temperature of indoor air, °F (°C)

\( t_{a} \)  Outside air temperature, °F (°C)

\( t_{wi} \)  Wet-bulb temperature of indoor air blowing over the evaporator coil of the air conditioner or heat pump, °F (°C)

\( \text{UPF} \)  Utility payback factor

\( W_{\text{aux}} \)  Power supplied to the auxiliary fans (kW)

\( W_{c} \)  Power supplied to the compressor of heat pump (kW)

\( W_{\text{eqm}} \)  Power supplied to the equipment, computers (kW)

\( W_{\text{light}} \)  Power supplied to the lights (kW)
CHAPTER I

INTRODUCTION

The United states is the largest total energy consumer, using 100 quadrillion Btu’s per year [1]. This has increased drastically in the past 30 years mainly due increase in population and expansion of metropolitan cities.

Major sectors of energy consumptions are industries, transportation, residential and commercial. Buildings in the United States consume 39% of the total energy and 72% of its is electricity [2]. Further more, buildings emit 38% of the carbon dioxide, the primary gas associated with the climate change and global warming [2]. Currently vast majority of energy is produced from non-renewable, fossil fuel resources. Major concerns of dwindling fuels, energy security, the impact of greenhouse gases on world climate, and the depletion of the ozone layer has forced the nations to reduce the load, increase the energy efficiency and the use of renewable energy resources.

The future global economy is likely to consume even more energy, especially due to the rising energy demand of developing countries such as China and India. At the same time, the tremendous risk of climate change associated with the use of fossil fuels makes supplying this energy increasingly difficult. Coal, nuclear and hydro power plants are used primarily to produce electricity. Natural gas is widely used for heating. If oil production remains constant, there is enough to last 42 years [2]. Oil wells produce less as they become depleted which will make it impossible to keep production constant.
Similarly, there is enough natural gas to last 61 years and there is enough coal to last 133 years [3]. Consideration, due to climate change would necessitates to drastically reduce the use of fossil fuels. Based on current and anticipated energy demand growth, India and China alone will account for half of the energy demand increase in the next quarter century. As for other regions, the population in Africa will increase by more than 60% by 2030 and could become an important factor for global energy shortage. Considering this projected sharp rise in the energy demand, the depletion of fossil fuels will be lot sooner than expected, may be within the life time of present generation. Inevitably, transitional strategic plans must be developed to sustain the energy resources to maintain the present level standard of living.

LEED certification of building, the solar decathlon competition and development of standards for energy efficiency improvements are some of the measures taken to emphasize on use of renewable energy resources, and the reduction in consumption of energy and materials.

**Global Warming**

Global warming is the increase of surface temperature of the earth and that of oceans and air near the earth’s surface. The main cause is due to increase in concentrations of the carbon dioxide (CO₂) gas in the earth’s atmosphere, which is also known as green house gas that is produced due to combustion of fossil fuels. Recent rise in CO₂ levels in atmosphere is the result of rapid growth in usage of fossil fuels to meet
the increased energy demands. If the rise of carbon dioxide is left unchecked it is expected to cause the irreversible damage to the environment around the world. This damage is likely to include the increase in urban air pollution, increased air borne disease, rise of sea levels, causing floods and erosion in the coastal areas, extreme weather, which leads to more intense draught and hurricanes, reduced productivity of some agriculture regions and economic imbalance.

![Graph showing increase in temperature and CO2 concentration](image)

**Figure 1** Increase of the Globe Temperature and Concentration of the CO2 [4]

Figure 1 indicates there is a $0.7^\circ$ C rise in the temperature of the Earth’s surface over the past hundred years due to increase in the concentrations of the green house gases. This rise in the temperature is the main reason for melting of polar ice and rise in sea levels and climatic changes. Burning of the fossil fuel not only raise the temperature of environment but also lead to depletion of the ozone layer, which protects earth from harmful ultraviolet radiation from the sun.
Practical solutions exist, which may not be economical at present, is the use of more renewable energy resources.

![World Energy Use by Fuel Type, Quadrillion Btu](image)

**Figure 2** Projected Growth of Worlds Energy Consumption

There is gradual increase in the use of renewable source such as wind, solar, geothermal and bio-mass as compared to other fuel sources as shown in Figure 2. Most of the developed countries are investing huge resources and offering incentives for the advancement of renewable energy (green) technologies. This commitment to increase the use of renewable energy resources at the higher level will have a significant impact on reducing $CO_2$ emissions. Several utilities in US and other nations have committed to produce 25 % of their electricity by renewable resources. In addition to realizing significant reduction of harmful emissions, it is also found that renewable standards and energy efficiency improvement measures are an effective means to help meet critical fuel diversity, energy security, and economic goals. Some of these efforts are also considered
to be an effective means of dealing with rejuvenating the economies of the nations and the world.

**Breakdown of Energy Production by Renewable Energy Resources**

Conventional energy sources such as oil, natural gas, coal and uranium are limited resources and have the potential to run out during course of time. Renewable energy resources can overcome this problem by using these resources that may last longer than the human race. Most of the renewable energy resources are environmental friendly and reduce global warming by reducing carbon emissions. Measures have been taken by many different nations to invest huge amounts of their energy budget in the three major energy sectors mainly wind power, solar photovoltaic and biomass as shown in Figure 3.

![Global Clean-Energy Projected Growth 2009-2019 (US Billions)](image)

**Figure 3** Renewable Energy Projected Growth [6]
These three major sectors kept up a blistering growth rate, increasing 53 percent from $75.8 million in 2007 to $115.9 million in revenue in 2008 according to Clean Energy Trends [5] and it keeps rising. By 2018 it is believed that these three sectors will have revenues of $325.1 billion according to a report in *Clean Edge*.

Wind energy is the most established of all renewable energy resources, constituting 1% of global energy production, or about $51.4 billion in 2008, and accounting for more than 50% of global investment in renewable energy in 2009 [7]. It can also be the most cost-effective, especially when large installations using large turbines can take advantage of economies of scale. On the other hand, solar power is the most expensive, least efficient form of renewable energy resources now. Increased spending in the recent past has caused tremendous improvements both in cost and efficiency, and this has caused firms from China, Europe, and the U.S to flood this sector, increasing competitiveness and driving even greater improvements; in 2008, $29.6 billion of the renewable market was made from solar [8]. Biofuels, however, have been all the rage, global production and wholesale pricing of biofuels reached $34.8 billion in 2008 and are projected to a total of $105.4 billion by 2018 [9]. Last year, the global biofuels market consisted of more than 19 billion gallons of ethanol and biodiesel production worldwide. In Brazil, ethanol from sugarcane surpassed petroleum use for the first time.

As seen in the Figure 4, the energy production using biomass is the largest among all the renewable energy resources, other than hydroelectric. Companies similar to Community
Power Corporation have come up with integrated products which are smaller in size and can be used for various biomass conversion applications such as CHP, production of hydrogen and biofuels. These equipment use biomass to produce syngas or producer gas which in turn is used to run internal combustion engine coupled to a generator. These small size equipments can be used in commercial building to produce electric power as well as the heat energy, recovered from exhaust gas. Recovered heat energy can be employed for space heating and meet their space cooling load through use of double – effect absorption chiller, thus making the building to be a Zero Energy Building (ZEB).

![Figure 4 Annual Energy Consumption of Different Energy Sources](image)

**Figure 4** Annual Energy Consumption of Different Energy Sources [10]

As can be seen, there is growth in the usage of renewable energy resources by 7% out of which 43% is accounted from the biomass. Power production from biomass is going to
increase further, as the new technological breakthroughs take place in increasing the quality of the syngas produced from gasification process of biomass.

**Zero Energy Building**

The term zero energy building is self explanatory, it says a building that uses no more energy than it creates. As commercial and residential building consume about 40% of the primary energy and 72% of the electricity in United States (EIA 2005) and these energy demands are excepted to increase by 50% by 2025 (EIA 2005) due to expansion of the cities and construction of new buildings, it is necessary to design and construct building which produce enough energy to offset the rising energy demands using low cost, locally available, non polluting renewable resources. At strictest level, a ZEB generates enough renewable energy onsite to equal or exceed its annual energy use. A ZEB typically uses conventional energy sources such as electricity from the grid and natural gas from local utilities when the onsite generation does not meet the loads. When the onsite generation is greater then building loads, excess electricity is exported to the utility grid.

**Biomass for ZEB**

In this investigation, biomass is employed as the renewable energy for the ZEB. The Thermo-chemical technique used is known as gasification or Thermo-chemical conversion process. Thermo-chemical biomass conversion does include a number of
possible forms of products from initial biomass feedstock to bio-fuels, chemicals or heat and electricity. In this study we will be using this technique to generate producer gas or syngas that eventually will be utilized to produce electricity. This process has undergone, intense research and development efforts that ultimately led to the development of much more efficient conversion of feedstock into bio-fuel, improving its calorific value and reducing the formation of unwanted by products during gasification. At the same time these efforts also resulted in optimized methods for reduction of impurities, the cost of gas cleaning and conditioning. This technique is more efficient than other renewable resources in producing electricity, required in any season under any kind of load. This flexibility is not possible with other renewable sources, such as photovoltaic and wind energy, since they dependent on the climatic conditions. Comparatively, gasifier power module occupies lesser space. In this process, the producer gas produced from the biomass is employed to operate an internal combustion engine and exhaust gases are used for CHP applications. In this study, the focus will be mainly on the use of BioMax 25, a product by CPC (Community Power Corporation) which uses gasification technology to produce producer gas and that leads to electricity production with use of engines and generators. Double effect absorption chiller can be used for CHP applications by use of recovered heat energy from the exhaust gas. Using double effect absorption refrigeration system will make the entire gasifier system more energy efficient. The details on this gasifier will be covered in the next two chapters.
CHAPTER II

BIOMASS

Biomass Overview

Biomass is the organic matter derived from the plants or any living organism. Biomass is a renewable energy resource, because the energy it contains comes from the Sun. Through the process of photosynthesis, chlorophyll, a chemical substance in plants captures the sun’s energy by converting carbon dioxide drawn from the air and water from the ground into complex compounds composed of hydrogen and oxygen. When these complex compounds are burned they release the sun’s energy, carbon dioxide and water. Annually, photosynthesis process stores 5 to 8 times of the energy in biomass than the world currently consume from all sources. It is predicted that there will be a strong increase in the use of biofuels derived from biomass between 2025 and 2050.

Biomass, particularly wood, has historically been an important energy source for oven, fires and stoves since the ancient times and even now in some under-developed countries. But the recent advances, in biomass research have shown that there are more efficient and cleaner ways to use biomass. It can be converted into liquid fuel for transportation or combustible gas or producer gas using technology called gasification. Biomass energy conversion could take advantage of many existing waste streams, but would also likely involve the cultivation and conversion of solar energy to biomass which leads to large requirement of land, water and nutrients. Certain crops such as switch grass
and willow trees are especially known as energy crops, which are plants grown especially for energy generation. The other sources that contribute to the biomass potential sources are shown in Figure 5. Life-cycle cost, energy content and greenhouse gas emissions and considerations such as fertilizer production, harvesting and feedstock transportation are barriers to the widespread use of energy crops. Increase in the yield of energy crops for given energy, water and nutrient inputs would decrease the associated life-cycle cost. Research utilizing modern biotechnology could increase efficiency with respect to each of the inputs and would enable more efficient and lower cost conversion methods that would benefit the application of biomass energy. Thermo-chemical conversion systems designed for coal could be adapted to accommodate biomass feedstock or new system can be designed to take advantage of the unique properties of biomass can be explored. Biological conversion systems have the potential for higher efficiency and lower cost as our understanding and control of this feed stocks improve.

**Figure 5** Distribution of the Feedstock [11]
Types of Feedstock

There are many types of plants in the world, and many techniques can be used for energy production suitable to each of these plants. In general there are two approaches: growing plants specifically for energy use, and using the residues from plants that are used for other applications. The best approaches vary from region to region according to climate, soils, geography, population and so on. Figure 6 shows the types of biomass resources available across the U.S. It can be seen from the Figure 6 that variety of biomass resources are available suitable to the climate of individual region. The customized plants are developed to suit the regional biomass stocks to avoid excess transportation costs involved in moving them to the plant.

Figure 6 Types of Biomass Resources Available Across U.S [12]
Forestry Wastes

Forestry wastes are the largest source of producing heat and electricity now, since lumber, pulp and paper mills use them to power their factories. One large source of wood waste is tree tops and branches normally left behind in the forest after timber-harvesting operation. Some of these must be left behind to recycle necessary nutrients to the forest and to provide habitat for birds and mammals, but some could be collected for energy production. Other sources of wood waste are sawdust and bark from sawmills, shavings produced during the manufacture of furniture and organic sludge (or Black Liquor) from the pulp in paper mills.

Agriculture Wastes

Agriculture activities generate large amount of biomass residue. While most crop residues are left in the field to maintain the nutrition of the soil and prevent it from erosion. But the residue like rice husk and other grass from the sugarcane can be used as they are abundantly available after harvest. Other wastes such as whey from cheese production and manure from livestock operations can also be profitably used to produce energy while reducing disposal cost and pollution.

Energy Crops

Crops produced only to generate energy are know as energy crops. Switch grass appears
to be most promising herbaceous energy crop. It produces high yield and can be harvested annually for several years before replanting. Other native varieties that grow quickly are big bluestem, reed canary grass could be profitably be grown. These perennial crops require less maintenance and fewer inputs than do annual row crops, so they are cheaper and more sustainable to produce. Beside these, even corn, sunflower and soybeans are also used to produce oil which can be used to make fuel.

**Other Forms of Biomass**

People generate biomass wastes in many forms, including urban wood waste, the biodegradable portion of garbage. Even the sewage can be considered as the biomass waste. It can be used to produce methane gas by storing it in large tanks for 2-3 weeks. Experimental results of these processes of converting sewage in producing the syngas are very encouraging and are near commercialization.

**Biomass Conversion Process**

The Figure 7 shows the bulk of the biomass can be broadly classified as source and wastes. This Figure also shows the two methods of conversion methods namely, Thermo-Chemical and Bio-Chemical conversion techniques that can be employed to produce, heat, electric power or fuels. Combined heat and power is a method under the category of Thermo-Chemical process in which both the heat energy and electric power can be produced through a single source of fuel.
Figure 7 Biomass Resources and Conversion Processes
Different Types of Conversions

Since ancient times, many different methods have been employed by the people to extract energy from the biomass. From simple methods of burning wood to cook meals and heat houses, to gasification of biomass through Thermo-Chemical process to produce fuels, heat and electricity. Technological improvements and breakthroughs in biomass conversion techniques led to the reduction in the usage of fossil fuels and greenhouse gas emissions.

Combustion

The old way of converting biomass to energy, practiced for thousands of years, is simply burn it to produce heat. The heat can be used directly for heating building, crop drying, dairy operation and industrial processes. It can also be used to produce steam and electricity. The problem with burning biomass is that much of the energy is wasted and that it can cause some additional pollution if it is not carefully controlled. It is found to be economical only when the biomass is available at little or no cost, and is used near the source. Logistics is one more major factor as transportation of un-processed raw material or biomass can be higher than fossil fuel, as they contain less energy per unit volume.

Anaerobic Digestion

Anaerobic digestion is a biological process that produces a gas principally composed of methane (CH₄) and carbon dioxide otherwise known as biogas. Anaerobic process can occur naturally or in controlled environment such as in land fill biogas plant. Biomass
such as animal manure and various type of other organic matter fermented in an air tight container called digester as shown in Figure 8. Biomass is decomposed by breaking down to usable size molecules of sugar, and different enzymes that are added to convert it into organic acids which are further fermented to produce methane gas or biogas. Biogas is used to produce electricity and heat. The process is much efficient on a small scale. As the time taken for decomposition and space used to store biomass is large making it unsuitable for large scale energy production.

![Anaerobic Digester Plant](image)

**Figure 8** Anaerobic Digester Plant [13]

**Thermo Chemical Conversion**

Process of breaking down of biomass into various gases using heat and insufficient supply of air or oxygen is called Thermo-Chemical conversion, which is also called gasification. Gasification technology represents the next generation of solid feedstock based energy production system. Many breakthroughs have been made in the research
and development efforts of the gasification technology to increase the heating value of the product gas by using different designs of gasifiers to suit the variety of feed stocks, and by varying pressure, temperature and moisture of the feedstock in the gasifier combustion. In this study, we will analyze, if this technique is economically feasible and make a commercial building of 10000 sq ft floor area a ZEB.

**Carbon Neutral**

Carbon neutral refers to achieving net zero, carbon emissions by balancing a measured amount of carbon dioxide released equal to the amount of absorption during photosynthesis process.

Combustion of the biomass is also considered as carbon neutral. Biomass also returns carbon dioxide to the atmosphere, once biomass or the fuels derived from it are used, and the agricultural plants absorb this carbon dioxide through photosynthesis process as seen in Figure 9. It is observed that growth of trees and plants over a period of few months or years absorb carbon dioxide released from combustion of biofuels and thus sustainable balance is maintained between carbon emitted and absorbed. The greenhouse gases emitted due to agricultural process of ploughing the land, harvesting and other processes are considered to be negligible.
**Figure 9** Carbon Neutral Accountability [14]

**Variety of Fuels**

Different variety of fuels can be obtained from biomass employing different chemical technologies. For example, ethanol from the sugar-cane or corn through Fisher-Troph process, producer gas from gasification process can even be converted to hydrogen gas for fuel cell application. Byproducts produced from biomass conversion processes are efficient, viable and clean burning.

**Demerits**

Biomass is still expensive in some cases compared to other renewable energy resources and fossil fuels. In direct combustion of biomass, produce huge amounts of
carbon dioxide which leads to global warming. Transportation of the feedstock is difficult it might increase the overall capital production cost of the energy, not suitable for large-scale energy production. Production of methane or biogas through anaerobic digestion process takes lot of time and space, which might not be suitable to metropolitan environment.

**Most Suitable Technology**

Gasification is considered to be the most reliable technology, when it comes to biomass conversion process. Gasification is reliable, flexible, clean and green technology that converts feedstock of low calorific value into highly efficient bio-fuels. All other process like anaerobic digestion and direct combustion are less efficient compared to gasification. An anaerobic digestion is very slow process and needs lot of land, fill space and can not respond spontaneously to energy demand when needed. On the other hand, direct combustion is not environmental friendly. Most of the research and development is done in gasification technologies for a long time to optimize this process. There are different types of the gasification technologies to suit the size or the amount of the output energy needed. In this study, we will be using downdraft gasifier in the biomass gasification as it is more suitable for small scale applications and produce less tar compared to others, and it needs less cleaning process of the producer gas, obtained from gasification. This makes it more economically feasible, as its capital cost is reduced by use of more advanced inexpensive cleaning technology.
CHAPTER III

Gasification

Overview

Gasification is the process to convert coal and other carbon feedstock such as biomass into relatively clean biogas often alternatively called as producer gas or syngas, basically consisting of mostly hydrogen and carbon monoxide by burning them with partial oxidation in high temperature environment. This syngas is inherently cleaner burning and has higher calorific value than the initial biomass, it is derived from, and is more useful in raising overall efficiency level in production of electric power and heat energy.

The gasification of the biomass began in about 1800 or 1850’s and the product was commonly used to light streets. Before the construction of natural gas pipelines, there were many gasifier plants serving larger town and cities in Europe and U.S. During the petroleum shortage of World War II, almost a million gasifiers were used to run cars, trucks and buses using primarily wood as fuel. After the war was over, usage of the gasifier came to halt due to availability of inexpensive petroleum fuels. However, oil crisis in 1970’s and 1980’s has inspired continued research and development of coal gasifiers. Since, 1990’s concern about global warming has shifted focus to biomass as a gaseous fuel. Due its higher overall efficiency levels, the gasification becomes increasingly more desirable than direct combustion. Gaseous fuels can be easily distributed for domestic and industrial use, can also be employed in electricity producing
devices such as engines, gas turbines and fuel cells or for chemical synthesis of liquid fuels and chemicals.

However, for technical and economic reasons it will be many decades before these resources can meet more than a fraction of current world energy demand. But the need for a clean, environmentally friendly and affordable energy of modern life might force nations to deploy it in near future. To that end, United States and other developing nations are striving to make green technologies a larger part in their overall energy mix.

Types of Gasifiers

The thirst and necessity of producing a cleaner syngas with a better calorific value, has led to the development of many gasifier designs. There are many hundreds of designs available to suit the variety of applications. But they all work on similar design principles. The four main gasifier designs are:

1. Fixed Bed Gasifier
2. Fluidized Bed Gasifier
3. Entrained Flow Gasifier
4. Downdraft Gasifier

Fixed Bed Gasifier

The fixed-bed gasification process involves a bed of fuel that is maintained at a
constant-depth by the addition of fuel from the top of the gasifier. Fixed bed gasifiers have a stationary reaction zone typically supported by a grate. This process is also referred to as moving bed since the fuel is continuously moving as it is reacted and slowly proceeding downward under gravity. The “gasification agent” which can be a mixture of oxygen and steam, or air, having the oxidant fed from the bottom, counter-current to the feed, is referred to as an “up draft” or counter-current fixed-bed gasifier. In order to maintain a permeable bed, the fuel itself must be non-caking and mechanically strong. Putting in the oxidant from the top is fittingly called a “down draft” or co-current fixed-bed gasifier. Gasification occurs near the bottom of the gasifier in a relatively shallow hot reaction zone composed of a layer of charcoal. Since the oxidant is being delivered from the top, the upper part of the bed must have heat provided from an external source. Gases leaving the bed at high temperature also provide heat to the oxidant, increasing efficiency. Both methods have the dry ash or molten slag flow out through the bottom to be collected. In order to run a fixed-bed dry bed gasifier, the temperature in the gasifier must be kept below that of ash fusion temperature range (1000\(^\circ\) – 1300\(^\circ\) C). Fixed-bed gasifiers have the advantages of high thermal efficiency and low temperatures (425\(^\circ\) -650\(^\circ\) C) for the outlet gas. But they also are plagued by low throughput and produce significant amounts of tar and methane that require separation. Maintaining gas flow and regulating impurities in product gas requires fuel to be a minimum particle size of about 6.4mm. The fixed-bed gasifier has long been shown to be a commercially capable solution in the production of syngas, mainly with coal as its fuel source.
Fluidized Bed Gasifier

Fluidized-bed gasifiers are characterized by their method of suspending the supplied feedstock and other solids using an upward flow of gasification agent that causes high-velocity, turbulent motion of the reactants as shown in the Figure 10. The turbulent mixing results in uniformity of the product gas and allows a maximum heat and mass transfer between the gases and solids. It also results in a high throughput, although not quite the level of entrained flow gasifiers. This also gives uniformly distributed temperature throughout the reactor and very low oil/tar production. Operating temperatures are around 600° -1000°C depending on heating methods, indirect heating being the lower. Fluidized-Bed implementations consist of a wide range of operating conditions. They are able to accept feedstock of varying sizes and densities, but the lower operating temperatures make fairly reactive fuels preferred, such as biomass or low-grade coals. Untreated feed and/or char are separated from the gas stream and bed material.

Figure 10 Fluidized Bed Gasifier [15]
via recycling cyclones and they are fed back into the reactor, thus sustaining satisfactory levels of carbon conversion and efficiency. Multiple beds can also be used in order to maintain optimum conditions for the reaction. One distinct advantage of the fluidized-bed gasifier is the available option of in-bed sulfur capture through the use of limestone that can reach levels of 90% sulfur removal, although this is mainly necessary for gasification of coal and it does add complexity to the process.

Depending on design, biomass can be fed into the top, bottom, or middle of the moving bed. Heat to drive the gasification reaction can be provided in a variety of ways in fluidized bed gasifiers. Direct heating occurs when air or oxygen in fluidizing gas partially oxidizes the biomass and heat is released by the exothermic reactions that occur. Indirect heating methods such as internal heat exchangers, using preheated bed material, or other means can also be used to drive the gasification reactions.

Fluidized bed gasifiers can be effectively sized for medium to large-scale facilities and are best suited to situations where there is a relatively constant demand for the gas product. Essentially, all of the larger-scale gasifier facilities built and tested in the last decade use fluidized bed designs. Since the gas is typically hot, it may also contain vaporized alkali salts. The amount of tars in the raw gas can vary significantly depending on the specific reactor design and feed stocks employed.

**Entrained Flow Gasifier**

Entrained flow gasifiers take a very different approach in the conversion of the fuel. Finely pulverized fuel, on the scale of 100-600 microns, is gasified within seconds at high temperatures of around 1500° – 1900°C. The feed is entrained with oxygen and steam in a
co-current flow, which requires an air separation unit, in turn increasing costs and energy use. The gasification process’s quick reaction time allows for a very high throughput, less problem with caking fuels, and highly efficient carbon conversion. And due to the high temperature of the outlet gas ($1250^0 - 1600^0$C), the product gas contains no tar or methane, but requires a large effort in gas cooling. The elevated temperatures produce only slag as seen Figure 11, since the entrained flow gasifiers operate above the point of ash fusion. The slag is usually solidified in a water bath after leaving the gasifier.

Preparation of fuels becomes a major hurdle for the operation of entrained flow gasifiers

![Texaco Entrained Flow Gasifier](image)

**Figure 11**  Texaco Entrained Flow Gasifier [16]
because of the required small size and need to either pump fuel-water slurry or feed a dry fuel that is in a dense phase of nitrogen. The pulverization of the fuel can add considerable energy costs to the operation. Certain types of biomasses can form corrosive slag which causes damage to ceramic inner walls of the gasifier. Use of biomass also does not allow for slurry fed gasifiers due to the hydrophilic nature of most biomasses, which causes an inability to form concentrated solid slurry.

**Downdraft Gasifier**

Downdraft gasifiers have been very successful in producing biogas that can be used for operating engines because of less tar content. Downdraft gasifier have a V-shapes throat in the middle as shown in Figure 12, here limited amount of air or reactive agent is supplied for the combustion to take place. In downdraft gasifier, the fuel is introduced at the top and the air or reactive agent is introduced through a set of nozzles at the throat in the middle on the side of the reactor. The pyrolysis products in downdraft gasifier are passed through the high temperature combustion zone where the further decomposition takes place. The moisture in the syngas evaporates and serves as reactive agent. The syngas leaves the gasifier from the bottom at a temperature of about 700°C and contains less tar (about 1%), therefore it requires less cleaning and can be used to run various power producing devices such as gas turbines and internal combustion engines.
Different Zones of Gasifier

During gasification, the fuel is introduced from the top and air is supplied through the fuel bed for gasification process to proceed at various levels. The process of gasification occurs in different zones namely, drying, pyrolysis, oxidation and reduction zones developed within the reactor. The fuel is dried and moisture is removed in the drying zone, just above pyrolysis zone as shown in the Figure 12. In the pyrolysis zone, the fuel is converted to volatile compounds and char. The char is gasified in the presence of reactive agents such as carbon dioxide, steam, hydrogen and oxygen in the gasification zone. Secondary reactions of primary gases and tar take place in the oxidation zone. Because the major product of biomass at temperatures below 600°C is char, biomass gasification requires high temperature in order to gasify char. The location of these zones within the reactor depends on relative movement of the fuel and air, and the zones are differentiated by the variety of the reactions and their temperatures. The depth and relative importance of each zone depend on the chemical composition of the fuel, its moisture content and particle size, the mass flow rate of the reactive agent, and bed temperature. The detail analysis of the each zone is discussed further.

Drying Zone

Drying zone is the top most layer or the zone, as the name says it takes out the moisture from the feedstock. The drying zone receives its energy through heat transfer
from the other zones. The rate of drying depends upon the temperature, velocity and moisture content of the drying gas, as well as the external surface area of the feed material and nature of bonding of moisture to the feedstock. As the feedstock enters the drying zone the temperature is increased to 100° to 150°C. Low density materials change dimensions slightly due to shrinkage and compression whereas negligible size changes are experienced by the fuel with high density. No chemical reaction takes place in this zone.

**Pyrolysis Zone**

Heat transfer from the adjacent hot combustion zone cause devolatilization of the fuel. Temperature in the pyrolysis zone increases rapidly due to large temperature difference between the relatively cold fuel material and hot gas. As the fuel pass through
this zone, rapid charring and reduction occurs, causing changes in the structure, physical and thermal properties of the fuel. Gases, liquids and solids are produced in this zone. Liquid production of tar is undesirable in this zone, that needs to be controlled. Tar cracking and char formation are two important conversions that takes place in this zone.

**Combustion Zone**

Combustion zone is high temperature zone due to exothermic reactions taking place in this zone. Physical and chemical changes are inhibited as the limited supply of oxygen or air is introduced into the fuel bed. The air burns the carbon in the fuel until nearly all free carbon is exhausted.

**Reduction Zone**

The products of partial combustion now pass through a red hot charcoal bed where the main reduction takes place water, carbon dioxide, and partially cracked combustion products takes place and are reduced to syngas or producer gas. As the reactions taking place are endothermic this helps in cooling gas or brings the gas temperature down to some extent.

**Factors Affecting the Biomass Gasification Process**

Although gasification process is highly developed, there are several limitations particularly to biomass gasification including moisture of the feedstock, temperature of the reactor bed and equivalence ratio. A number of studies have examined each of the factors and how they effect the composition of the product gas and its calorific value.
Effect of the Temperature

The most important factor affecting the gasification is the temperature. The rate of the gasification and performance of the gasification reactor is temperature dependent. Gasification reactions are generally reversible and changing temperature can shift the equilibrium point. Most of the gasification reaction in downdraft gasifier takes place between 600\(^0\) to 900\(^0\) C. Increasing temperatures increase the formation of combustible gases, decrease the yield of char and liquids, and leads to more complete conversion of the fuel from the feedstock [18]. At higher temperatures the onset of cracking and reforming reactions breaks down the higher molecular weight hydrocarbons into lighter components [19]. The energy content of the product gases increases steadily up to 700\(^0\)C due to increase in concentration of CO and H\(_2\) and then decreases probably due to cracking of the hydrocarbons [20].

Effect of Pressure

Increased pressure increases the percentage of the methane gas in the producer gas composition (liinaki 1985). Most of the downdraft gasifiers operate at atmospheric pressure or little below atmospheric pressure in order to avoid the gas to escape from the top. Gasification at high pressure helps to maintain the gas flow rate in whole system as they will exert pressure when the gases pass through each and every stage. High pressure appears to increase the char gasification rate and increase the heat transport properties of the biomass.
Moisture Content in Feedstock

Moisture in the feedstock also plays a vital role in the biomass gasification. High moisture content fuel will decrease the thermal efficiency of the gasifier which leads to less producer gas and more tar. The maximum moisture allowed for the best yield of the producer gas is found to be 15% - 30%.

Equivalence Ratio

Equivalence Ratio (ER) is the ratio of actual air fuel ratio of the gasification process to that of a stoichiometric reaction. Equivalence ratios play a significant role in the performance of biomass gasification. It affects the fuel bed temperature, gas quality and thermal efficiency of the reaction. High equivalence ratios increase the rate of producer gas production and increase reactor temperature due to more exothermic reactions [21]. Low equivalence ratio reduce the fuel bed temperature resulting in the production of less producer gas of low energy content and more tar [22]. An ER value of zero is typically employed for pyrolysis process basically to produce liquid fuels, an ER value of 0.25 is maintained for gasification process, while values greater than 1 are suitable for complete combustion process as shown in Figure 13.

This study only deals with use of this gasifier with trade name (BioMax) for CHP applications with a total energy efficiency (the Utilization Factor) reaching up to 80% to meet the total
Figure 13  Effect of Equivalence Ratio and Required Temperature of the Gasifier

building loads of a light commercial building.

BioMax Gasifier Power Module

The BioMax gasifier power module consists of several components on a trailer mounted base as shown in Figure 14. The components are the gasifier, the gas cooler, the gas cleaner, the power conversion, and the waste heat recovery units in series as shown in the following figure. These components are designed to operate in this sequence to maximize the overall efficiency that results from use of waste heat recovery unit. The recovered waste heat can be utilized for space heating and space cooling. The quality of waste heat energy expressed in terms of the temperature of the hot water produced, is although low, but this low level of recovered energy can be supplied to the state of the art absorption chiller to produce chilled water for space cooling during cooling season.
1. Gasifier

Gasification in the BioMax is fully automated and computer controlled. The flow rates of the feedstock, air for combustion and thermo-chemical reaction are totally controlled by computer algorithm. Feed stocks enter through the top of the downdraft gasifier as shown in Figure 15.

As the feed stock particles travel further down, they are heated to pyrolysis temperatures of about 900°C and begin to emit pyrolysis vapors. The combustion gases and residual tar
vapors travel down to the char oxidation zone, along with the char formed in the flaming pyrolysis zone. The following chemical reactions occur at various zones of the gasifier.

\[ C + H_2O \rightarrow H_2 + CO \quad \text{Endothermic reaction} \]

\[ CO + H_2O \rightleftharpoons H_2 + CO_2 \]

\[ C + 2H_2 \rightleftharpoons CH_4 \]

\[ 2C + O_2 \rightleftharpoons 2CO \]

\[ C + CO_2 \rightleftharpoons 2CO \]

\[ CO + 3H_2 \rightleftharpoons H_2O + CH_4 \quad \text{Exothermic reaction} \]

One of the goals of the computer controlled environment is to ensure that the above chemical reactions would occur to the completion that results in the chemical composition (by volume) of the producer gas exiting the gasifier:

- CO: 20%
- H\textsubscript{2}: 20%
- CH\textsubscript{4}: 2%
- N\textsubscript{2}: 58%

**Gas Cooling**

Hot producer gas form the gasifier at 700\(^0\) C enters the tube–and-shell heat exchanger and is cooled to 100\(^0\) C as shown in Figure 16. The fuel gas flows inside the tubes and a cooling fluid on the shell side. There are clean out ports to allow inspection and cleaning of the tubes due to accumulation of the tar and ash.
The speed of the pump to pump fluid or liquid is computer controlled to maintain the temperature of the exit gas from the heat exchanger. The heat given out or gained by the fluid can be used to dry feedstock or any other heating purposes.

**Gas Cleaning**

A dry filter system removes the char particles from the fuel gases. The ash and char are collected continuously in a drum as a dry powder for disposal. The level of 100 ppm is generally considered to be maximum acceptable level of tars for producer gas to be consumed in an internal combustion engine.

Tar levels in Biomax are consistently less than 20 parts per million (ppm) that implies it produces an exceptionally clean gas. For woody biomass clean fuel gas typically has energy or calorific value of about 120 to 165 btu/ft³. The fuel gas composed of about
produce about 50 ft$^3$ of producer gas. The amount of char/ash is found to be less than 2% of the feedstock rates. This low amount of char/ash of the producer gas, requires a very low maintenance and facilitates automatic operation of the unit, free of skilled manpower as shown in Figure 17. The factor that it needs no water makes it even more attractive module compared to the traditional power plants, where large amounts of cooling water is needed from cooling towers.

**Waste Heat Recovery**

The waste heat recovery module recovers heat from two sources: engine exhaust and engine coolant. The waste heat recovery module also houses a pollution control system reducing CO and NO$_x$ emissions from the engine.

The thermal energy can be supplied as clean heated air or hot water (near the boiling point and unpressurized) as shown in Figure 18. A pressurized water system can be employed with
appropriate size of heat recovery unit to obtain hot water temperatures up to 320°F. Obtaining water temperatures up to 320°F from the heat recovery unit is a critical hardware modification required to enable this power module to use the state-of-the-art double-effect absorption chiller, which would make it a perfect stand-alone unit that can meet all the building energy needs.
CHAPTER IV
CHEMICAL REACTIONS AND CALCULATIONS

Fuels are combustible substances that produce heat energy during combustion process. Combustion is complete when all the carbon of the fuel is converted to carbon dioxide, hydrogen ($H_2$) to water vapor ($H_2O$) and sulphur (S) to sulphur dioxide ($SO_2$) and all other combustible elements are fully oxidized. The combustion is in incomplete when these conditions are not met. The minimum amount of air that supplies sufficient oxygen for complete combustion is called theoretical or stiochiometric amount of air. Gasifier typically allows nearly quarter amount of theoretical air to react with feedstock under ideal pressure and temperature conditions. In some of the recently marketed commercial software controlled gasifier unit, ideal conditions of temperature, pressure and air flow rate are maintained through sensors to ensure the producer gas exit the gasifier at nearly constant volumetric chemical composition of $0.20 \text{ CO}$, $0.20 H_2$, $0.02 CH_4$ with remaining being $N_2$. The key parameters that affect the performance of fuels are it’s calorific value, the amount of heat transfer resulting from combustion of kmol of the fuel for a given exit temperature from the combustion chamber or engine, the amount of green house gas emissions and the dew point temperature of the product gases. The lower values of dew point permits greater heat energy recovery from the outgoing exhaust gases thereby improving the overall energy efficiency of the producer gases. The lower dew point also reduces the maintenance costs associated with corrosion problems.
In this chapter, chemical reactions associated with combustion of the producer gas emerging from the gasifier is presented for the stoichiometric air flow rates, the determination of the air flow rates, the volumetric composition and dew point of the product gases, the rates of heat transfer resulting from the combustion of the fuel at two specified outside air temperatures is presented. The typical lowest air temperature in winter and the highest temperature in summer are considered to assess the difference in heat transfer rates or power output of the gasifier power module. Finally, the green house gas emitted from use of producer gas is compared to that resulting from the use of coal and natural gas is presented.

**Combustion of Producer Gas**

Producer gas undergoes the combustion in the internal combustion engine under the stoichiometric amount of air. In the combustion of the producer gas temperature of the outside air plays an important role in the energy output.

**The Air-Fuel Ratio and Dew point Temperature of Producer Gas**

The chemical reaction of producer gas with stoichiometric amount of air is given as

$$0.2CO + 0.2H_2 + 0.02CH_4 + 0.58N_2 + 0.24(O_2 + 3.76N_2) \rightarrow 0.22CO_2 + 0.24H_2O + 1.4824N_2$$

The air fuel ratio of the above chemical combustion equation is therefore

$$AF = \frac{(0.24)(4.76)(29)}{(0.2 \times 28 + 0.2 \times 2 + 0.02 \times 16)} = 1.4685$$
mole fraction of water vapor  \[= \frac{0.24}{0.22 + 0.24 + 1.4824} = 0.1236\]

The partial pressure of water vapor, \(P_{H_2O} = (0.0.1236) (101.325)\) Kpa = 17.85 kpa

The dew point of the product is the saturated temperature corresponding to the saturated pressure of 17.88 kpa is obtained from steam tables as

\[T_{dew\text{point}} = 50^\circ C \text{ or } 121.8^\circ F\]

Since, the dew point of the products is relatively low, therefore greater amount of heat energy can be recovered.

**Combustion Reaction Calculations**

Composition of the producer gas by volume basis is 20% CO, 20% H\(_2\), 2% CH\(_4\), 58% N\(_2\). Combustion takes place in the internal combustion engine of electrical capacity 25 kW with stoichiometric air. Following cases have been consider according to typical ambient temperature in peak winter and summer days.

1. Air entering engine at -10\(^\circ\) C

2. Air entering engine at 45\(^\circ\) C

Temperature of the producer gas entering the engine is assumed to be 60\(^\circ\) C with the volumetric flow rate of 63 cubic meter per hour. These values of temperature and flow rates of producer gas are selected as specified in the gasifier product brochure.
Case 1

**Air Entering at -10°C**

Stoichiometric equation will be

\[ 0.20CO + 0.20H_2 + 0.02CH_4 + 0.58N_2 + a_{th}(O_2 + 3.76N_2) \Rightarrow XCO_2 + YH_2O + ZN_2 \]

Balancing each and every element we get

C : \( 0.2 + 0.02 = X \Rightarrow 0.22 \)

H : \( 0.40 + 0.08 = 2Y \Rightarrow 0.24 \)

O : \( 0.2 + 2a = 2X + Y \)

\[ \Rightarrow a = 0.24 \]

Now the balanced equation will be

\[ 0.20CO + 0.20H_2 + 0.02CH_4 + 0.58N_2 + 0.24O_2 + 0.9024N_2 \rightarrow 0.22CO_2 + 0.24H_2O + 1.48N_2 \]

Applying energy balance or the first law of thermodynamics for this chemical equation gives

\[ \sum N(h_f^* + \bar{h} - h)_{\text{react/tras}} = \sum N(h_f^* + h - h^*)_{\text{product}} + Q_{\text{out}} \]
Table 1  Enthalpy Values at Different Temperatures [15]

<table>
<thead>
<tr>
<th>Substance</th>
<th>( h_f )</th>
<th>( h_{333} )</th>
<th>( h_{263} )</th>
<th>( h_{473} )</th>
<th>( h_{298} )</th>
<th>( h_{318} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO</td>
<td>-11,0530</td>
<td>9,597</td>
<td>-</td>
<td>-</td>
<td>8,669</td>
<td>-</td>
</tr>
<tr>
<td>H₂</td>
<td>-</td>
<td>9,390</td>
<td>-</td>
<td>-</td>
<td>8,468</td>
<td>-</td>
</tr>
<tr>
<td>CH₄</td>
<td>-74,850</td>
<td>5,216.9</td>
<td>-</td>
<td>-</td>
<td>4,986.26</td>
<td>-</td>
</tr>
<tr>
<td>N₂</td>
<td>-</td>
<td>9,597</td>
<td>7,645.3</td>
<td>13,781.5</td>
<td>8,669</td>
<td>9,247.6</td>
</tr>
<tr>
<td>O₂</td>
<td>-</td>
<td>-</td>
<td>7,653.6</td>
<td>13,934.7</td>
<td>8,682</td>
<td>9,266</td>
</tr>
<tr>
<td>CO₂</td>
<td>-393,520</td>
<td>-</td>
<td>-</td>
<td>16,483</td>
<td>9,364</td>
<td>-</td>
</tr>
<tr>
<td>H₂O</td>
<td>-241,820</td>
<td>-</td>
<td>-</td>
<td>15,881.7</td>
<td>9,904</td>
<td>-</td>
</tr>
</tbody>
</table>

\[
\Rightarrow Q_{out} = \sum N(h_f^\circ + \bar{h} - \bar{h})_{reactants} - \sum N(h_f^\circ + \bar{h} - \bar{h})_{product}
\]

At Ambient Air Temperature = \(-10^0\) C

\[
\sum N(h_f^\circ + \bar{h} - \bar{h})_{reactants}
= 0.20[-110,530+9,684.3-8,669]_{CO} + 0.2 [ 0+9,447 - 8,468]_{H_2} + 0.02 [ -74,850 + 2.2537 \text{kJ/kg.K} \cdot 16 \text{kg/kmol} (333-298)\text{K}]_{CH_4} + 0.58 [0 + 9,684.3- 8,669]_{N_2}
+ 0.24 [0+7,653.6-8,682]_{O_2} +0.9024 [0+7,645.3-8,669]_{N_2}
\]
= -23,755 kJ/kmol

\[ \sum N(h_f + h - h^*)_{\text{product}} \]

= 0.22 [-393,520 + 16,483 - 9,364] \text{CO}_2 + 0.24 [-241,820 + 15,822 - 9,904] \text{H}_2\text{O} + 1.4824 [0 + 13,782 - 8,669] \text{N}_2

= -134,030 kJ/kmol

\[ \Rightarrow Q_{\text{out}} = 134,030 - 23,755 = 110,275 \text{ kJ/kmol} \]

Molar Mass of the fuel = 0.20 \times 28 + 0.2 \times 2 + 0.02 \times 16 + 0.58 \times 28

= 22.0 kg

1 mole of the Product gas weighs 22 kgs and produces 110,275 kJ of heat energy, then one kg of fuel produces

= 110,275 kJ / 22 kg

= 5,031 kJ/kg

As the flow rate of the producer gas is 63 m\(^3\) / hr at standard temperature and pressure and this gives after using ideal gas equation a mass flow rate of 0.01575 kg/s, therefore the rate of heat transfer from combustion = (5,031 kJ/kg) (0.01575 kg/s) = 79 kW

**Case 2**

**Air Entering the Engine at 45° C**

The balanced equation

\[ 0.20CO + 0.20H_2 + 0.02CH_4 + 0.58N_2 + 0.24O_2 + 0.9024N_2 \rightarrow 0.22CO_2 + 0.24H_2O + 1.48N_2 \]
\[ \sum N(h_f' + h - h)_{reactants} \]
\[ = 0.20\left[-110,530+9,684.3-8,669\right]_{CO} + 0.2 \left[ 0+9,447 - 8,468\right]_{H_2} + 0.02 \left[-74,850 \right. \\
+ 2.2537 \text{kJ/kg.K} \cdot 16 \text{kg/kmol} (333-298)K]_{CH_4} + 0.58 \left[ 0 + 9,684.3-8,669\right]_{N_2} \\
+ 0.24 \left[0+9,266-8,682\right]_{O_2} + 0.9024 \left[0+9,247.6-8,669\right]_{N_2} \]
\[ = -21,922 \text{kJ/kmol} \]

\[ \sum N(h_f + h - h')_{product} \]
\[ = 0.22 \left[-393,520 +16,483 -9,364\right]_{CO_2} + 0.24 \left[-241,820+15,822 - 9,904\right]_{H_2O} + 1.4824 \left[0 + 13,782-8,669\right]_{N_2} \]
\[ = -134,030 \text{kJ/kmol} \]
\[ \Rightarrow Q_{out} = 134,030 - 21,922 \]
\[ = 112,108 \text{kJ/kmol} \]
\[ = 112,108 \text{kJ/kmol}/22.\text{kg/kmol} = 5096.0 \text{kJ/kg} \]

As the flow rate of the producer gas is 63 m³/hr at standard temperature and pressure and this gives after using ideal gas equation a mass flow rate of 0.01575 kg/s, therefore the rate of heat transfer from combustion = (5,096 kJ/kg) (0.01575 kg/s ) = **80.26 kW**

The above two calculations indicate that there is a little difference in the rate of heat energy output due to combustion of producer gas. When the air temperature changes from -10°C to 45°C. Assuming the thermal efficiency remains constant as it is function of compression ratio, it can be concluded that the electrical power output drops a little on a bitter winter day compared to on hot summer day. The electrical power output of the gasifier power module is the product of the heat transfer rate during combustion and the thermal efficiency of the engine, which normally remains constant around 30 to 31
percent. Based on the heat transfer rates calculated for combustion above, the electrical power output comes to about 23.7-24.1 kW, however, if the temperature of the exit product gases were to be reduced to a slightly lower value than the assumed value of 200°C, then the rated electrical power output would be close to the 25 kW as claimed by the manufacturer in its product brochure. It is to be remembered that the feed stock employed in the gasifier varies from wood chips to switch grass and other animal manure, which may account for variation of electrical power output.

**Adiabatic Flame Temperature**

In the absence of any work interaction and any heat loss to the adjacent surrounding (Q = 0) the temperature of the exhaust gases in the combustion will reach to the maximum temperatures, which is known as adiabatic flame temperature.

The adiabatic flame temperature of the steady combustion is obtained by setting Q = 0 and W = 0 in the energy balance equation.

\[
\sum N(h_f^i + \Delta h - h)_{\text{reactants}} = \sum N(h_f^j + \Delta h - h)_{\text{products}}, \text{ then if } Q = 0, \text{ we get}
\]

\[
\Rightarrow \sum N(h_f^i + \Delta h - h)_{\text{reactants}} = \sum N(h_f^j + \Delta h - h)_{\text{products}}
\]

The adiabatic flame temperature for the product gas combustion can be calculated using the data from the previous section.

**Balanced Equation**

\[
0.20CO + 0.20H_2 + 0.02CH_4 + 0.58N_2 + 0.24O_2 + 0.9024N_2 \rightarrow 0.22CO_2 + 0.24H_2O + 1.4824N_2
\]
Enthalpy of the reactants will the same as we got in the precious calculation as the temperature and the flow rate is same .

\[ \Rightarrow \sum N(h_f^+ + h^- h)_{\text{reactants}} = -21,922 \text{ kJ/kmol} \]

\[ \sum N(h_f^+ + h^- h)_{\text{products}} = 0.22[-393,520 + h_{\text{co}_2} - 9,364] + \]
\[ + 0.28[-241,820 + h_{\text{H}_2} - 9,904] + 1.489 \left[ h_{\text{N}_2} - 8,669 \right] \]

\[ \Rightarrow \text{The adiabatic flame temperature is obtained from trial and error approach. A first} \]
\[ \text{guess is made about the product (adiabatic flame temperature) from which the} \]
\[ \text{enthalpies of the product components are obtained from the thermodynamic} \]
\[ \text{tables and based on these enthalpies substituted in the above equation to find the} \]
\[ \text{value of} \sum N(h_f^+ + h^- h)_{\text{products}}. \text{ If this value is close to the value of} \]
\[ \sum N(h_f^+ + h^- h)_{\text{reactants}} = -21,922 \text{ kJ/kmol, then} \text{the assumed value of the product} \]
\[ \text{gases is correct, otherwise a new value for product gas temperature is assumed} \]
\[ \text{and the above procedure is repeated until the following equation is satisfied.} \]

\[ \Rightarrow \sum N(h_f^+ + h^- h)_{\text{reactants}} = \sum N(h_f^+ + h^- h)_{\text{products}} = -21,922 \text{ kJ/kmol} \]

Based on this approach, the adiabatic flame temperature is found to be 2004 K or 1731\(^{\circ}\) C

This result show that the adiabatic flame temperature is fairly high, therefore requires
a very selective lining materials for the combustion chamber of the product gas.
Emissions from Combustion of Fossil Fuels and Producer Gas

Carbon dioxide is the most important green house gas produced by human activities, primarily through the combustion of the fossil fuels. Increasing energy demand due to rapid increase in the population has lead to burn more fossil fuels which in turn is bring the changes in the carbon dioxide percentage in the atmosphere. The temperature of the globe has increased in the recent years due to this reason, which is the main cause of melting of polar ice and change of climatic conditions. As we cannot stop the rise in energy demand but we can always have an alternative to meet these energy demands in the clean and green way, which is through renewable energy. In the further literature in this section we have calculated, analyzed and compared the combustion, chemical reaction and amount of the carbon dioxide emitted by burning different sources of energy.

Combustion of Fossil Fuels

Case 3: Coal

Coal has the ultimate analysis (by mass) as

\[ C : 84.36\% \]
\[ H_2 : 1.89 \% \]
\[ O_2 : 4.4\% \]
\[ N_2 : 0.63\% \]
\[ S : 0.89\% \]
\[ \text{Ash} : 7.83\% \]
Combustion is complete under stiochiometric amount of air and products of the combustion are $CO_2$, $H_2O$, $SO_2$, $N_2$.

Mass fraction of the constituents of the coal are

$$m_{f_c} = m_c / m_{total} = 84.36 / 92.17 = 0.91135$$

$$m_{f_{H_2}} = 1.89 / 92.17 = 0.020505$$

$$m_{O_2} = 4.40 / 92.17 = 0.04773$$

$$m_{N_2} = 0.63 / 92.17 = 0.00683$$

$$m_{S_2} = 0.89 / 92.17 = 0.009656$$

Mole Fraction is of each constituent is given by

$$N_i = m_i / \text{mol wt}$$

$$N_C = 91.13 / 12 = 7.5 \text{ kmol}$$

$$N_{H_2} = 2.05 / 2 = 1.025 \text{ kmol}$$

$$N_{O_2} = 4.773 / 32 = 0.14906 \text{ kmol}$$

$$N_{N_2} = 0.683 / 28 = 0.0243 \text{ kmol}$$

$$N_{S_2} = 0.9656 / 32 = 0.030175$$

Total number of moles in the Coal, $M = N_C + N_{H_2} + N_{O_2} + N_{N_2} + N_{S_2}$

$$= 7.5 + 1.025 + 0.14906 + 0.0243 + 0.030175$$

$$= 8.7284 \text{ kmol}$$

Now the mole fraction is $Y_i = N_i / M$

$$Y_C = 7.5 / 8.7284 = 0.8593$$

$$Y_{H_2} = 1.025 / 8.7284 = 0.117432$$

$$Y_{O_2} = 0.0243 / 8.7284 = 0.002784$$

$$Y_{N_2} = 0.030175 / 8.7284 = 0.003457$$
Now is know moles of the constituents in the reaction, the combustion chemical equation will be

\[0.8593C + 0.117432H_2 + 0.017077O_2 + 0.00278402N_2 + 0.003457S_2 + a_{sh}(O_2 + 3.76N_2) \rightarrow XCO_2 + YH_2O + ZSO_2 + KN_2\]

C: \( X = 0.8593 \)

H: \( Y = 0.234864 \)

Similarly all values are acquired

The balanced equation can be written as,

\[0.8593C + 0.117432H_2 + 0.017077O_2 + 0.00278402N_2 + 0.003457S_2 + 0.966569O_2 + 3.63407N_2 \rightarrow 0.8593CO_2 + 0.234864H_2O + 0.006914SO_2 + 3.63707N_2\]

The molecular weight of the coal is given by

\[M_{\text{coal}} = 0.8593 \text{(12 kg/kmol of C)} + 0.117432 \text{(2 kg/kmol of H}_2\text{)} + 0.017077 \text{(32 kg/kmol of O}_2\text{)} + 0.00278402 \text{(28 kg/kmol of N}_2\text{)} + 0.003457 \text{(32 kg/kmol of S}_2\text{)} \]

\[M_{\text{coal}} = 11.22 \text{ kg/kmol}\]

Amount of the \(CO_2\) released due to combustion of 1 kg coal is

\[= 0.8593 \text{ kmol} \times 11.22 \text{ kg/kmol of coal} = 3.36 \text{ kg}\]

\[=> \text{ When we burn 1 kg of coal we get 3.5368 kg of } CO_2 \text{ in the exhaust gases}\]

**Case 4 : Combustion of Natural Gas**

Volumetric analysis 65% of \(CH_4\), 8% \(H_2\), 18% \(N_2\), 3% \(O_2\), and 6% \(CO_2\).

Then the chemical equation will be

\[0.65CH_4 + 0.08H_2 + 0.18N_2 + 0.03O_2 + 0.06CO_2 + a_{sh}(O_2 + 3.76N_2) \rightarrow XCO_2 + YH_2O + ZN_2\]
C : 0.65 + 0.06 = X

X = 0.71

Similarly

Y = 1.38

a = 1.31

Z = 5.106

Balanced equation

\[0.65 \text{CH}_4 + 0.08 \text{H}_2 + 0.18 \text{N}_2 + 0.03 \text{O}_2 + 0.06 \text{CO}_2 + 1.31 \text{O}_2 + 4.92 \text{N}_2\]

\[\Rightarrow 0.71 \text{CO}_2 + 1.38 \text{H}_2\text{O} + 5.106 \text{N}_2\]

We can see from the chemical equation that 1 mol of the natural gas gives out

= (0.71 kmol of CO$_2$) \times (44 kg/kmol of CO$_2$)

= 31.24 kg of CO$_2$

\[\Rightarrow 19.62 \text{ kg of the natural gas emit } 31.24 \text{ kg of the CO}_2\]

\[\Rightarrow 1 \text{ kg of the natural gas emits } 31.24 \text{ kg CO}_2 / 19.62 \text{ kg of N.gas}\]

\[\Rightarrow = 1.59 \text{ kgs of the CO}_2\]

**Case 5 : Combustion of Producer Gas**

Chemical equation will be

\[0.20\text{CO} + 0.20\text{H}_2 + 0.02\text{CH}_4 + 0.58\text{N}_2 + a_{ab}(\text{O}_2 + 3.76\text{N}_2) \Rightarrow X\text{CO}_2 + Y\text{H}_2\text{O} + Z\text{N}_2\]

Balancing each and every element we get

C : 0.2 + 0.02 = X \Rightarrow 0.22
H : 0.40 +0.08 = 2Y \Rightarrow 0.24
O : 0.2 +2a = 2X+ Y
\Rightarrow a = 0.24

Now the Balanced equation will be

\[0.20CO + 0.20H_2 + 0.02CH_4 + 0.58N_2 + 0.24O_2 + 0.9024N_2 \rightarrow 0.22CO_2 + 0.24H_2O + 1.48N_2\]

Mass of the Producer Gas = \(0.20 \times 28 + 0.2 \times 2 + 0.02 \times 16 + 0.58 \times 28\)

= 22.56 kg

Amount of the carbon dioxide emitted by 22.56 kgs of the Producer Gas

= ( 0.22 kmol of CO\(_2\) ) ( 44 kg/kmol of CO\(_2\) ) = 9.68 kgs of CO\(_2\)

\[\Rightarrow 1 \text{ kg of the Producer Gas emits } = 9.68 \text{ kg } / 22.56 \text{ kg of Producer Gas}\]

\[\Rightarrow = 0.429 \text{ kgs of } CO_2.\]

The relative magnitudes of the carbon dioxide emissions are shown in the following figure. From the Figure 19 it is clear that billions if not trillions of tons of carbon dioxide

![Figure 19 Carbon Dioxide Emissions by Different Energy Sources](image-url)
must have been released into atmosphere since industrial revolution by the fossil fuels or conventional source of energy. It can be seen that the carbon dioxide emissions by producer gas is almost 8 times less than that emitted by the coal and is almost one–third emitted by the natural gas. Moreover, even though emissions are low, the carbon footprint of the producer gas will be zero, as the minute percentage of the carbon dioxide released is absorbed by the plants during the process of photosynthesis which makes it carbon neutral. That makes biomass truly a clean renewable energy resource.
The biomass is basically an organic matter that comes from different sources. The sources may originate from forest or agricultural residue, energy crops, municipal solid wastes, forest wood, wood chips, saw dust, animal manure and industrial wastes. The chemical formulae not only varies with each of these sources, obviously for all of these different sources. The bio-fuels are derived from these resources by either thermo-chemical or bio-chemical conversion processes. Some of the conversion processes require use of water, air, steam, oxygen and other chemicals along with bio-feed stocks. Hundreds of papers have been published in the biomass literature that deal with many such processes containing all of the aforementioned fluid streams. Almost all of the papers in literature, do not specify the flow rates or thermodynamic properties or even chemical formulae at each state of the process. The bulk of the published literature lack comprehensive data on formulation that can readily be extended for computer simulations to evaluate the economic feasibility of the entire system.

A gasifier product that is recently developed and undergoing commercialization process that requires only the specification of the flow rates of the feedstock is selected for this study. This gasifier is designed to be used for a variety of feed stocks, can produce a biogas with nearly constant chemical composition and flow rate. The biogas temperature at the exit of gasifier varies slightly with the amount of moisture content in the feed stock and with the type of feed stock employed during the gasification process.
The flow rate and chemical composition of the biogas is maintained constant for variety of bio stocks due to optimized computer controlled operational parameters of air flow rates and temperatures at various locations along the gasifier. The proprietary software and optimized shape of the gasifier along with the grate design to meet the variety of feed stocks, appears to be the result of many years of research and developmental efforts. The biogas of near constant chemical composition exiting the gasifier can be further processed to produce liquid fuels, shaft work or electricity, heat energy or combination of electric power and heat energy for a CHP application.

Due to proprietary nature of the BioMax gasifier power module fabricated by a commercial firm, not much operational data was available to develop a working model for the device.

Figure 20  Schematic of BioMax Power Module
Based on the data available on the company website, a schematic of the plant is prepared as shown in the Figure 20. It may be noted that the data on flow rates of the air and that of the water used in the heat recovery unit is missing.

**Principles of Reverse Engineering**

Working model for the BioMax gasifier power module is developed based on the techniques employed in reverse engineering. The feedstock flow rate of \( m = 22 \, \text{kg/hr, dry} \), the volume flow rate of the producer gas (63 m\(^3\)/hr at STP) generated, and the electrical power output (25 kW\(_e\)) of the power module is specified by the manufacturer for one its model.

Air is supplied in the power module at various locations, to the gasifier at different zones, through the gas cooler and to the internal combustion engine as shown in the Figure 20. The flow rate of the producer gas at state 13 just before entering the engine is specified by the manufacturer to be 63 m\(^3\)/hr. Based on this flow rate, and that of the feedstock supplied on top of the gasifier, the flow rate of air to the gasifier is determined to be just about the same as that of the feedstock to the gasifier.

Based on the electrical power output of 25 kW\(_e\), and the typical thermal efficiency of the internal combustion engine to be close to 30% to 31% range, the rates of heat energy supplied through combustion of producer gas is estimated to be around in the range of 80 to 84 kW as heat energy supply. This supply of heat energy input rate requires the input air supply to the engine to be that required by a stiochiometric reaction or 100% theoretical air as shown in Chapter IV. Based on the procedure presented in Chapter IV,
the required input heat transfer rates to the engine is possible, provided the exhaust gases leave the engine around the temperature of about $125^\circ C$, this will not only guarantee the electrical power output of $25 \text{ kW}_e$ but also the recovered heat energy rates of about 100,000 Btu/hr or (29.3 kW).

In the remaining portion of this chapter, mathematical models will be presented to estimate the hourly building loads, the equipment loads, the electrical power, recovered heat energy provided by the BioMax gasifier power module, and the expenses or revenue derived due to operation of the building equipment and BioMax gasifier power module. Finally, the required equations to estimate the simple payback period are presented.

**Specifications of Light Commercial Building**

Building is located in City of Chattanooga, Tennessee

Floor Area: 10,000 square feet

Occupants during Office Hours: 40

Occupants Non-Office Hours, Weekends & Holidays: 0

Electrical Plug Load during Office Hours: 10 kW

Electrical Plug Load Non-Office Hours: 0 kW

Infiltration Rate: 0.15 ACH

Ventilation Rates: 15 cfm

Lighting load: 1.0 W/ft$^2$

The building envelope is that typically employed for a medium-sized commercial buildings with window area of 100 ft$^2$ on south and north facing walls, in which the occupants work during week days from 8.00AM to 5.00 PM.
During the day, the roof, walls and windows of the building are exposed to the sunlight that would change the temperature of these components and also the heat transfer rates to or from the building. Depending upon the ambient conditions of the outside are such as temperature and humidity, the building needs to be supplied or extracted with heat energy to maintain the space temperature and humidity at the desired levels. This implies the transfer of heat to and from the building is essentially a transient phenomenon. The transient conditions are primarily effected by the thermal properties of the building materials such as density, specific heat, coefficient of thermal conductivity, thickness of the individual material that make up the composite wall or roof. Commercial softwares are available in the market that evaluate the transient thermal loads of the building. In this investigation, a software called TABLET, a in-house software developed by Dr. Dhamshala is employed to estimate the transient building loads, and equipment loads. This software is developed based on the principles of transfer function method (TFM). Essentially, this method is primarily centered around the following assumptions and equations.

Transfer function method takes into account the thermal storage effect of the solar energy, occupants, lights and equipment. For instance, the heating or cooling load $Q$ can be considered as the response of a building or room to the effects that the temperature of the space ($T_i$), the temperature of the environment outside ($T_o$), or adjoining spaces, and the solar heat transfer rate ($\dot{Q}_{sol}$), etc. have on that building or room. The temperature of the space, the temperature of the environment outside, or adjoining spaces, the solar heat transfer rate, heat energy from occupants, equipment, and lighting ($T_i, T_o, \dot{Q}_{sol}, etc$) are known as the driving terms. The Transfer Function Method calculates the response of a
system by making the following three assumptions:

1. **Discrete time steps**: all functions of time are represented as series of values at regular time steps. (Hourly in this case).

2. **Linearity**: the response of a system is a linear function of the driving terms and of the state of the system.

3. **Causality**: the response at time \( t \) can depend only on the past, not on the future.

Take into consideration, for example, the following driving term \( u(t) \) (or sometimes represented as \( u_t \)) and its response \( y(t) \) (or sometimes represented as \( y_t \)). To indicate the time dependence of the driving term and its response to make it more readable, a linear series relationship between the response and the driving term is assumed to be in the form:

\[
y_t = -\left(a_1 y_{t-1\Delta t} + a_2 y_{t-2\Delta t} + \ldots + a_n y_{t-n\Delta t}\right) + \left(b_0 u_t + b_1 u_{t-1\Delta t} + b_2 u_{t-2\Delta t} + \ldots + b_m u_{t-m\Delta t}\right) \tag{1}
\]

where the time step \( \Delta t = 1 \) hour and \( a_1 \) to \( a_n \) and \( b_0 \) to \( b_m \) are coefficients that characterize the system.

The coefficients \( a_1 \) to \( a_n \) and \( b_0 \) to \( b_m \) are independent of the driving term or response.

Equation (1) satisfies the assumption of causality because \( y_t \) depends only upon the past values of the response (\( y_{t-1\Delta t} \) to \( y_{t-n\Delta t} \)) and on present and past values of the driving terms (\( u_t \) to \( u_{t-n\Delta t} \)). The thermal inertia of the system is taken into account with the coefficients \( a_1 \) to \( a_n \) and \( b_0 \) to \( b_m \). If these coefficients are zero, then the response is instantaneous. The greater the number and magnitude of the coefficients, the greater the weight of the past has with the system. And, the accuracy of the model increases as the number of
coefficients increases and as the time step is reduced. Hourly time resolution and a
handful of coefficients per driving term will be enough for load calculations. The
coefficients are called transfer function coefficients. In the symmetric form, the
relationship between \( u \) and \( y \), as seen above, in Equation (1) becomes:

\[
a_{o} y_{t} + a_{1} y_{t-1 \Delta t} + \ldots + a_{n} y_{t-n \Delta t} = b_{o} u_{t} + b_{1} u_{t-1 \Delta t} + \ldots + b_{m} u_{t-m \Delta t}
\]

Equation (2) can be generalized to the case where there are many driving terms. For
example, in the case of heating and cooling load calculations, if the response of the
indoor temperature \( T_{i} \) is determined by two driving terms, heat input into the space \( \dot{Q} \),
and the temperature outside \( T_{o} \), then the transfer function model can be written as
follows:

\[
a_{i,0} T_{i,t} + a_{i,1} T_{i,t-1 \Delta t} + \ldots + a_{i,n} T_{i,t-n \Delta t} = a_{o,0} T_{o,t} + a_{o,1} T_{o,t-1 \Delta t} + \ldots + a_{o,m} T_{o,t-m \Delta t}
+ a_{Q,0} \dot{Q}_{t} + a_{Q,1} \dot{Q}_{t-1 \Delta t} + a_{Q,2} \dot{Q}_{t-2 \Delta t} + \ldots + a_{Q,r} \dot{Q}_{t-r \Delta t}
\]

Equation (3) can be considered as an algorithm for calculating \( T_{i,t} \), hour by hour, given
the previous value of \( T_{i} \) and the driving terms \( T_{o} \) and \( \dot{Q} \). Likewise, \( \dot{Q} \) could be
calculated as the response if \( T_{i} \) and \( T_{o} \) were given as the driving terms. Any set of
response and driving terms can be handled as above. In other-words, for any driving
terms such as meteorological data, building occupancy, heat gain schedules, etc: the
cooling and heating loads can be calculated hour by hour. Once the necessary numerical
values of the transfer function coefficients have been calculated, the calculation of the
peak loads is simple enough for a spreadsheet.

The Transfer Function Method (TFM) applies a series of weighting factors, or conduction
transfer function (CTF) coefficients to the various exterior opaque surfaces and to
differences between sol-air temperature and inside space temperature to determine the heat gain with the appropriate reflection of thermal inertia of such surfaces.

These CTF coefficients relate an output function at a given time to the value of one or more driving functions at a given time and at a set time immediately preceding. The TFM applies a second series of weighting factors known as Room Transfer Functions (RTF) to heat gain and cooling load values from all load elements that have radiant components. The purpose is to account for the thermal storage effect in converting heat gain to cooling load. RTF coefficients relate specifically to the special geometry, configuration, mass, and other characteristics of the defined space in order to reflect weighted variations in thermal storage effect on a time basis rather than a straight-line average.

Calculating the conductive heat gain (or loss), $\dot{Q}_{\text{cond},t}$ at time t through the roof and walls can done with the following relationship:

$$
\dot{Q}_{\text{cond},t} = -\sum_{n=1}^{d_n} b_n \dot{Q}_{\text{cond},n-\Delta t} + A \left( \sum_{n=0}^{b_n} T_{\text{os},n-\Delta t} - T_i \sum_{n=0}^{c_n} \right)
$$

(4)

where: $A =$ area of the roof or wall, can be in units of m$^2$ or ft$^2$

$\Delta t =$ time step, which is 1 hour

$T_{\text{os},n-\Delta t} =$ sol-air temperature of the outside surface at time t

$b_n, c_n, d_n$ are the coefficients of conduction transfer function

Typical values of conduction transform function are shown in the following Table. or walls, the layers of wall components employed in construction of the wall can be identified from a table like the example above and with the R-value of the dominant material.

**Cooling Load**
**Sensible**

\[
Q_r = Q_{rf} + Q_{sc}
\]  \hspace{1cm} (5)

\[
Q_{rf} = \sum_{i=1}^{i} (v_i q_{r,i} + v_1 q_{r,i-1} + v_2 q_{r,i-2} + \ldots) - (w_1 Q_{r-1} + w_2 Q_{r-2} + \ldots)
\]  \hspace{1cm} (6)

\[
Q_{sc} = \sum_{j=1}^{j} (q_{c,j})
\]  \hspace{1cm} (7)

Where: \(Q_{rf}\) = sensible cooling load from heat gain elements having radiant components.

\(v\) and \(w\) = room transfer function coefficients, selected per element type,

- circulation rate, mass, and/or fixture type.
- \(q_{r,i}\) = each of \(i\) heat gain elements having a radiant component; select appropriate fractions for processing,

- \(\delta\) = time interval (1 hr)

- \(Q_{sc}\) = sensible cooling load from heat gain elements having only convective components.

- \(q_{c,j}\) = each of \(j\) heat gain factors having only convective component

It is to be noted that latent heat gain is assumed to become cooling load instantly, whereas the sensible heat gain is partially delayed depending upon the nature of the conditioned space.

The sensible heat gain for people generally is assumed to be 30 percent convective (instant cooling load) and 70 percent radiative (the delayed portion of the sensible load). The radiant portion of fluorescent lamps is assumed to be 59 percent radiative and that of the incandescent lamps to be 80 percent radiative.
The details on this method as well as on the estimation of the building loads due to lights, equipment, occupants, infiltration and ventilation are presented in Appendix 2.

Based on the estimated buildings loads for the hour, the equipment loads are evaluated by use of the following mathematical models developed from the manufacturer’s catalog performance data:

**Model of Total, Latent Cooling Capacity, and Compressor Power of the 16 Ton Capacity Air-Conditioner**

\[
q_{tot} = \frac{(190.91931 - 0.040353767 t_w - 5.3174821 t_{wi} + 0.039058891 t_w^2)}{(1 - 0.0001094898 t_a + 4.67707 \times 10^{-7} t_a^2 - 0.027635477 t_{wi}}
\]

\[+ 0.00019847 t_w^2)\] (8)

\[
q_{lat} = \frac{(0.028933553 + 6.9280368 \times 10^{-6} t_a - 0.00045 t_{wi})}{(1 + 4.9265639 \times 10^{-7} t_a - 3.7609923 \times 10^{-9} t_a^2 - 0.028812691 t_{wi}}
\]

\[+ 0.00020726652 t_{wi})\] (9)

\[
W_c = \frac{(1.8599615 + 0.01157513 t_a - 0.85042074 \ln(t_{wi}))}{(1 + 0.0043016227 t_a - 2.6179003 \times 10^{-7} t_a^2 + 6.2150864 \times 10^{-8} t_a^3}
\]

\[+ 0.56274522 \ln(t_{wi}) + 0.062346101 (\ln(t_{wi}))^2)\] (10)

**Double-Effect Absorption Chiller**

Conventionally the cooling loads of buildings are met by operation of air-conditioners or centrifugal chillers depending upon the size of the buildings. The equations (8-10) describe the models to evaluate the capacity and compressor power requirements of a 16-ton capacity air-conditioner. These equations are derived from the performance data presented by the manufacturer in their catalog, and are employed to
estimate the cost of operating the building by conventional equipment consisting of an air-conditioner and air furnace heating up the space air during the heating season.

However, the use of BioMax gasifier power module eliminates the use of the air-conditioner as it can generate the needed chilled water by use of double effect absorption chiller. The essential difference between the air conditioner and absorption chiller are shown at the bottom of the Figure 21.

![Diagram of mechanical vapor compression cycle and absorption refrigeration cycle]

**Mechanical Vapor Compression Cycle**
- COP = 2.0 – 4.5
- Cost: $300-400 / RT
- Huge Electric Power Required for Compressor
- Electric Power

**Absorption Refrigeration Cycle**
- COP = 0.55 – 1.4
- Cost: $400-1100 / RT
- Low Grade (T= 320°F)
- Heat Energy Required with Negligible Electric Power

**Figure 21** Distinction Between the Mechanical Vapor Compression Cycle and Absorption Refrigeration Cycle

The absorption chiller shown in the Figure 21 is a single-effect absorption chiller whose COP values varies from 0.5 to 0.72. The state-of-art in double effect absorption chiller marketed recently claims to provide a COP of 1.4, provided it is supplied with hot water at 320°F. Effect of variation in cooling water temperature is assumed to be negligible on
the COP value. The use of double-effect absorption chiller along with the BioMax gasifier power module will improve its overall energy efficiency reaching up to more than 80%. The COP of 1.4 is assumed to remain constant irrespective of the load.

\[
\text{COP} = 1.4 \quad (11)
\]

The details on the evaluation of COP of a simple conventional double-effect absorption chiller is presented in the Appendix 3 [23]. The absorption chiller considered in this investigation is one of the most efficient unit, with optimized hardware design and configuration to yield a highest value of COP = 1.4.

**Building Peak-Loads and Part-Load Conditions**

The building loads are transient having a peak-heating and a peak-cooling load. These loads occur at a particular hour of the year and only once a year. The equipment for HVAC applications are selected based on these peak loads. The variation of building load and extraction load (equipment load) on a peak-cooling load is shown in Figure 22. It can be seen from this Figure that heat extraction load is highest during the office hours of the building and drops significantly at other hours. Although the rate of heat energy recovered from the engine exhaust as shown in Figure 18 may be lower than the peak
Figure 22 Variation of Cooling Load, Heat Extraction Rates on Peak Cooling Day

heat extraction rates. By use of hot / or cold or both storage tanks of appropriate sizes, the peak loads can be very easily met by the recovered heat energy for each day of the entire year. This can be verified based on the hourly heat extraction load as depicted in Figure 22 and the assumed rate of recovered heat energy of 29.3 kW based on the assumption that the BioMax gasifier power module is operated constantly throughout the year.

Economic Analysis

The annual cost of space heating ($C_{htg}$) is given by

$$C_{htg} = \sum C_h W_{hl} (t) \quad (12)$$

where,

$C_h = \text{cost of heating fuel (natural gas for a conventional/alternate system)}$

$W_{hl} (t) = \text{hourly heating load at the hour, } t$
The total annual cost of building operations ($C_{opr}$) is given by

$$C_{opr} = \sum C_c \{ W_c(t) + W_{eqm}(t) + W_{light}(t) + W_{aux}(t) \} + C_{htg} \quad (13)$$

where,

- $C_c = \text{cost of electricity used from grid (for a conventional/alternate system)}$
- $W_c(t) = \text{hourly compressor load of air-conditioner at the hour, } t$
- $W_{eqm}(t) = \text{hourly load of the equipment at the hour, } t$
- $W_{light}(t) = \text{hourly load of the lights at the hour, } t$
- $W_{aux}(t) = \text{hourly load of the auxiliary equipment at the hour, } t$

The simple payback period (PB) for Gasifier power module is given by

$$PB = \frac{P \left( \frac{ets}{100} \right)}{(C_{opr} - C_{htg})} \quad (14)$$

where,

- $P = \text{total capital cost of the BioMax gasifier power module plus the cost of feedstock employed per year}$
- $ets = \text{equivalent total subsidies}$

However, in this investigation there are two types of simple payback periods that are estimated by the TABLET software. One (PB1) is based on the condition that the excess recovered heat energy left unused after meeting the building heating/cooling loads as well as domestic hot water loads is sold at a cost, it takes to produce it and other simple payback period (PB2) is the one where excess recovered heat energy left unused is dumped. Obviously, the values of PB1 are lower than that of PB2.
The TABLET software performs the computer simulations for the specified building located at the specified city. The moment the city is selected the software downloads the hourly weather data for the location into the code and based on the type of the roof, wall, windows, the sizes of these, and other pertinent HVAC related data is entered by the user of the code along with the electrical load profile and local utility power rates, the code will evaluate the simple payback periods. The order of simulations is presented in the flowchart as shown in Figure 23.
Figure 23 Flow Chart for Computer Simulations
CHAPTER VI

RESULTS

The results obtained in this investigation are based on the computer simulations performed using the software called TABLET. The software allows to conduct parametric analysis by varying more than fifty variables that can effect the economic feasibility of employing the BioMax gasifier power module. Most of the variables deal with parameters related to the building and HVAC components and others related to the energy efficiency technologies as shown in Figure 24.

However, the major thrust of this research is on economic analysis of using gasifier to meet the building loads of a light commercial building, therefore only the variables that effect the economic outcome significantly are taken into consideration, such as capital cost of the gasifier power module, the local electrical energy power and gas costs, the equivalent total subsidies, the cost of feedstock, the utility purchase factor (UPF), night set back of thermostat (NSB), COP of the absorption chiller, the fraction of rated electrical power capacity, and the hours of operation of the commercial building. The factors such as those related to the building envelope, the type of roof or walls or window area, the rates of infiltration and ventilation rates are omitted due to the esthetic reasons. A building with face brick walls are commonly employed and the architects prefer to employ greater amounts of glass windows for better looks. However, in order to have a conservative estimate of payback period the minimum values of infiltration, ventilation and window areas are considered in this study. The following tables and graphs illustrate the influence of the above variables on the payback period of the investments.
Figure 24 The Input/Output Screen of the TABLET Computer Code
# Table 2 The Variation of Simple Payback Period with COP of Absorption Chiller and Capital Cost

<table>
<thead>
<tr>
<th>Capital Cost,$</th>
<th>Ann Cost 1 $/yr</th>
<th>PB1 Yrs</th>
<th>Ann Cost 2 $/yr</th>
<th>PB2 Yrs</th>
<th>Ann Cost 1 $/yr</th>
<th>PB1 Yrs</th>
<th>Ann Cost 2 $/yr</th>
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<td>-20,180</td>
<td>8.72</td>
<td>-11,773</td>
<td>18.15</td>
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</tbody>
</table>

Table 2: The Variation of Simple Payback Period with COP of Absorption Chiller and Capital Cost

Subs = 30%; Cost of Feedstock = $50/ton; Cost of Electric Power = 10.08 cents/kWh; Gas Cost = $1.23/therm; Fraction of Electric Power Capacity = 0.9, COP = 1.4, Utility Purchase Factor = 2

# Table 3 The Variation of Simple Payback Period with Utility Purchase factor (UPF) and Capital Cost

<table>
<thead>
<tr>
<th>Capital Cost,$</th>
<th>Ann Cost 1 $/yr</th>
<th>PB1 Yrs</th>
<th>Ann Cost 2 $/yr</th>
<th>PB2 Yrs</th>
<th>Ann Cost 1 $/yr</th>
<th>PB1 Yrs</th>
<th>Ann Cost 2 $/yr</th>
<th>PB2 Yrs</th>
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Table 3: The Variation of Simple Payback Period with Utility Purchase factor (UPF) and Capital Cost

Subs = 30%; Cost of Feedstock = $50/ton; Cost of Electric Power = 10.08 cents/kWh; Gas Cost = $1.23/therm; Fraction of Electric Power Capacity = 0.9, COP = 1.4
Table 4: The Variation of Simple Payback Period with Fraction of Electric Power Capacity, Cost of Feed Stock and Capital Cost

<table>
<thead>
<tr>
<th>Feedstock Cost, $/ton</th>
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Subs = 30%; Cost of Electric Power = 10.08 cents/kWh; Gas Cost = $ 1.23/therm; COP = 1.4, Utility Purchase Factor = 2

Table 5: The Variation of Simple Payback Period with Fraction of Electric Power Capacity, Cost of Feed Stock and Capital Cost

<table>
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<tr>
<th>Feedstock Cost, $/ton</th>
<th>Ann Cost 1 $/yr</th>
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Subs = 30%; Cost of Electric Power = 10.08 cents/kWh; Gas Cost = $ 1.23/therm; COP = 1.4, Utility Purchase Factor = 2
### Table 6 The Variation of Simple Payback Period with Electric Power & Gas Cost and Capital Cost

**Gas Cost of $ 1.00/therm**

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<th>Capital Cost, $</th>
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Subs = 30%; Cost of Feedstock = $ 50/ton; Fraction of Electric Power Capacity = 0.9, COP = 1.4, Utility Purchase Factor = 2

### Table 7 The Variation of Simple Payback Period with Electric Power & Gas Cost and Capital Cost

**Gas Cost of $ 1.50/therm**

<table>
<thead>
<tr>
<th>Capital Cost, $</th>
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Subs = 30%; Cost of Feedstock = $ 50/ton; fraction of Electric Power Capacity = 0.9, COP = 1.4, Utility Purchase Factor = 2
Table 8 The Variation of Simple Payback Period with Federal Subsidies and Capital Cost

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<thead>
<tr>
<th>Capital Cost, $</th>
<th>Ann Cost 1 $/yr</th>
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Cost of Feedstock = $ 50/ton; Cost of Electric Power = 10.08 cents/kWh; Gas Cost = $ 1.23/therm; Fraction of Electric Power Capacity = 0.9, COP = 1.4, UPF = 2
The important aspect of Biomax gasifier power module is its capability of using the recovered heat energy from the engine for useful or CHP applications. If this heat energy can be used for meeting the cooling load of the building then it will have a major impact on reducing the electrical demand by the building equipment, especially that required by the compressor of an air-conditioner typically employed for space cooling. Absorption chillers can utilize the recovered heat energy to produce the chilled water for space cooling purposes. The greater is the COP of the chiller, the lesser is the amount of heat energy required by the absorption chiller. The single and double-effect absorption chillers have a COP of 0.6 and 1.4, respectively. The cost of the absorption chiller is significantly lower than that of the gasifier power module, therefore its cost is not taken into account.

The Figure 25 shows the impact of capital cost of the gasifier which varies from $200,000 to $600,000 per unit, and that of COP on the payback period. The dotted
line in this graph refers to the case, where the excess amount of recovered heat energy after meeting the building loads is dumped.

In order to encourage development of renewable energy resources, certain utilities are paying twice (UPF = 2) the amount than they charge to the customers, who feed the excess electric power to the grid that is produced from use of renewable energy resources. In certain countries, such in Germany the utilities are paying up to nearly eight times. Figure 26 shows the impact of utility purchase factor (UPF) and the capital cost of the gasifier power module on the payback period. The dotted lines in this graph refer to the case, where the excess amount of recovered heat energy after meeting the building loads is dumped.

![Variation of Payback Period with Capital Cost and Utility Purchase Factor](image)

**Figure 26** Variation of Payback Period with Capital Cost and Utility Purchase Factor (UPF)
The use of gasifier power module is economically more cost effective, if it is employed close to the source of feed stock, otherwise the cost of transporting the feed stock might become cost prohibitive. In some cases, the feed stock cost may be negative implying the people are willing to pay their feedstock residue from their sites. The impact of cost of feedstock on the payback period is shown in Figure 27. The dotted lines in this graph refer to the case, where the excess amount of recovered heat energy after meeting the building loads is dumped.

**Variation of Payback Period with Capital Cost and Cost of Feedstock**

![Graph showing variation of payback period with capital cost and cost of feedstock.](image)

**Figure 27** Variation of Payback Period with Capital Cost and Cost of Feedstock

Most of the metropolitan cities with major business centers have high electric power and gas costs compared to some of the smaller cities. Figure 28 shows the impact of the local energy costs on the payback period. For example in the city of New York, the electric power cost is 16 cents/kWh. The larger values of energy costs gives lower payback period due to large operational costs avoided and the payback from the utilities would be high for excess energy sold. It may be noted that the dotted lines in this graph
refer to the case of varying gas costs from $0.75, $1.00 and $1.50 per therm (100,000 Btus or 29.3 kWh).

**Figure 28** Variation of Payback Period with Capital Cost, Electric Power and Gas Costs

Many utilities, cities, states, and federal governments are offering different levels

**Figure 29** Variation of Payback Period with Capital Cost, and Federal Subsidies
of subsidies to encourage the development of renewable energy resources and also to protect the environment from global warming. The level of subsidies varies from state to state and other factors. Figure 29 shows the impact of subsidies and capital cost on the payback period. The dotted lines in this graph refer to the case, where the excess amount of recovered heat energy after meeting the building loads is dumped.

The BioMax gasifier power module considered in this investigation can handle a variety of feedstocks such as woody mass, switch grass, nut shells to poultry manure. The greater the density and lower the moisture content, higher would be the electrical power output. Due to variations in the feedstock conditions, the electrical power output would falls short of the rated capacity. The fraction of rated electrical power capacity has a slight impact on the payback period as shown in Figure 30. The dotted lines in this graph refer to the case, where the excess amount of recovered heat energy after meeting the building loads is dumped.

![Variation of Payback Period with Capital Cost and Fraction of Rated Electrical Power Capacity](image)

**Figure 30** Variation of Payback Period with Capital Cost, and Fraction of Rated Electrical Power Capacity
Finally, the impact of operating the building with thermostat night setback (NSB) or without thermostat night setback (NNSB) and whether operation of the building with full load conditions 24 hours a day and 365 days year on payback period and operational costs are determined. The trends of this variation is shown in Figure 31.

![Variation of Operating Costs and Payback Period](image)

**Figure 31** Variation of Annual Operating Cost and Payback Period with Operation Mode

It may be noted that annual operational costs are negative for the cases of building operation with or without night setback. This implies the revenue derived from excess electrical and heat energy sold are higher than what it would have cost to operate the building from the conventional means. The payback period for both of these cases is little over four years. However, in case of continuous electrical loads existing for 24 hours of the day and 365 days of the year, the annual operational cost is positive, but significantly less than the conventional cost as shown in the Figure 31.
CHAPTER VII

CONCLUSIONS

Based on the data gathered during the investigation and also from the results of the computer simulations, the following conclusions may be drawn:

1. The capital cost of biomass gasifier plant that require no water to operate and also can employ a wide variety of feed stocks range from $200,000 to $600,000 for a rated capacity of 25 kWₑ of electric power plus nearly 100,000 Btu/hr or (29.3 kW) of heat energy at 100°C. The high capital cost is due to high level of automation for hands free operation, this feature practically eliminates the maintenance cost associated with labor or skilled manpower.

2. The key variables that affect the payback period for the gasifier power module plant are the capital cost, the utility purchase factor (UPF), the total equivalent subsidies, local electric power and gas costs, fraction of rated electrical power capacity generated. The other factors that affect the payback period are the COP of the absorption cycle and the cost of the feedstock.

3. The excess heat energy recovered after meeting the heating and cooling loads of the building, when utilized for other applications or marketing for local sales can significantly reduce the payback period.
4. The combined cycle power plants typically operate at greater thermal efficiency close to 45% using natural gas (as shown in the example presented in Appendix 1) and the newer units exceeding 55% compared to the assumed value of 30% for the present investigation. The producer gas if employed in a combined cycle power plant can potentially yield higher electrical power output than 25 kW_e obtained from BioMax gasifier power module. Due to lower calorific value of the producer gas, mass flow rates of the producer gas would be significantly higher than the case with natural gas, thus requiring the reconfiguration of combustor and other components of the combined cycle power plant system.

5. Due to high content of nitrogen in the producer gas, use of this biofuel in Graz cycle appears to be very limited.

6. The gaseous emissions from the use of producer gas is very low compared to the emissions from the use of coal or natural gas resources and therefore use of the biomass resources through the investigated gasier technology is extremely _environment friendly_, especially when one considers the carbon neutral aspect of the resources.
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APPENDIX 1

Combined-Cycle Power Plant

A combined gas steam power plant is considered. The topping cycle is a gas turbine cycle and the bottoming cycle is a non ideal reheat Rankine cycle. The mass flow rate of the air in the gas turbine cycle, the rate of total heat input, and thermal efficiency of the combined cycle is determined.

The analysis of gas cycle

\[ T_7 = 290 \text{ K} \rightarrow h_7 = 290 \text{ kJ/kg} \]

\[ P_{\text{r}} = 1.2311 \]

\[ P_{\text{r}} = \frac{P_{8,S}}{P_7} (8)(1.2311) = 9.849 \rightarrow h_{8,S} = 526.12 \text{ kJ/kg} \]

\[ \eta_c = \frac{h_{8,S} - h_7}{h_8 - h_7} \rightarrow h_8 = h_7 + \frac{(h_{8,S} - h_7)}{\eta_c} \]

\[ = 290.16 + (526.12 - 290.16) / (0.80) \]

\[ = 585.1 \text{ kJ/kg} \]
\[ T_9 = 1400\, K \implies h_9 = 1515.42\, \text{kJ/kg} \]
\[ p_{r_9} = 450.5 \]

\[ P_{r_{10}} = \frac{P_{10}}{P_9} = \left( \frac{1}{8} \right) (450.5) = 56.3 \implies h_{10} = 860.35 \, \text{kJ/kg} \]

\[ \eta_r = \frac{h_9 - h_{10}}{h_9 - h_{10_i}} \implies h_{10_i} = h_9 - \eta_r (h_9 - h_{10}) \]
\[ = 1515.42 - (0.85)(1515.42 - 860.35) \]
\[ = 958.4 \, \text{kJ/kg} \]

\[ T_{11} = 520\, K \implies h_{11} = 523.63 \, \text{kJ/kg} \]

From the steam tables

\[ h_1 = h_{f_{10\, \text{kPa}}} = 191.81 \]

\[ v_1 = v_{f_{10\, \text{kPa}}} = 0.00101 \, \text{m}^3/\text{kg} \]

\[ W_{pI, in} = v_1(P_2 - P_1) \]
\[ = (0.00101 \, \text{m}^3/\text{kg}) (15000 - 10\, \text{kPa}) \left( \frac{1\, \text{kJ}}{1\, \text{kPa} \cdot \text{m}^3} \right) \]
\[ = 15.14 \, \text{kJ/kg} \]

\[ h_2 = h_1 + w_{pI, in} = 191.81 + 15.14 = 206.95 \, \text{kJ/kg} \]

\[ P_3 = 15 \, \text{MPa} \implies h_3 = 3157.9 \, \text{kJ/kg} \]

\[ T_3 = 450^\circ \text{C} \implies s_3 = 6.1428 \, \text{kJ/kg \cdot K} \]

\[ P_4 = 3 \, \text{MPa} \implies x_{4_i} = \frac{S_{4_i} - S_f}{S_{fb}} = \frac{6.1434 - 2.6454}{3.5402} \]
\[ S_4 = S_3 \quad h_4 = h_f + x_4 h_{fg} \]
\[ \eta_f = \frac{h_3 - h_4}{h_3 - h_4} \Rightarrow h_4 = h_3 - \eta_f (h_3 - h_4) \]
\[ = 2838.1 \text{ kJ/kg} \]
\[ P_5 = 3 \text{ MPa} \quad \Rightarrow \quad h_5 = 3457.2 \text{ kJ/kg} \]
\[ T_5 = 500^\circ C \quad S_5 = 7.2359 \text{ kJ/kg.K} \]
\[ P_6 = 10 \text{ KPa} \quad \Rightarrow \quad x_{6s} = \frac{S_{6s} - S_f}{S_{fg}} = 0.8783 \]
\[ S_{6s} = S_5 \quad \Rightarrow \quad h_{6s} = h_f + x_{6s} h_{fg} = 2292.8 \text{ kJ/kg} \]
\[ \eta_f = \frac{h_5 - h_6}{h_5 - h_{6s}} \Rightarrow h_6 = h_5 - \eta_f (h_5 - h_{6s}) \]
\[ = 3457.2 - (0.85)(3457.2 - 2292.8) \]
\[ = 2467.5 \text{ kJ/kg} \]

Noting that \( \dot{Q} = \dot{W} = \Delta ke = \Delta pe = 0 \) for the heat exchanger, the steady flow energy balance equation yields

\[ \dot{E} - \dot{E}_{\text{out}} = \Delta \dot{E} \]
\[ \dot{E}_{\text{in}} = \dot{E}_{\text{out}} \]
\[ \sum \dot{m} h_i = \sum \dot{m} h_e \Rightarrow \dot{m}_s (h_3 - h_2) = \dot{m}_{\text{air}} (h_{10} - h_{11}) \]
\[ m_{\text{air}} = \frac{h_3 - h_2}{h_{10} - h_{11}} m_s = 203.65 \text{ kg/s} \]

b). \[ \dot{Q}_{\text{in}} = \dot{Q}_{\text{air}} + \dot{Q}_{\text{reheat}} = \dot{m}_{\text{air}} (h_9 - h_8) + \dot{m}_{\text{reheat}} (h_5 - h_4) \]
\[ = 207,986 \text{ kW} \]

c.) \[ \dot{Q}_{\text{out}} = \dot{Q}_{\text{out,air}} + \dot{Q}_{\text{out,steam}} = \dot{m}_{\text{air}} (h_{11} - h_7) + \dot{m}_s (h_6 - h_1) \]
\[ \eta_{th} = 1 - \frac{Q_{\text{out}}}{Q_{\text{in}}} = 1 - \frac{115,805\, kW}{207,986\, kW} = 44.3\% \]
APPENDIX 2

Transfer Function Method

The estimation of the annual or monthly energy requirements of a building to maintain it at comfortable conditions require the knowledge of building loads for each hour of the year. Chapter 4 in the main body of the text dealt with the materials to compute the solar angles for each hour provided the hourly weather data is available. After estimating the sol-air temperature for the hour, one can estimate the solar heat gain through the window or walls or roofs. The materials presented in this appendix deals with estimation of hourly cooling load that is required to estimate the energy cost associated with air-conditioning equipment. Approximate techniques to estimate the peak heating and cooling loads using CLTD method is also presented. Hourly loads are estimated using transfer function method, since the weather data is available for most of the cities on an hourly basis. The weather data collected for each city reflects the historical average for a period of 20 to 30 years.

Transfer function method takes into account the thermal storage effect of the solar energy, occupants, lights and equipment. For instance, the heating or cooling load \( Q \) can be considered as the response of a building or room to the effects that the temperature of the space \( (T_i) \), the temperature of the environment outside \( (T_o) \), or adjoining spaces, and the solar heat transfer rate \( (\dot{Q}_{sol}) \), etc. have on that building or room. The temperature of the space, the temperature of the environment outside, or adjoining spaces, the solar heat transfer rate, heat energy from occupants, equipment, and lighting \( (T_i, T_o, \dot{Q}_{sol}, etc) \) are known as the driving terms. The Transfer Function Method calculates the response of a system by making the following three assumptions:

4. **Discrete time steps**: all functions of time are represented as series of values at regular time steps. (Hourly in this case).

5. **Linearity**: the response of a system is a linear function of the driving terms and of the state of the system.

6. **Causality**: the response at time \( t \) can depend only on the past, not on the future.

Take into consideration, for example, the following driving term \( u(t) \) (or sometimes represented as \( u_t \)) and its response \( y(t) \) (or sometimes represented as \( y_t \)). To indicate the time dependence of the driving term and its response to make it more readable, a linear series relationship between the response and the driving term is assumed to be in the form:

\[
y_t = -(a_1 y_{t-1\Delta t} + a_2 y_{t-2\Delta t} + \ldots + a_n y_{t-n\Delta t}) + (b_0 u_t + b_1 u_{t-1\Delta t} + b_2 u_{t-2\Delta t} + \ldots + b_m u_{t-m\Delta t}) \quad (1)
\]

where the time step \( \Delta t = 1 \) hour and \( a_1 \) to \( a_n \) and \( b_0 \) to \( b_m \) are coefficients that characterize the system.
The coefficients $a_i$ to $a_n$ and $b_0$ to $b_m$ are independent of the driving term or response. Equation (F1) satisfies the assumption of causality because $y_t$ depends only upon the past values of the response ($y_{t-n\Delta t}$ to $y_{t-n\Delta t}$) and on present and past values of the driving terms ($u_t$ to $u_{t-n\Delta t}$). The thermal inertia of the system is taken into account with the coefficients $a_i$ to $a_n$ and $b_0$ to $b_m$. If these coefficients are zero, then the response is instantaneous. The greater the number and magnitude of the coefficients, the greater the weight of the past has with the system. And, the accuracy of the model increases as the number of coefficients increases and as the time step is reduced. Hourly time resolution and a handful of coefficients per driving term will be enough for load calculations. The coefficients are called transfer function coefficients.

In the symmetric form, the relationship between $u$ and $y$, as seen above, in Equation (1) becomes:

$$a_0 y_t + a_1 y_{t-\Delta t} + \ldots + a_n y_{t-n\Delta t} = b_0 u_t + b_1 u_{t-\Delta t} + \ldots + b_m u_{t-n\Delta t}$$  \hspace{1cm} (2)

Equation (F2) can be generalized to the case where there are many driving terms. For example, in the case of heating and cooling load calculations, if the response of the indoor temperature $T_i$ is determined by two driving terms, heat input into the space $\dot{Q}$, and the temperature outside $T_o$, then the transfer function model can be written as follows:

$$a_{i,0} T_i + a_{i,1} T_{i-\Delta t} + \ldots + a_{i,n} T_{i-n\Delta t} = a_{o,0} T_o + a_{o,1} T_{o-\Delta t} + \ldots + a_{o,m} T_{o-m\Delta t}$$

$$+ a_{D,0} \dot{Q} + a_{D,1} \dot{Q}_{t-\Delta t} + a_{D,2} \dot{Q}_{t-2\Delta t} + \ldots + a_{D,r} \dot{Q}_{t-r\Delta t}$$  \hspace{1cm} (3)

Equation (3) can be considered as an algorithm for calculating $T_i$, hour by hour, given the previous value of $T_i$ and the driving terms $T_o$ and $\dot{Q}$. Likewise, $\dot{Q}$ could be calculated as the response if $T_i$ and $T_o$ were given as the driving terms.

Any set of response and driving terms can be handled as above. In other-words, for any driving terms such as meteorological data, building occupancy, heat gain schedules, etc: the cooling and heating loads can be calculated hour by hour. Once the necessary numerical values of the transfer function coefficients have been calculated, the calculation of the peak loads is simple enough for a spreadsheet.

The Transfer Function Method (TFM) applies a series of weighting factors, or conduction transfer function (CTF) coefficients to the various exterior opaque surfaces and to differences between sol-air temperature and inside space temperature to determine the heat gain with the appropriate reflection of thermal inertia of such surfaces.

These CTF coefficients relate an output function at a given time to the value of one or more driving functions at a given time and at a set time immediately preceding. The TFM applies a second series of weighting factors known as Room Transfer Functions (RTF) to heat gain and cooling load values from all load elements that have radiant components. The purpose is to account for the thermal storage effect in converting heat gain to cooling load. RTF coefficients relate specifically to the special geometry, configuration, mass,
and other characteristics of the defined space in order to reflect weighted variations in thermal storage effect on a time basis rather than a straight-line average.

Calculating the conductive heat gain (or loss), \( \dot{Q}_{\text{cond},t} \) at time \( t \) through the roof and walls can be done with the following relationship:

\[
\dot{Q}_{\text{cond},t} = - \sum_{n \geq 1} d_n \dot{Q}_{\text{cond},t-n\Delta t} + A \left( \sum_{n \geq 0} b_n T_{\text{os},t-n\Delta t} - T_i \sum_{n \geq 0} c_n \right) \quad (4)
\]

where: \( A \) = area of the roof or wall, can be in units of \( \text{m}^2 \) or \( \text{ft}^2 \).
\( \Delta t \) = time step, which is 1 hour.
\( T_{\text{os},t-n\Delta t} \) = sol-air temperature of the outside surface at time \( t \)
\( b_n, c_n, d_n \) are the coefficients of conduction transfer function

Typical values of conduction transform function are shown in the following Table 3.

| Table 3: Roof Conduction Transfer Function Coefficients (b and d factors) |
|----------------|--------------------------|----------------|----------------|----------------|----------------|----------------|----------------|----------------|
| Roof Group     | Layer Sequence Left to Right = Inside to Outside | \( n = 0 \) | \( n = 1 \) | \( n = 2 \) | \( n = 3 \) | \( n = 4 \) | \( n = 5 \) | \( n = 6 \) |
| 1              | Layers E A3 B25 E A0    | bn            | 0.0049 | 0.0347 | 0.0137 | 0.00036 | 0  | 0  | 0  |
|                | Steel deck with 3.33 in insulation | dn            | 1 -0.3545 | 0.0227 | -5E-05 | 0  | 0  | 0  |
| 2              | Layers E A3 B14 E E2 A0 | bn            | 0.0006 | 0.0122 | 0.0128 | 0.00143 | 1E-05 | 0  | 0  |
|                | Steel deck with 5 in insulation | dn            | 1 -0.6006 | 0.0861 | -0.0016 | 0  | 0  | 0  |
| 3              | Layers E E5 E4 C12 E E2 A0 | bn            | 0.0061 | 0.0398 | 0.0138 | 0.00025 | 0  | 0  | 0  |
|                | 2 in. h.w. concrete deck with suspended ceiling | dn            | 1 -0.7562 | 0.0144 | -6E-05 | 0  | 0  | 0  |

For walls, the layers of wall components employed in construction of the wall can be identified from a table like the example above and with the R-value of the dominant material.

**Cooling Load**

**Sensible**

\[
Q_s = Q_{rf} + Q_{sc} \quad (5)
\]

\[
Q_{rf} = \sum_{j=1} v_a q_{t,j} + v_1 q_{t,i-1} + v_2 q_{t,i-2} + \ldots - (w_1 Q_{i-1} + w_2 Q_{i-2} + \ldots) \quad (6)
\]

\[
Q_{sc} = \sum_{j=1} q_{c,j} \quad (7)
\]

where: \( Q_{rf} \) = sensible cooling load from heat gain elements having radiant components.
\( v \) and \( w \) = room transfer function coefficients, selected per element type, circulation rate, mass, and/or fixture type.
\( q_i = \) each of \( i \) heat gain elements having a radiant component; select appropriate fractions for processing,
\( \delta = \) time interval (1 hr)
\( Q_{sc} = \) sensible cooling load from heat gain elements having only convective components.
\( q_c = \) each of \( j \) heat gain factors having only convective component

It is to be noted that latent heat gain is assumed to become cooling load instantly, whereas the sensible heat gain is partially delayed depending upon the nature of the conditioned space.

The sensible heat gain for people generally is assumed to be 30 percent convective (instant cooling load) and 70 percent radiative (the delayed portion of the sensible load).

The radiant portion of fluorescent lamps is assumed to be 59 percent radiative and that of the incandescent lamps to be 80 percent radiative.

**External Heat Gain Through Glass**

Conduction:

\[
q = UA(t_o - t_i)
\]  \hspace{1cm} (8)

Solar:

\[
q = A(SC)(SHGF)
\]  \hspace{1cm} (9)

where: \( U = \) design heat transfer coefficients, for glass
\( SC = \) shading coefficient
\( SHGF = \) solar heat gain factor by orientation (geographic location).

The equation to estimate the SHGF is presented in Chapter 4. The maximum value of SHGF for each month, direction and a few latitudes is also presented Table 13 in Appendix E.

**Partitions, Ceilings, Floors**

\[
q = UA(t_b - t_i)
\]  \hspace{1cm} (10)

where: \( t_b = \) temperature of the adjacent space
\( t_i = \) inside design temperature in conditioned space

**Internal Heat Gain**

**People**

\[
q_{\text{sensible}} = N(SHG)
\]  \hspace{1cm} (11)

\[
q_{\text{latent}} = N(LHG)
\]  \hspace{1cm} (12)

where: \( N = \) number of people occupying space
\( SHG = \) sensible heat gain
LHG = latent heat gain

**Lights**

\[ q_{el} = 3.41 W F_{ul} F_{sa} \]  \hspace{1cm} (13)

where: \( W \) = wattage input from electrical plans or lighting fixture data  
\( F_{ul} \) = lighting use factor  
\( F_{sa} \) = special allowance factor

**Appliances**

\[ q_{sensible} = q_{input} F_u F_L \]  \hspace{1cm} (14)

where: \( q_{input} \) = rated energy input from appliances  
\( F_u, F_L \) = usage factors, cooling load factors

**Ventilation and Infiltration Air**

**Sensible, Latent, and Total Loads**

\[ q_{sensible} = 1.10 Q (t_o - t_i) \]  \hspace{1cm} (15)

\[ q_{latent} = 4840 Q (W_o - W_i) \]  \hspace{1cm} (16)

\[ q_{total} = 4.5 Q (h_o - h_i) \]  \hspace{1cm} (17)

where: \( Q \) = ventilation airflow – as per ASHRAE Standard 62; infiltration cfm  
\( t_o, t_i \) = outside, inside air temperatures, \(^\circ\)F  
\( W_o, W_i \) = outside, inside air humidity ratio, lb (water) / lbm d.a  
\( h_o, h_i \) = outside, inside air enthalpy, Btu/lbm d.a

\[ Latent \ Q = \sum_{n=1}^{n} (q_{c,n}) \]  \hspace{1cm} (18)

\( q_c \) = each of \( n \) latent heat gain elements.

The space heat load will be the summation of the heat transfer from each of the above. Weather and climate data analysis will have to be performed to find the peak summer and winter loads so that the HVAC system can meet the peak loads.

The radiant portions of sensible heat gain from the above loads are converted to current hourly cooling loads using the Equation (6). Estimation of total hourly cooling loads from all the components along with the heat gain from ducts and fans would provide the hourly equipment load. Knowing the local energy costs and the performance data on the equipment one can estimate the hourly energy cost.
Estimation of Approximate Peak Loads Using CLTD method

Peak Heating Load

The peak heating load normally occurs on a very cool day. The ASHRAE hand book lists the design outside winter temperatures for most of the cities in the world. These temperature are selected based on the historical fact that outside air temperature drop below these values only 2.5 percent of the time during the heating season for the given location. The design temperatures for winter and summer are given in Table 24 in Appendix E for most of the cities in U.S. The heat gain into the space due to people, solar energy, equipment and lighting are often neglected to estimate the peak heating load. The Table 1 describes the methodology to estimate the peak heating load.

Peak Cooling Load

The estimation of peak cooling load is complicated due to the thermal storage effect of the building mass. For instance, a major portion of the solar gain in the early part of the day is absorbed by the building roof, walls and inside furnishings before it raises the building space temperature. The cooling load as measured by the rise in inside air temperature lags by certain amount of time depending upon the type of roof, walls and inside furnishings of the building. This effect of thermal storage also applies to the internal load caused by sensible heat released by the occupants, lights, and equipment. The time delay in peak load occurrence is taken into account by the cooling load factor and the heat gain by the sunlit walls, roofs due to solar energy are taken into account by the cooling load temperature difference (CLTD). The peak cooling building loads can be estimated with reasonable accuracy by the cooling load temperature difference (CLTD) method as illustrated in the Table 2. The methodology presented in Tables 1 and 2 are useful only in estimating the peak loads for the building by which the HVAC equipment can be sized.

The cooling load due to heat transfer through roof, windows and walls can be estimated from

\[ q \text{ (W, Btu/hr)} = U \times A \times (\text{CLTD}_c) \]  \hspace{1cm} (19)

where: U is heat transfer coeff, W/m².K or Btu/hr.ft².°F  
A is area calculated from building plans, m², or ft²

The Corrected Cooling Load Temperature Difference, CLTDₖ is given by

\[ \text{CLTD}_c = \{(\text{CLTD}+\text{LM})K + (78-t_i) + (t_{om} - 85)\} \]  \hspace{1cm} (20)

where  
\( t_i \) inside design temperature, °F  
CLTD cooling load temperature difference, °F obtained from Tables given in Appendix E [24] for sunlit walls, roofs, and windows  
LM correction for latitude and month see Tables in Appendix E [24]  
K color adjustment factor (k=1 for dark surface)  
\( T_m \) outdoor mean temperature = \( t_0 - DR/2 \)  
\( t_o \) outside design temperature, °F  
DR is the daily range

95
$U$ is heat transfer coeff, $W/m^2.K$, or $Btu/hr.ft^2.0^\circ F$

### Table 1  Summary of Calculating Design Heating Load for Buildings

<table>
<thead>
<tr>
<th>Heating Load Component</th>
<th>Equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Roofs, ceilings, walls, and floors</td>
<td>$q = U A (t_i - t_o)$ where: $U$ is heat transfer coeff, $Btu/hr.ft^2.0^\circ F$ \ A is area calculated from building plans \ $q$ is in Btu/hr \ $t_i$ inside temperature \ $t_o$ outside design temperature from table $^1$</td>
</tr>
<tr>
<td>Walls and floors below grade</td>
<td>$q = U A (t_i - t_g)$ where $U = 0.1 \frac{Btu}{hr ft^2.0^\circ F}$ for concrete floors \ $= 0.2$ for walls \ $t_g = 50^\circ F$ for most of the locations</td>
</tr>
<tr>
<td>Floors around the grade</td>
<td>$q = U' P (t_i - t_o)$ where: $P$ is the perimeter of the building from the drawing plans. \ $U' = 0.55 \frac{Btu}{hr ft.0^\circ F}$ for concrete floors with R-5 insulation along the edge \ $U' = 0.70 \frac{Btu}{hr ft.0^\circ F}$ for R-2.5 along edge</td>
</tr>
<tr>
<td>Infiltration and Ventilation Air.</td>
<td></td>
</tr>
<tr>
<td>Sensible</td>
<td>$q_s = 1.1 Q \Delta t$ where: $Q =$ volume flow rate (CFM) \ $\Delta t = t_o - t_i$, $^\circ F$, $q_s$ is in Btu/hr \ $t_i, t_o$ design inside and outside$^1$ temperature</td>
</tr>
<tr>
<td>Latent</td>
<td>$q_s = 1.23 Q \Delta t$ $\Delta t$ is in $^\circ C$, $q_s$ is in watts, $Q$ is in L/s</td>
</tr>
<tr>
<td></td>
<td>$q_l = 4840 Q \Delta w$ $\Delta w = w_o - w_i$, $q_l$ is in Btu/hr, $Q$ is in CFM \ $w_i, w_o$ design inside and outside humidity$^1$</td>
</tr>
<tr>
<td></td>
<td>$q_l = 3000 Q \Delta w$ $Q$ is in L/s and $q_l$ is in watts</td>
</tr>
</tbody>
</table>

The infiltration flow rates can be estimated using the CRACK METHOD or AIR CHANGE METHOD as shown below

#### AIR CHANGE METHOD
Recommended values range from 0.5 to 1.5 air changes per hour. One air change per hour would be a volumetric flow rate numerically equal to the internal volume of the space. Smaller values for well fitted windows or doors and higher values for loose fittings.

#### CRACK METHOD
Recommended allowed design infiltration rates through exterior windows and doors

<table>
<thead>
<tr>
<th>Item</th>
<th>Infiltration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Windows</td>
<td>0.5 CFM/ft sash crack</td>
</tr>
<tr>
<td>2. Sliding Glass Door (Residential)</td>
<td>0.5 CFM/ ft^2 door area</td>
</tr>
<tr>
<td>3. Swinging Door (Residential)</td>
<td>1.0 CFM/ft^2 door area</td>
</tr>
<tr>
<td>4. Sliding, swinging or revolving doors</td>
<td>11.0 CFM/ft door area</td>
</tr>
</tbody>
</table>

$^1$ these values are given in ASHRAE handbook of Fundamentals
### Table 2 Summary of Calculating Space Design Cooling Load

<table>
<thead>
<tr>
<th>Cooling Load Component</th>
<th>Equations</th>
</tr>
</thead>
</table>
| **Roofs**              | \( q = U A (CLTD_e) \) where: \( U \) is heat transfer coef, Btu/hr.ft\(^2\).\(^o\)F  
\[ CLTD_e = ((CLTD+LM)K + (78-t_i)+(t_om-85)) \]  
where,  
A is area calculated from building plans, ft\(^2\)  
\( q \) is in Btu/hr  
\( t_i \) inside design temperature, \(^o\)F  
CLTD cooling load temperature difference, \(^o\)F  
LM color correction for latitude and month  
K color adjustment factor (k=1 for dark surface)  
\( T_m \) outdoor mean temperature = \( t_0 - DR/2 \)  
\( t_o \) outside design temperature, \(^o\)F  
DR is the daily range |
| **Walls**              | \( q = U A (CLTD_e) \) where: \( U \) is heat transfer coef, Btu/hr.ft\(^2\).\(^o\)F  
\[ CLTD_e = ((CLTD+LM)K + (78-t_i)+(t_om-85)) \]  
where,  
A is area calculated from building plans, ft\(^2\)  
\( q \) is in Btu/hr  
CLTD cooling load temperature difference, \(^o\)F  
for the wall obtained from a table\(^1\) |
| **Glass**              | **Conduction** \( q = U A (CLTD_e) \) where: \( U \) is heat transfer coef, Btu/hr.ft\(^2\).\(^o\)F  
\[ CLTD_e = ((CLTD+LM)K + (78-t_i)+(t_om-85)) \]  
where,  
A is area calculated from building plans, ft\(^2\)  
\( q \) is in Btu/hr  
CLTD cooling load temperature difference, \(^o\)F  
for the glass\(^1\)  
\( q \) = \( A \cdot SC \cdot SHGF \cdot CLF \)  
\( A \) area of the glass window, ft\(^2\)  
SC shading coefficient\(^1\)  
SHGF maximum solar heat gain factor\(^1\), Btu/hr. ft\(^2\)  
CLF cooling load factor\(^1\) for solar heat gain |
| **Glass**              | **Solar** \( q = U A (CLTD_e) \) where: \( U \) is heat transfer coef, Btu/hr.ft\(^2\).\(^o\)F  
\[ CLTD_e = ((CLTD+LM)K + (78-t_i)+(t_om-85)) \]  
where,  
A is area calculated from building plans, ft\(^2\)  
\( q \) is in Btu/hr  
CLTD cooling load temperature difference, \(^o\)F  
for the glass\(^1\)  
\( q \) = \( A \cdot SC \cdot SHGF \cdot CLF \)  
\( A \) area of the glass window, ft\(^2\)  
SC shading coefficient\(^1\)  
SHGF maximum solar heat gain factor\(^1\), Btu/hr. ft\(^2\)  
CLF cooling load factor\(^1\) for solar heat gain |
| **People**             | Sensible  
\( q_s = No. Sens H.G. CLF \)  
where: \( No \) number of people  
Sens H.G sensible heat gain\(^1\), Btu/hr  
CLF cooling load factor\(^1\) for people |
| **People**             | Latent  
\( q_l = No. Lat H.G \)  
Lat H.G latent heat gain\(^1\), Btu/hr |
| **Internal Lights**    | \( q = 1.2 \cdot INPUT \cdot CLF \)  
INPUT input ratings from light fixtures\(^1\)  
CLF cooling factor\(^1\) for light fixtures |
| **Appliances**         | Sensible  
\( q_s = Heat Gain_s \cdot CLF \)  
Heat Gain\(_s\) recommended rate of heat gain\(^1\), Btu/hr  
CLF cooling factor\(^1\) for appliances |
| **Appliances**         | Latent  
\( q_l = Heat Gain_l \)  
Heat Gain\(_l\) recommended rate of heat gain\(^1\) |
| **Infiltration and Ventilation air.** | Sensible  
\( q_s = 1.1 Q \Delta t \)  
\( \Delta t = t_o - t_i \), \(^o\)F, \( q_s \) is in Btu/hr  
\( Q \) = volume flow rate (CFM)  
\( q_s = 1.23 Q \Delta t \)  
\( \Delta t \) in \(^\circ\)C, \( q_s \) is in watts, \( Q \) is in L/s  
\( q_l = 4840 Q \Delta w \)  
\( \Delta w = w_o - w_i \), \( Q \) is in BTU/hr, \( q_l \) is in CFM  
\( q_l = 3000 Q \Delta w \)  
\( Q \) is in L/s and \( q_l \) is in watts |
| **Infiltration and Ventilation air.** | Latent  
\( q_s = 1.1 Q \Delta t \)  
\( \Delta t \) in \(^\circ\)C, \( q_s \) is in watts, \( Q \) is in L/s  
\( q_l = 4840 Q \Delta w \)  
\( \Delta w = w_o - w_i \), \( Q \) is in BTU/hr, \( q_l \) is in CFM  
\( q_l = 3000 Q \Delta w \)  
\( Q \) is in L/s and \( q_l \) is in watts |
| **Infiltration and Ventilation air.** |  
The infiltration flow rates can be estimated using the CRACK METHOD or AIR CHANGE METHOD as presented in the Table III |
| **Space Cooling Load** | is equal to sum of all of the above, \( q = \sum q_i \) |

\(^1\) these values are given ASHRAE handbook of Fundamentals, 1997
Solar Energy

The earth rotates about its own axis causing day and nights and simultaneously revolving in an elliptical orbit at a mean radius of about 149.6 million kilometers around the sun causing the seasons as shown in the Figure 1. As seen from this figure, the summer solstice, the day of the longest daylight occurs about June 21, while the winter solstice, the day of the longest night occurs about December 21 for northern latitudes. The equinox characterized by a day of equal lengths of daylight and nights occurs about March 23 and September 21. Thus the amount of solar energy received by a surface on the earth varies with time during the day, as

**Figure 1** Transient Nature of Solar Energy
well as with the season and latitude of the location. The solar noon is the time of the day at which the rays from the sun are normal to the longitude of the location. The gases, dust particles and clouds present in the atmosphere scatter the beam radiation originating from the Sun. The scattered energy often called diffuse energy exists even on cloudy days. The actual energy reaching a surface on the earth is the sum of the beam (or also known as direct radiation) and diffuse radiation. The magnitude of diffuse energy reaching the surface on the earth is at the minimum on a clear sky conditions, while the energy is 100 percent diffuse on a day of fully cloudy conditions.

The average rate of solar energy also known as extra terrestrial irradiance or insolation striking a surface directed normal to the solar beam outside the earth’s atmosphere, is termed as Solar Constant and its value is given as 1373 W/m$^2$ or 435.2 Btu/hr.ft$^2$. The extra terrestrial irradiance is all beam radiation. However, due to slight eccentricity in the earth’s orbit, the actual value of extra terrestrial irradiance is related as

$$I_o \left( \frac{Btu}{hr. ft^2} \right) = 435.2 \left[ 1 + 0.033 \cos \left( \frac{360^\circ n}{365.25} \right) \right]$$  \hspace{1cm} (1)

$$I_o \left( \frac{W}{m^2} \right) = 1373 \left[ 1 + 0.033 \cos \left( \frac{360^\circ n}{365.25} \right) \right]$$  \hspace{1cm} (2)

where, n is the day of the year (= 1 for January 1).

Estimation of Total Incident Solar Irradiance on a Surface on the Earth

Due to the rotation of the earth about its own axis as well around the Sun, the estimate of incident solar irradiance, $I_t$ consisting of beam and diffuse components involves determining various solar angles, namely:

Declination angle ($\delta$) is the angle made by the equator with sun’s rays as shown in Figure 2b
Surface latitude angle ($\lambda$) is the angle between the radius vector of the location from the center of the earth and the equatorial plane indicated in the Figure 2a.

Hour angle ($\omega$) is the angle between the meridian plane of the sun’s rays with the local meridian at the center of the earth in an equatorial plane as shown in Figure 2a.

Solar azimuth angle ($\Phi_s$) is the angle between the projection of the sun’s rays on a local horizontal plane and the south direction as shown in Figure 3a.

Solar zenith angle ($\theta_s$) is the angle between the sun’s rays and the normal on the local horizontal plane as shown in Figure 3a.

Surface azimuth angle ($\Phi_p$) is the angle between the normal to the surface with the south direction as shown in Figure 3b.

Surface tilt angle ($\theta_p$) is the angle between the surface and the local horizontal as shown in Figure 3b.

Solar incident angle ($\theta_i$) is the angle between the normal to the surface with sun’s rays as shown in Figure 3b.
(a) Three-Dimensional View  
(b) Cross-Sectional View at Solar Noon

Figure 2 Illustration of Latitude, hour angle and solar declination

The declination angle ($\delta$) can be given as,

$$\sin \delta = -\sin 23.45^0 \cos \frac{360^0 (n + 10)}{365.25}$$

where, $n$ is the day of the year with January 1 being $n = 1$.

Figure 3. Solar Angles for a Tilted Surface

The hour angle ($\omega$) can be estimated in terms of solar time ($t_{sol}$) from,

$$\omega = \frac{360^0 (t_{sol} - 12h)}{24h}$$
The solar time, $t_{sol}$ is related to the local standard time, $t_{std}$ as

$$ t_{sol} = t_{std} + \frac{L_{std} - L_{loc}}{15^\circ / hr} + \frac{E_t}{60 \text{ min} / hr} \tag{5} $$

where $t_{std}$ = local standard time

$L_{std}$ = longitude of the standard time, for United States, Eastern = 75°, Central = 90°, Mountain = 105°, Pacific = 120°.

$L_{loc}$ = longitude of the location in degrees.

$E_t$ = equation of time is the difference between the solar noon and noon time based on local Time and it varies over the year.

It may be noted that solar noon refers to the time when sun reaches the highest point in the sky. The equation of time $E_t$ is obtained from

$$ E_t = 9.87 \sin \left( \frac{360^\circ (n-81)}{364} \right) - 7.53 \cos \left( \frac{360^\circ (n-81)}{364} \right) - 1.5 \sin \left( \frac{360^\circ (n-81)}{364} \right) \tag{6} $$

The solar zenith angle ($\theta_s$) as shown in the Figure 4 can be estimated from,

$$ \cos \theta_s = \cos \lambda \cos \delta \cos \omega + \sin \lambda \sin \delta \tag{7} $$

Now, the solar azimuth angle ($\phi_s$) in terms of solar zenith angle ($\theta_s$) is obtained as follows

$$ \phi_s = \frac{\cos \delta \sin \omega}{\sin \theta_s} \tag{8} $$

Finally, the solar incident angle ($\theta_i$) is given by

$$ \cos \theta_i = \sin \theta_s \sin \theta_p \cos (\phi_s - \phi_p) + \cos \theta_s \cos \theta_p \tag{9} $$
where, $\theta_s$ is the solar azimuth angle given by Equation (8) while the surface azimuth angle $\theta_p$ as shown in the Figure 2 is the angle made by the surface normal with the south direction. The tilt angle of the surface $\theta_p$ is the angle of inclination of the surface with local horizontal surface as shown in Figure 3.

Now the total incident solar load $I_t$ is the sum of

(i) the solar direct radiation ($I_{dir}$) incident normal to the surface

(ii) the solar diffuse radiation ($I_{dir}$), the diffuse radiation is the radiation scattered from the surroundings and the dust particles present in the atmosphere.

(iii) the solar radiation reflected from the ground

\[
I_t = I_{dir} \cos \theta_s + I_{dif,hor} \frac{1 + \cos \theta_p}{2} + I_{glo,hor} \rho_g \frac{1 - \cos \theta_p}{2} \quad (10)
\]

where, $I_{glo,hor}$ is the global horizontal radiation incident on the horizontal surface.

The weather stations in various major cities record hourly data consisting of $I_{dir}$, $I_{dif,hor}$, $I_{glo,hor}$, the ambient air temperature, the dew point temperature, the relative humidity, wind speed and direction, cloud cover factor and many other data. The meteorologists obtained the average of 25 to 30 years of such data and designated these data as the
typical meteorological year (TMY) for that city. Use of such data allows a more detailed and accurate estimate of solar loads.

**Estimation of Solar Irradiance from Approximate Models**

The meteorological data are being collected for various major towns in the United States and around the world. The collection of such data grew rapidly for many other towns, when the fuel cost escalated during the mid 70’s. In locations where such data are not measured, attempts have been made to estimate the solar data from approximate models related to extra terrestrial solar irradiance and variables such as percentage of sunshine or clearness index, and visibility. The following equations outline such procedures.

\[
\frac{I_{\text{diff}}}{I_{\text{glo}}} = \begin{cases} 
1 - 0.09 K_T & \text{if } K_T \leq 0.22 \\
0.951 - 0.1604 K_T + 4.388 K_T^2 - 16.638 K_T^3 + 12.336 K_T^4 & \text{if } 0.22 < K_T \leq 0.8 \\
0.165 & \text{if } K_T > 0.8 
\end{cases}
\]  

(11)

where \(K_T\) = hourly clearness index = \(I_{\text{glo}} / I_o \cos \theta_s\)

**Clear-Day Model**

The direct or beam diffusion, \(I_{\text{dir}}\) during a clear sky radiation is given by

\[
I_{\text{dir}} = I_o \left[ a_o + a_1 \exp \left( - \frac{k}{\cos \theta_s} \right) \right]
\]

(12)

where \(a_o\), \(a_1\) and \(k\) are constants as listed in Table 1.

The constants \(a_o\), \(a_1\) and \(k\) are dependent upon the visibility, the altitude and latitude of the location as seen from the Table 1.

The diffuse irradiance \(I_{\text{diff,hor}}\) is obtained from

\[
I_{\text{diff,hor}} = (0.271 I_o - 0.2939 I_{\text{dir}})
\]

(13)
Solar Heat Gain Factors

The energy interaction across a glass window of a building consists of reflection, absorption, and transmission of incoming solar energy and reflected energy from the ground, as well as heat conduction due to temperature difference between the outside and indoor air as shown in Figure 6.

Table 1 Coefficients of Clear-Day Model [72]

(a) The coefficients $a_0$, $a_1$, and $k$ are expressed as a function of altitude, $A$ above sea level (in km), for two levels of visibility

<table>
<thead>
<tr>
<th></th>
<th>23 km (14.3 mi) visibility</th>
<th>5 km (3.11 mi) visibility</th>
</tr>
</thead>
<tbody>
<tr>
<td>$a_0$</td>
<td>$r_o [0.4237 - 0.00821 (6.0 - A)^2]$</td>
<td>$r_o [0.2538 - 0.0063 (6.0 - A)^2]$</td>
</tr>
<tr>
<td>$a_1$</td>
<td>$r_1 [0.5055 + 0.00595 (6.5 - A)^2]$</td>
<td>$r_1 [0.7678 + 0.0010 (6.5 - A)^2]$</td>
</tr>
<tr>
<td>$k$</td>
<td>$r_k [0.2711 + 0.01858 (2.5 - A)^2]$</td>
<td>$r_k [0.2490 + 0.0810 (2.5 - A)^2]$</td>
</tr>
</tbody>
</table>

(b) The correction factors $r_o$, $r_1$, and $r_k$ are given as

<table>
<thead>
<tr>
<th>Climate Type</th>
<th>$r_o$</th>
<th>$r_1$</th>
<th>$r_k$</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>23 km</td>
<td>5 km</td>
<td></td>
</tr>
<tr>
<td>Tropical</td>
<td>0.95</td>
<td>0.92</td>
<td>0.98</td>
</tr>
<tr>
<td>Mid-latitude summer</td>
<td>0.97</td>
<td>0.96</td>
<td>0.99</td>
</tr>
<tr>
<td>Sub-arctic summer</td>
<td>0.99</td>
<td>0.98</td>
<td>0.99</td>
</tr>
<tr>
<td>Mid-latitude Winter</td>
<td>1.03</td>
<td>1.04</td>
<td>1.01</td>
</tr>
</tbody>
</table>
It is also indicated that a fraction of the absorbed energy in the glass is reradiated back to the indoor air.

The radiational properties of absorptivitty ($\alpha$) and transmittivty ($\tau$) of a double-strength sheet (DSA) are often employed as a standard in determining the fractions of energy absorbed and transmitted. The actual values of the energy interactions of a given glass sheet are adjusted by multiplying the values of a DSA glass with a shading coefficient. The direct absorption ($\alpha_D$) and transmission ($\tau_D$) coefficients of a DSA glass are related to the incident angle $\theta$, and are given as

$$\alpha_D = \sum_{j=0}^{5} a_j [\cos \theta]^j$$  \hspace{1cm} (14a) 

$$\tau_D = \sum_{j=0}^{5} t_j [\cos \theta]^j$$  \hspace{1cm} (14b)
The diffuse absorption ($\alpha_d$) and transmission ($\tau_d$) coefficients of a DSA glass are also related to the incident angle $\theta$, and are given as

\[
\alpha_d = 2 \sum_{j=0}^{5} a_j /[j + 2]
\] (15a)

\[
\tau_d = 2 \sum_{j=0}^{5} t_j /[j + 2]
\] (15b)

The values of coefficients $a_j$, and $t_j$ are as listed in the Table 2.

The direct irradiance striking the window, $I_D$ is given as,

\[
I_D = I_{dir} \cos \theta \quad \text{if } \cos \theta > 0; \quad \text{otherwise } I_D = 0
\] (16)

The ratio, $Y$, of the sky diffuse irradiance to that on the horizontal surface is given as

\[
Y = 0.55 + 0.437 \cos \theta + 0.313 \cos^2 \theta \quad \text{for } \cos \theta > -0.2;
\]

\[
\text{otherwise } Y = 0.45
\] (17)

**Table 2** The Coefficients for DSA Glass for Calculation of Absorbtance and Transmittance

<table>
<thead>
<tr>
<th>$j$</th>
<th>$a_j$</th>
<th>$t_j$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.01154</td>
<td>-0.00885</td>
</tr>
<tr>
<td>1</td>
<td>0.77674</td>
<td>2.71235</td>
</tr>
<tr>
<td>2</td>
<td>-3.94657</td>
<td>-0.62062</td>
</tr>
<tr>
<td>3</td>
<td>8.57881</td>
<td>7.07329</td>
</tr>
<tr>
<td>4</td>
<td>-8.38135</td>
<td>9.75995</td>
</tr>
<tr>
<td>5</td>
<td>3.01188</td>
<td>-3.89922</td>
</tr>
</tbody>
</table>
The diffuse radiation, \( I_{\text{dif}} \) = diffuse sky irradiance, \( I_{\text{ds}} \) + diffuse ground reflected irradiance, \( I_{\text{dg}} \)

For vertical surfaces, \( I_{\text{ds}} = Y I_{\text{dif,hor}} \) (18)

For other surfaces, \( I_{\text{ds}} = I_{\text{dif,hor}} (1 + \cos \theta_p) / 2 \) (19)

Diffuse ground reflected irradiance, \( I_{\text{dg}} = \{I_{\text{dif,hor}} + I_{\text{dir}} \cos \theta_s \} \rho_g [1 - \cos \theta_p] / 2 \) (20)

where \( \rho_g \) is the reflectance of the ground normally assumed to be 0.2.

The transmitted component of solar energy, \( I_{\text{tra}} = \tau_D I_D + 2 \tau_d \{ I_{\text{ds}} + I_{\text{dg}} \} \) (21)

The absorbed component of solar energy, \( I_{\text{abs}} = \alpha_D I_D + 2 \alpha_d \{ I_{\text{ds}} + I_{\text{dg}} \} \) (22)

The hourly solar heat gain per unit area of a DSA glass is known as Solar Heat Gain Factor, SHGF and is given as

\[
\text{SHGF} = I_{\text{tra}} + N_i I_{\text{abs}}
\] (23)

where \( N_i \) represents the fraction of absorbed energy radiated back indoors and is given by

\[
N_i = \frac{h_i}{h_i + h_o}
\] (24)

where \( h_i \) and \( h_o \) represent heat transfer coefficients at the inner and outer surfaces of the glass window, respectively.
Evaluation of Rate of Heat Transfer Based on Transfer Function Method [72]

The following example illustrates the use of transfer function method to evaluate the rate of heat transfer from a wall exposed to the solar radiation. Consider a vertical wall, of dark color and facing west, consisting of 0.10 m (4 in) concrete with 0.05-m (2-in) insulation on the outside with sol-air temperatures as given in the second column of Table 3 for summer design conditions (July 2) for $T_i = 25^0$ C. The transfer function coefficients for the wall as given in the ASHRAE Handbook of fundamentals [73] are as follows:

\[
\begin{align*}
    b_0 &= 0.00312 & \sum c_n &= 0.0734 & d_0 &= 1.0000 \\
    b_1 &= 0.04173 & d_1 &= -0.94420 \\
    b_2 &= 0.02736 & d_2 &= 0.05025 \\
    b_3 &= 0.00119 & d_3 &= -0.00008
\end{align*}
\]

The d’s are dimensionless, and b’s are in W/m$^2$.K. All other coefficients are zero, and the U value is 0.693 W/m$^2$.K.

For time $t < 0$, it is assumed that the heat transfer to the wall $Q_{\text{cond}, t} = 0$.

From Equation (28) of Chapter IV,

\[
\dot{Q}_{\text{cond}, t} = -\sum_{n=1}^{\infty} d_n \dot{Q}_{\text{cond}, t-n\Delta t} + A \left( \sum_{n=0}^{\infty} b_n T_{\text{os}, t-n\Delta t} - T_i \sum_{n=0}^{\infty} c_n \right)
\]

For unit area of the wall with area $A = 1$ m$^2$, for the time $t = 1$ hr, the above Equation reduces to

\[
Q_{\text{cond}, 1} = -d_1 Q_{\text{cond}, 1-1} - d_2 Q_{\text{cond}, 1-2} - d_3 Q_{\text{cond}, 1-3} + b_0 T_{\text{os}, 1-0} + b_1 T_{\text{os}, 1-1} + b_2 T_{\text{os}, 1-2} + b_3 T_{\text{os}, 1-3} - T_i \sum c_n
\]

After Substituting the known values into the above equation gives

\[
Q_{\text{cond}, 1} = -(-0.94420) (0.00) - 0.05025 (0.00) - (-0.00008) (0.00) + 0.00312 (24.4)
\]
\[ + 0.04173 \times 25.0 + 0.02736 \times 26.1 + 0.00119 \times 27.2 - 2.0 \times 0.0734 \]

\[ Q_{\text{cond}, 1} = 0.03 \text{ W/m}^2. \]

The above process of calculation for \( Q_{\text{cond}, 1} \) is repeated for four or five days, the value appears to be stabilized by the fourth day.

It may be noted that the conductive heat gain reaches maximum at \( t = 19.00 \) hrs, which is 3 hours after the peak of the sol-air temperature indicating the effect of thermal storage.

**Table 3 Iterative Process of Transfer Function Method in Calculation of Heat Gain through a Wall**

<table>
<thead>
<tr>
<th>T</th>
<th>( T_{\text{os}, t} )</th>
<th>( Q_{\text{cond}, t} )</th>
<th>( Q_{\text{cond}, t + 24} )</th>
<th>( Q_{\text{cond}, t + 48} )</th>
<th>( Q_{\text{cond}, t + 72} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hr</td>
<td>(^\circ\text{C})</td>
<td>W/m(^2)</td>
<td>W/m(^2)</td>
<td>W/m(^2)</td>
<td>W/m(^2)</td>
</tr>
<tr>
<td>-2</td>
<td>27.2</td>
<td>0.00</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-1</td>
<td>26.1</td>
<td>0.00</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>25.0</td>
<td>0.00</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>24.4</td>
<td>0.03</td>
<td>9.29</td>
<td>9.82</td>
<td>9.85</td>
</tr>
<tr>
<td>2</td>
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<td>8.22</td>
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<td>8.72</td>
</tr>
<tr>
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<td>23.8</td>
<td>-0.04</td>
<td>7.25</td>
<td>7.67</td>
<td>7.70</td>
</tr>
<tr>
<td>4</td>
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<tr>
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<td>4.48</td>
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<tr>
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<td>-0.17</td>
<td>3.85</td>
<td>4.08</td>
<td>4.10</td>
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</tr>
<tr>
<td>10</td>
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<td>3.83</td>
<td>4.01</td>
<td>4.02</td>
</tr>
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<td>11</td>
<td>37.7</td>
<td>1.29</td>
<td>4.10</td>
<td>4.26</td>
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</tr>
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<td>12</td>
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<td>5.29</td>
</tr>
<tr>
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<td>6.48</td>
<td>6.49</td>
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<td>15</td>
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<td>8.44</td>
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<td>16</td>
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<td>9.26</td>
<td>10.81</td>
<td>10.9</td>
<td>10.91</td>
</tr>
<tr>
<td>17</td>
<td>72.2</td>
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<td>13.4</td>
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<td>18</td>
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<td>15.69</td>
<td>15.69</td>
</tr>
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<td>19</td>
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<td>13.64</td>
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<td>12.28</td>
<td>12.32</td>
<td>12.32</td>
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<td>10.42</td>
<td>11.02</td>
<td>11.05</td>
<td>11.06</td>
</tr>
</tbody>
</table>
APPENDIX 3

Double-Effect Absorption Chillers

Single-effect absorption technology provides a peak cooling COP of approximately 0.7 and operates with heat input temperatures in the range 190°F to 250°F. When higher temperature heat sources are available, absorption technology can be reconfigured to achieve higher COP. One such configuration is a parallel flow double-effect design as shown schematically in Figure 19 and 20. By adding an additional stage as a topping cycle on a single effect cycle, the machine does a cascade in which the heat rejection from the high temperature stage fires the lower temperature stage. The results, in practice, is cooling COP values in the range of 1.0 to 1.2, depending on the design and application. Double-effect calculations can be performed in a manner similar to that illustrated for the single effect cycle. Mass and energy balances of the model shown in Figure were calculated using the inputs and assumptions listed in Table [ ]. The results are shown in Table [ ]. Note that the COP value obtained is high compared to what is obtained in practice.

The COP is quite sensitive to several inputs and assumptions. In particular, the effectiveness of the solution heat exchangers and the driving temperature difference between the high temperature condenser and low temperature generator are two parameters that influence the COP strongly.
Figure 1 Double Effect Absorption Refrigeration

Inputs and Assumptions for Double-Effect Lithium Bromide/Water Model

<table>
<thead>
<tr>
<th>Inputs</th>
<th>Assumptions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacity $Q_c$</td>
<td>Steady State</td>
</tr>
<tr>
<td>500 tons refig</td>
<td>Refrigerant is pure water</td>
</tr>
<tr>
<td>Evaporator temperature $t_{10}$</td>
<td></td>
</tr>
<tr>
<td>41.1 °F</td>
<td></td>
</tr>
<tr>
<td>Desorber solution exit temperature $t_{14}$</td>
<td></td>
</tr>
<tr>
<td>339.3 °F</td>
<td></td>
</tr>
<tr>
<td>Condenser/absorber low temperature $t_1 = t_8$</td>
<td></td>
</tr>
<tr>
<td>108.3 °F</td>
<td></td>
</tr>
<tr>
<td>Solution heat exchanger effectiveness $\varepsilon$</td>
<td></td>
</tr>
<tr>
<td>0.6</td>
<td></td>
</tr>
</tbody>
</table>
No pressure changes except through the flow restrictors and the pump

States at points 1, 4, 8, 11, 14 and 18 are saturated

States at point 10 is saturated vapour

Temperature difference between high temperature condenser and low temperature generator 9°F

Parallel Flow

Both solution heat exchangers have same effectiveness

Upper loop solution flow rate is selected such that the upper condenser heat exactly matches the lower generator heat requirement

Flow restrictors are adiabatic

Pumps are isentropic

No jackets heat loses

No liquids carryover from evaporator to absorber

Vapor leaving both generators is at the equilibrium temperature
Table 1 State Point Data for Double Effect Lithium Bromide/Water Cycle

<table>
<thead>
<tr>
<th>No</th>
<th>$h$ Btu/hr</th>
<th>$m$ lb/min</th>
<th>$p$ psia</th>
<th>$Q$ Fraction</th>
<th>$t$ F</th>
<th>$x$ %LiBr</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>50.6</td>
<td>1263.4</td>
<td>0.13</td>
<td>0.0</td>
<td>108.3</td>
<td>-</td>
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<td>50.6</td>
<td>1263.4</td>
<td>1.21</td>
<td>108.3</td>
<td>59.5</td>
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<tr>
<td>3</td>
<td>78.3</td>
<td>1263.4</td>
<td>1.21</td>
<td>168.1</td>
<td>59.5</td>
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<td>1163.7</td>
<td>1.21</td>
<td>0.0</td>
<td>208.0</td>
<td>64.6</td>
</tr>
<tr>
<td>5</td>
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<td>137.9</td>
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<td>8</td>
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<tr>
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<td>99.8</td>
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<td>186.2</td>
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<td>86.7</td>
<td>727.3</td>
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<tr>
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<td>1.21</td>
<td>0.105</td>
<td>108.3</td>
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</tr>
</tbody>
</table>

$COP_c = 1.195$

$Q_e = 6 \times 10^6$ Btu/h

$\Delta t = 9.0^\circ F$

$Q_{gh} = 5.019 \times 10^6$ Btu/h
\[ \varepsilon = 0.600 \]

\[ Q_a = 7.936 \times 10^6 \text{ Btu/hr} \]

\[ Q_{sg} = 3.488 \times 10^6 \text{ Btu/hr} \]

\[ Q_c = 3.085 \times 10^6 \text{ Btu/hr} \]

\[ Q_{shx1} = 2.103 \times 10^6 \text{ Btu/hr} \]

\[ Q_{shx2} = 1.862 \times 10^6 \text{ Btu/hr} \]

\[ W_{p1} = 0.032 \text{ hp} \]

\[ W_{p2} = 0.2589 \text{ hp} \]