International Journal of Engineering Research and Applications (IJERA) ISSN: 2248-9622
International Conference on Humming Bird (01st March 2014)

RESEARCH ARTICLE OPEN ACCESS

Design of Tri-Generation for a Hotel

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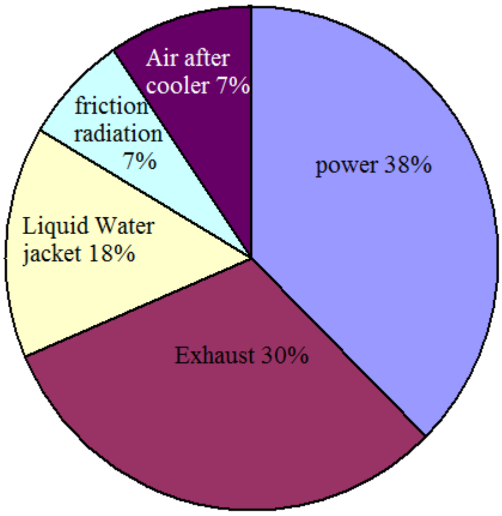
Abstract
Trigeneration is defined as the production of three useful forms of energy—heat, cold and power—from a primary source of energy such as natural gas or oil. The heat produced can be totally or partially used to fuel absorption refrigerators. Therefore, trigeneration systems enjoy an inherently high efficiency and have the potential to significantly reduce the energy-related operation costs of facilities.
Availability of Electricity is a big problem in India. Hotels and big buildings should have their own power generator system rather than paying for commercial electricity to power plants. These buildings can afford these systems. They usually have their own Diesel Generator (DG) which is used as a secondarypower source during power cuts. Here our proposal is to use 'Trigenerative System' as a primary source for our building and main supply power from grid as a standby.

I. Introduction

Nowadays, hotels take their electricity from the grid for fulfilling their requirement
like lighting, cooling and heating and for purposes. This electricity generally comes from thermal power plant. In thermal power stations, mechanical power is produced by a heat engine that transforms thermal energy, often from combustion of a fuel, into rotational energy. Most thermal power stations produce steam, and these are sometimes called steam power stations. Not all thermal energy can be transformed into mechanical power, according to the second law of thermodynamics. Therefore, there is always heat lost to the environment. Tones of coal are burnt to produce electricity to meet our demand. The flue gas from combustion of the fossil fuels is discharged to the air; this contains carbon dioxide and water vapour, as well as other substances such as nitrogen , nitrogen oxides, sulfur oxides, and (in the case of coal-fired plants) fly ash, mercury and traces of other metals. Solid waste ash from coal-fired boilers must also be removed. Some coal ash can be recycled for building materials. Fossil fuel powered stations are major emitters of greenhouse gases (GHG) which according to the consensus of scientific organisations are a major contributor to the global warming observed over the last 100 years. Brown coal emits 3 times as much GHG as natural gas, black coal emits twice as much per unit of electric energy.

Reciprocating engine rejects a considerable amount of energy to the ambience through the exhaust gas. Significant reduction of engine fuel consumption could be attained by recovery of exhaust heat.



Diesel engines reject a considerable amount of energy into the ambient atmosphere in the form of high temperature exhaust gases. Significant reduction of engine brake specific fuel consumption (bsfc) could be attained by recovering a significant part of exhaust gas heat.30-40% of heat generated in the process of fuel combustion into useful mechanical work. The remaining heat is emitted to the environment through the exhaust gases and the engine cooling systems. Therefore, even partial recovery of the waste heat would allow a significant increase of the overall combustion engine performance. The combination of absorption refrigeration with a trigeneration plant allows using all generated heat for the production of cooling and heating in summer and winter months of the year.

Heat from the jacket-water and from the exhaust can be extracted with the use of efficient heat recovery system to provide hot water to run the vapour absorption system in summer and hot air blower in winter. This way we can utilize the waste heat and overall efficiency can be increased up to 85%.

II. Data collection and Hotel survey:

a) Load of the hotel:

For our design of tri-generative system we have chosen a U.P tourism "Hotel Ilawart" , a a government hotel, located in Civil Line in Allahabad. We took specification of all the equipment used there and calculated maximum peak demand of electricity(inclusive of lighting, computers, water pumps etc.) , demand of cooling load (mainly in air-conditioning the rooms), demand of heating load(in Domestic hot water (DHW) and Geysers) .

Peak electricity demand = 23.7 kW
Peak heating load = 120.0 kW
Peak cooling load = 104.25 kW

Electricity bill of June month 2010 was Rs. 160,587.00 and unit consumed was 31209 units @ 4.1 Rs. Record shows that average bill varies between Rs. 1.5 Lakhs – 1.75 Lakhs monthly.

Roof area which is available is 2721.6 m^2 can be utilized for Solar Hot System.

b) Standby Diesel Generator reading:

Hotel has 3 Diesel Generator set to fulfill the requirement of hotel in case of power failure.

Power output	Avg. fuel consumption
20 kW	4 l/hour
63 kW	10 l/hour
125 kW	18 l/hour

III. Model Generation:

Using the '*design builder'* software, an approximate model of the hotel was generated. The specifics incorporated for generation of the model included RCC roof of 1.5% steel as per the Indian trend , dimensions of brick wall are 25.4cm (23cm (brick) +12mm +12mm (cement on both sides). The windows are Aluminum framed 3mm clear glass thickened, the projection over windows being 0.5m.

The design temperatures of the rooms were taken as 25 deg. C in summer and 18 deg. C ,in winter. The temperature of 28 deg. C was not taken although it would have less electrical load as some foreigners prefer a temperature of 25 deg. C and in the winter as we generally wear woolen clothes 18 deg. C was taken as the design temp.

We also observed that the average occupancy at any time of the year was 60% and the rooms occupancy were not occupied from 10:00 am to 3:00 pm during due to sightseeing by the tourists

These are the output obtained from the software for month wise energy consumption

IV. Comparison :

Total electrical load of the building in June is 33036 kWh from the model and the actual consumption in June was 31209 kWh. (Units) that are comparatively close.

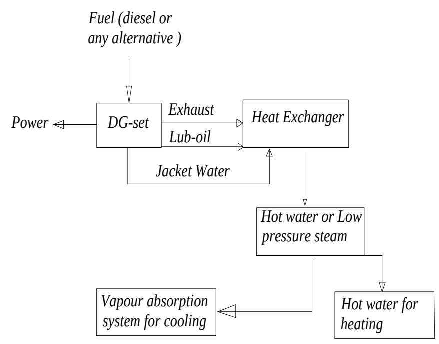
So we can conclude that model generated from the software it quite satisfactory.

And we can use it for further study.

Design of Trigeneration system for hotel:

For design of trigeneration main points are follows

1. Selection of power generator to fulfill electricity requirement
2. Design of proper heat exchanger to extract heat from exhaust and from jacket water used to cool the engine.
3. From extracted heat we can provide heat to VAM to provide cooling and to hot air blower for heating or domestic heating purpose as required.



V. Selection of Diesel Generator

The total power consumption of June is highest and equals 33,040 units.

- The electricity load is 15216.15 units of the month =21.13kW(avg)
- The heating load for domestic hot water is 411.19 units of June = 0.57 kW (avg).
- Cooling load is 17382.62 units = 24.142 kW(avg)

Taking COP of VAM as 0.7 and efficiency of the heat exchanger 66.7% (2/3 of waste heat) , heat required for VAM is 51.7 kW to fulfill the cooling requirement. Heat and power ratio remains from 1.0-1.5 of DG-set as efficiency varies from 35-42%. Then selection a DG-set with rating 50 kW with 40% efficiency and Air required is 15 times of fuel input i.e. Air-fuel ratio is 15.

1) For recovering heat from jacket water we have designed a Shell and tube type heat exchanger.

2) Then water from shell and tube heat exchanger goes to Gas-liquid heat exchanger to gain heat from exhaust gases.

1. Shell-and-Tube Heat Exchangers :

Shell and tube heat exchangers are built of round tubes mounted in cylindrical shells with their axes parallel to that of the shell.

a)- Flow configuration

A uniform spacing between the tubes, so that the flow and heat transfer distribution through the tube bundle are reasonably uniform.

(b)Header-Sheet Hole Patterns

The three principal tube arrays employed in shell-and–tube heat exchangers, namely, equilateral triangular, square, and staggered square.

The equilateral triangular arrangements gives the strongest header sheet for a given shell-side flow passage area, whereas the square arrangements simplifies some fabrication and some maintenance operations.

(c)Estimate of number of Tubes Required

The total number of tubes n is dependent on tube-side flow conditions. It is related to the tube length and diameter together with the allowable pressure drop and the total tube-side flow rate.

(d) Relation from Heat Balance Considerations:

These the shell-side fluid mass flow rate across the tube matrix, the tube length N2, the number of passes for the cross flow fluid. Three independent equations are required to solve for these three unknowns. One such relation can be obtain by expressing the total no of heat added to or taken from the shell side fluid in the terms of the product of the cross passage area per pass, the shell side fluid mass flow rate per unit of flow passage area, and the temperature rise or drop in the shell side fluid.

Parameter for Shell-and-tube heat exchanger:

- Flow configuration : Two-pass tube side multiplicity of passes shell side
- Matrix geometry : Equilateral triangular pitch
- Tube size : 0.25-in OD , 0.028-in thick, internal flow area 0.0295-in2
  - Tube spacing : 1.25*tube OD
  - Tube material : Admiralty (conductance = 112.7 W/m2*C)

Design Calculations

Quantity	
N1, number of tube-side passes	2
n, total number of tubes	32
d_s, shell diameter	0.458 Cm
C_m	1.05
ds/l	1
H/ds, ratio of the baffle window height to shell ID	0.46
s/d_0, ratio of tube pitch to tube OD	1.25
N_h	0.37
N_p	0.25
F_h	0.485
F_p	0.660
M	0.88
Y	6.5
$(Pr)^{1/3}$ (shell side)	1.234
a, slope of curve for h	0.56
b, slope of curve for f2	-0.53
Ca, coefficient in h=Ca*G2(a)	0.0555
Cb, coefficient in f = Cb*G(b)	1.820
t_m, LMTD	11.14
LMTD correction factor	1
G2, shell-side flow rate	765.8 $kg/m^2.s$
R_{ep}-Reynolds number for friction factor calculations	10522.8
f2, shell-side friction factor	0.5
l baffle spacing m	0.077 m
N2 No. of shell side fluid pass	15
L =l*N2	1.155 m
F1 tube side friction factor	0.23

Design of water-gas heat exchanger to recover heat from exhaust:

Internal combustion engines reject heat to a heat recovery exchanger from exhaust. The produced hot water is the heat source for VAM. Heat is exchanged between a liquid and gas, usually water and air. In most instances the heat transfer coefficient on the gas side are much lower than those on the liquid side, and hence finned gas-side surface are advantageous.

If the pumping power available is fixed and matrix volume is to be kept to a minimum, the aerodynamically clean flattened tube and the flat plate-fin matrix is likely to give close to the best performance obtainable. On the other hand , if the heat transfer matrix cost or weight is the prime consideration, it is usually best to employ some mild turbulating device to increase the heat transfer coefficient on the gas side and thus reduce the amount of surface area required.

Parameter fixed for heat exchanger

	Shell side	Tube side
Fluid	Jacket water	Circulating water
Temperature in (C)	110 C	87 C
Temperature out (C)	95 C	95 C
LMTD	11.14	
Density	998 kg/m3	998 kg/m3
Specific heat	4.2 KJ/kg	4.2 KJ/kg
Viscosity	3.05*10(-4)	3.05*10(-4)
Thermal conductivity	0.6802 W/m2*C	0.6802 W/m2*C
Prandtl number	1.88	1.88
Allowable pressure	34474 Pa/m	34474 Pa/m

A preliminary survey shows that tube matrix showed in figure with OD 1.121-in, is well suited to the application. Note that the collar at the root of the fins increases the apparent tube OD to 0.645 in.

The high heating effectiveness required indicates that something close to a counter flow configuration should be employed. The multipass crossflow configuration seems to meet this requirement. The length-diameter ratio for continuous circular passage on the air ought be roughly 300. The higher heat transfer coefficient for crossflow over finned tubes should cut this to about one-half. Using an l/d=150,

For building heating system the air velocity should be kept below 20 ft/s (6.1 m/s)-but for this application noise is not objectionable. A value of 5 ft/s for water velocity through the tubes is recommended. The total flow rate can be calculated from the heat load, the fluid temp rise (or drop) and the specific heat for the water and air.

Total flow rate= (heat load)/(specific heat*temp. rise (or drop)).

The number of parallel passages required on the water side can be calculated by dividing the total water flow rate by flow rate per tube and a round number must be used

Design Calculation for a Water heater by Using Exhaust Gas

- Design heating load : 50 kW
- Water : in tubes
- exhaust gases : outside
- Heat transfer matrix : Helically finned Cu tubes 0.625 in OD , 0.035 inch wall thickness, Cu fins 1.121 inch OD.

Fluid	Water	Exhaust Gases
Mean Pressure	2 bar	1.013 bar
Specific Heat	4.187 kJ/kg*C	1 kJ/kg*C
Internal flow area	156 m m^2	
Tube surface area ,m2/m	0.044 m^2/m	0.571 m^2/m
Inlet temperature	95 C	540 C
Outlet temperature	100 C	170 C
Greatest temperature difference	440 C	
Lowest temperature	80 C	
LMTD	206.30 C	
Density	998 kg/m3	1.138 kg/m3
Flow velocity	5 ft/s(1.52m/s)	22.2ft/s(6.76m/s)
Flow rate kg/(s* m^2)	1520 kg/(s*m^2)	7.71 kg/(s* m^2)
Viscosity (Pa.s)	$3.3*10^{-4}$ Pa.s	$.248*10^{-4}$ Pa.s
Reynolds No. (Re)	37500	3700
Equivalent Diameter mm	14.1	5.47
Friction factor	0.03	0.05
Prandtl No	1.88	0.683
$(Pr)^{2/3}$	1.523	.775
Heat transfer coefficient h	63.6 kW/m^2	95.5 W/m^2
Fouling factor	.001	
1/U	.11495	
U (W/m^2*C)	87.05 W/m^2*C	
Total Flow rate	0.942 kg/s	0.0533 kg/s

Total No. of passage required for water	0.942/flow per pass=	**4**
Free flow area/frontal area	0.443	
Matrix Intel face area	Flow rate per passage/ (Free flow area factor*flow rate)=	0.0156 m2
Total Matrix length		
Matrix surface per unit volume	324 m2/m3	
Tube matrix length	Surface required/(matrix frontal area*matrix surface per unit volume)=	.55 m
No. of tubes banks	**16**	

This way we find total number for one pass of water = 4 and number of banks of tubes required is 16. So 64 tubes are required to cool the exhaust gas from 540 to 170 C connecting in U shape to each other.

VI. RESULT

Energy extracted by heat-exchanger :

- Energy recovered by shell-and-tube heat exchanger= Mw*Cp*(temp. gain) =31kW
- Energy recovered from exhaust gases = 20 kW
- Total heat recover = 51 kW

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