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DEVELOPMENT OF LNG FIRED GAS TURBINE CYCLE FOR LOCOMOTIVE APPLICATION

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ABSTRACT

A feasibility study is undertaken to replace diesel engine with natural gas fuelled Gas Turbine for a locomotive or for any industrial purpose. In the present work a methodology is arrived at to design a gas turbine cycle, which can be used to power a locomotive using natural gas as its fuel. The cycle is designed for selected power output of 9 MW. A program is developed to estimate the state variables at inlet and outlet conditions of constituent equipment and predict the performance of the cycle for varied operating conditions.

NOMENCLATURE

Cp	Specific heat (kJ/kg.K)
$_{ex.air}$	Excess air
Far	Fuel to air ratio
fg	Flue gas
H	Actual Enthalpy (kJ/kg)
m_a	Mass flow rate of air (kg/s)
$m_{th.air}$	Mass of theoretical air for combustion (kg/s)
$M_{ex.air}$	Mass of excess air for combustion (kg/s)
m_f	Mass flow rate of fuel (kg/s)
mol.frac	Mole fraction of gas component
Mol.wt	Molecular weight of gas component
n	Polytropic Index of air
P	Pressure (MPa)
P_{inj}	CNG injection pressure into combustor (MPa)
P_{flash}	LNG Pressure at inlet of flash chamber (MPa)
PR	Pressure Ratio
T	Temperature (K)
T_{sat}	LNG Saturation temperature (K)
Q_{comb}	Heat of combustion (kJ/kmol)
R	Gas constant (kJ/kg.K)
wf	Weight fraction of gas component
W	Work done/consumed (kW)
η	Efficiency

INTRODUCTION

Indian Ministry of Railways - RDSO (Research Design and Standards Organisation), Lucknow[1],is planning to develop a high horsepower (8.5-9.5 MW) gas turbine based locomotive for heavy haul and long haul goods train operations for shortly coming up dedicated freight corridors of Indian Railways. The need for development of high horse power gas turbine based locomotive is mainly due to availability of large reserves of Natural Gas (NG) in India.

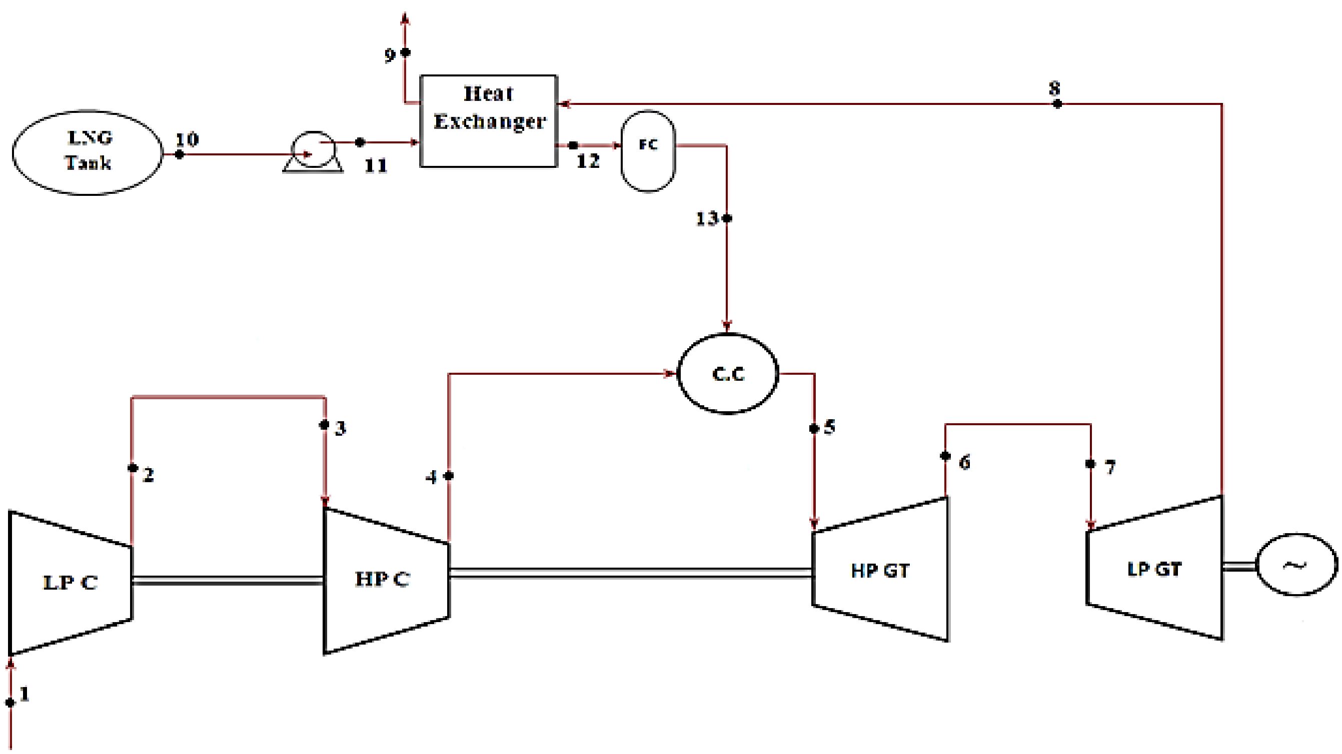
The proposed Gas turbine cycle is shown in Fig. 1. Air from low pressure (LP) compressor through a purification device is further compressed by the high pressure (HP) compressor enters combustion chamber (CC) for combustion of CNG. Energy released during the process of combustion is used to drive the turbine. The pressure ratio of the high pressure turbine is selected such that it is entirely supplied to drive LP and HP compressors. The HP turbine exhaust enters LP turbine which drives an alternator. The alternator drives the traction motor of the locomotive.

The exhaust flue gas from LP turbine through a heat exchanger shall preheat the Liquefied Natural Gas (LNG) stored at 1 atm and 111 K (T_{store}).This LNG is stored and supplied from a cryogenic tank. A cryogenic pump is used to transfer the LNG from storage tank to the combustor through the heat exchanger and flash chamber. This when released into the flash chamber (FC) changes its phase to enter the combustion chamber. The pressure ratios for compressors and gas turbines are suitably selected for the application such that the exhaust gas temperature to environment is 550 K (277^{o}C). Mass of LNG undergoing this process is determined accurately to achieve the desired power output. Cycle efficiency and parametric sensitivity is presented with the help of developed computer program.

LITERATURE REVIEW

The majority of gas turbine locomotives have had electric transmission but mechanical transmission has also been used, particularly in the early days.

Work leading to the emergence of the gas turbine locomotive began in France and Sweden in the 1920s but the first locomotive did not appear until 1933. High fuel consumption was a major factor in the decline of conventional gas-turbine locomotives and the use of a piston engine as a gas generator would probably give better fuel economy than a turbine-type compressor, especially when running at less than full load.



Legend

No.	Description
1	Inlet to Low Pressure (LP) compressor
2	Exit from LP compressor
3	Inlet to High Pressure (HP) compressor
4	Exit from HP compressor to Combustion Chamber
5	Exit from combustion chamber to HP turbine
6	Exit from HP turbine
7	Inlet to LP turbine
8	Exit from LP turbine to Heat Exchanger
9	Exhaust of flue gases to atmosphere
10	LNG fuel tank outlet to cryogenic pump
11	Cryogenic pump outlet to heat exchanger
12	Heat exchanger outlet to flash chamber
13	Flash chamber outlet to combustion chamber

FIGURE 1. SCHEMATIC OF PROPOSED GAS TURBINE CYCLE

Despite the first GT locomotive being successfully developed in 1933, it wasn't until 1952 that commercial application of GT locomotive started with Union Pacific's UP-18. This engine was developed by General Electric which was capable to deliver power of 8500 hp (6338 kW).

Robert C Allen[2] described the gas turbine locomotive as a self-propelling vehicle and discussed the aspects of maneuvering and traction effort of the engine. John et al[3], gave the intake design of air to the gas turbine such that foreign impurities intake is avoided and the size of the structure is reduced. Hans[4] described the design of the supporting structure of a gas turbine locomotive. Further improvements to the existing designs were made by Andre[5], in which he added an air turbine to a gas turbine locomotive which provided additional torque to accelerate or decelerate the locomotive. Donnelly[7], described gas turbine locomotive fuelled by compressed natural gas, in which he explained the electrical connectivity to the alternator to form a high speed electric traction generator.

The most recent and advanced gas turbine locomotive was built by the Russians named GT1h-002[6], which runs on LNG.

METHODOLOGY

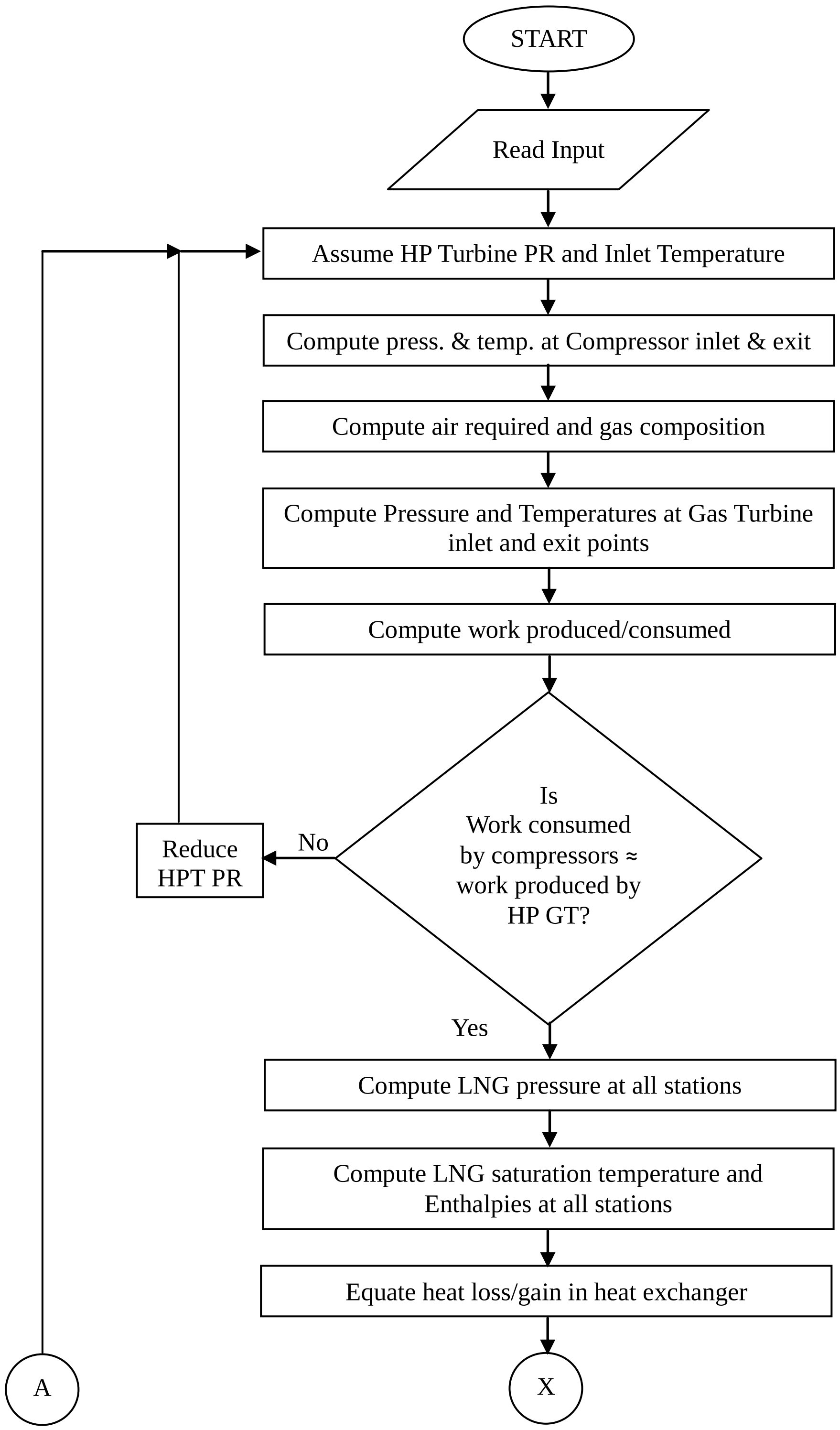
Following are the list of assumptions that were safely assumed from different sources in developing the computer program for gas turbine cycle:

a. Mass of air entering the compressor is considered as moisture free.
b. Amount of Sulphur and Phosphorus present in the fuel is considered to be negligible and is not considered in the calculations.
c. Pressure drop between the two compressors was assumed to be 0.025 MPa.[8]
d. Pressure drop between the HP compressor and combustion chamber was assumed to be 0.05 MPa.[8]
e. Pressure drop between LP turbine exit and the heat exchanger was considered to be 0.05 MPa.[8]
f. Pressure difference between heat exchanger hot gas exhaust and the ambient pressure was safely assumed to be 0.05 MPa.
g. Enthalpies of air and its constituents were calculated as the product of specific heat at constant pressure[9] and the temperature at that point. The resultant enthalpy value was accurate with an error margin of 0.001%.[9]
h. Margin of error in the turbine inlet temperature set in the program is 1 K.

i. Pressure ratio of both turbines calculated by the computer program is accurate up to 0.001 decimal.

j. Equations developed to calculate the enthalpy values of methane[15] are accurate up to 0.001 kJ/kg.

Figure 2 explains in brief the steps involved in developing the program.



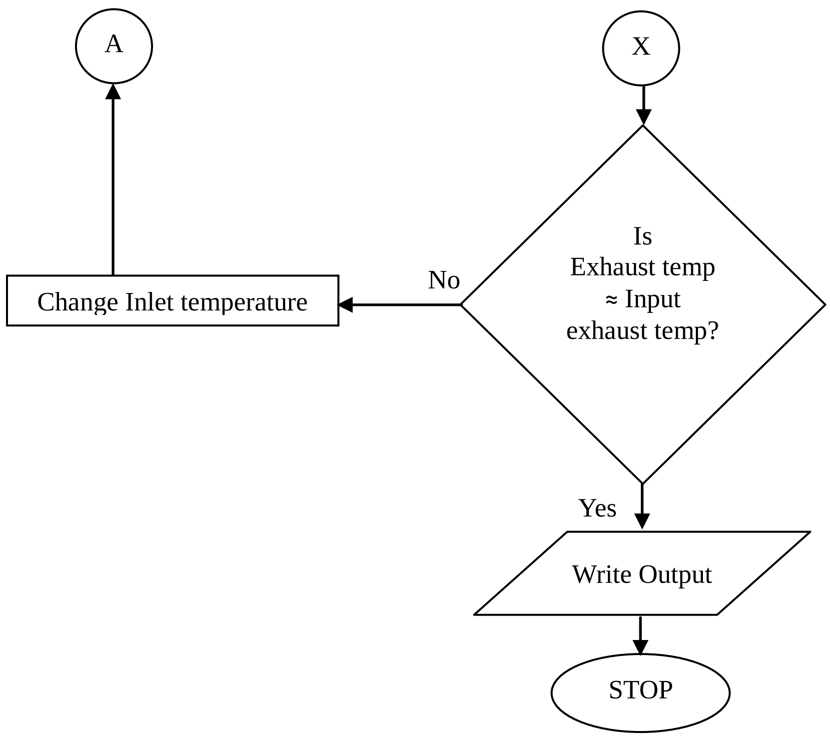


FIGURE 2. FLOWCHART OF THE PROPOSED PROGRAM

The logic behind developing the computer program is explained in brief below.

1) Input parameters.
   i. Fuel composition.
   ii. Ambient Inlet air pressure and temperature (P_1 and T_1).
   iii. Low pressure and High pressure compressor pressure ratios (PR_{LPC} and PR_{HPC}) and isentropic efficiencies (η_{LPC} and η_{HPC}).
   iv. High pressure and low pressure turbine efficiencies (η_{HPT} and η_{LPT}).
   v. Alternator efficiency (η_a).
   vi. Exhaust gas temperature (T_{exit}).
   vii. Power required (Pw).

2) Read guess values of HP Turbine Pressure Ratio (PR_{HPT}) and inlet temperature (T_5).

3) Calculate the pressure and temperatures at the inlet and exit of both compressors.
   For example for LP compressor,

$$P_2 = PR_{LPC} * P_1 \quad (1)$$
$$T_2 = T_1 * (PR_{LPC})^{(n-1)/n} \quad (2)$$

where n = Cp/(Cp – R)

where $R = \sum_{i=1}^{j} wf_i * 8.3143/Mol.wt_i$, here j represents the number of species present in the gas considered, in this case the gas is air.

Actual exit temperature,

$$T_2' = [(T_2 - T_1) / \eta_{LPC}] + T_1 \quad (3)$$

4) Compute heat of combustion for given composition of LNG using following formula.

$$Q_{comb} = \sum_{i=1}^{j} (mol.frac_i * Mol.wt_i * Q_{comb\ .i}) \quad (4)$$

where j is the number of species present in LNG.

5) Calculate the theoretical air ($m_{th.air}$) required for complete combustion of given composition of LNG.

$$C_xH_y + (x+\frac{y}{4})(O_2 + 3.76\ N_2) \longrightarrow xCO_2 + \frac{y}{2}H_2O + 3.76(x+\frac{y}{4})N_2 \quad (5)$$

So

$$m_{th.air} = (x+\frac{y}{4})*(O_2 + 3.76\ N_2) \quad (6)$$

6) Weight fractions of flue gas constituents at stoichiometric conditions are computed.

7) Calculate the excess air ($m_{ex.air}$) to be added to above stoichiometric air to obtain desired HP turbine inlet temperature (T_5).

$$T_5 = Q_{comb} / [(\sum_{i=1}^{j}(Cp_i * wf_{fgi})_{stoich} + \sum_{i=1}^{j}(Cp_i * wf_i)_{ex.air}] \quad (7)$$

$$m_{ex.air} = \frac{\left(\frac{Qcomb}{T_5}\right)-(wfCO_2*Cp5(CO_2))-(wfH_2O*Cp5(H_2O))-(wfN_2*Cp5(N_2))}{(0.23*Cp5(O_2))+(0.77*Cp5(N_2))} \quad (8)$$

Where i to j are the species present in flue gas and air

8) Compute the fuel to air ratio (far).

$$far = m_f / m_a \quad (9)$$

where $m_a = m_{th.air} + m_{ex.air}$

9) Compute the gas composition after combustion (i.e., weight fractions of CO_2, O_2, N_2, and H_2O).

10) Calculate the pressure and temperatures at inlet and exit of both turbines. For example, pressure of gases at exit of High Pressure turbine (HPT) is calculated as

$$P_6 = PR_{HPT} * P_5 \quad (10)$$

$$T_6 = T_5*(PR_{HPT})^{(n-1)/n} \quad (11)$$

$$T_6' = T_5 - [\eta_{HPT} * (T_5 - T_6)] \quad (12)$$

11) Calculate Specific Heat at constant Pressure and Enthalpy at each state points of the cycle considering the temperature and gas composition at each stage using the formulae given in [9].

12) Calculate the work produced by turbine and consumed by compressor for 1 kg/s of LNG. Work consumed by compressor (W_c) and turbine (W_T) is calculated as

$$W_c = m_a * (H_4 - H_3 + H_2 - H_1) \quad (13)$$

$$W_T = (m_a + m_f) * (H_5 - H_6 + H_7 - H_6) \quad (14)$$

13) Calculate the air flow and fuel flow rates required to obtain desired work output.

$$m_f = Pw / \{\eta_a * [((1+far^{-1}) * (\Delta H_{5-6} + \Delta H_{7-8})) - (\Delta H_{2-1} + \Delta H_{4-3})/far]\} \quad (15)$$

$$m_a = m_f / far \quad (16)$$

14) Is the work consumed by compressors and the work produced by the HP turbine nearly equal? If yes, then continue. Else go to 2 and reduce the HPT PR by 0.001 and start at 3.

15) CNG injection pressure (P_{inj}) into the combustion chamber is assumed to be 0.005 MPa greater than the compressed air pressure at the exit of HP compressor. Also the pressure drop in the flash chamber (P_{flash}) is assumed to be 1 MPa.

$$P_{inj} = P_4 + 0.005 \quad (17)$$

$$P_{flash} = P_{inj} + 1 \quad (18)$$

16) LNG from the storage tank is raised to above P_{flash} with the help of cryogenic pump.

17) Saturation temperature for LNG is calculated at P_{flash} using the relations developed by data given in [15].

$$T_{sat} = -0.355*P_{inj}^4 + 4.227*P_{inj}^3 - 19.23*P_{inj}^2 + 50.07*P_{inj} + 114.4 \quad (19)$$

This should be the temperature of LNG at the end of heat exchanger and will be the injection temperature of fuel since flashing takes place at constant temperature.

18) Pressure-Temperature-Enthalpy relations for LNG were developed using the data given in [15].
Enthalpy gain by LNG in the heat exchanger for pressures ranging from 0.5 to 8 MPa (as given in [15]) are given below as a function of temperature. The 12 relations given below are for pressures 0.5, 0.8, 1, 1.5, 2, 2.5, 3, 3.5, 4, 4.5, 5 and 8 MPa respectively.

$\Delta H(1) = (3.532*dt)$
$\Delta H(2) = (3.597*dt)$
$\Delta H(3) = (3.583*dt)$
$\Delta H(4) = (3.678*dt)$
$\Delta H(5) = (3.721*dt)$
$\Delta H(6) = (3.812*dt)$
$\Delta H(7) = (0.011*((tsat^2) - (tstore^2))) + (0.765*dt)$
$\Delta H(8) = (0.013*((tsat^2) - (tstore^2))) + (0.259*dt)$
$\Delta H(9) = (0.01*((tsat^2) - (tstore^2))) + (0.986*dt)$
$\Delta H(10) = (3.804*dt)$
$\Delta H(11) = (3.776*dt)$
$\Delta H(12) = (0.01*((tsat^2) - (tstore^2))) + (0.82*dt)$
(20)

where $dt = T_{sat} - T_{store}$

19) Enthalpy calculated above is interpolated to calculate value of enthalpy gain by LNG in the heat exchanger (ΔH_{LNG}).

20) LNG is to be heated in the heat exchanger from its storage temperature to above calculated saturation temperature. The amount of heat required by LNG during this process is given out by exhaust gas from the LP turbine. Balancing the energy loss and gain, the enthalpy of exhaust gases at exit of heat exchanger is calculated.

$$H_{exhaust} = H_8 - \Delta H_{LNG} \quad (21)$$

21) Temperature corresponding to above computed $H_{exhaust}$ enthalpy is computed using the relations given in [9].

22) Exhaust gas temperature is compared with specified value. If it is different, then HP Turbine inlet temperature is varied accordingly till such time they are within specified error limit.

OUTPUT RESULTS

The output of the program comprises of the following parameters:

1. Pressure, temperature and enthalpy values at inlet and exit of constituent equipment in the cycle.
2. Mass flow rate of air and fuel.
3. Exhaust gas composition.
4. Calculated pressure ratio of high pressure turbine.
5. Work done by compressor, turbine and the net useful work.
6. Thermal efficiency of the cycle.

PARAMETRIC SENSITIVITY

Different input parameters are varied and their effect on the output variables is studied below. The selected parameters were varied reasonably while the other input parameters remained same. For example, the inlet temperature variation was studied at full load and ISO conditions. Similarly, load requirement was varied in the input while the other parameters remained same at ISO conditions.

Variation of Inlet Temperature

Efficiency of the cycle is dependent on the inlet temperature of air which is sucked through the compressor. The lower the air inlet temperature, higher is the cycle thermal efficiency. This is shown in Fig.3 where the compressor inlet temperature was varied from 97-112% of the designed inlet temperature. The thermal efficiency is decreasing as the inlet temperature is increased because the heat supplied to the cycle is decreased by the program by increasing the mass flow rate of the gases.

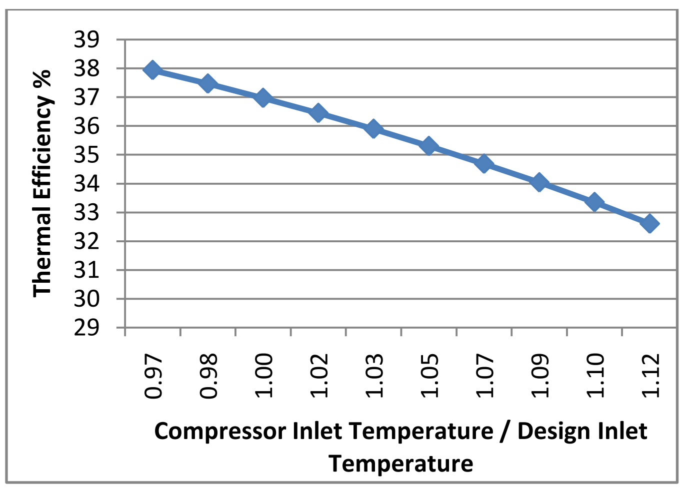


FIGURE 3. THERMAL EFFICIENCY VS. NON DIMENSIONAL INLET TEMPERATURE

Variation of Cycle Pressure Ratio

The overall cycle pressure ratios which were given as input variables to the program in terms of individual compressor pressure ratios are varied below.

Fig.4 shows that pressure ratio is directly proportional to the cycle thermal efficiency. Increasing the pressure ratio will improve the thermal efficiency of the cycle since the net work produced by the flue gases when they expand at higher pressure in the turbine is higher compared to the same at lower pressure. At pressure ratio of 53% of designed PR the GT gives 30.75% thermal efficiency which is the minimum value[1] that the cycle can be allowed to operate at.

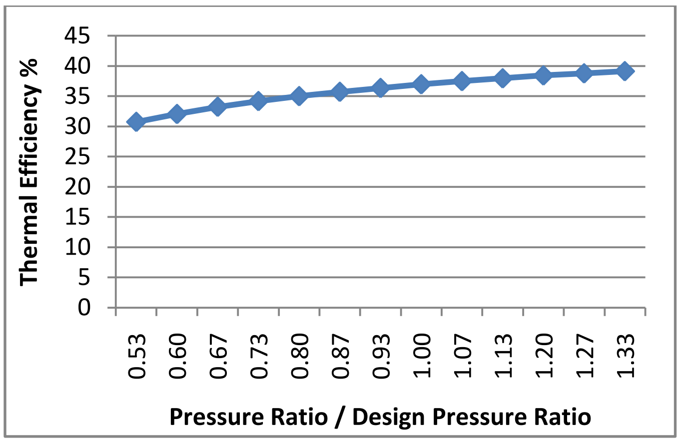


FIGURE 4. THERMAL EFFICIENCY VS. NON DIMENSIONAL PRESSURE RATIO

The maximum value of pressure ratio (i.e., 133% of designed PR) is selected such that the pressure of the air after complete compression is lower than the pressure of the LNG fuel so that fuel can be injected easily into the combustion chamber. Hence the maximum value of PR is limited by the pressure at the end of flashing of LNG.

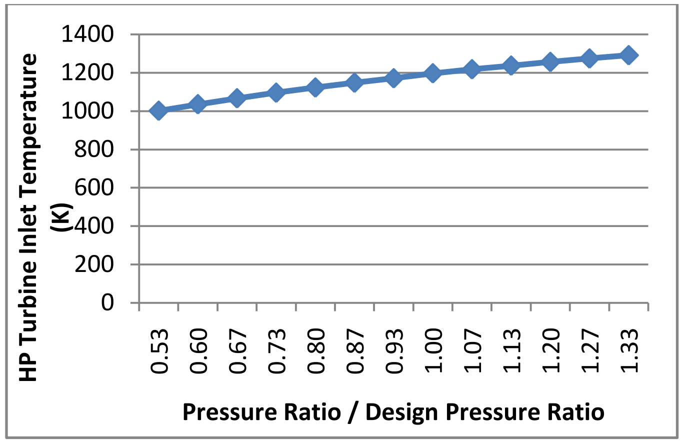


FIGURE 5. HP TURBINE INLET TEMPERATURE VS. NON DIMENSIONAL PRESSURE RATIO

Increase in pressure ratio implies higher amount of work is done by the compressor for which the turbine must produce equally higher amount of work to generate the power required. This is done by increasing the mass flow rates which affects the turbine inlet temperature and fuel-to-air ratio as shown in Fig.5 and Fig.6 respectively. The turbine blade material properties are to be considered while designing the turbine so that the material can withstand the designed turbine inlet temperature. Modern turbines which are steam cooled can withstand a maximum temperature of 1600°C[12] when the blades are made of Silicon Carbide (SiC).

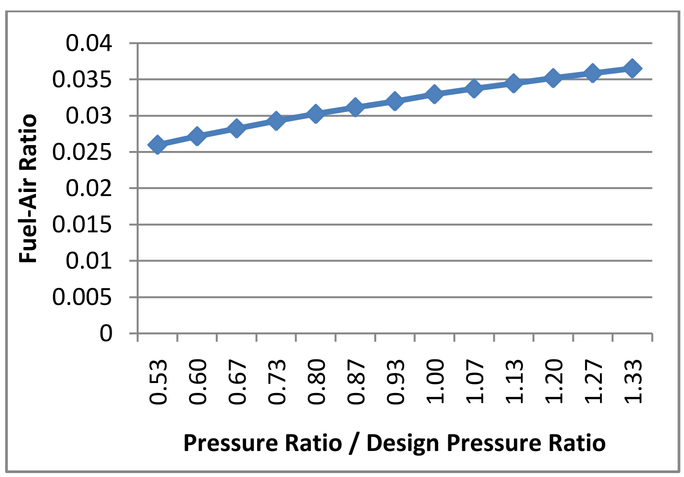


FIGURE 6. FUEL AIR RATIO VS. NON DIMENSIONAL PRESSURE RATIO

The HP turbine pressure ratio must also be increased accordingly so that the LP turbine alone acts as the power turbine in the cycle. This is shown in Fig.7 below where the HP turbine pressure drop ratio increases linearly as the pressure ratio of the cycle is varied. It is interesting to note that the HP turbine PR is nearly equal to the square root of the cycle pressure ratio.

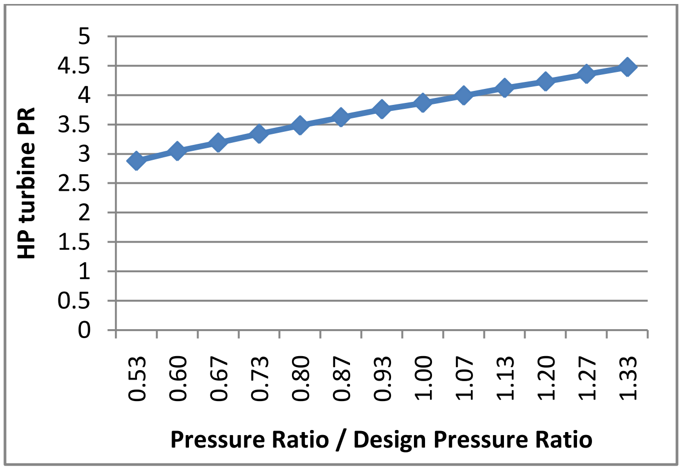


FIGURE 7. HIGH PRESSURE TURBINE PRESSURE RATIO VS. NON DIMENSIONAL PRESSURE RATIO

Variation of Fuel Composition

LNG may be classified taking into account several criteria: Density, Heat Value, Wobbe Index, Methane or Nitrogen amount, etc. Normally, its density is the most usual parameter used for classification. Thus, is spoken of heavy or light LNG's. Table 1 presents three typical LNG qualities due to its density[13].

TABLE 1. LNG CLASSIFICATION BASED ON COMPOSITION

Composition (%)	LNG light	LNG medium	LNG heavy
Methane	98.0	92.0	87.0
Ethane	1.4	6.0	9.5
Propane	0.4	1.0	2.5
Butane	0.1	0.0	0.5
Nitrogen	0.1	1.0	0.5

The above composition was given as the input of LNG composition to record its effect on cycle performance and the output is shown in table 2.

TABLE 2. OUTPUT RESULTS FOR DIFFERENT LNG COMPOSITION

LNG type	Light	Medium	Heavy
Thermal Efficiency (%)	37.04	36.94	36.91
Heat of combustion (kW)	25579.13	25645.82	25666.
Fuel-Air ratio	0.0300	0.0314	0.0335

Surprisingly the thermal efficiency for all the three inputs was nearly equal. The only major difference was observed in the heat supplied by the fuel. To lower the temperature given out by different compositions due to their difference in heat of combustion, fuel and air mass flow rates were increased and decreased respectively by the program which is reflected in the rise of fuel to air ratio.

Variation of Load

The power requirement of the GT locomotive depends on the load it is hauling. The Power and mass flow rate of air were calculated for a given shaft speeds in eight steps (for the eight notches) to obtain the various notch positions as per ISO conditions.

The amount of air sucked in through the compressor determines the power output of the engine, which can be monitored and controlled by the engineer driving the locomotive through these eight notches as shown by the eight points in Fig.8.

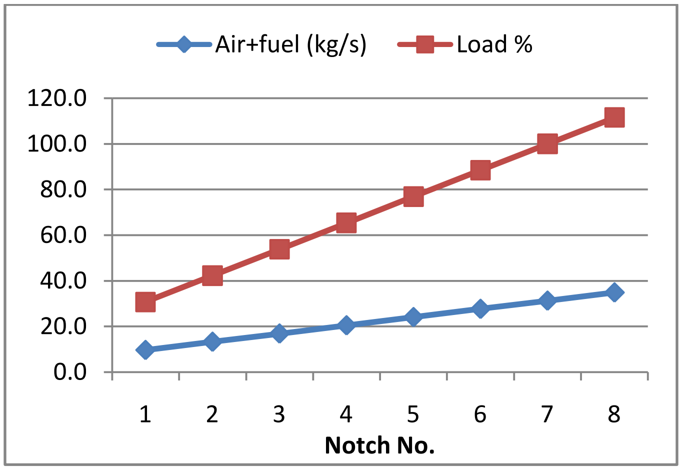


FIGURE 8. GAS FLOW RATE AND LOAD% VS. NOTCH POSITION

Variation of Exhaust Temperature

In this section the exhaust temperature was varied. Allowing the gases to exit the heat exchanger at high temperatures may be harmful to the environment because of which it was decided by RDSO[1] that the exhaust at a distance of 1 meter must not exceed 300ºC.

As shown in Fig. 9, thermal efficiency is increasing since the heat supplied to the cycle is reduced by the program to balance the heat.

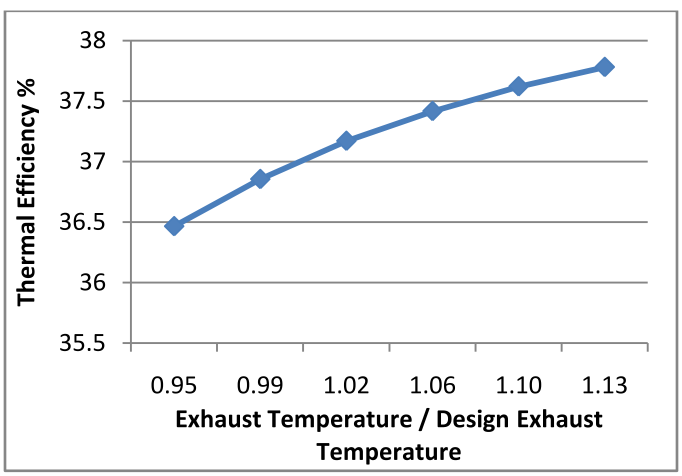


FIGURE 9. THERMAL EFFICIENCY VS. NON DIMENSIONAL EXHAUST TEMPERATURE

Fig. 10 shows that the fuel to air ratio increases as the exhaust temperature is increased. This is because the turbine inlet temperature increases when the exhaust temperature is increased.

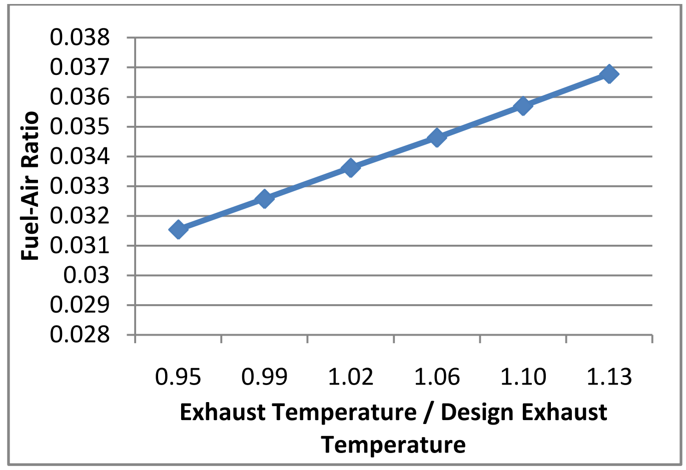


FIGURE 10. FUEL TO AIR RATIO VS. NON DIMENSIONAL EXHAUST TEMPERATURE

CONCLUSIONS

The following are the lists of conclusions that can be drawn from this study.

1) Brayton cycle suitable for Gas Turbine locomotive is conceived based on Indian Railways-RDSO specifications.
2) A computer program is developed for the cycle.
3) Cycle analysis and heat balance are carried out for different load conditions.
4) Parametric sensitivity of different input conditions is studied and respective conclusions based on the observations are established.
5) The thermal efficiency of the cycle drops as the inlet temperature of the atmospheric air is increased for rated load.
6) Increase in overall cycle pressure ratio raises the thermal efficiency, turbine inlet temperature, fuel to air ratio and high pressure turbine pressure ratio for rated load.
7) Slight rise in thermal efficiency is seen for different densities of LNG, however, increase in fuel to air ratio and significant rise in heat of combustion was observed for rated load.

8) The amount of fuel and air flow rate increases when the load is varied at ISO conditions.
9) Increase in the exhaust temperature of flue gases from the heat exchanger increases the thermal efficiency, turbine inlet temperature and fuel to air ratio of the cycle for rated load.
10) Since the program was originally developed to predict the performance of a gas turbine engine using LNG as fuel, in future, it can be modified to use any type of fuel as its input. Also the program can be further developed for different industrial applications.

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