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# Inlet Air cooling for Gas turbines

***By : Arun Dharap and Anil Ghanwat***

*Cogeneration & Captive Power Business Group*

*Larsen & Toubro Limited, Mumbai*

In a typical gas turbine ambient air is drawn through the air inlet plenum assembly, filtered and compressed in a multi stage axial compressor. Compressed air from the compressor enters the combustor chamber. A measured quantity of fuel, at a rate consistent with the speed and load of the turbine, is injected in the combustor chamber. The hot gases from the combustor chamber expand and flow to a multistage turbine section. Each stage consists of a row of fixed nozzles followed by a row of rotary turbine buckets. In each nozzle row, the kinetic energy of the jet is increase with an associated pressure drop and in each following row of moving buckets a portion of kinetic energy of the jet is absorbed as useful work on the turbine rotor. The resulting shaft rotation is used to turn the generator to generate electric power.

The heat from the hot gases leaving the gas turbine is recovered in a Heat Recovery Stream Generator (HRSG) before the gases are let out into the atmosphere through a stack. The steam generated in the HRSG may be used for process as a heating medium (Co-generation plants) or expanded in a steam turbine to generate power (Combined Cycle plants)

The performance of a gas turbine viz. the power output, and the heat rate (measure of efficiency, i.e. the amount of energy consumed per kWh of electricity produced) depends on the following major factors:

1. Site altitude i.e. atmospheric pressure
2. Inlet pressure drop in the filters and intake system

3. Outlet pressure drop in the HRSG.
4. Site design temperature.
5. Site design relative humidity corresponding to site design temperature.

A gas turbine is a constant volume machine i.e. the volume of air compressed is fixed, irrespective of ambient temperature. Hence, as the temperature of air rises, the density of air decreases and the mass flow rate of compressed air is reduced. As the power output of the gas turbine is proportional to the mass flow rate of air, power output reduces as the ambient temperature increases. Further, the efficiency of the gas turbine also falls as more power is required to compress warmer air. For a given site and the configuration of the plant, the first three parameters are fixed and cannot be changed. However it is possible to change the other two parameters and obtain a higher output and improved efficiency by cooling the air before it is admitted in the gas turbine compressor section.

**Table 1 : Comparison of performance of indutrial and aero-derivatives gas turbines at different inlet temperatures**

| | | **Frame 5 (5371 PA)** | | | **Frame 6 (6561 B)** | | | **LM 6000 PC** | | |
|---|---|---|---|---|---|---|---|---|---|---|
| Inlet temp | °C | 15 | 35 | 40 | 15 | 35 | 40 | 15 | 35 | 40 |
| GT power | kW | 25247 | 2143 | 20391 | 38564 | 33520 | 32268 | 40856 | 31870 | 27406 |
| Exhaust flow rate | tph | 44.7 | 408.3 | 398.2 | 520.9 | 476.3 | 464.7 | 442.2 | 370.3 | 359.6 |
| Exhaust temp | °C | 491 | 507 | 510 | 535 | 549 | 553 | 461 | 489 | 476 |
| Fuel consumption | tph | 7.767 | 6.909 | 6.704 | 10.4 | 9.395 | 9.161 | 8.554 | 7.178 | 6.591 |
| GTG efficiency | % | 28 | 26.37 | 25.86 | 31.54 | 30.34 | 29.95 | 40.61 | 37.75 | 35.36 |
| Mass flow of air | per kW | 17.72 | 19.05 | 19.52 | 13.5 | 14.2 | 14.4 | 10.82 | 11.61 | 13.12 |
| % Decrease in power w.r.t 15°C | % | 0.00 | 15.11 | 19.23 | 0.00 | 13.08 | 16.33 | 0.00 | 21.99 | 32.92 |
| % Decrease in efficiency w.r.t 15°C | % | 0.00 | 5.82 | 7.64 | 0.00 | 3.80 | 5.04 | 0.00 | 7.04 | 12.93 |

**Note:**

Basis for above analysis;

| | |
|---|---|
| Fuel | Distillate |
| Relative humidity | 60% |
| Elevation | 0 metres |
| Inlet pressure loss | 10 mbar |
| Exhaust pressure loss | 22 mbar |
| Frame 5 & Frame 6 | : Industrial Gas Turbine |
| LM 6000 PC | : Aero-derivative gas turbine |

**Table 1** gives the power output and efficiency of typical frame gas turbine (industrial gas turbine) and LM 6000 PC gas turbine (aero-derivative type gas turbine) at various ambient temperatures. From a study of this Table, one will note that the:

- **Percentage drop in power output, n case air temperature rises from 15°C to 40°C in case of industrial gas turbines viz. for Frame 5 it is 19.23% and for Frame 6 it is 16.33% whereas in aero-derivative machines viz. LM 6000 PC, it is 32.92%**.
- **Percentage drop in efficiency in case air temperature rises from 15°C to 40°C in case of industrial gas turbines viz. for Frame 5 it is 7.64% and for Frames 6 it is 5.04% whereas in aero-derivative machines viz. LM 6000 PC, it is 12.93%**.

Thus, cooling of the inlet air gives the following advantages:

- Improves the power output and efficiency (heat rate) of the turbine
- The output of the gas turbine is independent of the ambient temperature and does not decrease with increase in the ambient temperature.
- By careful selection of the temperature, to which he inlet air is cooled it is possible to ensure that the gas turbine operates at its highest efficiency through-out the year irrespective of ambient temperature.

## Factors Affecting Commercial Viability of Inlet Air Cooling

Inlet air cooling can be commercially attractive and viable, if the benefit of increased power output and decreased heat rate (improved efficiency) because of inlet air cooling is commensurate with the investment required in capital cost and the running cost of such equipment.

The main factors which decide the commercial viability of inlet air cooling are:

- **Plant configuration and schedule of operation of the plant** - Gas turbines are used either in open cycle mode / co-generation mode / combined cycle mode for either peaking duty / intermittent duty / continuous base load duty. Hence the commercial viability depends on the number of hours of operation of the plant in a year, desired output etc. It also depends upon the power to the steam ratio requirement of a co-generation application, the rate at which fuel is available and the rate at which power is sold to the grid.

- **Amount of air compressed per kW** - Lower the air flow per kW of output, lesser the refrigeration capacity and hence lesser the operating and owing costs. Aero-derivative gas turbines require lesser flow of air per kW of power generated than frame type heavy duty gas turbines (see **Table 1**) because of higher compression ratios, firing temperatures and number of stages of expansion in the turbines. **Table 1** shows a comparison of various makes and models of gas turbines, the amount of air required per kW of power generated, exhaust gas temperatures etc. Hence, once the model of gas turbine and type of gas turbine is selected, based on power and steam requirements, the scope for inlet air-cooling gets determined.
- **Enthalpy of ambient air at design conditions** - Higher the enthalpy of air per kg of dry air, higher the refrigeration capacity required to cool the air to desired conditions, hence more the operating and owning costs. It is important to judiciously select the outside design dry bulb temperature and concurrent relative humidity to size the capacities of inlet air cooling plant and equipment.
- **The temperature to which inlet air cooling is done** - There is an optimum temperature for a given turbine model and design ambient conditions to which air should be cooled. By cooling the air below this optimum temperature, the incremental additional benefit, by way of additional power & steam, is not commensurate with additional incremental investments required to achieve lower temperatures.

The design inlet temperature at the gas turbine is also affected by the capabilities of the equipment available. The minimum chilled water temperature available from lithium bromide absorption chillers or mechanical chillers is around 5°C. Thus, typical air temperatures at the outlet of the cooling coil and the inlet of gas turbine compressor will be around 10°C.

Furthers the inlet air cooling system must be designed to avoid icing at the compressor inlet or anywhere in the air intake system. Ice fragments a sucked into the compressor can cause serious structural damage. Icing is a potential problem, inlet Air Cooling, any time the ambient temperature drops to near the freezing mark. The problem is exacerbated for inlet air cooling systems because warm ambient air will almost always be saturated after passing through the inlet air cooling coils. When the air is drawn into the mouth of the compressor its velocity increases and its temperature drops further as air enthalpy is transformed into kinetic energy in an adiabatic process.

## Methods of Inlet Air Cooling

Inlet air cooling can be achieved by any of the following methods:

### Indirect Cooling

**Using chilled water** - In this case, the air is cooled by circulating chilled water through cooling coils. The cooling coils are installed in the intake air path and chilled water is produced using a vapor compression refrigeration cycle or absorption cycle.

Disadvantages of this method - There is penalty on the turbine performance because of pressure drop in the air stream. Also this being an indirect method of cooling, the temperature of air leaving the coil will be approximately 3 to 5°C more than the outlet chilled water temperature. The advantage of this type of system is that it uses standard, proven, factory tested equipment such as centrifugal / screw chillers or absorption machines.

**Direct expansion of refrigerant** - In this case the air is cooled by direct expansion of refrigerant such as ammonia or R134a, in cooling coils. The type of refrigeration system can be single stage / cascaded vapor compression system with liquid overfed air cooling coils. It is also possible to have multi stage cooling thereby consuming lesser power consumption per ton of refrigeration.

The advantages of this system over mechanical chillers / absorption machines are :

- Eliminating the auxiliary power consumption in circulating the chilled water, as the power consumption in circulating refrigerant is substantially lower.
- The heat rejection duty in this case is substantially lower than absorption machines, thereby saving on cooling tower and cooling water pumping costs.

The disadvantages are, in case of accidental leakage of ammonia it could affect the down stream equipment. The compressor systems also require electric power to drive the compressor performance because of pressure drop in the cooling coils.

### Direct Cooling:

**Evaporative Cooling Systems** - Evaporative cooling works on the principle of reducing the temperature of an air stream through water evaporation. The process of converting water from a liquid to a vapor state requires energy. This energy is drawn from the air stream, the result being cooler and more humid air. The effectiveness of an evaporative cooling system depends on the surface area of the water exposed to the air stream and the residence time. Conventional media type evaporative coolers use a wetted honeycomb like medium to maximize the evaporative surface area and cooling potential. However this has several drawbacks, such as the media cause a pressure drop in the inlet air duct as well as the installation requires substantial inlet air ducting modifications and the amount of cooling that can be achieved can be fairly small in humid climates.

Fig. 1 Tube array containing impaction pin fog nozzles

**High pressure fogging systems** - It is a more recent technology employed in inlet aircooling. It is similar to evaporative cooling but instead of using water as an evaporative medium, the water is atomized into billions of super-small droplets thereby creating a large evaporative surface area. In these systems, the evaporative surfaces are the frog droplets themselves. See **Figure 1**. For this reason the size of a droplet generated by the fog system is a critical factor. For instance water atomized into 10 micron droplets yields ten times more surface area than the same volume atomized into 100 micron droplets. A water droplet less than 40 microns is a fog and over 40 microns it is called mist.

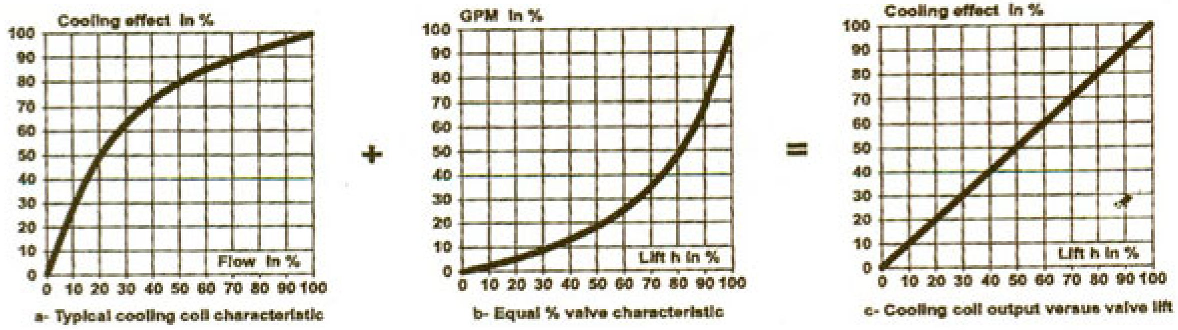

Fig 2: Combination of cooling coil characteristic and equal % control valve to obtain theoretical linear cooling transfer.

Fog systems use high pressure water pumps to pressure demineralised water to between 70 to 210 bar. The water then flows through a network of stainless tubes to fog nozzle manifolds that are installed in the air steam. In order to make droplets small enough to create the fog, impaction pin nozzles are normally used. See **Figure 2**. These nozzle orifices have diameter of 0.0006 in (152 microns) and produce fog droplets in the 3 to 30 micron range.

These nozzles atomize the water into micro-cine fog droplets which evaporate quickly thereby cooling the inlet air. Other factors being equal, the speed of evaporation of water depends on the surface area of water exposed to the air.

Another interesting development is "overcooling". In overcooling more fog is injected into the air stream than can be evaporated. Un-evaporated fog droplets are carried into the first stages of the turbine compressor section, where the air is hot due to the work of compression. Higher temperatures increase the moisture holding capacity of air so the fog droplets that would not evaporate in the inlet air duct, do so in the compressor. Once the fog evaporates in the compressor, it cools and makes the air more dense. This accelerates the total mass flow of air through the turbine giving an additional power boost.

The limits of fog overcooling have not been fully investigated

**Chilled water air washer cooling systems** - In this case the air is cooled by bringing it in direct contact with chilled water in an air washer. As the pressure drop in the air stream is minimal, there is no significant penalty on the performance of the GTG. Further as the air is in direct contact with chilled water, the temperature of air leaving the air washer is very close to the outlet chilled water temperature. Here also the chilled water is produced using a vapor compression refrigeration cycle or absorption cycle.

In all types of direct cooling the quality of water, an regards contaminants, needs to be controlled very accurately e.g. the total maximum limit on Na + K ions which can be tolerated, from all sources, for aero-derivative gas turbine is of the order of 0.1 ppm. Hence extremely pure DM (demineralised) water is required.

There is also the danger of carry-over of bigger water droplets / moisture in the compressor section, which could cause damage to the compressor section of the gas turbine. Larger droplets could have enough mass to damage the compressor blading due to liquid impaction caused by impaction of water droplets.

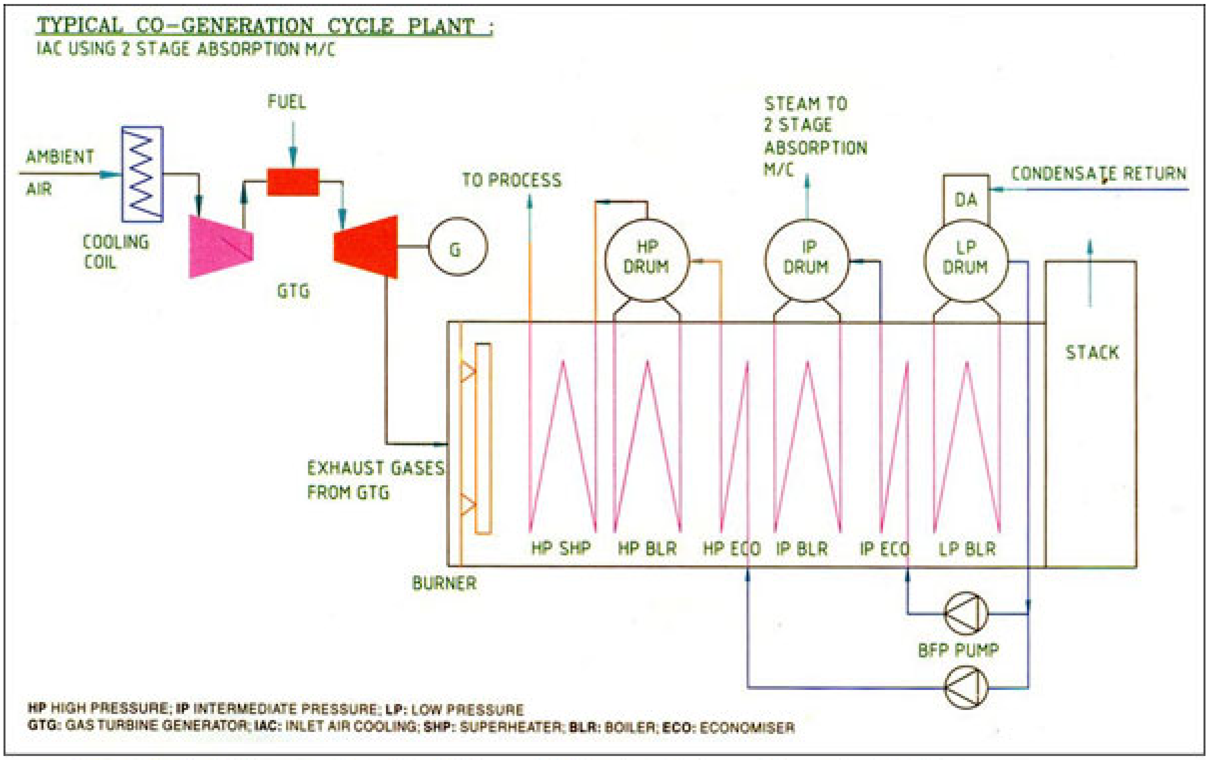

Fig 3 Typical Co-generation Cycle Plant: IAC using 2 Stage Absorption Machine

## Case Study 1

### Performance of the LM 6000 PC on Co-Generation mode

The performance of LM 6000 PC gas turbines of General Electric Company Co was studied in cogeneration mode under the following typical requirements of a refinery:

Net power 40 to 44 MW, maximum possible.

Net exports stream 120 tph

Steam pressure 115 kg / cm$^2$g

Steam temperature 515°C

**Table 2 : Performance of Co-Generation plant using different types of Chillers**

| Type of chiller | | Option 1<br>Mechanical | Option 2<br>Double stage absorption machine | Option 3<br>Single stage absorption machine |
|---|---|---|---|---|
| Gross Power output, at 10°C inlet air | kW | 44,650 | 44,650 | 44,650 |
| Auxilary power consumption,* | kW | 2,987 | 1,356 | 1,465 |
| Net Power output at 10°C inlet air | kW | 41,663 | 43,294 | 43,185 |
| Fuel fixed in GTG | tph | 9.843 | 9.843 | 9.843 |
| Fuel fixed in duct burner | tph | 5.317 | 5.89 | 6.374 |

| | | | | |
|---|---|---|---|---|
| Total fuel fired Co-generation efficiency | % | 86.87 | 84.59 | 82.00 |

**Auxiliary power consumption is the power required to drive all the auxiliary equipment necessary for running of the plant.*

The system configuration considered is, one LM6000 PC gas turbine with inlet air cooled to 10°C, with performance of gas turbine as per **Table 4**, and one fixed HTSG. Based on the site conditions, the refrigeration load worked out to 2250 TR. To bring down the temperature of air to 10°C, it was proposed to use chilled water at 5°C, at inlet of cooling coil, installed in the air path of the gas turbine. The temperature of water leaving the cooling coil was considered as 13°C.

The chilled water was proposed to be produced using either:

**Option 1** : Mechanical Chiller - The HRSG shall be dial pressure level HSRG (HP - 115 kg/cm$^2$g) with integral deaerator. The steam required for process will be provided from HP level of HSRG. The net power available for export will be gross power generated minus the power for driving the mechanical chiller compressor motor.

**Option 2** : Double stage absorption machine - The HSRG shall be triple pressure level HSRG, ( HO - 115 kg/cm$^2$g, LP - 10 kg/cm3 g) with integral deaerator. The steam required for process will be provided from HP level of HSRG. The steam required for single stage absorption machine will be taken from the LP drum at 10 kg/cm$^2$g.

**Option 3** : Single stage absorption machine - The HSRG shall be dual pressure level HSRG, ( HP - 115 kg/cm$^2$g, LP- 3 kg/cm$^2$g) with stand alone deaerator. The steam required for process would be provided from HP level of HSRG. The steam required for single stage absorption machine will be taken from the LP drum at e kg/cm$^2$g.

Please refer to **Figure 3** for system configuration under Option 2 above.

The results of the analysis are presented in **Table 2**.

## Comment and Conclusions:

1. By using mechanical chillers for inlet air cooling, the net power output from the co-generation plant (41,663 kW) is less than that using double stage vapor absorption machines (43,294 kW) by 3.76% and 3.5% respectively.

2. The co - generation efficiency of the plant using mechanical chillers (86.87%0 is higher than that using double stage vapor absorption machines (84.59%) as well as using single stage vapor absorption machines (82%) by 2.69% and 5.9% respectively.

3. If the net power, from the plant using mechanical chillers (viz. 41,663 kW,) meets one's requirement, then it is advantageous to select mechanical chiller as the option for

inlet air cooling. This will ensure minimum running fuel costs.

4. If the net power requirement is higher than 41,663 kW, then double stage absorption machines option (viz. 43,185 kW) should be selected. However this shall have a higher running cost.

## Case Study 2

### Performance of the LM 6000 PC in Combined Cycle mode

The performance of the LM 6000 PC was studied in a combined cycle mode. The system configuration considered is one LM6000 PC gas turbine with inlet air cooled to 10°C, and one unfired HSRG. The steam generated in the HSRG will be admitted in the steam turbine to produce power. The steam turbine will operate in a sliding mode of operation.

The air will be cooled to 10°C, using chilled water, before it is admitted into the gas turbine. Based on the site conditions, the refrigeration load worked out to 2250 TR. To bring down the temperature of air to 10°C, it was proposed to use chilled water at 5°C, at inlet of cooling coil, installed in the air path of the gas turbine. The temperature of water leaving the cooling coil was considered as 13°C.

The chilled water was proposed to be produced using either.

**Table 3 : Performance of Combined Cycle plant using different types of Chillers**

| Description<br>Type of chiller | Units | Option 1<br>Mechanical | Option 2<br>Double stage absorption | Option 3<br>Single stage absorption |
|---|---|---|---|---|
| GTG output | kW | 44,650 | 44,650 | 44,650 |
| STG output | kW | 12,720 | 11,110 | 9,944 |
| Total gross power | kW | 57,370 | 55,760 | 54,594 |
| Auxiliary power Consumption | kW | 3,034 | 1,289 | 1,206 |
| Net power | kW | 54,336 | 54,471 | 53,393 |
| Net heat rate | kcal/kWh | 1,831.9 | 1,827.3 | 1,864.2 |

**Option 1** : Mechanical Chiller - The HSRG will be triple pressure type (HP- 64 kg/cm$^2$g, 510°C and LP 6 kg /cm$^2$g, 250°C ) with integral deaerator. The HP steam generated in the HRSG will be admitted into the condensing steam turbine. The LP steam

generated will also be rejected into the steam turbine. Mechanical chillers will be used for producing chilled water necessary for inlet air cooling.

**Option 2** : Double stage absorption machine - The HRSG will be triple pressure type (HP-64 kg/cm$^2$g, LP- 10 kg/cm$^2$g ) with integral deaerator. The HP steam generated in the HRSG will be admitted into the condensing stem turbine. A part of LP steam generated in HRSG will be injected into the steam turbine and part used for double stage absorption machine to produce chilled water necessary for inlet air cooling.

**Option 3** : Single stage absorption machine - The HRSG will be triple pressure level type (HP- 64 kg/cm$^2$ g, LP-6 kg/cm$^2$ g ) with integral deaerator. The HP steam generated in the HRSG will be admitted into the condensing steam turbine. The LP steam generated ill also be injected into the steam turbine. The steam required for single stage absorption machine will be taken from the steam turbine by having a controlled extraction at 3 kg/cm$^2$g.

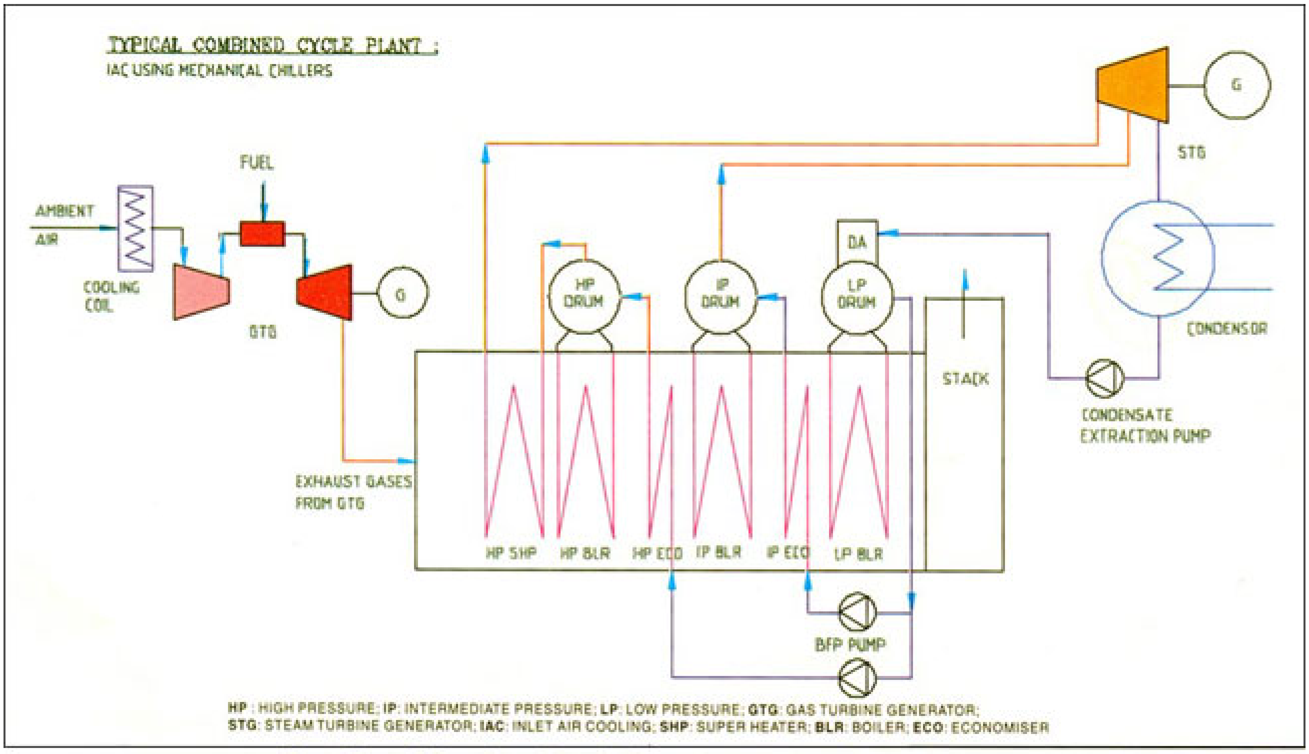

Fig 4 Typical Combined Cycle Plant : IAC using Mechanical Chillers

Refer to **Figure 4** for system configuration under Option 1 above.

All the above options were compared for the net power and heat rate and the results of the analysis are as shown in **Table 3**

## Comments and Conclusions:

1. The net power generated using mechanical chillers or double stage absorption machines is almost the same.

2. The net power generated using single stage absorption machines is less than when using mechanical chillers by (54,336-53,393=943 kW) i,e., approximately 1.73%

3. The net heat rate in using mechanical chillers or double stage absorption machines almost the same.

4. From this analysis it is evident that whether one uses mechanical chillers or double stage absorption machines the net power and heat rate is almost the same.

**Table 4 - Assumptions made for Case Study 1 & 2**

| | | |
|---|---|---|
| Outside design conditions | 35° C dry bulb temperature<br>60% relative humidity | |
| Cooling water temperature | 32°C | |
| **Performance of LM 6000 PC Gas Turbine generator.** | | |
| Site elevation | m | 0 |
| Inlet loss in duct | mm of wc | 140 |
| Outlet loss in duct | mm of wc | 250 |
| Inlet temperature to GTG | °C | 10 |
| Inlet RH to GTG | % | 95 |
| GTG fuel: | | Distillate |
| Net calorific value | kj/kg | 42,332 |
| Gross power output | kW | 44,650 |
| Heat rate | kcal per kW-h | 2,229 |
| Exhaust gas flow | kg/sec | 131.5 |
| Exhaust temperature | °C | 443 |
| Fuel flow | tph | 9.843 |

*Notes*: *Co- generation efficiency is defined as ratio of (total of net power output + net steam output) divided by net heat input. The power required for mechanical chillers is considered as 0.6 kW per ton of refrigeration, i.e., COP of 5.0*

*The specific steam consumption for double stage absorption machine is considered as 4.6 kg/hour / ton of refrigeration at a pressure of 8.5 kg / $cm^2$ g at inlet to machine i.e., COP of 1.16.*

*The specific steam consumption for single stage absorption machine is considered as 8.2 kg / hour / ton of refrigeration at a pressure of 1.5 kg / $cm^2$g at inlet to machine ie COP of 0.58 That* *above analysis has been done using GTPRO software of Thermoflow Inc USA.*