45th ASME/IGTI Congress
ASME TURBO EXPO 2000
May 8 - May 11, 2000 Munich Germany

ASME 2000-GT-0166

# AN ERICSSON CYCLE GT DESIGN BY LNG CRYOGENIC HEAT UTILIZATION

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## ABSTRACT

In many LNG receiving terminals worldwide, the cryogenic heat of imported LNG which was liquefied by using 10% energy of natural gas supply[1), 2)], has been wasted into the sea water mainly through heat exchangers like ORVs (Open Rack Vaporizer)[3)]. This cryogenic heat of 110 K (-256 F) class is considered, however, as an excellent energy source to apply thermodynamic cycles. Several literature, accordingly, are found to improve such high-grade energy potential of LNG regasification process as a low temperature sink, combining with fired heater at 1,100 K (1520 F) class or GT main exhaust gas at 700 K (800 F) class as a high temperature source, through Brayton and Rankine cycles[5), 6), 7), 8), 9)].

This paper presents a typical example of closed "Ericsson" cycle which has the minimum cycle temperature of 157 K (-176 F) from LNG cryogenic heat and the maximum of 550 K (531 F) from the partial HRSG exit heat mixed with the partial GT exit gas. This closed gas turbine, from viewpoints of minor modification to existing power plants and no energy impacts for high temperature source, which would be better than the above-described idea, is able to offer 35% thermal efficiency. And it is recognized that this system would be superior to existing cryogenic generation systems of 20% class operated by Rankine Cycle.

## NOMENCLATURE

| | |
|---|---|
| GEN | Electric Generator |
| GT | Gas Turbine |
| HEXH | High Temp. Exchanger (Pre-Heater) |
| HEXM | Medium Pressure Reheater |
| HEXL | Low Pressure Reheater |
| HPC | High Pressure Comp. |
| HPT | High Pressure Turbine |
| HRSG | Heat Recovery Steam Generator |
| LEXL | Low Temp. Exchanger (Pre-cooler) |
| LEXM | Medium Pressure Intercooler |
| LEXH | High Pressure Intercooler |
| LPC | Low Pressure Comp. |
| LPT | Low Pressure Turbine |
| MPC | Medium Pressure Comp. |
| MPE | Medium Pressure Evaporator |
| MPT | Medium Pressure Turbine |
| PW | Shaft Horsepower at GT Coupling |
| RHEX | Regenerator |

Numbering suffix

| | |
|---|---|
| 2 | Compressor Inlet(LP Compressor Inlet) |
| 24 | LPC Outlet |
| 25 | MP Compressor Inlet |
| 27 | MP Compressor Outlet |
| 28 | HP Compressor Inlet |
| 3 | Compressor Outlet(HP Compressor Outlet) |
| 31 | Regenerator Inlet |
| 32 | Regenerator Outlet |
| 34 | Pre-Heater Inlet |
| 4 | Pre-Heater Outlet |
| 41 | Turbine Inlet(HP Turbine Inlet) |
| 42 | HP Turbine Outlet |
| 44 | MP Turbine Inlet |
| 46 | MP Turbine Outlet |
| 49 | Power Turbine Inlet(LP Turbine Inlet) |
| 8 | Turbine Outlet(PWT Outlet) |
| 91 | Regenerator Inlet |
| 92 | Regenerator Outlet |

## INTRODUCTION

The gaseous fuel with minimal contents of sulfur compounds, solid particles etc. looks so far suitable for diminishing pollutants, and it is an ideal fossil fuel to decrease carbon dioxides emission by lower carbon composition as gaseous conditions.

The natural gas, sharing major parts of gaseous fuel supply, is treated as sweet gas to eliminate corrosive sour materials like sulfur dioxides prior to the liquefaction process followed by export, and accordingly, the imported LNG is very pure hydrocarbon fuel, in which the methane being normally 80% or higher percentage.

The liquefaction energy is counted as approximately 10% of the natural gas heat value[1), 2)], however, the imported LNG is normally gasified through ORVs(open rack vaporizers) [3)] by using the sea water and the considerable cryogenic heat is wasted into the sea.

There are some examples of the cryogenic generating plants mainly applying expansion turbines at the downstream of LNG pumps, which means the reduction of wasting energy by approximately 20%[4)].

The other example is to pre-cool the intake air and this concept looks attractive to recover the power shortage in summer time, however, it is hard to apply in cold winter time. And it might be inappropriate for the high potential exergy of 110 K (-261 F) to be applied in the ambient, due to a big temperature jump.

In this paper, the advantage of the closed cycle incorporation to LNG burning GT/ST combined cycle power plants is to be analyzed.

The combined cycles with the maximum temperature of 1,473 K (2,200 F) class have been holding the thermal efficiency of 52 - 58% as LHV basis. And now tried 1,773 - 1,973 K (2,700 - 3,100 F) class GTs offer the resultant cycle efficiency of 60 % or higher.

LNG receiving terminals are, normally, adjacent to large scale power plants and the utilization of LNG cryogenic heat shall be focused, from view points of energy and environmental savings.
Key factors for successful LNG heat utilization are as follows.

1) No fuel inputs to avoid additional exhaust emissions
2) High efficiency operation by the better cycle

The merits and advantages of closed cycle plants have been discussed for various applications[10), 11)]. And the preliminary parametric study for "Ericsson Cycle" was conducted for LNG cryogenic power generating application[12)].

When considering the heat sink of 110 K (-260 F) class by LNG and the supposed heat source of 600 K (621 F) class, the application of "Ericsson" closed cycle might be appropriate. This system with 157 K (-176 F) and 550 K (531 F) can offer better efficiency than that of steam bottoming cycle of the GT/ST combined cycle plant.

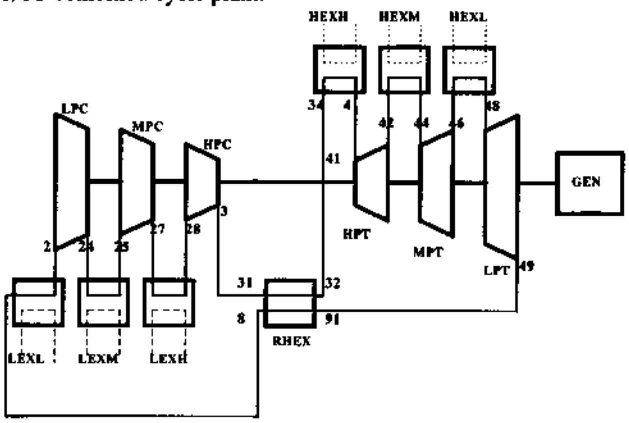
## ADVANTAGE OF CLOSED GT SYSTEM

The theoretical efficiency of Ericsson cycle shall be equivalent to that of an ideal Carnot cycle and be defined only by maximum /minimum temperature ratio. When adopting min. of 157 K (-176 F) and max. of 550 K (531 F), this would be 71%.
In the case of widely used open Brayton cycle, where the maximum temperature would be 1,473 K class (2,200 F), the efficiency would be 45% even in an ideal isentropic compression/expansion.

The proposed flow schematic and material/heat balance are shown in Fig.1 as below for the closed loop gas turbine "Ericsson" system with 157 K (-176 F) and 550 K (531 F).

The adoption of three-stage compression and expansion shown is proposed to meet the isothermal change required by pseudo Ericsson Cycle definition, as a practical approach.

The type of compressors and turbines might be of centri-fugal and -petal type, because of easier ducting capability for side-stream installations, although the efficiencies are less than those of axial flow type.




Main Flow Compression Side

| Station | 2 | 24 | 25 | 27 | 28 | 3 | 31 | 32 | 34 |
|---|---|---|---|---|---|---|---|---|---|
| Pressure (MPa) | 0.250 | 0.687 | 0.673 | 1.853 | 1.816 | 5.000 | 5.000 | 4.900 | 4.900 |
| Temp. (K) | 157.4 | 225.0 | 157.4 | 225.0 | 157.4 | 225.0 | 225.0 | 411.5 | 411.5 |
| Flow (kg/s) | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 |

Main Flow Expansion Side

| Station | 4 | 41 | 42 | 44 | 46 | 48 | 49 | 91 | 8 |
|---|---|---|---|---|---|---|---|---|---|
| Pressure (MPa) | 4.802 | 4.802 | 1.841 | 1.804 | 0.692 | 0.678 | 0.260 | 0.260 | 0.255 |
| Temp. (K) | 550.0 | 550.0 | 437.0 | 550.0 | 437.0 | 550.0 | 437.0 | 437.0 | 250.4 |
| Flow (kg/s) | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 | 116.5 |

Coolant Flow (LNG) / Heating Flow (Exhaust Gas)

| Station | LEXLi | LEXLo | LEXMi LEXHi | LEXMo LEXHo | Station | HEXHi | HEXHo | HEXMi HEXHi | HEXMo HEXHo |
|---|---|---|---|---|---|---|---|---|---|
| Pressure (MPa) | 5.00 | 4.17 | 5.00 | 4.17 | Pressure (MPa) | 0.1023 | 0.1018 | 0.1023 | 0.1018 |
| Temp. (K) | 110.0 | 200.0 | 110.0 | 200.0 | Temp. (K) | 600.0 | 447.0 | 600.0 | 447.0 |
| Flow (kg/s) | 18.3 | 18.3 | 13.3 | 13.3 | Flow (kg/s) | 101.6 | 101.6 | 83.0 | 83.0 |

**Fig. 1 Flow Schematic of Proposed Closed Cycle Gas Turbine**

The one-shaft configuration directly coupled with the electric generator is considered for compact and simple design, as the estimated power output is in the range of 15 - 17 MW.

Several candidates are checked as the working fluid for this system, and the normal dry air is selected from view points of compact design possibility, excellent safety and easy make-up provisions, although the helium gas is usually selected as a working fluid, because of perfect gas characteristics even in very low temperature. The compressibility factor of the dry air in this process is varying from 0.98 to 0. 90, the influence of which is neglected in this study.

Utilized LNG delivery is still in the range of 200 K (-100 F) as listed in Fig.1, which might be applicable to pre-cool the GT intake air, as the cascade heat utilization. Such total system might improve the efficiency of energy usage and also reduce the cold water emission near the LNG vaporizers in receiving terminals.

## STUDY CONDITIONS

### LNG CRYOGENIC HEAT UTILIZATION

The LNG from the storage tanks is pressurized up to 5.0 MPa (725 PSIA) @ 110K (-262 F) and is fed to power plants under 4.0 MPa (580 PSIA) pressure condition, which would be compatible for advanced GT/ST combined cycle plants of 1773 K TIT class.

The T-S and I-S diagrams are shown in Fig. 2a, and Fig. 2b, respectively, and the usable heat capacity is calculated as follows.

- Latent Heat Approx. 176 kJ/kg ( 76 BTU/lb)
- Sensible Heat Approx. 416 kJ/kg ( 179 BTU/lb)

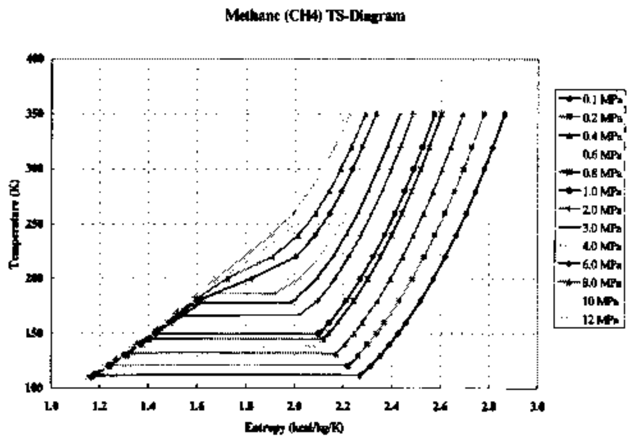



Fig. 2a TS-Diagram of Pure Methane (CH4)

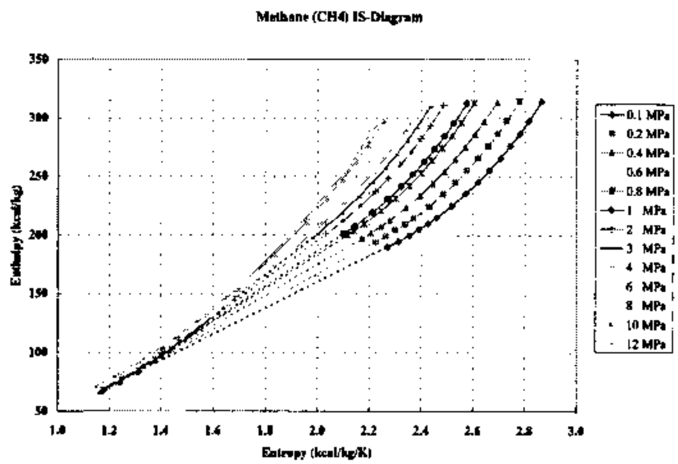



Fig. 2b IS-Diagram of Pure Methane (CH4)

This cryogenic heat might be absorbed by the once-through heat exchanger, the pinch point of which is settled at 10 K (18 F).

This temperature profile shown in Fig. 3a must coincide with the compressor characteristics, where the discharge temperature from the intercooler shall be identical to the suction temperature of the compressor and vice versa. This means automatically to determine the compressor pressure ratio, by assumed polytropic efficiency. It looks difficult to apply the proven technologies of compressors for the gas turbine, in the case of the overall pressure ratio more than 30, and it is recommended to keep the suction temperature more than 157 K (-176 F), the overall pressure ratio being less than 20, i.e., the pressure ratio in the single-casing approximately 2.75.

The low pressure exit air from the regenerator is precooled prior to the suction of the low pressure compressor, where the closed cycle air discharge is kept same to the inter-cooler design to avoid mixing problem of different temperature natural gases at the downstream of this system as shown in Fig. 3b, and the pinch point is more than 20 K (36 F), which corresponds to more conservative design criteria.

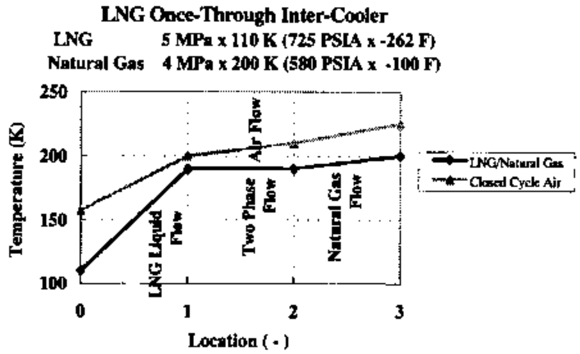



Fig. 3a Temp. Profile in LNG/Air Inter-Cooler

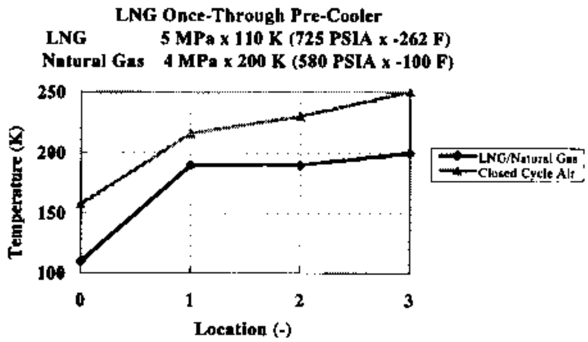



Fig. 3b Temp. Profile in LNG/Air Pre-Cooler

### HOT HEAT UTILIZATION

One potential hot heat source is the atmospheric air or sea water, the temperature of which might not be high enough to make efficient Ericsson Cycle. Other potential hot heat source is the exhaust gas from GT/ST combined cycle plants, the locations of which should be a little bit distant from LNG receiving terminal due to the assured safety reasons. The exhaust temperature might be approximately 373 K (212 F), and the another available hot heat source is the ventilation exit air for GT enclosure, also 373 K (212 F). As the result of preliminary studies, such temperature range might be still ineffective to have efficient cycle figures. The maximum cycle temperature, therefore, is settled as 550 K (531 F), taken into account partly (7 - 8 %) mixing by GT exhaust gas of 773 K (930 F) in case, as more effective heat utilization can be possible than conventional steam bottoming Rankine cycle, shown in Figure 1.

To minimize the radiation losses of hot and cold heat sources, the Ericsson Cycle plant shall be built in the vicinity of receiving terminal rather than GT/ST plant location.

Temperature efficiency of these heaters are counted as 68%, marginal from existing heat exchanger technology.

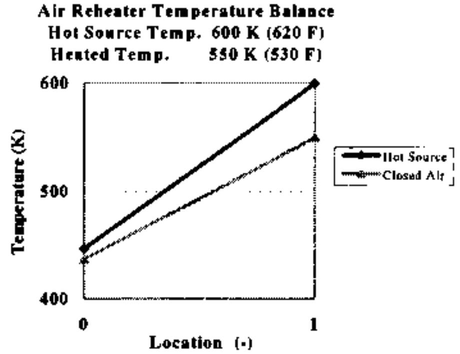



**Fig. 4a Temperature Profile in Inter-Heater**

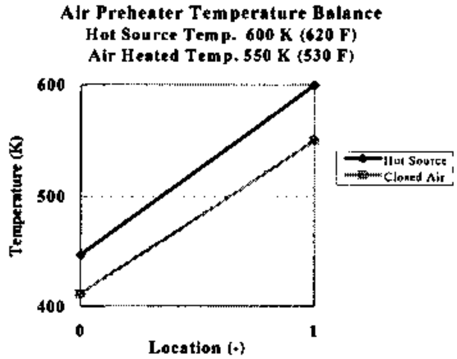



**Fig. 4b Temperature Profile in Pre-Heater**

## REGENERATOR INSTALLATION

The temperature efficiency of the regenerator is settled as 88%, taken into consideration best available heat exchangers technology.

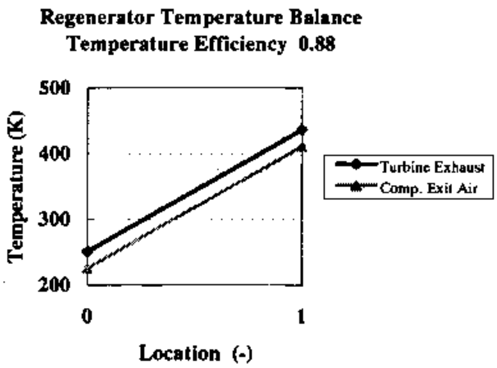



**Fig. 5 Temperature Profile of Regenerator**

## MAXIMUM PRESSURE CONDITIONS

Whatever the pressure ratio is settled by the optimization, the maximum pressure figure of the cycle is to be adopted as 8 MPa, considering the as-is application of existing open cycle gas turbines, high-pressure vessel and pipings design experiences.

## OTHER STUDY CONDITIONS

The parameters are varied to check the performance of practical closed GT cycles, based upon the flow diagram shown in Fig. 1.

| | |
|---|---|
| ① Pressure losses for each pre- and inter-cooler | 2 % |
| ② Compressor polytropic efficiency | 81% |
| ③ Pressure loss for the hot air side of regenerator | 2 % |
| ④ Pressure losses for each pre- and re-heater | 2 % |
| ⑤ Turbine polytropic efficiency | 84 % |
| ⑥ Pressure loss for the cold air side of regenerator | 2 % |
| ⑦ Generator efficiency | 94 % |

## PARAMETRIC STUDY OF ERICSSON CYCLE

As mentioned, the theoretical efficiency of Ericsson cycle shall be equivalent to that of an ideal Carnot cycle, based upon 100% heat exchange between compressor discharge and turbine discharge. And from view points of environmental protection, this closed cycle is superior to open cycles, being free from any pollutants emissions, as the hot heating source has no fuel inputs.

There are no existing power plants by Ericsson Cycle application in view of strict definition. Major reasons are considered as difficulties for applying large scale power generating plants like 100 MW or bigger capacity, due to the necessity of big air ducting for lower pressure sides. Such ducting are unnecessary for open Brayton cycle gas turbine, as the ambient air itself is in low pressure side.

The potential applications of Ericsson cycle might be limited in the small microturbine and/to medium scale capacity class, although very attractive features are pointed out. Before starting the design of closed Ericsson cycle gas turbine, it has to be checked that the status of existing technologies related to components design experiences.

Many experiences of intercooled centrifical compressors are found, including real isothermal type, where coolants are kinds of industrial water. And experiences on reheated turbines are much limited except ABB's proposing this adoption for the performance improvement.

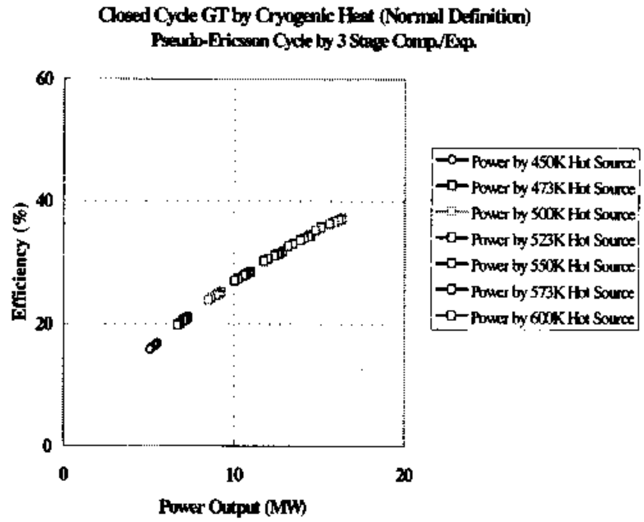



**Fig. 6a Power vs. Efficiency for Various Hot Sources**

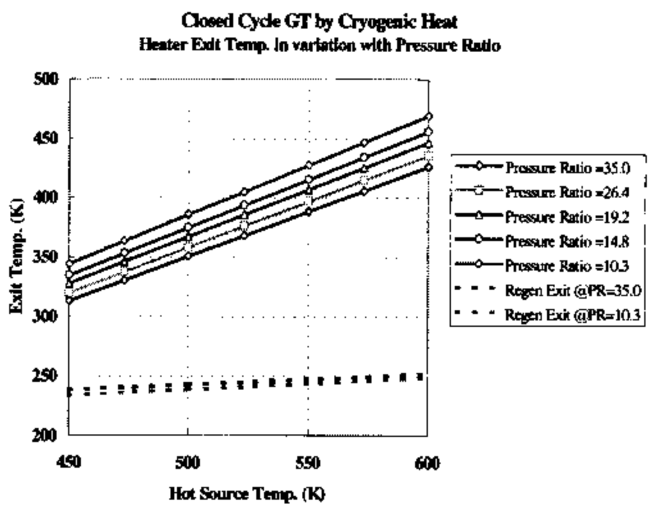



**Fig. 6b Exit Temperature of Hot Source**

The thermal efficiency of Ericsson Cycle , when applying the hot source of 450 - 623 K (350 - 662 F), is shown as in Fig. 6a. This efficiency level is close to that of 1473 K (2200 F) class Brayton Cycle GT plant in simple cycle. The application of Ericsson Cycle with cryogenic heat utilization is based upon the utilization of exhaust heat from the GT/ST combined cycle plant, as described in **STUDY CONDITIONS.** When the hot source of 500 K (440 F) or lower is adopted, the exit temperature of 370 - 380 K (206 - 224 F) would be expected as shown in Fig 6b, similar to the exhaust temperature of the combined GT/ST plants. The thermal efficiency in the range of 23 - 31% might not be excellent enough, as the criterion of this study is that the applied Ericsson cycle shall be superior to conventional steam bottoming Rankine cycle, the efficiency of which is supposed to be in the range of 30 - 35 %.

It can be seen that the system design would be possible, however, far from the optimum design point for TIT of 450 K (350 F) in Fig. 7a and, on the contrary, the optimum design could be easily conducted for any suction temperature range of 150 - 168 K (-190 - -157 F) when choosing TIT of 600 K (620 F) or higher, as in Fig. 7d. And it should be noted that the higher TIT yields in the more flat efficiency curve, meaning not sensitive for design point selection.

To attain the excellent efficiency when applying the Ericsson Cycle plant by using cryogenic heat utilization, therefore, the hot source shall be adopted from the 550 K (530 F) or higher temperature, as shown in Fig. 7a through 7d.

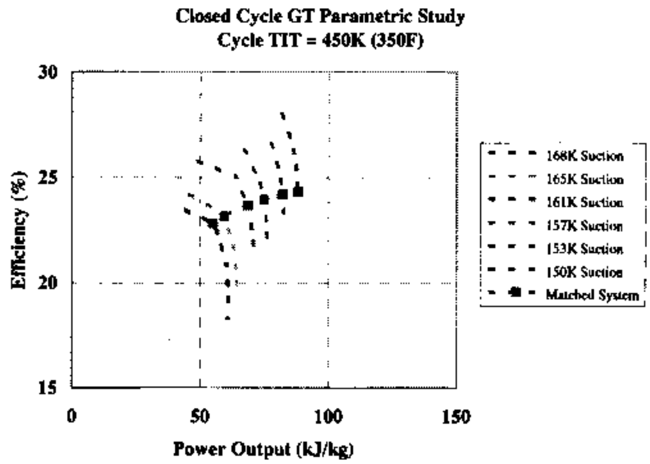



**Fig. 7a Power vs. Efficiency for TIT of 450K**

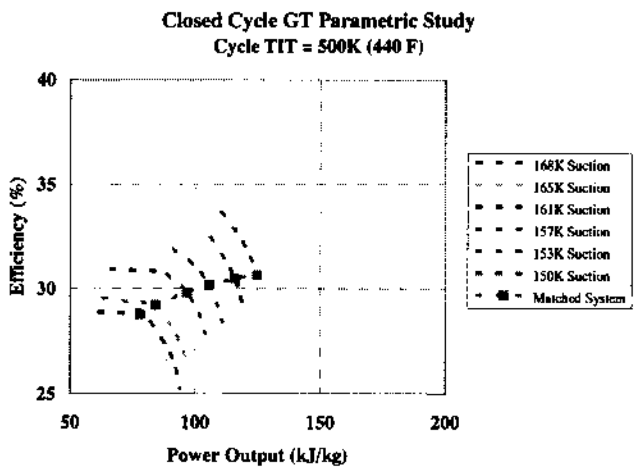



**Fig. 7b Power vs. Efficiency for TIT of 500 K**

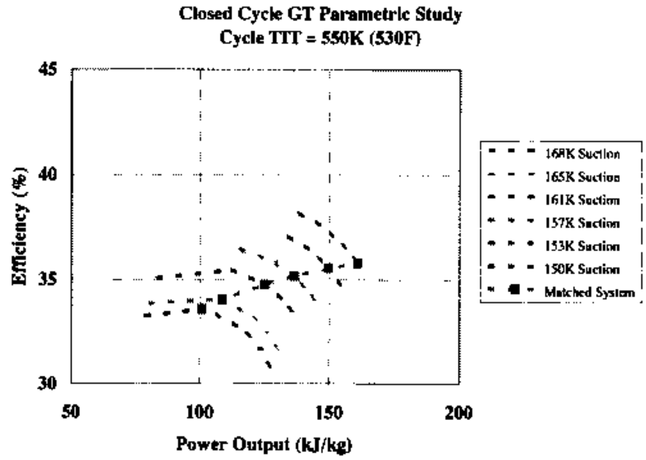



**Fig. 7c Power vs. Efficiency for TIT of 550 K**

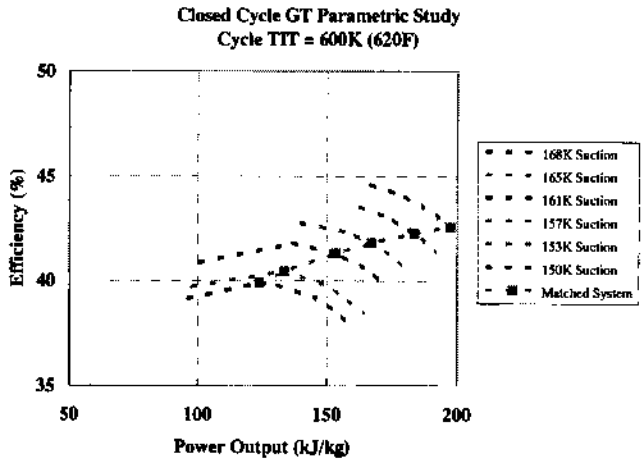



**Fig. 7d Power vs. Efficiency for TIT of 600 K**

There are restrictions for selection of pressure ratio figures, when settling the very same particulars for three identical compressors of LP, MP, & HP stages, as the pressure ratio in each compressor is automatically determined from the inlet/outlet temperature conditions of the pre-cooler and inter-coolers. That is to say, the inlet/outlet temperature figures of coolers correspond to the discharge/suction temperature figures for the compressors and, accordingly, temperature conditions converted automatically into the pressure ratio, subject to practically estimated compressor polytropic efficiency.

This restrictions are shown as dotted line indicated as "Matched System" even in Fig. 7a, 7b, 7c and 7d. Under the conditions of suction temperature over 165 K (-163 F), the matched point can be selected near the optimum efficiency point of Ericsson cycle application. And it can be seen that the matched pressure ratio goes up when the adopted suction temperature goes down, getting far away from the optimum cycle efficiency point. It is noted that the efficiency figure itself goes up in variation with the selection of the lower suction temperature.

It is recommended, therefore, that the suction temperature is to be selected in the range of 155 - 160 K, considering the application of the practical design experience, i.e., the overall pressure ratio less than 20, although the matched points are far from the optimum efficiency point.

Under such adoption of the cycle maximum temperature (TIT) and the cycle minimum temperature (Suction), it would be estimated that the target efficiency level of 35% or higher could be attained for Ericsson Cycle application studies.

## RESULTS OF STUDIES

### SETTLED CLOSED CYCLE GAS TURBINE

The target efficiency of closed cycle GT shall be 35% as minimum, because the existing steam bottoming Rankine cycle have been offering approximately 30% efficiency level. The maximum temperature of 277 C (531 F) would be appropriate, considering GT exhaust gas (723 – 773 K level) mixing with HRSG flue gas (373 – 423 K level). A typical TS chart of this closed cycle is shown in Fig. 8, where the cycle is approaching to Ericsson Cycle. It can be seen that 37% thermal efficiency would be expected. The overall pressure ratio is selected in the vicinity of nineteen (19), and the pressure ratio of an individual compressor is approximately 2.7, which has no difficulty in designing twin stage configuration of centrifugal type. The 37% efficiency might be reasonable with real components and 24% pressure losses in total. In addition, this system can play a role of diminishing environmental impacts.

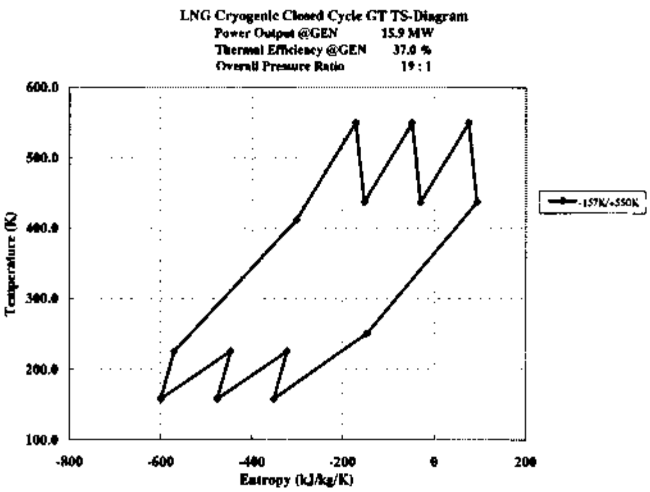



**Fig. 8 TS-Chart of Settled Closed Cycle GT**

### SELECTION OF CYCLE INLET TEMPERATURE

The selection of suction temperature yields in deciding the optimum pressure ratio, in relationship with the cycle maximum temperature. It would be difficult to design the associated hardware in a small scale turbomachinery, when the pressure ratio goes up. The pressure ratio of 35 would be considered as maximum, checking GT technology experiences for industrial purposes of advanced aero-derivative GTs of multiple-shaft type. Accordingly, the suction temperature of 150 K (-190 F), corresponding to such the pressure ratio, might be lowest minimum temperature for this study.

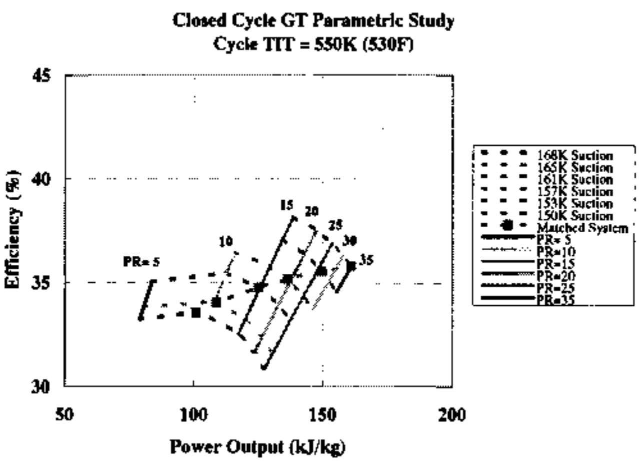



**Fig. 9 Performance Influence in variation with Suction Temp.**

### SELECTION OF CYCLE MAXIMUM PRESSURE

The selection of 8 MPa maximum pressure yields in getting the variation of suction pressure from 0.23 MPa till 1.0 MPa, corresponding to 150 K – 168 K suction temperature range.

The typical example adopts 0.40 MPa as a suction pressure of the cycle, where the designing of rotating machinery might be more compact by eight (8) times than conventional conditions.

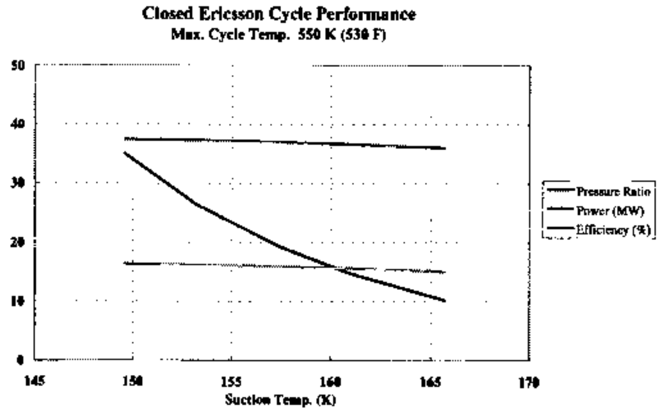



**Fig. 10 Pressure Ratio vs. Suction Temperature**

The other key components are LNG heat exchangers, which are to be replacing existing current ORVs (open rack vaporizers).

The pressure difference is to be very important when designing the flow passages of the heat exchanger, and in this case, the pressure difference between both working fluid is only 3.5 MPa at most, whichever positive or negative pressure inside. This pressure difference levels might be almost half of those in existing ORVs. This would be very advantageous for cutting down the equipment related cost of the plant.

### INFLUENCE OF PRESSURE LOSSES

This influence is significantly big due to eight (8) heat exchangers installation in this cycle and in the case of 2% pressure loss for each passage in the heat exchanger, resultant pressure losses amount to 8%*2=16% in total. In the case of 5% pressure losses for all the heat exchangers, it is very difficult to obtain the target efficiency of 35%.

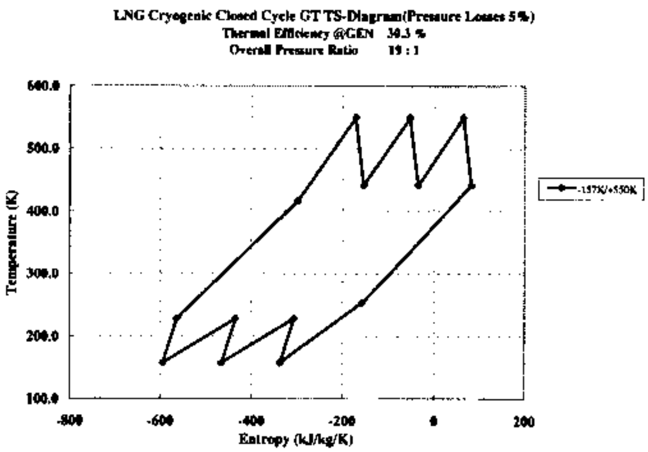



**Fig. 11 TS-Chart for Settled Closed Cycle with 5% Losses**

## CONSIDERATIONS FOR IMPROVEMNT

### POSSIBILITY OF MORE EFFICIENT SYSTEM

An environment-friendly power generating plant could be proposed, by recovering cryogenic heat of advanced LNG-burning GT/ST combined power generating plants. These are of 1,000 MW class generating capacity with thermal efficiency of 50%, the required LNG supply for which is amounting to 45 kg/s (99 lb/s) as LHV of 44,000 kJ/kg (19,000 BTU/lb). The cryogenic heat unit potential is estimated as 820 kJ/kg (353 BTU/lb), and the resultant cryogenic usable are counted to 36.9 MW. The LNG outlet temperature from the precooler and inter coolers is expected to 200 K (-262 F), and accordingly, 68% of potential energy is utilized as follows.

- Utilized Latent Heat 176 kJ/kg
- Utilized Sensible Heat 4.2*(200-110)=378 kJ/kg
- Totally Utilized Heat 554 kJ/kg
- Heat Utilization Percentage 554/820= 0.68

The total utilized heat of 24.9 MW from 36.9 MW cryogenic exergy would be the base for generation efficiency and the expected power output might be 15.9 MW considering the thermal efficiency of 37% at a gas turbine coupling.

The remaining 32% exergy is still usable for the other less efficient system like the pre-cooling system for GT intake air for the 273 K (32 F) estimated outlet temp. , as follows.

- Utilized Sensible Heat 3.5*(273-200)=256 kJ/kg
- Pre-cooling Effectiveness 0.4
- Totally Utilized Heat 256*0.4=102 kJ/kg
- Heat Utilization Percentage 102/820= 0.12

The utilized heat for this pre-cooling system is 4.6 MW, which can have improvement impacts of the gas turbine performance. The air flow for a 1,000 MW GT/ST combined cycle plant is estimated at approx. 2,000 kg/s (4,400 lb/s) at the reasonable specific output of 500 kJ/kg, and resultant pre-cooling temperature difference would be 2.3 K (4.1 F), which yields in performance improvement 1.1% in power output and 0.4% in thermal efficiency, especially effective impacts in summer time to avoid power supply shortage problems.

In this case, it should be noted that the large initial investment to incorporate the pre-cooling system and to install the heat exchanger should be necessary, for example, 800 m2 (8,600 ft2) at the reasonable air flow speed through pre-cooler of 2.5 m/s (8.2 ft/s).

The installation of both systems results in 80% utilization of the cryogenic heat exergy and more environment-friendly influences.

### POSSIBILITY OF HOT HEAT SUPPLY

The required hot gas flow would be 267.5 kg/s (590 lb/s) as indicated in the table of Fig. 1, which corresponds only to 13% of total GT intake air flow.

It might be rather difficult to acquire the 600 K (621 F) hot gas, because the actual exhaust gas temperature for the advanced GT/ST combined cycle plant is in the range of 373 K (212 F).

It is appropriate to by-pass HRSG passage at a specified rate from the GT exhaust of 723 – 773 K (840 – 930 F) main flue gas, to make cryogenic Ericsson cycle plant of 37% efficiency. In the case of say 57% by-passing flue gas with 43% plant exhaust, the mixed gas temperature would be in the range of 573 – 603 K (572 – 626 F).

It should be noted, however, that relatively 57% HRSG by-passing flue gas is equivalent to only 7.6% of GT main flue gas flow and such bleed attainment might be allowable with minor modifications of existing plant lay-out.

If major modifications are possible, it can be seen that the thermal efficiency of Ericsson cycle will go up to 50% level, as shown in Fig. 12, which is much better than most advanced gas turbine Brayton cycle.

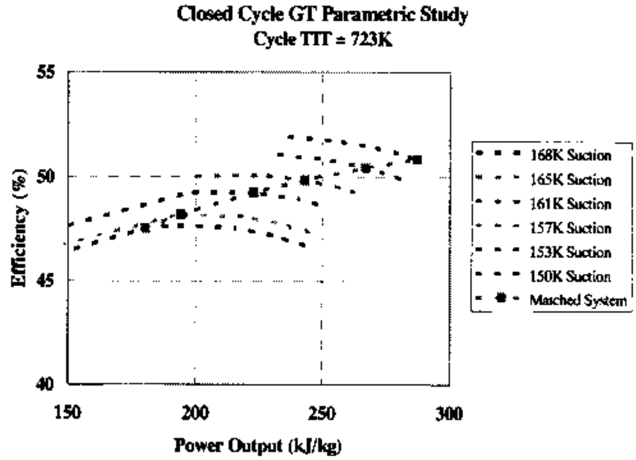



**Fig. 12 High Efficiency in case of 723 K as Max. Temp.**

### HEAT EXCHANGERS DESIGN REQUIREMENTS

The once-through LNG heat exchangers as a precooler and intercoolers shall be in the appropriate range for design conditions.

The logarithmic mean temperature difference for gas/gas passages, which are key parameter for designing the heat exchangers, can be calculated as follows.

$LMTD_{precool}$ = 45.5 K (81.9 F) for the precooler gas/gas passage
$LMTD_{interco}$ = 25.5 K (45.9 F) for the intercooler gas/gas passage
(refer to Fig. 3a and Fig. 3b )

The other key parameter of the pinch point for an once-through heat exchanger is settled as follows.

$Pinch_{precool}$ = 24.2 K (43.6 F).
$Pinch_{interco}$ = 10.0 K (18.0 F)

The LMTD for heaters are as follows.

$LMTD_{preheat}$ = 42.3 K (76.1 F)
$LMTD_{interhe}$ = 24.9 K (44.8 F)
(refer to Fig. 4a and Fig. 4b )

All heat exchangers with outsources are to be within the appropriate design range, and to be of counter flow design, i.e., approximately one third of the flue gas flow flowing for each exchanger.

The LMTD of the regenerator is also in similar conditions with others, derived from settled 88% temperature efficiency, as follows.

$LMTD_{kyo}$ = 25.9 K (46.6 F)
(refer to Fig. 5 )

The design of this regenerator is also to be conducted within the appropriate range.

### ENVIRONMENTAL IMPACTS

The cryogenic heat can be recovered up to 273 K (32 F), combining GT precooling system, which results in offering minimum environmental influences.

The exhaust heat, on the contrary, is cooled down by 383 - 423 K (200 - 270 F), and the resultant exit temperature of 363 - 393 K (194 - 248 F) looks reasonable for stack dispersion, considering the similar stack temperature figures in existing GT/ST plants. The environmental impacts due to exhaust gas emissions would be same as the existing GT/ST combined cycle plants.

The installment of this Ericsson Cycle plant prior to HRSG might be considered as one of improvement idea, although the exit temperature of 20% bled gas is over than 600 K. After this cycle, 80% main gas of 800K and this 20% bled gas could be mixed, and then fed to HRSG for steam bottoming cycle.

This kind of studies has to be conducted to improve more thermal efficiency of thermal generating plant as total.

## CONCEPTUAL DESIGN OF CLOSED CYCLE GT

It might be rather difficult to design the reliable rotating turbomachinery without the plenty feed-back design data and/or operational experiences. The big jump from the existing reliable type is not recommended, as the many and time-consuming verification testing will be mandatory to proceed the conceptual design works.

The gas turbine design like flow cascade, structural design and etc., for matching this proposed small-scale closed cycle is within the verified range in the past. And also the closed loop of working fluid offers the flexibility to choose the suction pressure level freely, which yields in proceeding the design to match the appropriate size.

This study was based upon selection of centrifical type for both compressors and turbines, because of the side-stream capability, the efficiency of which would be 85% at most.

Selected compressor in conceptual design work is of centrifugal type and of three casings with two-side stream, and in each casing, twin series impellers are installed in the size of approximately 710 mm (28 in.) OD.

The selection of axial flow type have had rich experiences for both compressors and turbines mainly in the larger capacity range, and offers much better component efficiency up to 88 – 90%.

## CONCLUSIONS

1) It is ascertained that the application of pseudo-Ericsson cycle with 35 % or higher thermal efficiency for LNG cryogenic heat utilization is more advantageous than existing the cryogenic direct expansion and/or indirect chemical Rankine cycles, without any additional fuel inputs for high temperature energy source.

2) The conceptual design of pseudo-Ericsson Cycle power plant could be presented. And this plant is supposed to be located in the vicinity of both LNG receiving terminal and GT/ST combined cycle power generating plant, in order to get the appropriate heat source and sink.

3) 50 % thermal efficiency can be attainable, in the case of supplying directly of the GT exhaust gas bypassing ST bottoming cycle. It should be noted that the power capacity might be expected to be small, as the cryogenic heat sink is considerably less than exhaust gas heat source (approximately 5 -10 % range).

4) In this study, the application of centri-fugal and -petal turbomachinery is considered. It may be, however, meaningful to make feasibility study to compare the axial flow type with the centrifical type, weighing the efficiency figures as first priority, to aim overcoming the GT/ST combined cycle plant efficiency.

## REFERENCES

1) Sutopo et al. "20 Years' Operating Experience of Steam-Driven Sea water Cooling in the BADAK LNG Plant" 3-1, LNG 12, 1998

2) K. J. Link et al. "Comparison of Baseload Liquefaction Processes" 3-6, LNG 12, 1998

3) M. Sugiyama et al. "The Operation Technology of LNG Terminals of Tokyo Electric Power Company" 5-3, LNG 12, 1998

4) K. Ohnishi "The Power Generation by LNG Cryogenic Heat Utilization" Reference Materials No.9 of GTSJ, Jan., 1981

5) Greipentrog, H., et al. "Vaporization of LNG with Closed Cycle Gas Turbine" ASME Paper 76-GT-38, 1976

6) Krey, G., et al. "Utilization of the Cold by LNG Vaporization with Closed Cycle Gas Turbines" ASME Paper 79-GT-84, 1979

7) G. Bisio et al. "Combined Helium and Combustion Gas Turbine Plant Exploiting Liquid Hydrogen (LH2) Physical Exergy" ASME Trans., J. Eng. for Gas Turbines and Power, April Vol. 118, 1996

8) P. Chiesa "LNG Receiving Terminal Associated with Gas Cycle Power Plants" ASME Paper 97-GT-441, 1997

9) A. F. Agazzani et al. "An Assessment of the Performance of Closed Cycles with and without Hear rejection at Cryogenic Temperatures" ASME Trans., J. Eng. For Gas Turbines and Power, July Vol. 121, 1999

10) C. F. McDonald "A Nuclear Gas Turbine Perspective The Indirect Cycle (IDC) Offers A Practical Solution", No. 96005, IEEE, 1996

11) D. H. Coers "LNG Thermal Storage at Receiving Terminals", GASTECH 98, Nov., 1998

12) K. Hanawa "Closed Gas Turbine Design by LNG Cryogenic Heat Utilization", 115c, AIChE 99 Spring Meeting, March, 1999