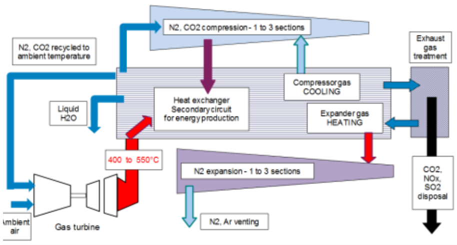
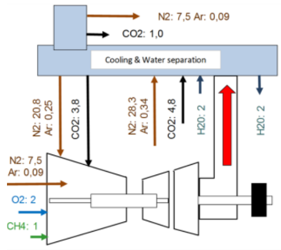
Fossil Fuel Power Plants

CONTENT





1_Gas turbine types

A **gas turbine** or more precisely a **combustion turbine** is a thermodynamic machine with internal combustion. It is used to provide mechanical power. The terminology “gas turbine” is improperly used considering that the machine can burn either a gas fuel (natural gas, butane or a mixture of hydrocarbon gases) or a liquid fuel (from relatively volatile to more heavy components but, more commonly, kerosene).

In the simplest configuration, the combustion turbine operates according to a cycle of Joule, including **four phases**:

- A polytropic compression providing an increase in entropy during the compression process. Elements 1 to 2 on figure 1.
- An isobar heating taking place during the fuel combustion. Elements 2 to 3.
- A polytropic expansion (let down with entropy increase) until ambient pressure. This expansion produces mechanical energy. Elements 3 to 4.
- A constant pressure heat rejection.

This cycle is called the **Brayton cycle**.

The **thermal efficiency** is the ratio between the mechanical energy produced and the combustion heat. This efficiency increases with the compressor pressure ratio (combustion chamber pressure) and the combustion temperature.

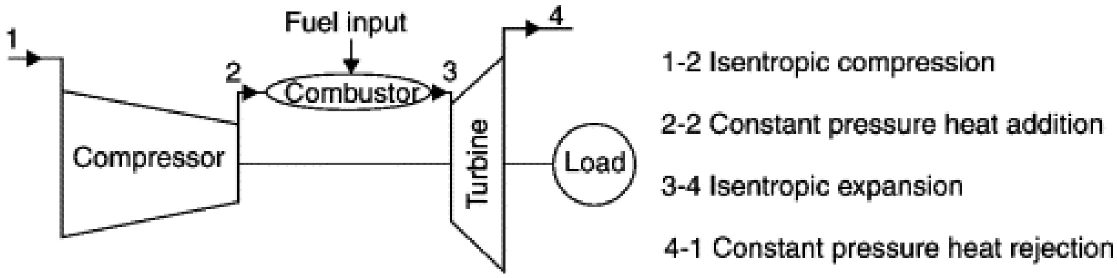


Figure 1 – Schematic representation of a simple gas turbine cycle.

The combustion turbine operates usually in an **open cycle** (gas released to the atmosphere) and with an **internal combustion**. The combustion turbine may also operate in a **closed cycle** with an **external combustion**, heating and cooling being provided by heat exchangers. This last configuration allows the use of specific gases and to operate at a pressure lower than the atmospheric pressure.

In the most frequent configuration (open cycle – internal combustion), the thermal efficiency may be improved in several ways:

- ***Heat recovery at the gas turbine exhaust***. The heat recovered may be transferred through an exchanger to the gas leaving the compression before its admission into the combustion chamber

- ***Compression cooling***. In that instance, the compression includes several stages with intermediate cooling. In that manner, the absorbed compression power is reduced (towards isothermal compression – Less entropy increase).
- ***Expansion heating*** or post combustion or multi stage combustion. In that instance, the expansion includes several stages with intermediate heating. In that manner, the delivered expansion power is increased (approaching an isothermal expansion – Also less entropy increase).

The last two features are justified to combustion turbines with a very high pressure ratio. All these features permit an increase in the turbine efficiency. Their use is however limited due to the complexity to industrialize these thermal processes.

There are two main types of gas turbines differing by their application and mode of construction: the aero derivative and the heavy duty (or industrial) gas turbines.

As suggested by their name, **aero derivative gas turbines** are based on a central core engine derived from the aircraft industry (jet engine) discharging hot gases with a high momentum (backward engine thrust displacing forward the aircraft) towards a last turbine stage usually named Power turbine or **Free turbine**. The aircraft engine may use several shafts connecting separately some compression sections to some expansion sections with similar **specific velocity**, a lower velocity shaft running inside a faster running shaft. In that configuration, the lowest pressure compression section is driven by the lowest pressure expansion section. Same principle is adopted for the highest pressure compression and expansion sections. This allows speed adaptation to the sections with a large diameter running relatively slowly and the ones with a small diameter running relatively fast. These combustion engines may include up to three shafts (also called **spools)** embedded in each other.

*Figure 2 – **Rolls-Royce RB211** Aero derivative gas turbine.*

The Rolls-Royce RB211 (figure 2) engine was the first production of three-spool engine*.* CC BY-SA 3.0, https://commons.wikimedia.org/w/index.php?curid=1416380.

Industrial gas turbines differ from aeronautical designs in that the frames, bearings and blading are of heavier construction. They include only one driving shaft from the extreme compression end (air entrance) to the extreme expansion end (air release to the atmosphere) preventing running speed adaptation to the different turbine elements. As such, the gas turbine efficiency is generally smaller (often, of the order of 30% - Table 1). For this reason, these turbines are often used in a combined cycle (Combined Heat and Power – CHP) to significantly improve the efficiency of the whole system. A **Siemens *SGT5-4000f*** Heavy Duty gas turbine is presented on figure 3.

https://new.siemens.com/global/en/products/energy/power-generation/gas-turbines/sgt5-4000f.html

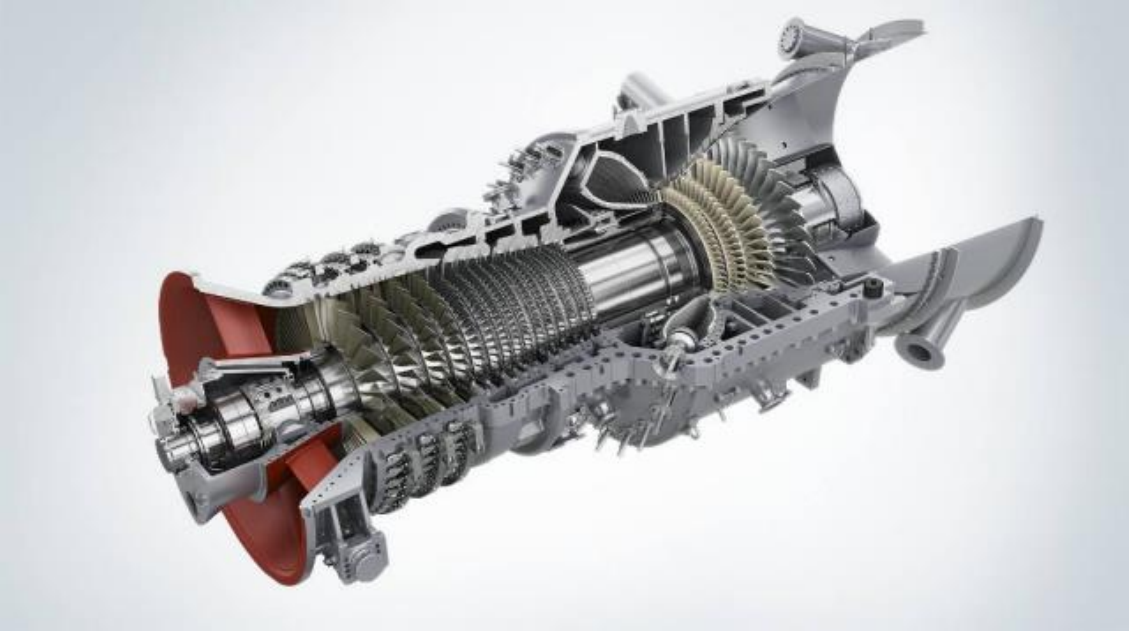


*Figure 3 – **Siemens SGT5-4000f** Heavy duty gas turbine.*

It can be seen from table 1 that the aero derivative turbines are characterized, as an average, by a lower exhaust temperature, 460 °C instead of 535 °C in the case of the heavy duty turbines. This induces a higher thermal efficiency, 39.3 % instead of 32.8% in the case of heavy duty turbines. This also corresponds to a lower heat flux rejection in the case of the aero derivatives. This is the reason why heavy duty gas turbines are often used in a combined cycle.

For the same reason, heavy duty turbines are good candidates for the process system proposed in this document.

The gas turbine G6 (table 1) providing a **mechanical power of 39.7 MW** with an **exhaust temperature of 503 °C** and a **thermal efficiency of 32.5%** is taken as an example in this document. The study is also conducted for **two other values of exhaust temperature, 450 and 550 °C** representing the exhaust temperature range for these two types of turbine.

	ISO CONDITIONS								
Fuel heat rate	Power	Thermal	Exhaust	Exhaust	Exhaust	Fuel	CO2	CO2	
kJ/kg	shaft end	efficency	mass flow	vol. flow	temp.	flow	flow	Ms Frct	
47000	hp	Btu/hp-hr	lb/s	m3/s	°F	-------kg/s-------			
	MW	**%**	**kg/s**	**Mm3/hr**	**°C**	**----MT/year----**		**%**	
F8 1992	34500	6535	186	65	830	1,41	3,87		Average
Aero derivative	25,7	38,9	84	234	443	44	122	4,6	Eff %
L25 1995	37000	6819	183	64	952	1,57	4,33		**39,3**
Aero derivative	27,6	37,3	83	230	511	50	137	5,2	Exhst °C
Rb 1974	39500	6695	208	73	914	1,65	4,54		**464**
Aero derivative	29,5	38,0	94	261	490	52	143	4,8	CO2 Frac.
L60 1992	56130	6370	277	97	836	2,23	6,14		**4,81**
Aero derivative	41,9	39,9	126	348	447	70	193	4,9	
Rt 1996	70470	5993	350	122	802	2,64	7,25		
Aero derivative	52,5	42,4	159	440	428	83	229	4,6	
G5 1972	38000	8700	278	97	960	2,06	5,67		Eff %
Heavy Duty	28,3	29,2	126	349	516	65	179	4,5	**32,8**
G6 1978	53200	7820	302	105	938	2,60	7,14		Exhst °C
Heavy Duty	39,7	32,5	137	379	503	82	225	5,2	**535**
G7 1984	108200	7790	617	215	982	5,26	14,46		CO2 Frac.
Heavy Duty	80,7	32,6	280	775	528	166	456	5,2	**5,27**
Pg1 1987	165482	7531	890	311	1001	7,78	21,39		
Heavy Duty	123	33,8	404	1118	538	245	674	5,3	
Pg3 1991	303740	7136	1327	463	1093	13,53	37,19		
Heavy Duty	226	35,6	602	1667	589	427	1173	6,2	

Table 1 – Examples of aero derivative and heavy duty gas turbines. The name of the gas turbines has been changed. Data from the World Turbine Handbook.

Gas turbines may cause several types of **pollution**:

- They produce a significant amount of **nitrogen dioxides** if the combustion temperature is greater than 1300 °C which comes in conflict with an increase in thermal efficiency requiring a higher combustion temperature. This pollution may be reduced by water vapour injection in the combustion chamber.
- Although **carbon monoxide** is produced in small quantity; it is extremely dangerous to human beings. It is usually reduced by using special injectors.
- **Sulphur dioxides** are produced in smaller quantity by the use of natural gas (usually containing very little sulphur) and liquid fuel with little sulphur content following severe refining treatment.

2_Gas turbine assembly and associated units

As shown in table 1 above, the rejected heat at a gas turbine exhaust is generally 60 to 70 % of the combustion heat, the rest of the heat being converted to mechanical energy to drive an electric generator or any mechanical equipment.

The large quantity of heat usually rejected to the atmosphere is sometimes exploited in a combined system providing water steam.

Alternatively, it could be used to boost the exhaust gas containing a large fraction of carbon dioxide into a gas treatment system operating at high pressure in order to dissolve the sour gas into a physical solvent. The residual heat could also be used to activate a motor cycle. See figure 2.

In order to boost the largest fraction of carbon dioxide into the treatment unit without any entrainment of oxygen molecules, the exhaust gas is recycled (after cooling) inside the gas turbine to provide a stoichiometric combustion (number of oxygen molecules matching the number of hydrocarbons molecules). As a consequence, the gas leaving the turbine exhaust contains mostly nitrogen, carbon dioxide, a small amount of argon and nitrogen oxides and traces of other molecules like xenon and krypton. This is described in section 3.1.

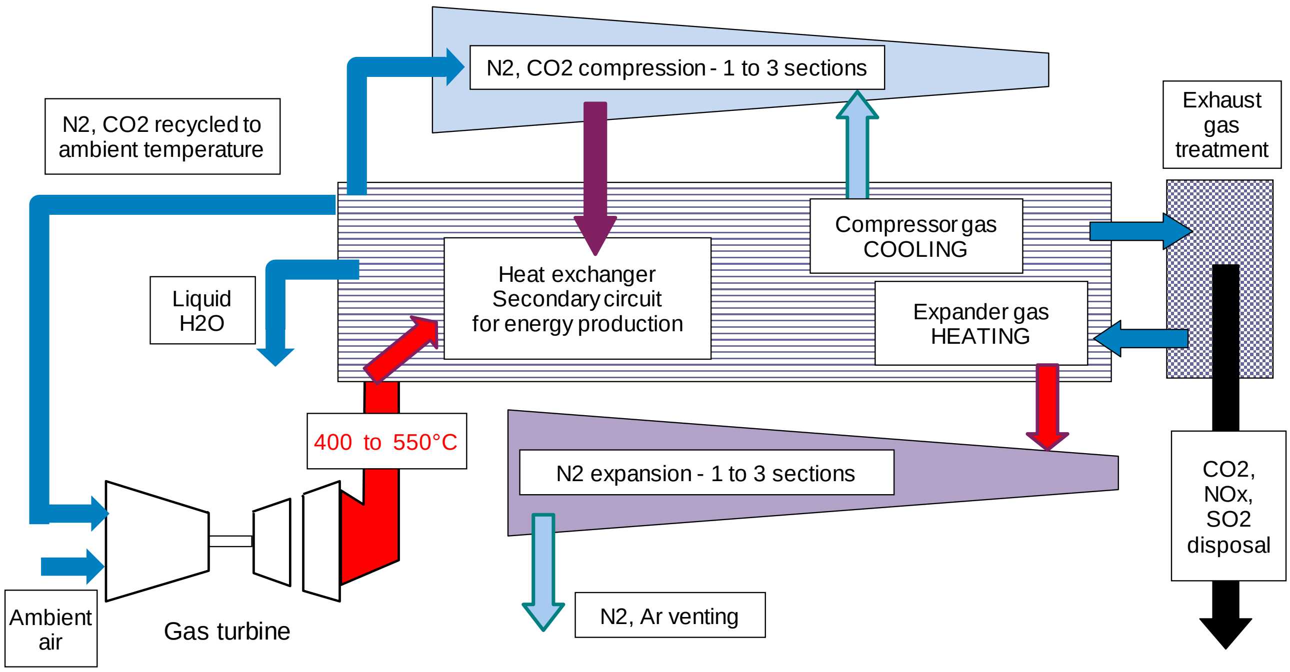


Figure 2 – General assembly for gas turbine with recycling, exhaust gas boosting, sour gas captation and energy recovery

The boosting of the gas fraction not returning to the gas turbine is performed in a compression unit including up to three sections. After gas treatment, the high pressure gas is let down in an expansion unit including also up to three sections, the expander unit driving mechanically the compression unit.

In order that the compression unit absorbs the minimum power, each compressor inlet is cooled down to ambient temperature. To the contrary, in order that the expansion unit provides the maximum power to the compression unit, each expander inlet is heated up to the turbine exhaust temperature. Considering the mass flow rate of each unit (larger for the compression section) and the large temperature difference between the expander and compressor inlets (beneficial to the expansion unit) an equilibrium condition (running speed, pressure and temperature) is reached when the absorbed and supplied power have equalized. This is described in section 3.2.

The gas treatment unit is briefly presented in section 3.3.

3_Sour gas boosting and captation

Sour gases are capted at high pressure and cold temperature by dissolution in a physical solvent as schematically represented in figure 2. The system includes the following elements:

a) A gas turbine with partial recycling of the exhaust gases after cooling;

b) A multi compression unit including up to 4 sections (3 in this document);

c) A multi expansion unit including up to 4 sections (3 in this document);

d) A gas treatment unit operating at the compressor discharge pressure;

e) A heat exchanger unit providing air cooling at each compressor section inlet (40 °C in the present document – A conservative value);

f) A heat exchanger unit providing heating at each expansion section inlet. The heating is provided by the hot exhaust gases.

To summarize, the system includes, outside the gas turbine and treatment units, two main energy based circuits, designated as follows: the **"Primary Circuit"** (sour gas boosting to high pressure followed by let down) and the **"Secondary Circuit"** (a motor cycle providing additional energy to the main turbine) described in section 4. Both the Primary and Secondary circuits are activated by the large heat released at the gas turbine exhaust.

3.1_Gas recycling in the gas turbine unit

In a reciprocating engine (using pistons), **the combustion is stoichiometric**, that is, the number of oxygen molecules match exactly the number of hydrocarbon molecules to get a perfect combustion. When the fuel gas is methane, two molecules of oxygen are required to burn completely one molecule of methane. This in turn provides one molecule of carbon dioxide and two molecules of water. Supplying the oxygen from the ambient air, 3.7 molecules of nitrogen are added to the combustion gases to each molecule of oxygen. At the engine exhaust, other gases may be found in smaller quantities like argon and nitrogen dioxides (as a result of the combustion). See figure 3.1.1 – Left.

Note: The air composition is 78.08 % of nitrogen, 20.95 % of oxygen, 0.93 % of argon, 0.04 % of carbon dioxide (January 2019) and 0.0018 % of neon. The air molecular weight is 28.976 g / mol.

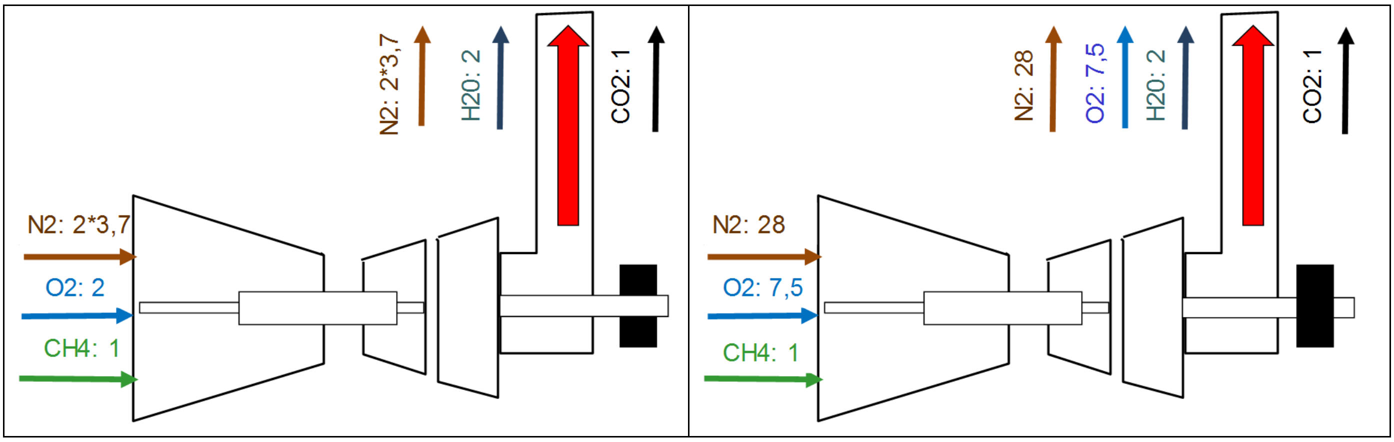


Figure 3.1.1 – Left: Combustion in stoichiometric condition. Right: Combustion with air dilution

Contrary to reciprocating engines, **gas turbines operate with an excess of air**. As an average, the number of oxygen molecules is 3 to 5 times greater than the number required to produce a stoichiometric combustion. The number of carbon oxide, nitrogen oxide and water vapour molecules are unchanged while the number of nitrogen and argon molecules is proportional to the amount of oxygen. See figure 3.1.1 – Right.

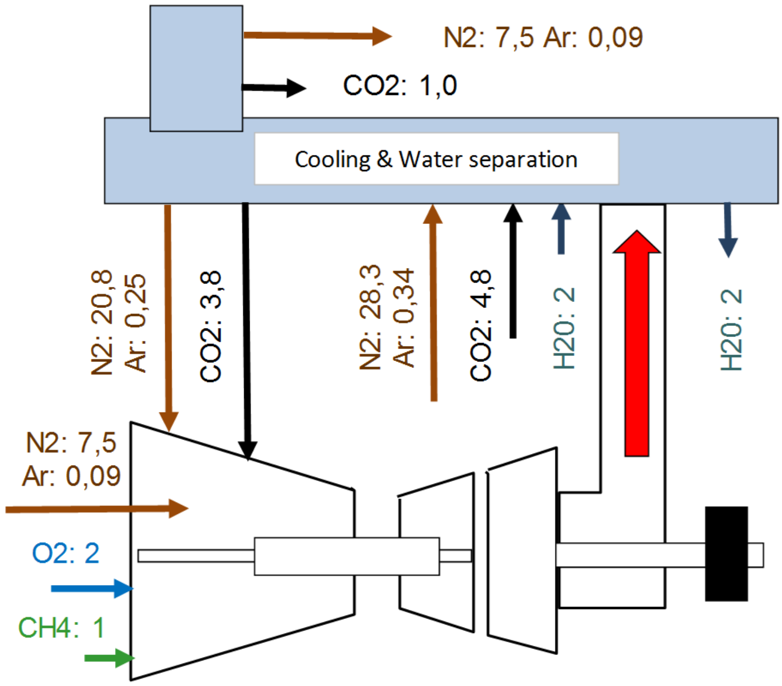


Figure 3.1.2 – Combustion with exhaust gas recycling and dilution at inlet

As stated in section 2, in order to boost the largest fraction of carbon dioxide into the treatment unit without any entrainment of oxygen molecules, a fraction of the exhaust gas is recycled towards the gas turbine inlet. As a consequence, the gas leaving the turbine exhaust contains mostly nitrogen, carbon dioxide and the gases listed in section 2.

The number of molecules participating to the stoichiometric combustion is unchanged (oxygen, methane and water). It should be noted that most of the water vapour condenses after the first air cooling leaving a relatively small fraction of recycled water vapour in the downstream compression - cooling sections. The amount of nitrogen and carbon dioxide entering the compression unit (not recycled) corresponds to a stoichiometric combustion. The amount of recycled nitrogen, argon and carbon dioxide is as represented in figure 3.1.2.

3.2_Compression and expansion units – Primary circuit

3.2.1_Description of the system

The operation of the compression and expansion units may be described as follows.

After cooling in heat exchangers E0 (high temperature exchanger) and E0a (low temperature exchanger) the exhaust gas is directed towards the first compression section (C1). Afterwards, the gas is cooled again (E1 and E1a), compressed (C2), cooled again (E2 and E2a), compressed (C3) then cooled down again (E3 and E3a) before entering the gas treatment unit. See figure 3.2.1.

After heating in heat exchanger E4 (high temperature exchanger) the treated exhaust gas is directed towards the first expansion stage (T3). Following which, the gas is heated again (E5) and expanded (T2) then heated again (T6) and finally expanded (T1) before venting. In the above cases, the heat supplied to the expanders (E4 to E6) is provided by the heat exchanger E0.

The exhaust gas being recycled into the gas turbine (stoichiometric combustion), the gas flowing through the compression units is composed mainly of nitrogen, carbon dioxide and much smaller quantities of argon and nitrogen oxides. During the gas treatment process, sour gases (mainly, carbon dioxide, nitrogen oxides and sulphur oxides if any) are extracted at high pressure from the compressed gas. As a consequence, the flow rate of the expanded gas is smaller than the one of the compressed gas. This lower mass flow rate of the expanded gas ***tends to reduce the rotating speed*** of both the expansion and compression units.

To the contrary, the discharge temperature of the compression units is considerably lower than the inlet temperature of the expansion units indicating that the potential power (enthalpy) of the expansion units is greater than the absorbed power of the compression units ***tending to rise the rotating speed*** of all units.

These combined features indicate that equilibrium is reached when the absorbed power matches exactly the supplied power. To summarise the energy balance mechanism: **The compression unit provides potential energy to the expansion unit while the latest provides mechanical energy to the first.**

For practical reasons, the system cannot reach this equilibrium point when starting from a no speed situation. From that initial configuration, energy must be provided to the system to break solid friction but also to initiate gas compression which, once heated, can generate energy through gas expansion. This energy is provided by a speed – torque controller which may be energetically linked to the main turbine shat end. The inlet temperature of the expansion sections being considerably greater than the compression ones, the system is accelerated. During acceleration, the energy provided by the controller is progressively reduced. It is annealed when energy equilibrium is reached. The controller may be designed to absorb energy in order to decelerate the compression – expansion unit when it is required.

3.2.2_Types of compression and expansion modes

Calculations have been carried out in a semi theoretical case then in a real case.

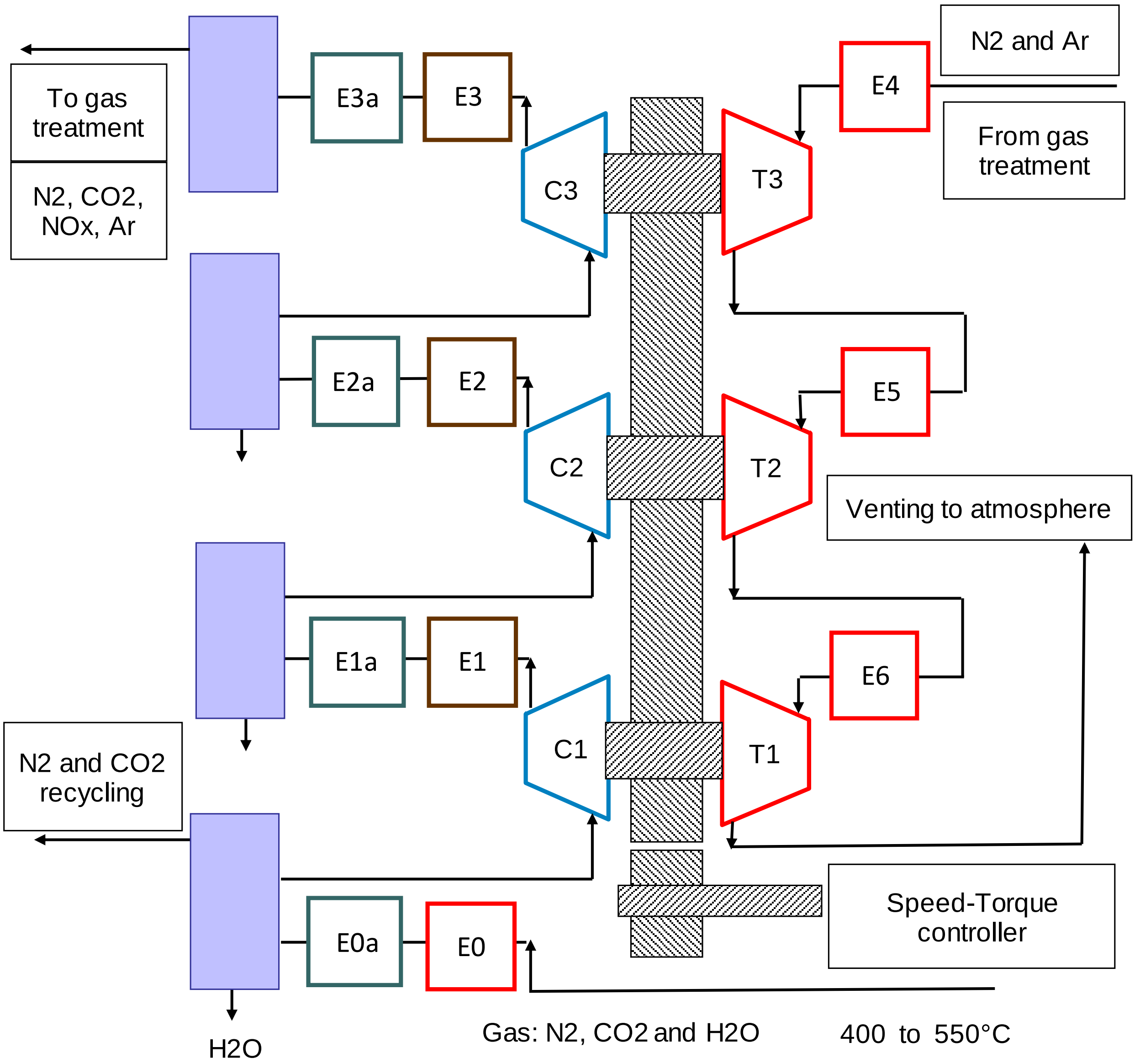


Figure 3.2.1 – Compression and expansion units with heat exchanger assembly

But before these two cases are analysed, a special arrangement needs to be looked at. It is called the "*closed loop case" (a* theoretical one). ***In the closed loop – theoretical case***, losses are totally disregarded. In particular, the compression and expansion processes are adiabatic with efficiencies equal to 100%, intermediate pressure drops and heat losses are nil, the last compressor outlet is directly connected to the first expander inlet while the last expander outlet is connected to the first compressor inlet. Also the heat generated by the compression is entirely transferred to the expansion sections. In this situation, all compression and expansion data match each other: power, rotating speed, temperatures and pressures. This means that the system is unstable and that any condition can be met, in particular, infinite data (power, rotating speed, temperatures and pressures).

This exercise indicates that the closest this condition is approached, case in particular of large adiabatic efficiencies (small energy dissipation), less the unit is stable. In that case, equilibrium is met for large operating values. The system inertia and the controller have to be designed to stabilize the overall unit. To the contrary, when energy dissipation is large, the unit is considerably more stable. In that case, equilibrium is met for lower operating values, therefore, the system inertia and the controller will rapidly stabilize the overall units.

In the semi theoretical case, as per the "closed loop" case, all pressure and heat losses are nil. The semi theoretical case differs from the "closed loop" case only by the polytropic efficiencies which are not nil.

In the real case, pressure losses between the turbine exhaust and the first compression section are assumed equal to 2 % of the absolute pressure. The same rule is applied to the intermediate pressure losses and pressure losses between the final compression section and the gas treatment unit. The same rule is also applied from the gas treatment unit to the expansion outlet.

A hot temperature is reduced by 5 % (relative temperature) to reflect the real operation of a heat exchanger. The water vapour is condensed by 60 % upstream the first compression section, the rest being condensed at the entrance of the second compression section.

For these two cases (semi theoretical and real) same efficiency values are used.

Gas turbine characteristics

Fuel gas: 2.6 kg/s mass flow rate and 47.0 MJ/kg; Combustion heat: 122 MW; Thermal efficiency: 32.5 %; Power at shat end: 39.7 MW; heat rejected: 82.3 MW. This case is considered in the data presented below except where specified.

Fuel $CH4$ or natural gas: 2.6 kg/s mass flow rate and approx. 52.0 MJ/kg; Combustion heat: 135 MW; Thermal efficiency: 32.5 %; Power at shat end: 43.9 MW; heat rejected: 91.1 MW

Compression data: Mass flow rate: 41.6 kg/s; Mol. weight: 30.0; Polytropic eff.: 85%

Expansion data: Mass flow rate: 34.4 kg/s; Mol. weight: 28.1; Polytropic eff.: 90%

3.2.3_Semi theoretical cases

Column	A	B	C
Compression: **Expansion:**	Polytropic Polytropic	Partial Isothermal Polytropic	Partial Isothermal Partial Isothermal
	1 and 2 sections	2 sections	2 sections
Compress. Pd/Ps Expansion Ps/Pd	8.12	20.6	65.8
Compress. Ts&Td	**40** and 324 °C	**40**-226°C each	**40**-324°C each
Expansion Ts/Td	**550** and 232 °C	**550**-305;305-133°C	**550**-232°C each
Compress. Power Expansion Power	12.5 MW	16.3 MW	25.0 MW
C&E power ref. (1)	15.2 %	19.9 %	30.3 %

Note (1): The compression and expansion power is compared to the heat rejected (82.3 MW)

*Table 3.1 – Compression and expansion results for 1 and 2 sections – Intermediate pressure and temperature losses **not taken** into account*

The assumptions taken for the semi theoretical case facilitate the analysis of the calculation results.

It has to be noted that when ***the situation is strictly polytropic***, the same operating condition is met whatever the number of compression and expansion sections (columns A and D). In that instance, the pressure ratio is equal to 8.12 (8.12 bar abs discharge pressure) and the equilibrium power is met at 15 % of the rejected heat or 31 % of the gas turbine mechanical power.

When the ***compression sections operate in a semi isothermal condition*** (section inlet temperatures of 40 °C), the pressure ratio is increased to 20.6 and the balanced power to 16.3 MW (Column B) by doubling the number of sections. The pressure ratio is increased to 35.0 and the balanced power to 18.2 MW (Column E) by tripling the number of sections. The effect of the number of sections is therefore more important on the pressure ratio than on the balanced power with a beneficial effect on the gas treatment unit without a too strong penalty on the mechanical sizing of the units.

When both the ***compression and the expansion sections operate in a semi isothermal condition*** (section inlet temperatures of, respectively, 40 and 550 °C), the pressure ratio is increased to 65.8 and the balanced power to 25.0 MW (Column C) by doubling the number of sections. The pressure ratio is increased to 536 and the balanced power to 37.5 MW (Column F - Roughly gas turbine mechanical power) by tripling the number of sections. Same observation is made as before concluding that the closer the isothermal condition is reached, the greater the discharge pressure. Despite this condition is met for a very large number of sections, an optimized arrangement is probably met for a number of sections not exceeding three.

Column	D	E	F
Compression: **Expansion:**	Polytropic Polytropic	Partial Isothermal Polytropic	Partial Isothermal Partial Isothermal
	3 sections	3 sections	3 sections
Compress. Pd/Ps Expansion Ps/Pd	8.12	35.0	536
Compress. Ts&Td	**40** and 324 °C	**40**-178°C each	**40**-324°C each
Expansion Ts/Td	**550** and 232 °C	**550**-351;351-200; 200-86°C	**550**-232°C each
Compress. Power Expansion Power	12.5 MW	18.2 MW	37.5 MW
C&E Power (1)	15.2 %	22.1 %	45.5 %

*Table 3.2 – Compression and expansion results for 3 sections with main expansion inlet temperature of 550°C – Intermediate pressure and temperature losses **not taken** into account*

Column	G	H	I
Compression: **Expansion:**	Partial Isothermal Partial Isothermal	Partial Isothermal Partial Isothermal	Partial Isothermal Partial Isothermal
	3 sections	3 sections	3 sections
Compress. Pd/Ps Expansion Ps/Pd	536	272	131
Compress. Ts&Td	**40**-324°C each	**40**-284°C each	**40**-244°C each
Expansion Ts/Td	**550**-232°C each	**500**-227°C each	**450**-222°C each
Compress. Power Expansion Power	37.5 MW	32.2 MW	26.9 MW
C&E Power (1)	45.5 %	39.1 %	32.7 %

*Table 3.3 – Compression and expansion for 3 sections with expansion inlet temperature varying from 550 to 450 °C – Intermediate pressure and temperature losses **not taken** into account*

The same exercise has been carried out for ***three different expander inlet temperature values: 550, 500 and 450 °C***, the compression and the expansion sections still operating in a semi isothermal condition. When the temperature is reduced from 550 to 500°C, the pressure ratio is reduced from 536 to 272 (roughly divided by 2) and the balanced power from 37.5 to 32.2 MW (Column H – 14 % less). When the temperature is reduced from 550 to 450°C, the pressure ratio is reduced from 536 to 131 (roughly divided by 4) and the balanced power to 26.9 MW (Column I – 28 % less).

3.2.4_Real case and variation of a few parameters

The above exercise was repeated but **taking into account some losses**, particularly, the pressure drop and heat loss when passing from one section to another. This exercise was carried out for the above three expansion temperature values, the compression and the expansion sections still operating in a semi isothermal condition.

When the temperature is equal to 550°C, consideration for the above losses reduces the pressure ratio from 536 to 222 (divided by more than 2) and the balanced power is reduced to 31.6 MW (Column J – 16 % less).

When the temperature is equal to 500°C, consideration for the above losses reduces the pressure ratio from 272 to 111 (divided by more than 2) and the balanced power is reduced to 26.5 MW (Column K – 18 % less).

When the temperature is equal to 450°C, consideration for the above losses reduces the pressure ratio from 131 to 51 (divided by more than 2) and the balanced power is reduced to 21.2 MW (Column L – 19 % less).

Column	**J**	**K**	**L**
Compression: **Expansion:**	Partial Isothermal Partial Isothermal	Partial Isothermal Partial Isothermal	Partial Isothermal Partial Isothermal
	3 sections	3 sections	3 sections
Compress. Pd/Ps Expansion Ps/Pd	225	111	51.5
Compress. Ts&Td	**40**-273°C each	**40**-235°C each	**40**-196°C each
Expansion Ts/Td	**550**-254°C each	**500**-250°C each	**450**-247°C each
Compress. Power Expansion Power	31.6 MW	26.5 MW	21.2 MW
C&E Power (1)	38.4 %	32.1 %	25.7 %

*Table 3.4 – Compression and expansion for 3 sections with expansion inlet temperature varying from 550 to 450 °C – Intermediate pressure and temperature losses and partial H20 condensation **TAKEN** into account*

The same exercise was repeated **by doubling and tripling the pressure losses** (4 % and 6 % instead of 2 %). This exercise was carried out only for the highest expansion temperature value (550 °C), the compression and the expansion sections operating as previously.

By doubling the pressure losses, the pressure ratio is reduced from 225 to 177 and the exchanged power from 31.6 MW to 29.8 MW.

By tripling the pressure losses, the pressure ratio is reduced from 225 to 132 and the exchanged power from 31.6 MW to 27.7 MW.

We should consider that the present compression and expansion efficiencies are adapted to medium – large gas turbines (between 20 and 50 MW mechanical shaft end). Should gas turbines be used in association or the gas turbine increases significantly in mechanical power (above 200 MW), both compression and expansion efficiencies would increase. The above exercise was repeated **by increasing the compression efficiency** (from 85 % to 90 %). **and the expansion efficiency** (from 90 % to 95 % - this last value is realistic according from actual installations). This exercise was carried out only for the highest expansion temperature value (550 °C), compression and expansion sections operating as previously also pressure losses equal to 2 % of the absolute pressure.

As a consequence, the pressure ratio is increased from 225 to 860 and the exchanged power from 31.6 MW to 47.8 MW. In other words, this increase in efficiency tends to operate the system towards the "closed loop – theoretical" case (section 3.2.1).

In conclusion, exhaust temperatures of 550, 500 and 450 °C tend to provide pressure ratios of the order of, respectively, 200, 100 and 50 representing approximately the pressure level in bars of the gas treatment unit. However, each application should be studied specifically.

3.3_Gas treatment unit with sour gas captation

Following final compression and before entering the first stage of the high pressure turbine, the gas is cooled down and sent to a high pressure separator to dissolve the sour gas in a physical solvent. This solvent may be water, methanol or other types of physical solvent. Fine droplets of solvent are injected into the separator to absorb the sour gas. These droplets then fall down at the bottom of the separator. In a separate chamber, the solvent is heated up to release the sour gas maintaining the sour gas pressure relatively constant. Then the purified solvent is re-injected into the separator for further sour gas contacting.

The mass flow rate of sour gases (mostly, carbon and nitrogen oxides) is 41.58 (compression) minus 34.44 kg/s (expansion) i.e. 7.14 kg/s.

Depending on the exhaust gas temperature, sour gases are treated at a different pressure level requiring different compression power. In the case of a 550 °C exhaust temperature, 5.3 MW are required to boost these gases from 1 to 225 bar abs.

Treatment pressure	Exhaust temperature	Sour gas absorbed power
225 bar	550 °C	5.3 MW
111 bar	500 °C	4.5 MW
51.5 bar	450 °C	3.7 MW

Table 3 – Absorbed power for the compression of sour gases to boost them from the atmospheric to the treatment pressure for three exhaust gas temperatures.

Further details of sour gas treatment with physical solvents may be found in other documents.

4_Secondary system for energy recovery

Heat available at each compression and expansion section outlet and also the residual heat of the exhaust gas (not transmitted to the expansion unit) are used to operate a motor cycle.

> Note: The above is a ***mathematical approach*** where heat is removed at the expander outlet leaving a lower residual heat flux from the exhaust gas for the motor cycle.
>
> - In the ***physical approach*** (real case), no heat is removed at expander outlets leaving a higher residual heat flux from the exhaust gas for the motor cycle. In reality, the outlet of an expander section is heated up and not cooled down before the next expander inlet. Only, the last expander section outlet is cooled down.
>
> - Mathematical and physical approaches provide same results. They have both their justification: The mathematical approach provides some simplicity in the calculation process while the physical approach optimizes the heat exchanger arrangement.

The motor cycle is a classical system including four elements in which a refrigerant fluid circulates in a closed loop. Downstream a pressurizing pump, the liquid refrigerant is vaporized in an evaporator heated externally by a flux of hot gases. The vaporized refrigerant, at high pressure and high temperature, is let down through an expander providing mechanical energy. The gas expansion provides some cooling to the refrigerant. At the expander outlet, the low pressure refrigerant is further cooled down to lower the outlet pressure of the expander and to convert the refrigerant from a gas to a liquid phase. At the condenser outlet, the liquid refrigerant is pumped to a high pressure level in order to repeat a new refrigerant cycle.

The mode of operation of the motor cycle is presented on figure 4 and its performance in table 4.1.

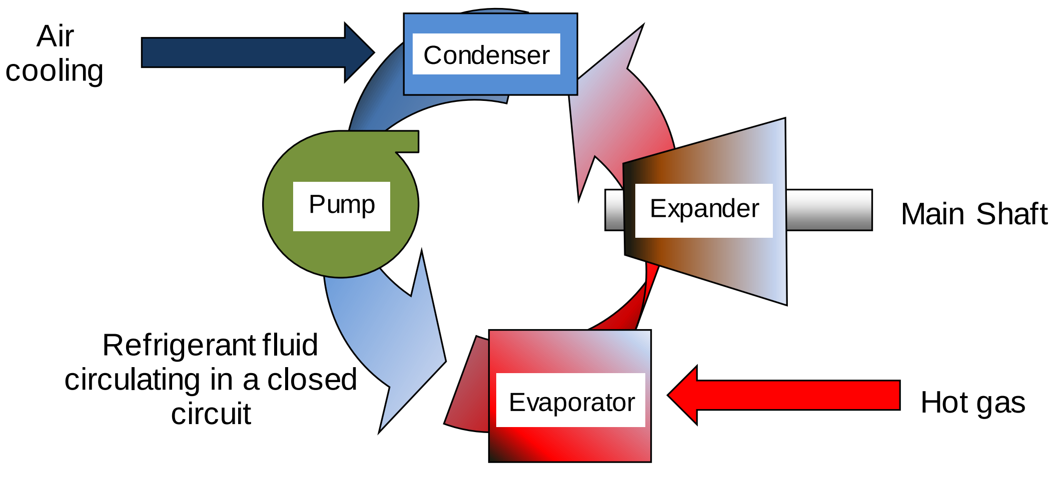


Figure 4 – Schematic of a motor cycle.

Heat is removed from the outlet of the three compression sections handling 41.6 kg/s of a nitrogen – carbon dioxide mixture at 273 °C (see table 3.4). Taking into account heat losses, the temperature is lowered to 259°C. By cooling the gas to 40°C, the potential heat recoverable at each compression section outlet is 9.1 MW totalizing 27.3 MW for the entire compression unit.

Heat is removed from the outlet of the three expansion sections handling 34.4 kg/s of nitrogen (also argon) with 254°C outlet temperature (see table 3.4). Taking into account heat losses, the temperature is lowered to 240°C. By cooling the gas to 40°C, the potential heat recoverable at each expansion section outlet is 7.2 MW totalizing 21.5 MW for the entire expansion unit.

Heat is also provided by the residual heat available from the cooling of the recycled exhaust gas before it re-enters the main gas turbine inlet (figures 2 and 3.1.2).

	Mass kg/s	Temp diff (3)	Heat_MW recoverable	Carnot efficiency	Energy_MW recovered
Compression (1)	41.6 – 3 sections	259 – 40°C	27.3	36.4 %	5.0
Expansion (1)	34.4 – 3 sections	240 – 40°C	21.5	34.0 %	3.6
Residual gas heat (1)	67.3	522 - 40°C	32.5	58.6 %	9.5
					18.1
Exhaust gas Total (1-2)	170.6	522 – 40°C	81.2 (2)	58.6 %	

Note (1): the heat flux from the compressor and expander outlets (3 sections) and the residual heat flux (roughly recycled gas) should match the heat available at the gas turbine exhaust.

Note (2): heat rejected is 81 MW according to above calculation or 82 MW according to gas turbine efficiency, an exhaust temperature of 550 °C and a heat rate of 47 MJ/kg. It is to be compared to 91 MW if a heat rate of 52 MJ/kg is considered.

Note (3): Hot temperatures take into account for some temperature losses

Table 4.1 – Performance of a motor cycle in the case of a gas turbine with 550 °C exhaust gas temperature with a 3 section compression and 3 section expansion system.

The estimated power recovered is based on Carnot efficiency values listed in table 4.1 and by applying a correction factor of 0.5. On this basis the power recoverable is 18.1 MW i.e. 46 % of the shat end power of the main turbine. It is also 22 % of the rejected heat and 15 % of the combustion heat.

	Mass kg/s	Temp diff (3)	Heat_MW recoverable	Carnot efficiency	Energy_MW recovered
Compression	41.6 – 3 sections	223 – 40°C	22.8	31.4 %	3.6
Expansion (1)	34.4 – 3 sections	238 – 40°C	21.5	33.5 %	3.5
Residual gas heat (1)	67.3	475 - 40°C	29.3	55.8 %	8.2
					15.3
Exhaust gas Total (1-2)	170.6	475 – 40°C	73.2 (2)	55.8 %	

Notes (1), (2) and (3): see table 4.1.

Table 4.2 – As table 4.1 but for a gas turbine with 500 °C exhaust gas temperature.

In the case of a 500 °C exhaust temperature, the estimated power recovered is 15.3 MW i.e. 20.6 % of the rejected heat. It is based on Carnot efficiency values listed in table 4.2 and by applying a correction factor of 0.5.

	Mass kg/s	Temp diff (3)	Heat_MW recoverable	Carnot efficiency	Energy_MW recovered
Compression	41.6 – 3 sections	186 – 40°C	18.2	25.4 %	2.3
Expansion (1)	34.4 – 3 sections	235 – 40°C	20.8	33.1 %	3.4
Residual gas heat (1)	67.3	427 - 40°C	26.1	52.6 %	6.9
					12.6
Exhaust gas Total (1-2)	170.6	427 – 40°C	65.2 (2)	55.8 %	

Notes (1), (2) and (3): see table 4.1.

Table 4.3 – As table 4.1 but for a gas turbine with 450 °C exhaust gas temperature.

In the case of a 450 °C exhaust temperature, the estimated power recovered is 12.6 MW i.e. 19.1 % of the rejected heat. It is based on Carnot efficiency values listed in table 4.3 and by applying a correction factor of 0.5.

5_Conclusion

The heat released to the atmosphere by most gas turbines is extremely large. It is 60 to 70% of the combustion heat, this percentage depending on the gas turbine type (aero derivative or heavy duty).

This heat could be converted into energy, firstly, to boost sour gases to a high pressure level (primary circuit for gas treatment) and, secondly, to activate a motor cycle (secondary circuit) providing additional energy to the gas turbine.

Below is an energy summary including the gas turbine rejected heat, the energy recoverable if there was a single motor cycle (no primary circuit), the absorbed power to boost the carbon dioxide to a high pressure level (primary circuit) and the delivered power by the motor cycle (secondary circuit). All these parameters are presented versus the exhaust temperature, an important parameter for converting heat into energy.

Exhaust temperature °C	550	500	450
Rejected heat - MW	82.4	74.3	66.2
Recoverable power without circuits 1 and 2 - MW	24.4	20.7	17.4
1 - Carbon dioxide – Absorbed power - MW	5.3	4.5	3.7
2 - Motor cycle – Delivered power - MW	18.1	15.3	12.6

Roughly, 6 % of the rejected heat can be used to boost sour gases (mainly carbon dioxide) to the gas treatment unit and 20 % of the rejected heat can be used to produce additional energy to the main turbine. These percentages are relatively independent of the exhaust temperature.

If the motor cycle was operated alone (no sour gas boosting) the recovered power would represent 27.8 % of the rejected heat that is approximately the same amount that is recovered by the primary and the secondary circuits, 26.6 %.