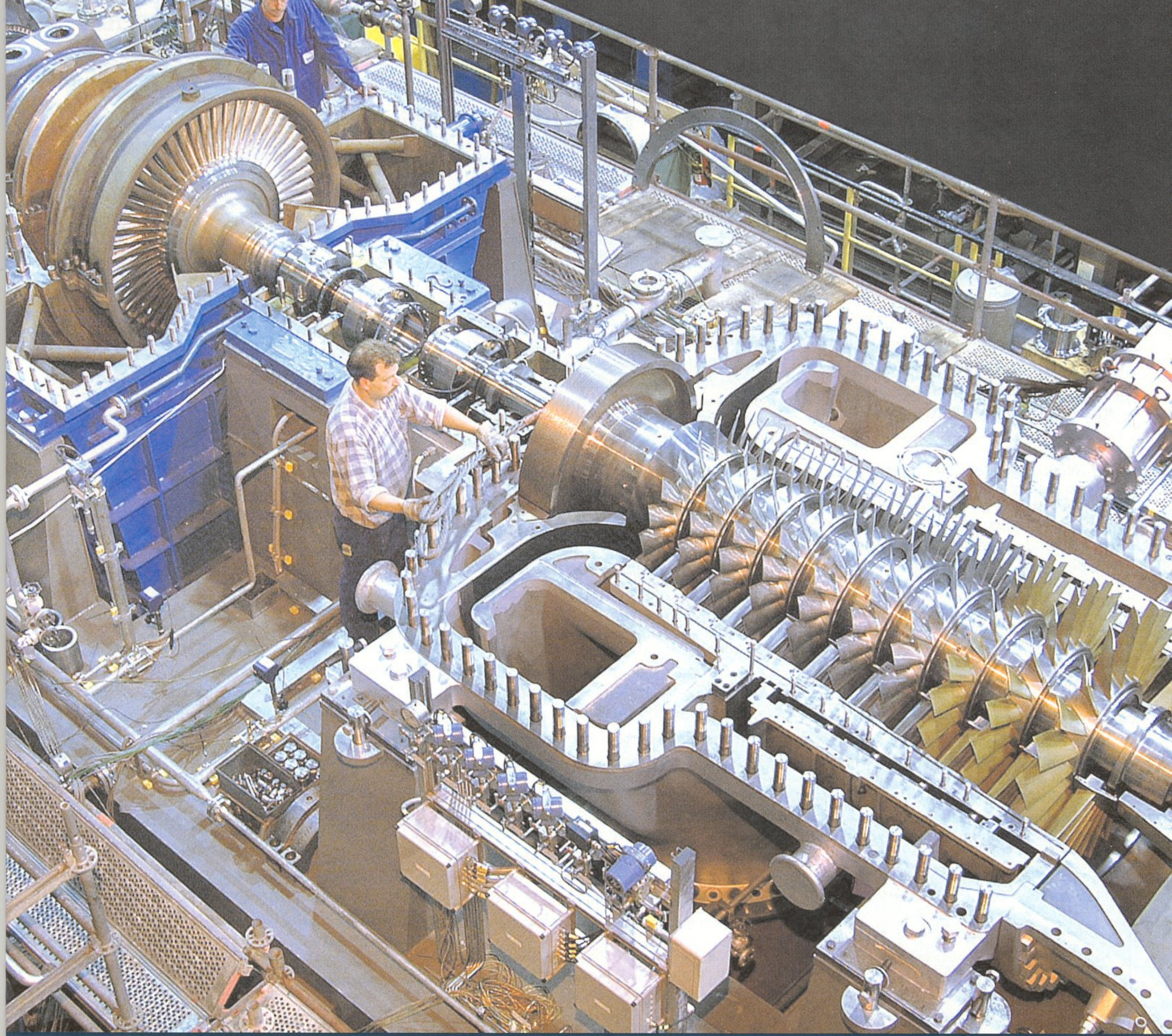

Conceptual Solutions Regarding the Influence of the Oxy-Fuel-Process on Conventional Power Plants

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MAN TURBO AG

Conceptual Solutions for the Integration of the Oxy-fuel Process

Kurzfassung

Konzeptüberlegungen bezüglich der Auswirkungen des Oxy-Fuel-Prozesses auf konventionelle Kraftwerksanlagen

Nach Schätzungen der Internationalen Energieagentur wird der Bedarf an elektrischer Energie weltweit in den nächsten Jahrzehnten um ca. 50 % ansteigen. Die Deckung des oben genannten wachsenden Strombedarfs kann nur durch einen erheblichen Ersatz und Neubau von Kraftwerkskapazitäten ermöglicht werden.

Im Zeitraum von 2010 bis 2020 sind z. B. in Europa (EU 15) rund 300 GW und in Deutschland ungefähr 40 GW an zusätzlicher Kraftwerksleistung zu erwarten. Darüber hinaus müssen die Kapazitäten aus den 70er Jahren altersbedingt ersetzt werden müssen.

Diesen enormen Bedarf an zusätzlicher Kraftwerkskapazität bei gleichzeitigem Ausstieg aus der Atomenergie kann selbst bei massivem Ausbau der "Regenerativen Primärenergien" nur durch fossil-gefeuerte Kraftwerke erreicht werden. Gleichzeitig muss dem Kyoto-Protokoll Rechnung getragen werden, die CO_2-Emissionen in den EU-Industriestaaten bis zum Jahre 2012 um ca. 8 % gegenüber 1990 zu senken [1].

Auch wenn nach den bisherigen Erkenntnissen der Klimaforschung die CO_2-Emissionen nicht zweifelsfrei einen klimarelevanten Einfluss haben, so bleibt dennoch ein schonender Umgang mit den vorhandenen knapper werdenden Reserven an fossilen Primärenergien und der Ausnutzung des CO_2-Reduktionspotentials eine wichtige Aufgabenstellung.

Die Anforderungen an die zukünftige Entwicklung der Kraftwerkstechnologie werden nach dem derzeitigen Stand der Technik durch zwei Maßnahmen gekennzeichnet [2] und [3].

– Erhöhung des Prozesswirkungsgrades

– Energiekonzepte mit CO_2-Abtrennung und -Verwendung aus dem Kraftwerksprozess.

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Introduction

According to International Energy Agency estimates, the worldwide demand for electrical power is set to increase by approximately 50 % over the coming decades. Satisfaction of these growing energy demands can only be achieved by significant replacement and new-build projects to increase power plant capacities.

For instance, between 2010 and 2020 approximately 300 GW of additional power plant capacity is expected to be built in Europe (EU 15), some 40 GW of which in Germany. However, existing power plant dating back to the 1970's must also be replaced due to ageing (Figure 1).

Even with a massive expansion in "renewables", this enormous demand for additional power plant capacity coupled with a simultaneous retreat from atomic energy can only be achieved through fossil-fuel power plants. At the same time, the Kyoto Protocol stipulates that CO_2 emissions in EU industrial countries must, by 2012, be lowered by approximately 8 % (compared to 1990).

Even if current findings from climate research do not irrefutably prove that CO_2 emissions effect climate change, it is still important to conserve dwindling reserves of primary fossil fuels and to utilise CO_2 reduction potentials.

In line with the current state of the art, the requirements relating to the future development of power plant technology are characterised by two measures [2] and [3].

- Increasing process efficiency
- Energy concepts with CO_2 capture in the power plant process

It is known that power plant efficiency can be increased by elevating the top average process temperature. With a combined gas and steam turbine process, this can only achieve the desired results with the use of higher firing temperatures and pressure ratios as well as improved cooling and/or coating technologies [4] to [8].

With a steam power plant, the determining criteria for reducing CO_2 can be seen in high steam parameters with one or two stages of intermediate superheating. In addition to the

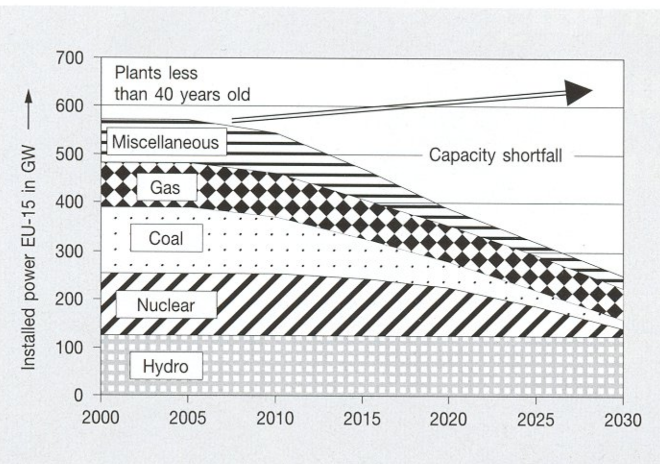

Figure 1. Future capacity replacement and new-build in Europe according to [1].

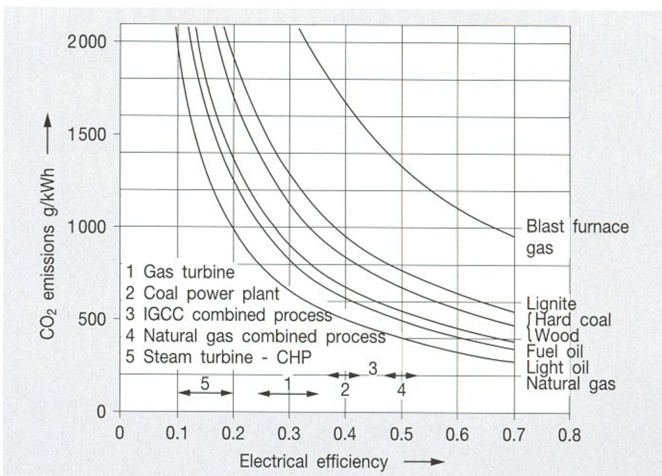

Figure 2. CO_2 emissions as a function of electrical efficiency with various fuels as parameters.

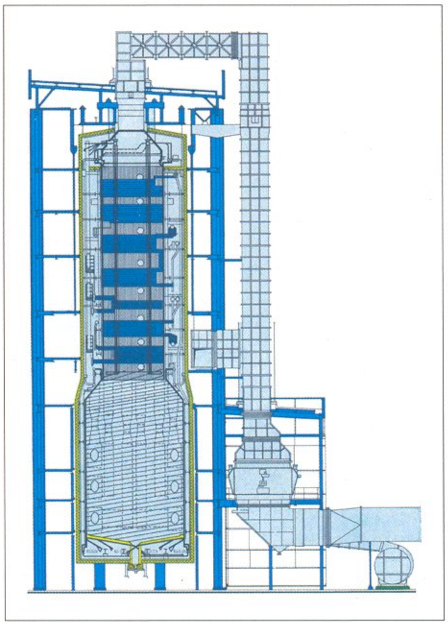

Figure 3. Once through steam generator for a municipal CHP (design: Walther & Cie, now ALSTOM POWER).

Steam capacity	*320 t/h*
HP pressure	*189 bar*
HP temperature	*535 °C*
Feed water temperature	*260 °C*
Hot air temperature	*360 °C*
Int. super heater temperature	*535 °C*
Int. super heater pressure	*47 bar*
Cold int. super heater temperature	*355 °C*
Heavy fuel oil "S" Natural gas	

above-mentioned process efficiency levels, the CO_2 emission can also be affected by the fossil fuel used (Figure 2). For instance, CO_2 emissions from coal are about twice that of natural gas, assuming the same general conditions.

It is not at all clear in what direction energy policies will move from 2012 onward [9], the success or failure of emissions trading will show in which direction the use of coal and natural gas will develop.

A possible further tightening of CO_2 reduction targets will necessitate new CO_2 capture routes, as well as, increased cycle efficiency.

The oxygen-based IGCC process (Integrated Gasification Combined Cycle) with innovative membrane technology for CO_2 capture and the oxy-fuel process and subsequent flue gas condensation are the power plant technologies currently needing development.

Flue gas condensation has been proven for years in the petrochemical industry, (e.g. methanol and ammonia syngas coolers), and the required membrane technology in the IGCC process for CO_2 capture is new territory. In terms of the oxygen technology, there is a considerable experience over the oxy-fuel process in the power plant sector in Europe with the operational experiences of the Buggenum and Portellano [10] and [11] IGCC units.

Highly intensive research in the area of materials technology and innovative power plant concepts are prerequisites for the above-mentioned measures in order to be able to maintain the relevant climate protection requirements over the long term. At least 10 to 20 years of high investment are required for this considerable R&D challenge in order to be able to operate power plants commercially with one of the above-mentioned technologies.

A concept is developed in this paper that takes into account CO_2 capture with the introduction of the oxy-fuel process in conventional natural-gas-fired power stations without adversely affecting possible operation using primary air.

Generally these conceptual solutions are also applicable for coal-fired power plants.

Basic Power Plant Concept Using the Oxy-fuel Process

Due to the size of the single-train oxy-fuel-process turbo machinery units already in existence (without the need for major development costs), and the need for CO_2 capture, only 100 to 400 MW power plant units can be considered. Parallel turbo machinery trains are more suitable for larger power plant outputs.

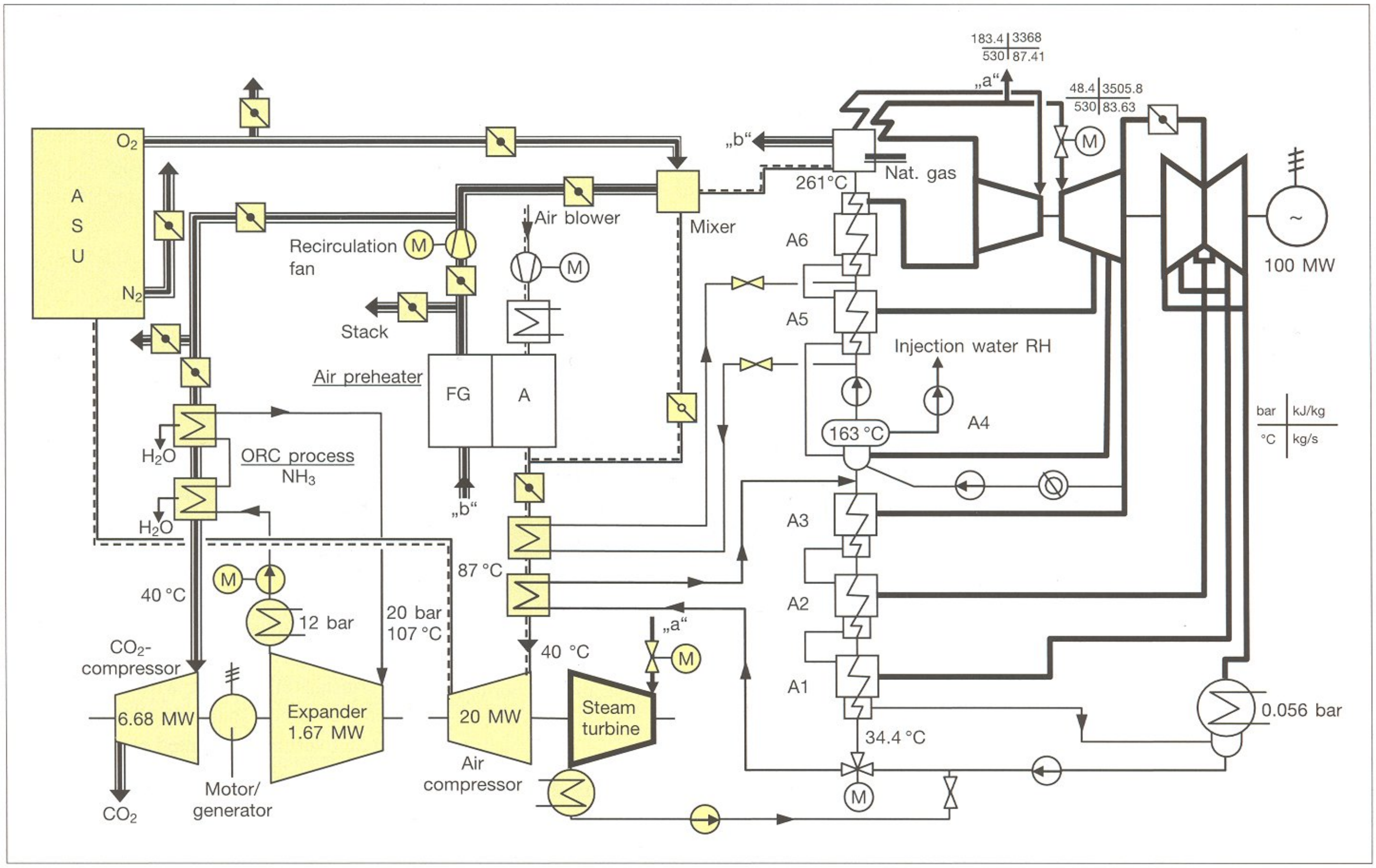

Figure 4. Expanded circuit diagram for retrofitting a 100 MW natural gas power plant to an oxy-fuel plant (pending MAN TURBO patent No. 102005026534.0).

Against this background the following criteria for oxy-fuel integration in conventionally-fired power plants have been assumed:

- existing regenerative feedwater preheating
- intermediate superheater units (RH)
- after plant modification, fresh air operation must continue to be possible without restrictions
- combustion air preheating on the exhaust gas side of the steam generator must be present
- there must be present design data or dimensional results of the plant, particularly for the steam generator
- scaling must be possible for larger units

An operational natural-gas-fired power plant consisting of a Benson boiler and a 100 MW intermediate superheater steam turbine was used as the basis power plant design (Figure 3).

The once through boiler shown in Figure 3 was used for extending a cogeneration plant and was specifically designed for peak current cover. A neighbouring refinery and a district heating network were also supplied with steam.

The steam generator fitted with heavy oil-gas firing and intermediate superheating can be operated in block operation and manifold steam main operation in combination with a 100 MW extraction condensing turbine.

The firing system, which is designed as a pressure oil return atomiser with gas burner devices, is arranged in three stages in the boiler combustion chamber as tangential burners (Figure 3).

The combustion chamber pipes are coiled whereas the wall pipes are vertical in the convection pass.

In order to increase the efficiency of the steam process, a six-stage regenerative feedwater preheating system is provided.

The preheaters in the feedwater train are supplied with steam at the requisite pressures at the discharge points via the extraction ports (A1 to A6) on the 100 MW steam turbine (Figure 4).

The necessary reduction in flue gas stack entry temperatures is performed by two Ljungström combustion air preheater working in parallel, which are connected with the end of the boiler by a vertical open pass (Figure 3). In order to prevent sulphuric acid corrosion during the combustion of heavy oil, a steam-heated air preheater is installed upstream for preheating the combustion air to 70 °C (Figure 4). The right-hand side of Figure 4 shows the existing power plant process, and the new oxy-fuel infrastructure is shown on the coloured left-hand side.

Integration of the Air Supply of the Basic Power Plant Utilising the Oxy-fuel Process

The requirement to operate the power plant in both air and oxygen mode, necessitates the retention of the existing air supply to the steam generator.

Because of the specified minimum air volume for the air separation unit (ASU) (60 % of total combustion air) and for the once through boiler without steam separator (40 % of total combustion air), the existing forced draft fan always works within a load range of 60 to 100 %.

The steam generator works in 40 % fresh air mode until the required oxygen quality is reached in the ASU. This plant is then switched from air mode to oxy-fuel mode. In the event of < 60 % load, the excess oxygen is vented to atmosphere (Figure 4). Further load increases of the plant are then undertaken taking into account the permissible values of the ASU. The switching from oxygen to air mode is also possible in the reverse direction.

The air required for oxy-fuel mode is cooled downstream of the combustion air preheater (Lj-Luvo) by means of high-pressure feedwater (extraction port A5) and steam turbine condensate (extraction ports A1 to A3) from approximately 400 °C to 40 °C and compressed in the air compressor to approximately 6 bar. Depending on the load conditions, the preheaters work on the air side as partial or full flow economisers in the same way as has already been performed several times with the re-powering of coal power plants by including the waste heat from topping-cycle gas turbines (TCGT) on the feedwater preheating side [12] to [15].

Extraction ports A4 and A6 retain their functionality in order to ensure that the de-aerator and the intermediate super heater characteristics are maintained (Figure 4).

The resulting reduction in extraction steam mass flows for regenerative feedwater preheating increases the mass flow throughput in the turbine and therefore the electrical output of the unit. Both lead to higher generator load and require a corresponding load capability of the power plant turbine.

To reduce the load on these two components, steam can be extracted on the "hot" intermediate super heater line to drive the air compressor steam turbine.

The power shortfall on the plant is then only 5.5 % (according to Table 2) with the intermediate superheater steam extraction. Transfer of the combustion air heat to the condensation cycle is partially compensated in terms of a reduction in extraction steam mass flow (closed or partially open LP or HP extraction ports).

If the load capability of the power plant steam turbine is sufficient and the generator also has additional reserve, it would be possible to decouple the air compressor from the intermediate superheater line of the power plant turbine. For the compressor drive, it makes sense here to use an electric motor or gas turbine with waste heat recovery, or alternatively, a simple steam turbine process with a directly fired steam generator. The advantage of such an approach is firstly in the free choice of steam parameters and secondly in the improved dynamics of the switching the plant from oxy-fuel to fresh air operation in the event that one of the turbo machinery units in the oxy-fuel process has tripped.

The air cooled by the high-pressure feedwater is compressed by means of "quasi-isothermal" compression to the necessary pressure of the ASU, whereby the power output of the compressor is considerably reduced. Figure 5 shows a single-shaft centrifugal compressor from the MAN TURBO RIKT series with axial air intake and integrated intercoolers after each stage.

This design principle of the isotherm compressor is characterised by the following features:

- Adjustable stator blades in the first stage enable energy-saving part load operation
- Cooling tube bundles with finned tubes for optimum heat transfer
- Integration of additional, proven water condensate separators, also suitable for large quantities of water condensed in the coolers to be removed from the compressed medium.
- Compact design and low space requirement

The large number of intercoolers naturally results in a relatively low compressor discharge temperature after each stage so that waste heat utilisation makes little sense.

In power plants in the 200 to 400 MW category, a single-casing axial/centrifugal compressor with intercooling is a good solution for air volumes up to 1,200,000 Nm^3/h.

MAN TURBO AG axial compressors can be constructed as compact and space-saving elements by using two axial stage groups in opposed configuration each with a radial end stage on a common shaft. By additionally installing an overhung-mounted centrifugal impeller, pressure conditions up to 12 can be achieved with two stages of intercooling (Figures 6 and 7).

In order to optimise efficiency, speed control or stator blade control may be more benefici-

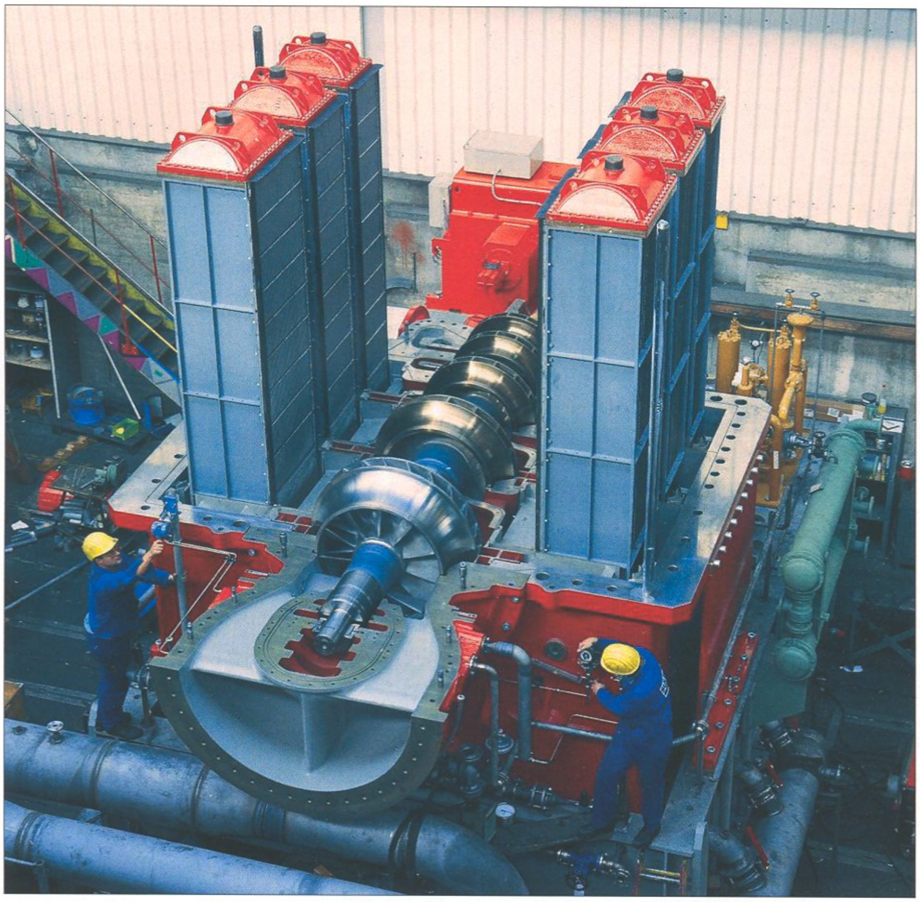

Figure 5. Centrifugal air compressor RIKT (MAN TURBO design) with intercoolers downstream of each stage ("quasi" isotherm compression).

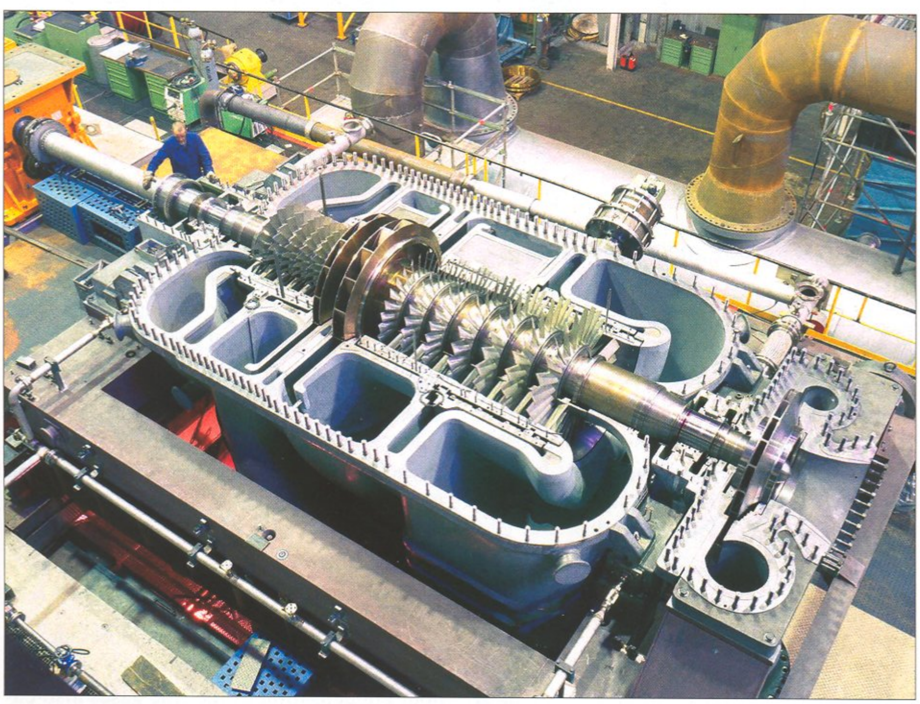

Figure 6. Combination of axial and centrifugal compressors with intercooling for max. 1.2 · 10^6 Nm^3/h air volume (MAN TURBO design).

al depending on the position of the part load operating points specified. Speed control requires no design measures on the compressor side and only causes minor losses on the drive side (steam turbine, gas turbine or speed-regulated electric motor).

If an axial compressor is driven at constant speed, capacity control is achieved by means of adjustment of the stator blades on the front stages of the compressor; by this means a wide turndown range is achieved at high levels of efficiency.

The surge limit for these axial compressors is approximately 65 % of the design mass flow, which approximately corresponds to the minimum load of the ASU; this can be achieved whether the compressor is operated with fixed or variable speed. With compression at constant speed, approximately 40 % of the stator blade rows are of an adjustable design in order to maintain the same operating range as with speed control.

With the axial/centrifugal compressors mentioned above, the air-side temperature level downstream of each compressor is considerably higher than with the isotherm compressor - for such a solution waste heat utilisation makes sense both for the compressor intercoolers and for the aftercooler (i.e. prior to the ASU).

The integration of the ORC process (Organic Rankine Cycle), necessary for flue gas condensation, on the waste-gas side of the basis power plant (Figure 4) and the boiling temperature (50 °C) for ammonia (NH_3) selected for flue gas condensation reasons, provide optimal waste heat utilisation on the compressor side, too (Figure 9).

The selection of the waste steam pressure for the NH3 expander is based both on the conditions of the existing water cooling cycle of the power plant unit and on the required minimum air temperature at the waste heat boiler outlet of the intercooler. (See also the section "Integration of the air supply of the basic power plant utilising the oxy-fuel process").

Figure 8 shows a simplified circuit diagram with integrated waste heat flow for the air compressor train through the ORC process. In order to maintain as low as possible the air inlet temperature to the second compressor (centrifugal compressor), the NH3 preheater must be installed in the intercooler (Figures 8 and 9).

The process loop described above ensures that the required minimum temperature at the centrifugal compressor inlet is maintained. The air temperature downstream of the aftercooler (60 °C) is, however, significantly lower compared to the isotherm compression (Figures 4 and 9).

The air pressures downstream of the individual compressors (3 bar and 6 bar) require pressurised waste heat boilers (WHB). Figure 10 shows an example of such a WHB, which is installed as an intercooler in the compressor train. The air from the axial compressor flows from above into the pressurised shell where the heating surfaces (here evaporator and economiser) are suspended in an inner convection duct. The mechanical loading of the convection duct is caused solely by the pressure differential between inlet and outlet, whereas the pressurised shell of

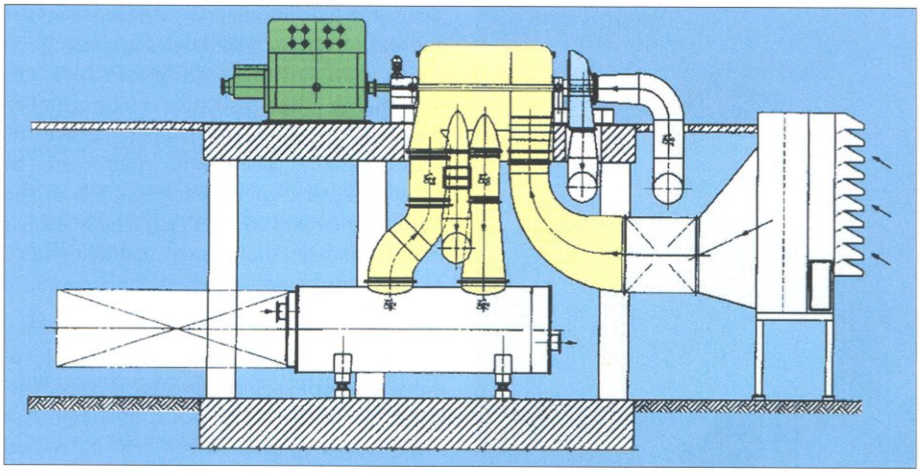

Figure 7. Schematic diagram of the air path in the combined axial/centrifugal compressor with intercoolers and the centrifugal final stage on one shaft (corresponding to Fig. 6).

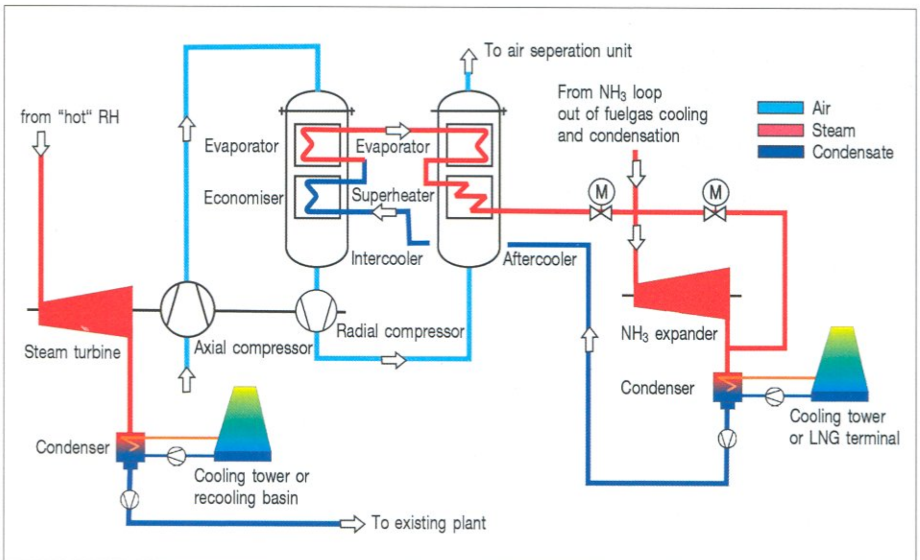

Figure 8. Simplified diagram for integrated waste heat generation in the air compressor train (NH_3-based ORC process) (pending MAN TURBO patent No. 102004020753.4).

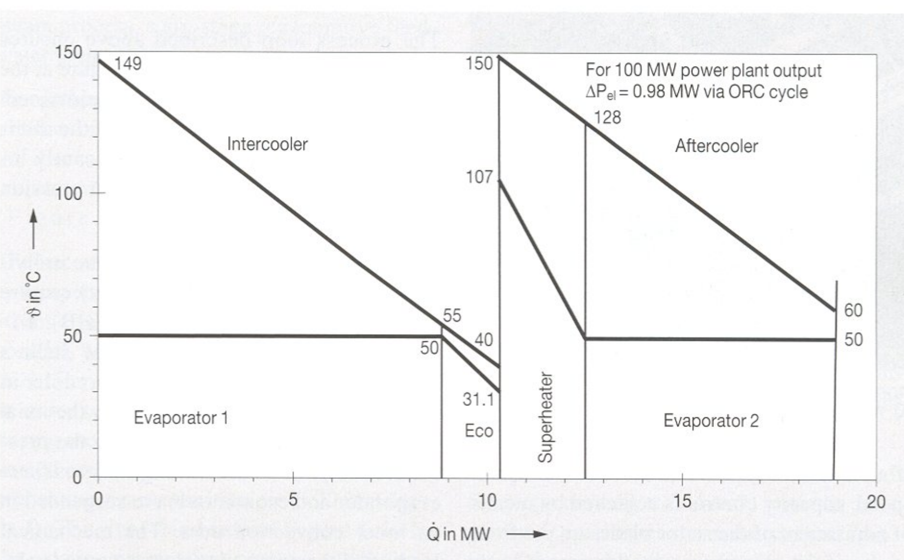

Figure 9. $\dot{Q}$-ϑ diagram for the ammonia cooling cycle (ORC-NH_3) of intercooler and aftercooler of the air compressor (100 MW natural gas power plant).

the heat exchanger sees the full air pressure, but lower air temperature (Figures 9 and 10). An analogous situation applies for the WHB of the aftercooler.

The relatively high heat transfer figures of boiling ammonia (NH_3) enable the use of both bare and finned tubes for the heat exchange surfaces [16], [17] and [18]. Since the temperature differences, and therefore the heat flux densities, are very moderate (Figure 9), both natural and forced circulation systems as well as once through flow can be used for these evaporators [6], [19] and [20].

During the installation of circulation systems for the ORC process, expensive separators (vertical vessels or drums) with associated connection pipe-work (down comers and overflow pipes) must be installed to separate the fluid phase, which leads to higher investment costs.

With once through systems, preheating, evaporation and superheating are performed in a single process, whereby reduced heat transfer figures may occur in the end stage of the boiling zone (drying out of the heat exchanger tube inner surface), which leads to higher construction volumes. The decisive factor here is the permissible pressure drop on the air/coolant side.

Integration of the Waste-gas Side of the Basic Power Plant in the Oxy-fuel Process

In the ASU the compressed air is separated into nitrogen (N_2) and oxygen (O_2) and, after reaching the required levels of oxygen purity, the oxygen is combined with the necessary flue gas recirculation volume (RECI) of the steam generator in the mixer (Figure 4) (see also the section "Firing concept of the basic power plant for oxy-fuel operation").

The combustion gases in oxy-fuel operation, which consist of 66.5 % water vapour (H_2O) and 33.5 % carbon dioxide (CO_2), are cooled down by an ORC (NH_3-based) installed parallel to the compressor air cooler. The NH_3 input parameters are selected, such that cooling to 40 °C can take place far below the water dew point of the flue gases (Figure 11).

Apart from cooling the air compressor, the high thermal component from the water vapour condensation of the flue gases also enables additional expander performance from low-temperature waste heat to drive the CO_2 compressor (Figure 4). Both NH_3 mass flows from flue gas water vapour condensation and compressor air cooling are fed to the common expander (Figure 8).

In 1982 a plant using the Clausius-Rankine process with high ammonia parameters

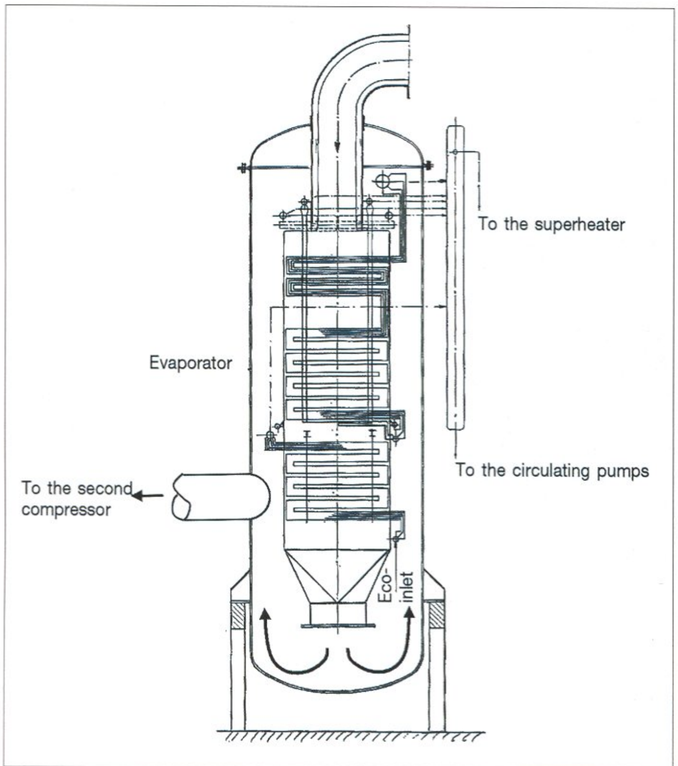

Figure 10. Pressurised waste heat generator integrated in the inter-cooler of the air compressor train with forced circulation (100 MW natural gas power plant).

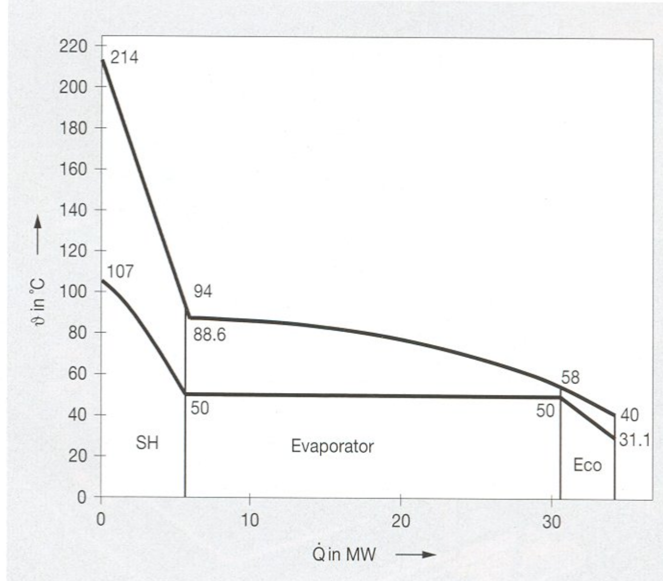

Figure 11. Q̇-ϑ diagram for cooling of water vapour saturated flue gases below the water dew point (100 MW natural gas power plant).
pH_2O = 66.5 %, pCO_2 = 33.5 %)
Coolant: ammonia (NH_3)
pNH_3 = 20 bar, ϑNH_3 = 107 °C = 380 °K (expander inlet)

(129 bar, 240 °C) was successfully put into operation for the first time by Borsig GmbH (now MAN TURBO AG) for recovery of gas turbine waste heat in the natural gas compressor station in Clearwater/Canada. From the waste heat from an LM 2500 gas turbine (16 MW), approx. 5.2 MW power output was generated with an NH_3 expander (design by Borsig, now MAN TURBO AG) for powering an additional natural gas compressor. The waste heat boiler built then by Borsig without flue gas condensation was designed according to the once through flow principle [27].

The design of the waste heat steam generator with flue gas condensation is similar to the systems used in synthesis gas cooling below the water dew point in the steam reforming process [24] to [26]. In oxy-fuel plants with a gas temperature of 40 °C before the CO_2 compressor inlet, the water vapour volume in the CO_2 waste gas is still 7.38 vol. %.

With such parameters the compression of CO_2 is normally performed in a two alt. three casing single-shaft centrifugal compressors, which however have significant disadvantages compared to machines of a multi-shaft design. Optimal compression of CO_2 can be achieved in multi-shaft gear compressors (MAN TURBO design) (Figure 12 with motor drive). The above-mentioned gear compressor generates a high discharge pressure of 187 bar and is used e.g. to transport CO_2 from the coal gasification plant in North Dakota/USA to the oil fields (Enhanced Oil Recovery) of Saskatchewan in Canada (370 km) and has been in operation since 2000 [21] and 22].

Multi-shaft gear compressors have the following advantages over single-shaft centrifugal compressors:

- High levels of efficiency because of optimal speed adjustment
- Axial intake and intercooling up/downstream of each stage ("quasi-isothermal" compression)
- Flexible operation through inlet guide vane control
- Reduced construction volume and lower investment costs

If, because of downstream processes, lower discharge pressures are required, the power outputs in Table 1 are reduced accordingly. In addition, the mechanical energy provided by the expander can be significantly increased if, instead of the specified cooling circuit of the power plant, there is a different heat sink with lower cooling temperatures.

Liquid Natural Gas (LNG) is particularly suitable here as after liquefaction the LNG is supplied at 113 °K (-160 °C) to the relevant LNG terminals in sea ports.

With the aid of this heat sink the pressure ratio of the expander can be increased by approximately 2 (Figure 4) to approximately 4 so that – at constant inlet parameters – the necessary cooling effect is achieved despite the low NH_3 waste steam temperature of approximately 5 °C.

This means that part of heat required for vaporisation of the LNG can be obtained from the cold end of the ORC process. Table 2 shows the losses of the oxy-fuel process, taking into account the same cooling cycle parameters of the existing power plant.

Taking into account the above-mentioned LNG heat sink, the expander would be able to supply the total mechanical power output

Table 1. Oxy-fuel retrofitting for 100 MW natural gas power plant. Mechanical power output of the CO_2 compressor at different discharge pressures:
CO_2 = 92.62 %, H_2O = 7.38 %,
ϑ = 40 °C, $\dot{m}$ = 15.039 kg/s.

Discharge pressure in bar	Power output in kW
200	6700
150	6600
100	6000
50	5300

Table 2. Oxy-fuel retrofitting for 100 MW natural gas power plant. Comparison of net power plant output in fresh air and oxy-fuel operation for a 100 MW natural gas plant.

	Frisch air in MW	Oxy fuel in MW
conventional power plant	100	94.5
NH_3 expander (air cooler)	–	+ 1.0
NH_3 expander (flue gas condensation)	–	+ 1.7
CO_2 compressor[1]	–	- 6.7
Net output	100	90.5

[1] CO_2 compression to 200 bar

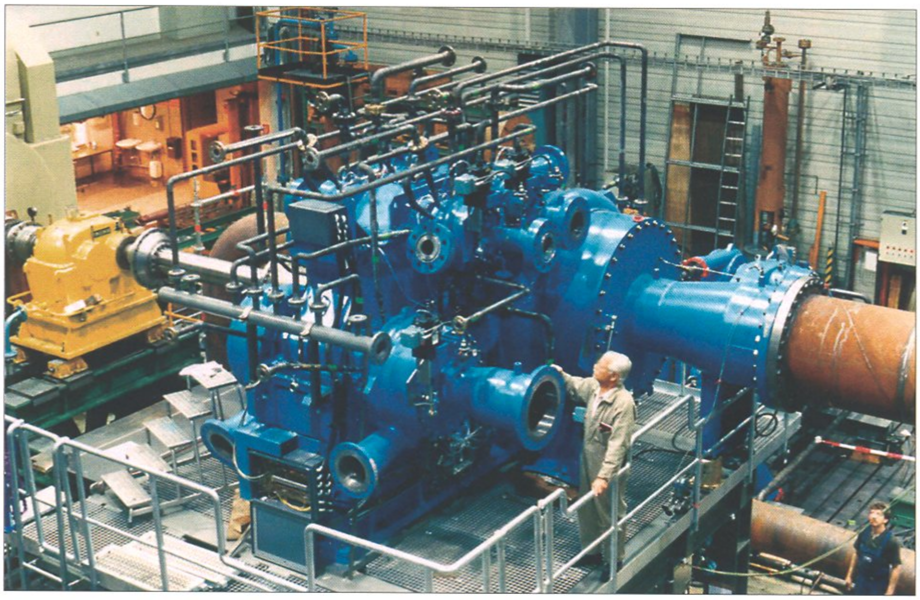

Figure 12. 8-stage gear compressor for CO_2 compression (Design: MAN TURBO).
Mass flow CO_2 = 37.8 kg/s; Pressures = 1.15 / 187 bar; Power output 13.2 MW

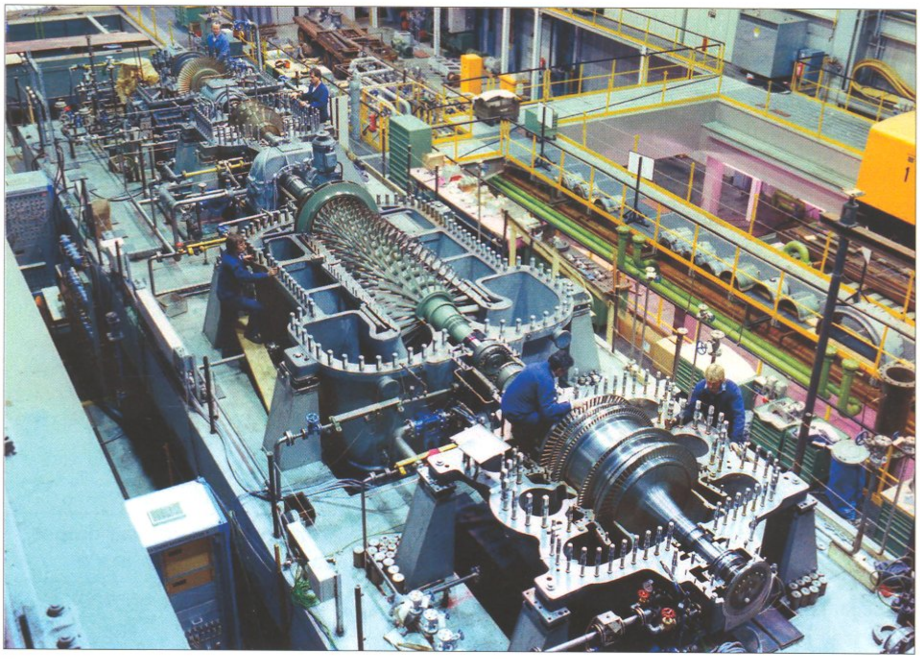

Figure 13. Single-shaft machinery train (steam turbine, air and NO compressor and residual gas expander) for nitric acid plant.

Air $\dot{V}$ = 273,064 m^3/h
π = 5,26
P_K = 17.878 kW
NO gas $\dot{V}$ = 54,675 m^3/h

π = 2.45
P_K = 8,793 kW
NO residual $\dot{m}$ = 244,015 kg/h
p_1/p_2 = 10.28/1.03 bar
P_K = 20,093 kW

Steam turbine
$\dot{m}$ = 30,200 kg/h
p_1/p_2 = 18/0.08 bar
P_K = 6,753 kW

of the CO_2 compressor because of the significantly higher pressure ratio, such that CO_2 compression would be covered just from the flue gas condensation and air cooling without external energy requirements. Since there is sufficient natural gas available (LNG terminal), combined operation between the natural gas power plant and LNG terminal would be feasible. The nearby sea would be a possible store for the separated CO_2 or pumping would have to take place over a great distance – as described above – to the relevant consumers (Figure 8).

The power situation of the steam turbine and air compressor (shown in Figure 4) as well as of the NH_3 expander and CO_2 compressor with motor/generator concept can also be executed as a single-shaft train. Figure 13 shows this sort of single-shaft train, which is integrated in a nitric acid plant. This single-shaft concept consisting of steam turbine, air compressor, NO compressor and a residual gas expander has been operating successfully in numerous nitric acid plants for years. The necessary drive energy for the compressors is provided from the steam turbine (steam generation from NH_3 combustion elements) and from the expansion of residual gases from the absorption tower [23]. The thermodynamic design of the machinery train is such that no additional external energy is required.

If, instead of steam turbine power, a gas turbine is envisaged for the air compressor (Figure 4) or for the above-mentioned single-shaft train, additional drive energy can be generated with an NH_3 expander through the recovery of gas turbine waste heat, as described in [27].

Since each component in the above-mentioned machinery train is integrated in the various intermediate stages of the nitric acid process, this concept is also suitable for oxy-fuel plants. Because of the special start-up situation with air separation plants, additional SSS couplings must then be installed in the area of the NH_3 expander and the CO_2 compressor. In addition, the rotor dynamics must be taken into account when adjusting the speed of the individual turbo machinery elements.

Firing Concept of the Basis Power Plant for Oxy-fuel Operation

One of the critical components when integrating oxy-fuel operation in conventional power plants is the combustion chamber of the steam generator.

Due to the fact that there is no nitrogen in the oxy-fuel process, as opposed to fresh-air operation, there is a corresponding reduction in flue gas mass flows in the flue gas pass of the boiler together with a considerable increase in firing temperatures. This would lead to significant thermal stresses in the tubes of the boiler combustion chamber from the power plant boiler.

By injecting a correspondingly high level of flue gas recirculation into the firing system of the steam generator (mixer), similar values are achieved as in fresh-air operation for the mass flows and combustion temperatures (Figure 4).

In order to be able to assess the operational behaviour of the combustion chamber in oxygen operation, fresh-air operation was modelled using flame geometry based on tangential firing for gas and oil over the variation of absorption coefficients in the boiler combustion chamber, such that a good correspondence to the design data was reached. In the adjacent radiation chamber the flame transmission into the superheater pass was also taken into account with the absorption coefficient of the gas core for water vapour (H_2O) and carbon dioxide (CO_2) for subsequent heat liberation.

With the absorption coefficients determined in this way, the combustion chamber was simulated for oxygen operation fuelled by

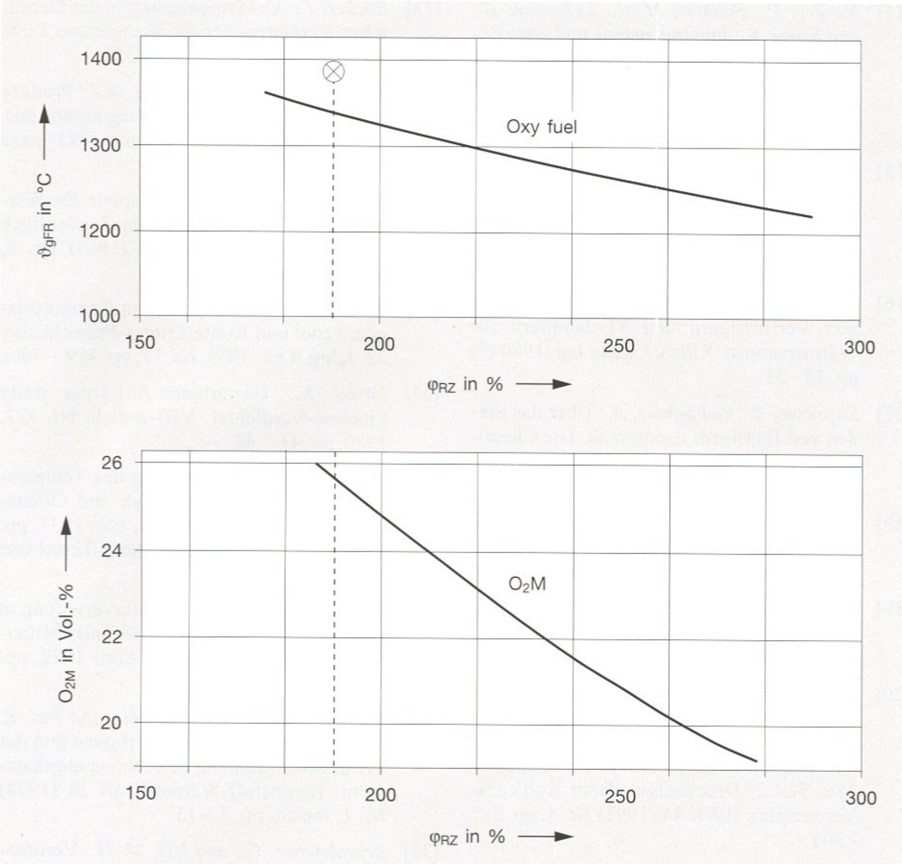

Figure 14. Operating behaviour of the boiler combustion chamber of the once through boiler dependent on re-circulated flue gas mass flow (φ_{RZ} in %) with oxy-fuel operation.
⊗ Combustion chamber end temperature (fresh-air operation); $\varphi_{RZ} = (\dot{m}_{RZ}/\dot{m}_g)$ 100 %; $\dot{m}_{RZ}$ = recirculated gas mass flow (kg/s); $\dot{m}_g$ = combustion gas mass flow (kg/s)

natural gas in accordance with the above-mentioned modelling, whereby the increased selective radiation of H_2O and CO_2 in the area between the shape of flame and wall tubes and the adjacent radiation chamber were taken into account in the calculation [28] to [31].

Since there is very little operational experience with oxygen-fired steam generators in terms of the flame characteristics, the modelling of the combustion chamber via the combination of flame chamber and radiation chamber is not assured. For this reason the entire combustion chamber was divided into two sections with simple gas core radiation, whereby an integral average gas temperature was defined in the area of the previous flame chamber.

The results showed a negligible difference of approximately 20 K lower combustion chamber end temperature compared with the division of combustion chamber into flame chamber and radiation chamber.

Figure 14 shows the operating behaviour of the combustion chamber in oxy-fuel operation (natural gas operation) with the above-mentioned division of flame and radiation chambers. Since for thermodynamic reasons the flue gas recirculation mass flow must be taken downstream of the combustion air preheater, the re-circulated flue gas mass flows here are lower than the values given in the literature because of the lower gas temperatures (downstream of the combustion air preheater) [32] and [33] (Figure 4).

With approximately double the mass flow of re-circulated gas and an approximately 65 K reduction in combustion chamber end temperature, the heat absorption in the combustion chamber is identical to fresh-air operation (Figure 14). The resulting reduction in the amount of heat provided for the convection heat surfaces (temperature and flue gas mass flow) is partially compensated by the higher heat transfer figures (high CO_2 and H_2O) and thermal capacity of the flue gas. The changes in steam-side temperature for once through boilers are primarily determined by the heat absorption in the combustion chamber, so that in oxy-fuel operation there should be at least be no significant heat shifts on the high-pressure superheater side (provided the valence of the convection heat surfaces corresponds to the design status). Only on the intermediate super heater side could there be higher injection water mass flows, which with corresponding heat surface changes can be unproblematic, since the possibility of subsequent heat surface extensions/reductions are normally included in the original design.

By combining oxygen (99.5 % purity) and re-circulated flue gas in the mixer, it needs to be borne in mind that there will be higher oxygen content in the relevant recirculation zone for firing the steam generator than in fresh-air operation (Figure 14). With these conditions, despite a lack of air heat from the combustion air preheater (Figure 4), high flammability should be expected, whereby the adiabatic combustion temperatures are somewhat below the temperatures in fresh-air operation because of the lack of air heat. Excess oxygen, which is needed in fresh-air operation for firing reasons, can be dispensed with here.

Dissociation effects caused by hot spots or CO post-combustions before the convection heat surfaces are not envisaged with this firing concept [32].

Summary

With the expected increase in energy requirements over next two decades, there needs to be considerable investment and development of new techniques involving subsequent CO_2 capture within the power plant sector in order to comply with the Kyoto Protocol. The technical design and timescale will both play a significant role here.

This paper investigates the feasibility of such a concept, while also taking into account minimal efficiency losses in the power plant process. This is achieved by the integration of effective turbo machinery in the oxy-fuel process with simultaneous power generation from the waste heat potential of the air compressor and flue gas condensation system.

Because of the high degree of complexity involved in the task, power plant operators, universities, plant constructors and turbo machinery manufacturers should work together closely to create the pre-requisites for the development within an appropriate timescale of concepts that enable the separation of CO_2 and oxy-fuel operation on a commercial basis.

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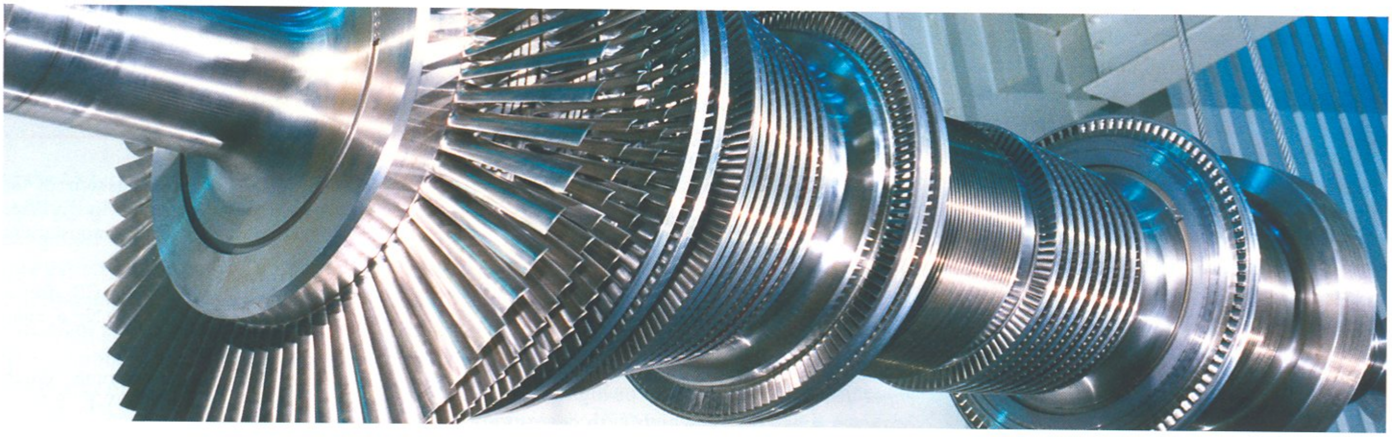

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