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Combined Heat and Air Generation – A Technical Study

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ABSTRACT

Compressed air generation represents one of the most significant hidden costs in manufacturing plants. While it has been shown that centralized compressed air generation may not be the most cost effective or energy efficient configuration, centralized air generation is most common in large scale facilities. Thus, from a total cost viewpoint, generating compressed air in a single or at most in a small number, of installations within a factory is accepted as the standard, if not the best, practice. This concentrating of plant offers the possibility of generating compressed air by directly coupling the compressor shaft to a heat engine, in an approach similar to combined heat and power. Currently the compressor is typically driven by an electric motor, which is typically supplied by a grid connection to a thermal generating plant; thus in this combined heat and air (CHA) scenario, the electricity generation and transmission stage is removed. This is not a new proposal. This approach of generating compressed air is commonly used in small scale applications where the point of use is far removed from an electricity supply (e.g. in the agricultural sector, or in civil engineering construction) and increasingly even in manufacturing environments. It is shown here that using a gas turbine, with regeneration, will yield a 43% improvement in operating efficiency. This does not require a balanced heat load demand, as the waste heat is reused in the generation of shaft power, although in principle, the total energy efficiency could be improved further by utilization of the low grade heat, in space heating, for example. It is concluded that a Combined Heat and Air (CHA) approach deserves further study.

NOMENCLATURE

c	specific heat	J/kg/K
Ex	exergy	J
h	enthalpy	J/kg

l_c	condensation heat	J/kg
$\dot{m}$	mass flow rate	kg/s
p	pressure	Pa
r	ideal gas constant	J/kg/K
T	absolute temperature	K
η	efficiency	[-]
φ	relative humidity	[-]

1 INTRODUCTION

Despite the awareness that compressed air is a major source of inefficient energy use in an industrial setting, it remains a mainstay of many installations for a variety of reasons. While legacy issues are certainly part of the reason for the continued use of compressed air, this does not fully capture the rationale for compressed air use, and so there is ongoing capital investment in compressed air generation in industry. The compressor device itself is technologically very mature, and so is unlike to yield dramatic improvements in efficiency. However, the compressor supplying compressed air system is a potential source of two forms of energy: pressure energy of compressed air; and heat energy rejected during compression, either through the inherent heat of compression, or inefficiencies in the machinery. As the compressor is certainly the biggest input power consumer in compressed air system, it would be reasonable in turn to utilize pressure energy of air as best as possible and to recover heat energy as much as possible. Unfortunately, pressure energy of compressed air, which is capable of work, is wasted in many applications such as cooling/drying, sealing, purging or cleaning. A considerable amount of rejected heat can be recovered using recent advanced heat recovery systems for heating purposes, but heat generated from the compressor has usually no onward utilization, for example cooling water often circulates between the compressor unit and cooling towers, where heat content is rejected. Furthermore, the most typical situation is centralized generation, i.e. the compressor(s) is located in one energy centre usually far away from end-use applications. If both energies (heat and pressure) are supposed to be utilized, then they must be distributed to the final applications resulting in energy losses.

The overall performance, in terms of energy efficiency, of the compressed air systems can be partially improved already in connection with the generation side, i.e. a more suitable compressor location, better utilization of heat energy or alternative compressor drive. For example, in the context of compressor location, based on a complete economic analysis, Yuan et al./Yua06/ have recently shown that a better configuration would be local compressed air generation (a certain number of compressed air consumers are grouped together and supplied by a local smaller compressor). Although the authors did not consider it, this arrangement might also provide a better configuration for waste heat utilization, although this will be very application specific. In the three scenarios envisaged by Yuan et al., the compressors are driven by electric motors. In practice, these motors have been developed to a high level of operational energy efficiency, especially with the recent prevalence of advanced technologies (such as variable speed drives or permanent magnet rotors). However, there is still the possibility that an alternative prime mover, such as thermal engine, would be more efficient in total as the heat generated would be at a higher temperature. This results from the fact that the heat lost during electricity generation in a thermal power plant is still available as is the transmission losses associated with centralized electricity generation and distribution. The philosophy shares much with Combined Heat and Power (CHP) generation, which is becoming common in many industrial and even domestic settings. In this paper the potential benefit of a Combined Heat and Air (CHA) generation plant is assessed.

2 ENERGY TRANSFORMATION IN COMPRESSOR

There are principally two energy inputs into the compressor, mechanical compressor drive power (i.e. shaft power) and energy coming with the intake air as shown in the simplified schematic in **Figure 1**. The main purpose of the compressor prime driver power is to increase work ability (exergy) of the inlet air, i.e. to increase pressure of the air during compression. The secondary effect of the compression is production of heat, which can be recovered and used for some heating purposes, but unlike high pressure energy this energy has no (or very little) work ability.

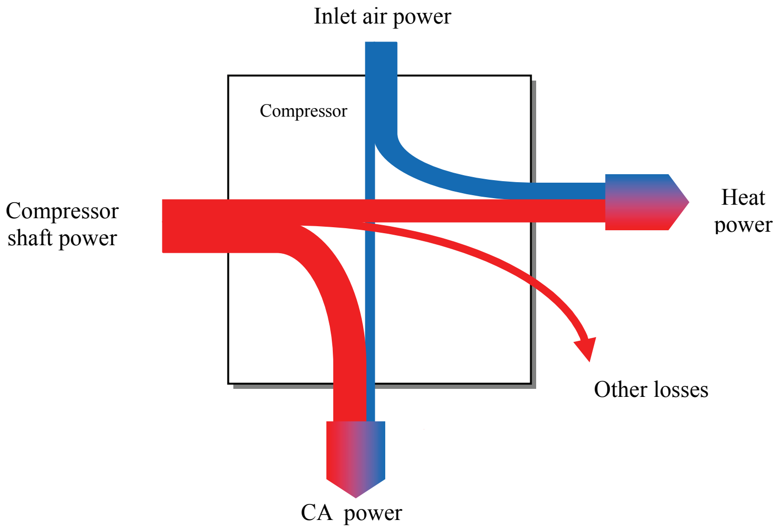

Figure 1: Energy balance of compressor

The part of compressor drive power transferred to compressed air power capable of work depends on the type, technical/working conditions and quality of the compressor. The air pressure energy increase can be formulated using the thermo-dynamical quantity the exergy Ex (i.e. work ability referenced to ambient conditions p_a, T_a). Thus the efficiency of transformation of compressor drive energy E_{in} into exergy leaving compressor Ex_{out} is expressed using Equation 1. A similar assessment of compressed air power can be found in Cai et al. /Cai06/. Experience shows, that the efficiency can likely range in value from 0.25 to 0.7, and the remaining part of the input energy is rejected as heat including losses.

$$\eta_{com} = \frac{Ex_{out}}{E_{in}} = \frac{\dot{E}x_{out}}{\dot{E}_{in}} = \frac{\dot{m} r T_a \ln \frac{p_{out}}{p_a}}{\dot{E}_{in}} \tag{1}$$

The energy of the incoming air is related to the enthalpy of the moist air and depends on temperature of the air and water mass fraction as shown in the Equation 2. The enthalpy h of gas mix of *1* kg of dry air (subscript da) and x kg of water vapour (subscript wv) referenced to the absolute temperature T of the gas also contains condensation heat l_c (with a typical value of 2500 kJ/kg), which can contribute to compressor heat balance as this heat is released during compression. The water mass fraction x can be expressed using relative humidity $\varphi = p_{wv} / p_{sat}$, where p_{sat} is partial pressure of water vapour in saturated moist air at actual temperature T.

$$h_{1+x} = c_{da}T + 0.622\,\frac{\varphi p_{sat}}{p - \varphi\, p_{sat}}\,(c_{wv}T + l_c) \tag{2}$$

For example, the latest oil-free compressor of a common compressor OEM, Atlas Copco, is able to recover heat energy, which is 100% equal to the electrical energy input, under specific design conditions of inlet air temperature 40ºC, 70% relative humidity, and temperature of cooling water 20ºC /Atl09/. In this case, the advanced heat recovery system captures condensation heat, which compensates radiation losses.

Although the heat rejected from compressor drive input power has higher temperature potential and inlet air thermal energy is contributing only when a suitable cooling system is used, both energy inputs are generally affecting the output heat load available from the compressor. The contribution of energy contained in inlet air to the heat balance is independent of compressor load, so only rejected heat from compressor drive is discussed further in this paper, as this determines the dynamic behaviour of the compressor as a heat generator.

3 HEAT GENERATION UNDER VARIOUS COMPRESSOR CONTROL

The overall heat power generation is a function of another crucial factor: compressor control mode related to the system demand. If compressor is operating at full load, then total amount of heat generated is obviously higher than at part load. Typical controls installed on positive displacement compressors - VSD, load/unload and inlet modulation - are detailed below. The description of first two modes are accompanied by data collected from Irish plants.

3.1 Variable speed drive

Figure 2 shows the long term revolution spectrum of 2-stage rotary screw oil-free air-cooled VSD compressor, which works as a base unit. This advanced control matches system demand by changing quantity of the delivered compressed air using motor revolution modification while maintaining relatively constant system pressure level. The mass flow rate change is directly proportional to the input compressor drive power, which has a rated power of 266 kW (including power consumption of the VSD itself and minor mechanical losses). So, the compressor rated power is 250 kW at a maximum speed of 2740rpm.

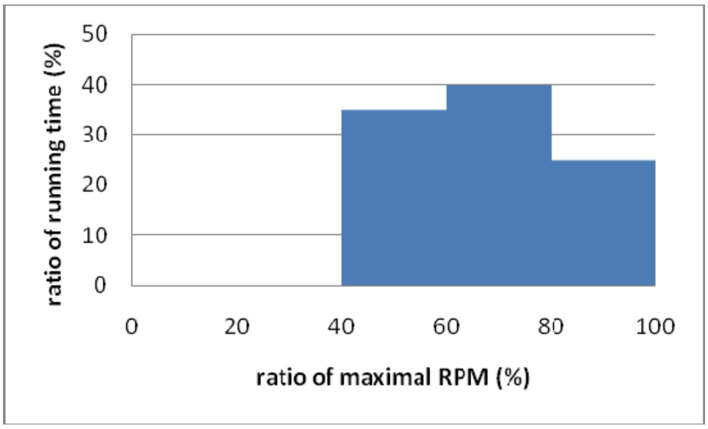

Figure 2: RPM spectrum of VSD

For sake of simplicity, the ratio of the mass flow rate to the power consumption is assumed to be the same over the investigated revolution range and the power is assumed to be constant over the segment of the RPM distribution. In this case, the compressor is consuming 175 kW for 40% of total operating time, 125 kW for 35% of total operating time and 225 kW for 25% of total operating time. Based on the long term averaged data available, approximately 60% of the input compressor power in each situation is transformed into useful pressure energy (exergy) of compressed air. So according to the energy transformation processes explained previously, less than 40% of the input energy can be taken for heating purpose and this energy is variable due to the nature of the control. In fact, there is no sophisticated usage of the warm air. Moreover, inlet air energy is also affected by mass flow rate change related to variable revolutions of the compressor shaft, but as the compressor is cooled by ambient air, only a minor contribution to the total output heat balance can be expected. The result is a significant rejection of energy in the form of heat.

3.2 Load/Unload

The next compressor control mode considered is load/unload which is typical of fixed speed compressors. **Figure 3** shows the time profile record of 900 seconds for three 2-stage oil-free rotary screw water-cooled compressors that are sequenced using the controller. The base unit is loaded all the time resulting in a potentially stable source of heat energy as nearly 45% of the input compressor energy is rejected in the form of heat and minor losses. In this particular case, two other compressors are loaded 60% of the running time in a regular symmetric pattern as shown in Figure 3, which makes them periodically cycling heat generators since there is no heat production from compression during unloaded time because the inlet valve is closed. The heat captured by cooling water is released in cooling towers.

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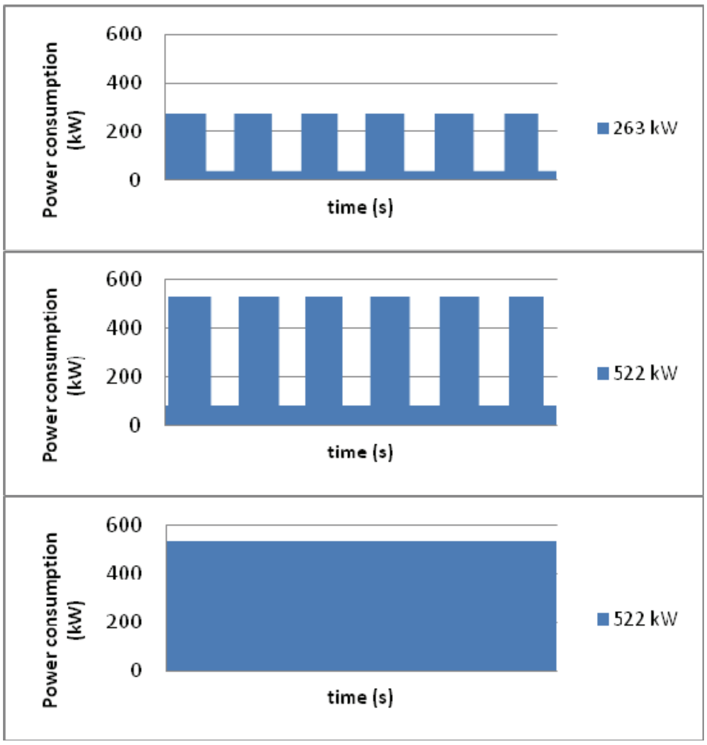

Figure 3: Load/unload profiles

3.3 Inlet modulation

The last traditional compressor control mode to mention briefly is inlet modulation, although this has not been observed in an industrial setting in this study. As the inlet valve modifies the mass flow rate going through compressor, the situation would be similar to VSD control, but revolutions of the motor are not modified and hence the power consumption is not changing so considerably. The efficiency of the transformation of the input energy into useful pressure energy decreases with closing the inlet valve, which means an increase of the rejected heat energy and mainly process losses. Inlet modulation is the least efficient control mode of positive displacement compressors and is not widely applied. On other hand, it is an efficient control strategy for dynamic compressors though in a limited turndown range.

3.4 Summary.

The above described controls are more familiar in connection with an electric motor as compressor drive, primarily because electric motors have historically dominated the market. However, as previously noted, the compressor can be driven by more types of prime mover; especially natural gas mechanical drives. Although these alternative engines have been available for over 20 years, they are not yet widespread. Note that regardless of type of compressor prime mover, the compressor will generate heat in the same way for each control mode as discussed above. Thus, based on the data preented above, there is a substantial heat source available, albeit potentially intermittent. Moreover if employed at all, heat rejected from a compressor driven by electric motor is usually used for some heating purposes and is not typically regenerated. Whereas heat utilization by regeneration, which will yield a higher efficiency, is possible with other compressor drives and is discussed in the next section.

4 HEAT UTILIZATION FROM COMPRESSOR MECHANICAL DRIVES

In order to demonstrate the motivation for this analysis consider a simple calculation: consider a compressor (of any power) driven by an electric motor with a power consumption 5% higher than rated compressor power, to allow for losses in the electric drive, working loaded 8000 hours per year. Using industrial energy prices in Ireland of 0.12 €/kWh for electric energy and 9.72 €/GJ for natural gas /SEI08/, the required thermal efficiency of an replacement gas cycle would be η_{th} = 0.28 to achieve the same delivered compressed air at the same operating costs. This is based on the long term average power consumption and air supply data that have been collected at several industrial sites. Thus, a gas engine with higher efficiency would result in better operating economics than an electric motor, assuming the total cost of ownership, e.g. finance and maintenance, as comparable. Principally, thermal engines work on three different basic types of ideal gas cycle: Otto cycle (spark-ignition engine), Diesel cycle (compression-ignition engine) and Brayton cycle (gas turbines). The advantage of some thermal cycles is to regenerate heat within the cycle, which is the best way of heat utilization, leading to the higher efficiency of the process. This is typical of gas turbine cycle with the regenerator located behind the last turbine stage using gas leaving turbine as source of heat. The **Figure 4** shows the T-s diagram of gas-turbine cycle with inter-cooling and reheating with ideal 2-stage compression and 2-stage expansion.

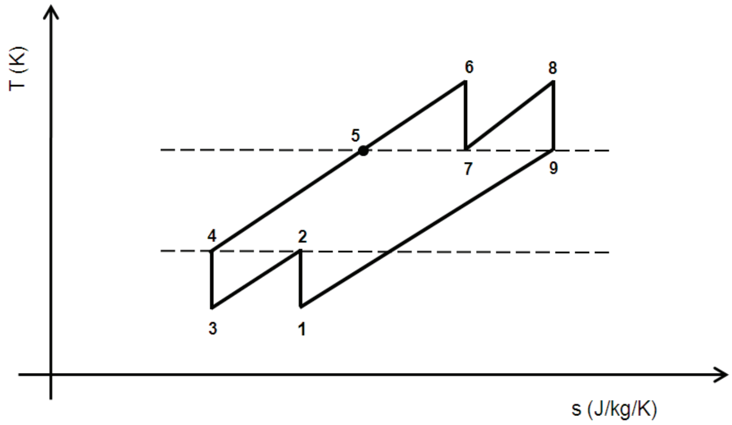

Figure 4: Ideal gas-turbine cycle

After implementation of the regenerator, thermal efficiency of the simplified actual gas cycle is expressed using Equation 3, which shows, that the regenerator does not improve the work of the cycle (numerator), but it does reduce the amount of the input energy to the cycle. Equation 3 also shows the effect of inefficiencies of the other main components.

$$\eta_{th} = \frac{\eta_t[(T_6-T_7)+(T_8-T_9)] - \frac{1}{\eta_C}[(T_2-T_1)+(T_4-T_3)]}{(1-\eta_{reg})(T_9-T_4)+(T_6-T_5)+(T_8-T_7)} \tag{3}$$

A representative value of thermal efficiency of the mentioned gas turbine cycle could be around 0.4, which gives an efficiency increase of almost 43% compared to an electric motor scenario. Moreover, Dellenback /Del02/ has proposed an alternative regenerator configuration between turbine stages leading to the higher efficiency of the gas turbine for practical ranges of operational parameters, which could bring even more energy savings associated with compressed air generation. Nevertheless, thermodynamic efficiencies of the turbine η_t and compressor η_c together with efficiency of the regenerator η_{reg} play crucial roles in the overall thermal efficiency of the cycle and improvement of these parameters has been subject to the long term research effort in gas turbines. In the case of the regenerator, a minimal pressure drop is desired. As result, there has been a significant increase in gas turbine efficiencies in recent years making them more than competitive alternatives to the Diesel engines or Otto engines.

In general, Diesel engines are slightly more efficient than spark-ignition engines due to higher operating compression ratios. Although, the benefit of the regenerator is evident, it is also theoretically limited and can be used only within a range of lower pressure ratios depending on the ratio of the maximum to minimum temperature of the gas in the course of cycle. However, high pressure ratios (e.g. over 20) are not typical for gas-turbine engines, and so this is not a practical limitation. It can be also noted that if the large balanced heat load exists, the compressor of gas turbine cycle is a potential source of heat as described in the previous section. The main difference is that, no heat can be recovered after the final compressor stage, because the effect of high-temperature (and high-pressure) air is necessary for the process of fuel burning in the combustion chamber. It should also be noted that the low grade heat residue leaving the regenerator could be further used for some minor heating purposes.

In the case of a Diesel engine working as a compressor mechanical drive, exhaust heat utilization for heating can be typical, but there is also the possibility of increasing power output from the engine i.e. efficiency of the process by waste heat driven power cycles as already shown /Aly87/ .

Unfortunately, to the authors' knowledge none of these alternatives are used in compressed air systems installed in industrial sites in Ireland. This huge energy saving potential related to the generation of the compressed air and also improvements of the application side of the compressed air system, where the biggest research opportunities have been found, will contribute to increase of overall efficiency of the compressed air systems.

5 CONCLUSION

In terms of compressed air systems optimization and operating energy/cost reduction, it is important to understand correctly all processes related to the generation of compressed air. As a part of it, energy transformations within the compressor and heat generation under various compressor loads are detailed. Moreover, heat from alternative compressor drive and its best utilization are briefly discussed.

Surveys taken over several companies in Ireland revealed that compressor generated heat is not generally integrated into manufacturing or utility processes significantly and an option of a more efficient natural gas driven engine as compressor prime mover is still unused, although relevant solutions such as more advanced heat recovery systems or efficient mechanical drive with regenerator are extant. Thus, there is a potentially large energy saving potential related to generation of the compressed air which can be readily exploited.

ACKNOWLEDGEMENT

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