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The Potential of Energy Savings in Oxygen Production by Pressure Swing Adsorption

Radek Šulc*, Pavel Ditl

Czech Technical University in Prague, Faculty of Mechanical Engineering, Department of Process Engineering, Technicka 4, Prague, Czech Republic
Radek.Sulc@fs.cvut.cz

The pressure swing adsorption (PSA) units are widely used as an oxygen source. The second PSA unit outgoing stream contains a dry mixture of O_2 in N_2 that is mostly exhausted into the atmosphere. Utilization of this stream's potential and waste compression heat leads to an increase in the unit's overall energetic capacity and the ecologically more friendly processing. This paper aims to demonstrate the above-described possibilities in the case of the oxy-fuel combustion unit. The following four options for integrating PSA technology were analyzed: i) single or dual compression, ii) utilization of waste compression heat for coal or biomass dewatering, iii) utilization of dry waste gas from the PSA unit for coal or biomass dewatering, and iv) energy recovery by an expansion of pressurized oxygen before combustion. The greatest potential for practical application was found for the usage of dual compression (saving of 10 % of electricity) and utilization of waste compression heat for coal or biomass dewatering. In this case, the saving of 5.3 % and 10.4 % of lignite and wood respectively can be reached depending on fuel moisture for reference fuel conditions.

1. Introduction

The oxy-fuel combustion and biomass co-firing under oxy-fuel discharge used for power generation are promising technologies reducing CO_2 emissions (Fogarasi and Cormos, 2017). This combustion technology utilizing oxygen instead of air (oxy-fuel combustion) enables us to increase the CO_2 concentration in flue gases and, in this way, to affect the CO_2 separation and purification costs. The process requires oxygen purity of at least 95 %, significantly affecting the investment and operational costs. The oxygen of the needed purity can be obtained by cryogenic air separation (ASU) or pressure swing adsorption (PSA). The case studies are usually performed for big power plants exceeding 500 MW_{el} and cryogenically produced oxygen (e.g., Fogarasi and Cormos, 2017). Fu et al. (2014) proposed an advanced cryogenic the ASU based on self-heat recuperation technology and one distillation column producing 95 % oxygen purity for the IGCC system. Soundararajan et al. (2014) analyzed the effect of operating pressure, oxygen purity, and downstream purification parameters on the performance of oxy-combustion coal-based power plants and simultaneous capture of at least 90 % of the CO_2 produced. The power demand also depends on the ASU configuration (Manenti et al., 2013).
The question arises of the attractive technology option of oxygen supply for cleaner heat and power production from small oxy-combustion units, typical examples of decentralized power supply systems.
Small coal-fired or biomass-fired heat and power generating units are typical examples of decentralized power supply units where the oxygen supply by pressure swing adsorption can be recommended. Šulc and Ditl (2020) calculated the specific energy consumption of approx. 0.803 kWh kg_{O2}^{-1} for the required 95 % oxygen purity for the POC8900 PSA unit (manufacturer INMATEC Inc., Germany).

2. The potential of energy savings

Skarstrom (1960) designed the first PSA technology for oxygen enrichment using nitrogen selective zeolite 5A. A comprehensive overview and insight into the science and technology of pressure swing adsorption are presented e.g. by Ruthven et al. (1993). In the PSA unit, the oxygen is separated from pressurized air; the energy needed for air compression is the largest cost item. The dry waste stream from the PSA unit, which is mostly exhausted into the atmosphere, enables fuel pre-drying. The utilization of waste compression heat is also a way to increase energy efficiency.

Banaszkiewicz et al. (2014) analyzed three options for integrating PSA technology with an oxy-combustion power plant: i) electricity produced by a power plant for PSA technology only, ii) combined pressure-temperature swing adsorption (PTSA) technology with the use of electricity and waste heat at an elevated temperature (90°C), and iii) PTSA technology using power and waste heat at a lower temperature (40°C), brought to a high temperature transferred by an absorption heat pump. They found that the utilization of waste heat slightly improves oxygen production economics. Chorowski and Gizicki (2015) analyzed technical and economic aspects of oxygen separation for oxy-combustion of black coal and lignite. They analyzed capabilities for utilization of waste energy fluxes and waste nitrogen from ASU for coal drying. They found that the use of waste nitrogen at ambient temperature enables to increase of calorific values of lignite by 6 % and to decrease fuel consumption by about 5.6 %. When heating waste nitrogen by waste heat from compressors up to 55°C, the moisture of lignite can be reduced to about 10 % that corresponds to the moisture content reached in a standard drying unit.

This paper aims to analyze the following four options for integrating PSA technology: i) single or dual compression, ii) utilization of waste compression heat for coal or biomass dewatering, iii) utilization of dry waste gas from the PSA unit for coal or biomass dewatering, and iv) energy recovery by an expansion of pressurized oxygen before combustion.

3. Oxy-fuel combustion

According to ISO 8573-1:2010, PSA units require an inlet air quality of class 1.4.1 (solid particles, air humidity, oil). It is necessary to treat the compressed air to achieve entering air quality. A refrigeration air drying produces an air of the required pressure dew point of +3 °C.

Hrdlička et al. (2016) investigated oxy-fuel combustion of lignite type coal experimentally in a bubbling fluidized bed in a laboratory-scale combustor. The pure oxygen consumption of 6.7 Nm^3 h^{-1} was found for a thermal power load of 35 kW. The analysis was carried out for on-site oxygen production using two-bed Pressure Swing Adsorption, thermal power load of 500 kW, and oxygen recovery characterized by air ratio AirF of 10 Nm^3 Nm^{-3} for 95 % purity (the POC8900 PSA unit; manufacturer INMATEC Inc., Germany). In this case, the flow rate of 95 % oxygen (GOX) required is 101 Nm^3 h^{-1}, and the corresponding compressed air flowrate is 1,010 Nm^3 h^{-1} with a pressure dew point of +3°C.

4. Oxygen by Pressure Swing Adsorption

4.1 Air compression

We formed the following model for calculating the electric power supply needed for the required oxygen production. The volumetric air flowrate at the PSA unit inlet at standard conditions is calculated as it follows:

$$\dot{V}_{PSA}^{N} = AirF \cdot \dot{V}_{GOX}^{N} , \tag{1}$$

where $\dot{V}^{N}_{PSA}$ is the volumetric airflow rate at PSA unit inlet at standard conditions (Nm^3 s^{-1}), AirF is air ratio defined as the ratio of airflow rate at PSA inlet and produced gaseous oxygen stream (Nm^3 Nm^{-3}), $\dot{V}^{N}_{GOX}$ is the volumetric flowrate of gaseous oxygen stream produced by PSA unit at standard conditions (Nm^3 s^{-1}).

The mass flow rate of dry air at a compressed air line is calculated as it follows:

$$\dot{m}_{d.a..} = \frac{M_{d.a.}}{((M_{d.a.}/M_{water}) \cdot x_{water-PSA} + 1)} \cdot \frac{\dot{V}_{PSA}^{N}}{22.4135} , \tag{2}$$

where $\dot{m}_{d.a.}$ is the mass flow rate of dry air (kg s^{-1}), $x_{water\text{-}PSA}$ is the specific air humidity at PSA unit inlet (kg $_{water}$ kg $_{d.a.}^{-1}$), $M_{d.a.}$ is the molar mass of dry air (kg kmol^{-1}), and M_{water} is the molar mass of water (kg kmol^{-1}).

The volumetric air flowrate at the compressor inlet at standard conditions is calculated as follows

$$\dot{V}_{comp}^{N} = 22.4135 \cdot \frac{((M_{d.a.}/M_{water}) \cdot x_{water-comp} + 1)}{M_{d.a.}} \cdot \dot{m}_{d.a..} , \tag{3}$$

where $\dot{V}^{N}_{comp}$ is the volumetric air flowrate at compressor inlet at standard conditions (Nm^3 s^{-1}) and $x_{water\text{-}comp}$ is the specific air humidity at compressor inlet (kg $_{water}$ kg $_{d.a.}^{-1}$).

The specific humidity at a given stream is calculated as it follows:

$$x_{water} = \frac{M_{water}}{M_{d.a.}} \cdot \frac{\varphi \cdot p_{water}^{SAT}}{(p - \varphi \cdot p_{water}^{SAT})}, \tag{4}$$

where φ is relative humidity (-), p^{SAT}_{water} is saturated water vapor pressure at stream temperature (kPa), p is air stream pressure (kPa).
The compressor electrical power input is calculated as follows:

$$P_M = -\frac{f_P \cdot (P_{ise} / \eta_c)}{\eta_M} = -\frac{f_P}{\eta_c \cdot \eta_M} \cdot p_1 \cdot \dot{V}_{air1} \cdot \frac{\kappa}{\kappa - 1} \cdot \left[1 - \left(\frac{p_2}{p_1}\right)^{(\kappa-1)/\kappa}\right], \tag{5}$$

where P_M is compressor electrical power input (kW), P_{ise} is an isentropic power input (kW), η_c is compressor efficiency (-),η_M is electromechanical efficiency of the gearbox and electrical motor (-), f_P is power factor representing power input of compressor accessories (-), p_1 is inlet air pressure (kPa), $V^{\bullet}_{air1}$ is the air flowrate (m^3 s^{-1}) at inlet conditions (temperature t_1, pressure p_1), p_2 is outlet air pressure (kPa), κ is Poisson constant (-).
The outlet temperature of compressed air which has been usually reached in standard compressor units is approx. 10° C above inlet air temperature. To ensure temperature difference for heat transfer we assume installation of compressed air/oil cooler (A/O cooler) for waste heat extraction and compressed air/ ambient air cooler (A/A cooler) for compressed air cooling onto expected outgoing temperature. The cooling capacity is calculated as it follows:

$$\dot{Q}_c = \dot{m}_{d.a.} \cdot (h_{in}^{air}(t_{in}, x_{in}) - h_{out}^{air}(t_{out}, x_{out})), \tag{7}$$

where $Q^{\bullet}_c$ is the cooling capacity of the cooler (kW), h^{air}_{in} (t_{in}, x_{in}) is the wet air enthalpy (kJ kg^{-1}) at cooler inlet at inlet conditions (temperature t_{in}, humidity x_{in}) and h^{air}_{out} (t_{out}, x_{out}) is the moist air enthalpy (kJ kg^{-1}) at the cooler outlet at outlet conditions (temperature t_{out}, humidity x_{out}). The enthalpy of moist air is calculated as it follows:

$$h^{air} = c_{pd.a.} \cdot t + x_{water} \cdot (c_{pwv} \cdot t + l_{water}) = 1.010 \cdot t + x_{water} \cdot (1.840 \cdot t + 2500), \tag{8}$$

where h^{air} is the enthalpy of wet air (kJ kg^{-1}), t is the air temperature (°C), $c_{pd.a.}$ is the specific heat capacity of dry air (kJ kg^{-1} K^{-1}), $c_{pwv.}$ is the specific heat capacity of water vapors (kJ kg^{-1} K^{-1}), and l_{water} is the latent heat of water (kJ kg^{-1}).

Table 1: Air compression – single- and dual compression

Description	Unit	Single compression	Dual compression[*3]	
			1st stage	2nd stage
Air temperature – inlet	°C	20	20	30
Air pressure – inlet	kPa	100	100	305
Temperature after compression	°C	250.5	133.7	133.6
Pressure after compression	kPa	762	315	762
A/O cooler – outlet temperature	°C	40	40	40
A/A cooler – outlet temperature	°C	30	not installed	30
Compressed air pressure - outlet	kPa	750	x	750
A/O cooler – heat duty	kW	82.4	34.9	38.8
A/A cooler – heat duty	kW	6.13	not installed	6.13
Compressor power input [*1,2]	kW	106.6	52.7	43.4

Note: ambient air: temperature of 20 °C, pressure of 1 bar (a), relative humidity of 70 %.
Note: [*1] Compressed air: outlet temperature of 30 °C, outlet pressure of 7.5 bar (a).
Note: [*2] Electrical efficiency: single compression η_{EM} = 0.954, dual compression η_{EM} = 0.952.
Note: [*3] Dual compression: overall compressor power input = 96.1 kW, overall A/O cooler duty = 73.7 kW.

The compressor electrical power input was calculated for intake air temperature of 20°C, pressure of 100 kPa and humidity of 70 %, κ = 1.4, η_c = 0.90, η_{GB} = 0.96. The compressor efficiency was taken for an oil-injected screw compressor (Renner, 2020). The electrical efficiency η_{EM} was estimated according to EN 60034-2-1 for a 4-pole asynchronous electrical motor of the IE3 class. For dual compression, the single shaft arrangement is

assumed. The power input of accessories (fans, pumps) was taken into account by a factor of 1.03. The water condensation during air compression was taken into account.
The calculated data for single and dual compression are summarized in Table 1. For single compression, the air/oil cooler and the air/air cooler are installed as after-cooler. For dual compression, the air/oil cooler is installed as an inter-cooler between the 1st and 2nd stages. After the 2nd stage, the air/oil cooler and the air/air cooler are installed as after-cooler. The output pressure of the1st stage was fitted to obtain the same temperature after compression to maximize heat recovery. The pressure losses of 10 kPa and 2 kPa were assumed for the air/oil cooler and the air/air cooler respectively

4.2 Refrigeration drying

The cooling capacity of dryer $Q^{\bullet}_{c\text{-}DR}$ without heat regeneration was calculated analogically using Eqs. (7) and (8). The refrigeration dryer electrical power input is calculated as it follows:

$$P_{RD} = \frac{f_{P-RD}}{\eta_{c-RD} \cdot \eta_{M-RD}} \cdot \dot{Q}_{c-DR} \cdot \left[f_{CF} \cdot \left(\frac{h_{11} - h_{12}}{h_{10} - h_{13}} \right) - 1 \right], \tag{9}$$

where P_{RD} is dryer compressor electrical power input (kW), $\eta_{c\text{-}RD}$ is compressor efficiency (-), $\eta_{M\text{-}RD}$ is electromechanical efficiency of gearbox and electrical motor (-),f_P is power factor representing power input of compressor accessories (-), f_{CF} is coolant factor representing coolant excess due to heat transfer resistances (-), (h_{11} – h_{12}) is the coolant enthalpy change in refrigerator condenser (kJ kg^{-1}) and (h_{13} – h_{10}) is the coolant enthalpy change in refrigerator evaporator (kJ kg^{-1}). The dryer compressor electrical power input was calculated for following parameters: (h_{11} – h_{12}) =190.38 kJ kg^{-1}, (h_{10} – h_{13}) = 171.72 kJ kg^{-1}, $\eta_{c\text{-}RD}$ = 0.76 (scroll compressor), $\eta_{GB\text{-}RD}$ = 0.96, $\eta_{EM\text{-}RD}$ = 0.877, $f_{P\text{-}RD}$ = 1.05, f_{CF} = 1.05. The electrical efficiency was estimated according to EN 60034-2-1 for a 4-pole IE3 class asynchronous electrical motor. The compressor efficiency was estimated for a coolant scroll compressor (Ingersoll-Rand, 2020). The dryer electrical power input calculated is presented in Table 2.

4.3 Total electrical power input

The total electrical power input for single- and dual compression technology is presented in Table 2. The electrical power input of the PSA unit is declared as 0.15 kW by the manufacturer. Using dual compression, the 10 % of electrical energy can be saved comparing with single compression as a reference case.

Table 2: Air compression – total electrical power input

Description	Unit	Single compression	Dual compression
Compressor power input	kW	106.6	96.1
Refrigeration dryer power input*1	kW	3.4	3.4
PSA unit	kW	0.15	0.15
Total electrical power input	kW	110.2	99.7
Specific energy consumption	kWh Nm^{-3}_{GOX}	1.091	0.987
Specific energy consumption	kWh kg^{-1}_{O2}	0.805	0.728
Energy saving	%	-	-9.5

Note: *1 Dryer: compressed air: pressure dew point of +3 °C.

Table 3: Energy recovery by expansion of pre-heated pressurized oxygen (GOX)

Description	Unit	Oil 80/30 °C *1	Oil 123/30 °C *2	Oil 240/30 °C *3
GOX temperature – inlet	°C	20	20	20
GOX pressure – inlet	kPa	500	500	500
GOX temperature – after preheating	°C	70	113	230
Pressure after expansion	kPa	150	150	150
Temperature after expansion	°C	-30	0.6	83.6
GOX/O pre-heater – heat duty	kW	2	3.7	8.3
Expander power output *4	kW	2.6 4a	3 4b	3.9 4c

Note: *1 Heating medium oil 80/30 °C: utilization of waste heat from both single- and dual compression unit.
Note: *2 Heating medium oil 123/30 °C: utilization of waste heat from both single- and dual compression unit.
Note: *3 Heating medium oil 240/30 °C: utilization of waste heat from single compression unit only.
Note: *4 Electrical efficiency: 4a η_{EM} = 0.867 for oil 30/80 °C, 4b η_{EM} = 0.877 for oil 30/123 °C, 4c η_{EM} = 0.877 for oil 30/240 °C.

5. Energy recovery by an expansion of pressurized oxygen before combustion

The pressurized oxygen (GOX) produced by the PSA unit is stored in a product tank at elevated pressure around 4-6 bars. The oxy-fuel combustion usually occurs at the pressure near atmospheric pressure. Thus, the GOX is throttled before combustion. Instead of energy losing throttling, the GOX reduction by adiabatic expansion has been assumed. The GOX preheating using waste compression heat was analyzed for three heating media of different inlet temperatures: i) 80 °C, ii) 123 °C, and iii) 240 °C. A 10° C difference between hot and cold streams is assumed in a preheater. The expander electrical power output and heat duty for GOX preheating was calculated analogically as in previous parts. The computed data for inlet GOX pressure of 500 kPa and expansion pressure of 150 kPa are summarized in Table 3.

6. Fuel dewatering

Assuming that water entering the PSA unit outflows in the waste PSA stream the mass flowrate of 1,158 kg h^{-1} of dry waste PSA stream and humidity of 0.0007 kg_{H2O} $kg_{d.g}^{-1}$ can be obtained by mass balance. To utilize the waste compression heat the ambient air and the mixture of ambient air and waste PSA stream were considered for fuel dewatering. The specific drying capacities and available drying capacities for three drying media calculated assuming the ideal isenthalpic drying and fully saturated outgoing drying medium are presented in Tables 4 and 5 respectively.

Table 4: Specific drying capacity for various drying gases and heating media

Heating medium	Description	Unit	Air *1	Waste PSA *2	Mixture
	Drying gas temperature – inlet	°C	20	20	20
	Drying gas pressure – inlet	kPa	100	100	100
	Drying gas humidity – inlet	kg_{H2O} $kg_{d.g.}^{-1}$	0.0103	0.0007	0.0067
80°C/30 °C	Drying gas temperature – after preheating	°C	70	70	70
	Drying gas temperature after drying	°C	29.6	24.7	28.7
	Specific drying capacity	kg_{H2O} $kg_{d.g.}^{-1}$	0.0163	0.0185	0.0169
123°C/30 °C	Drying gas temperature – after preheating	°C	113	113	113
	Drying gas temperature after drying	°C	36.8	33.2	35.6
	Specific drying capacity	kg_{H2O} $kg_{d.g.}^{-1}$	0.0305	0.0324	0.0316
240°C/30 °C	Drying gas temperature – after preheating	°C	230	230	230
	Drying gas temperature after drying	°C	48.9	46.9	47.2
	Specific drying capacity	kg_{H2O} $kg_{d.g.}^{-1}$	0.0720	0.0736	0.0735

Note: *1,2 Molar weight (kg $kmol^{-1}$) of drying gas (on dry basis): 28.968 for air*1, 28.595 for waste PSA stream*2.
Note: Drying gas outgoing from a dryer: relative humidity φ = 100 %.

Table 5: Drying capacity utilizing waste compression heat

Description		Unit	Air *1	Waste PSA *2	Mixture *3
Single compression					
80 °C/30 °C	Heater – duty	kW	82.4	16.7	82.4
	Drying capacity	kg_{H2O} h^{-1}	93.9	21.4	95.4
123 °C/30 °C	Heater – duty	kW	82.4	31	82.4
	Drying capacity	kg_{H2O} h^{-1}	94.6	37.5	96
230 °C/30 °C	Heater – duty	kW	82.4	70	82.4
	Drying capacity	kg_{H2O} h^{-1}	98.8	85.2	99.8
Dual compression					
80 °C/30 °C	Heater – duty	kW	73.7	16.7	73.7
	Drying capacity	kg_{H2O} h^{-1}	84	21.4	85.6
123 °C/30 °C	Heater – duty	kW	73.7	31.1	73.7
	Drying capacity	kg_{H2O} h^{-1}	84.7	37.5	86.1

The lignite coal is usually very moist with a moisture content up to 50 wt. % which is reduced in standard drying units to about 10 wt. % representing a minimal value of free water content (Chorowski and Gizicki, 2015). For raw lignite coal of 40 wt. % moisture and LHV of 14.8 MJ kg^{-1} dried up to 10 wt. % moisture the drying capacity of 48.7 kg_{H2O} h^{-1} is required for 500 kW of thermal load. For drying of raw wood of 50 wt. % moisture and LHV of 8.074 MJ kg^{-1} to 20 wt. % moisture which is suitable for combustion (Gebreegziabher et al, 2013) the drying capacity of 86.5 kg_{H2O} h^{-1} is required for 500 kW of thermal load. As follows, the waste compression heat is capable to dry a consumed fuel and to save 5.3 % or 10.4 % of lignite or wood respectively.

7. Conclusions

This paper aimed to demonstrate the potential of consumed energy-saving integrating PSA technology as an oxygen source for the oxy-fuel combustion unit to reach the ecologically more friendly processing.
The following four options were analyzed: i) single or dual compression, ii) utilization of waste compression heat for coal or biomass dewatering, iii) utilization of dry waste gas from the PSA unit for coal or biomass dewatering, and iv) energy recovery by an expansion of pressurized oxygen (GOX) before combustion.
The following conclusions can be drawn from the technical analysis presented in this paper:

- Using dual compression, 10 % of electrical energy may be saved, and specific energy consumption decreases from 0.805 kWh kg_{O2}^{-1} to 0.728 kWh kg_{O2}^{-1} compared to a single compression. It represents a saving of 85.68 MWh y^{-1} for the annual operating time of 8,160 hours.
- The GOX expansion is capable to recuperate 2.7 % of electrical energy.
- Utilization of low-grade waste compression heat for fuel drying enables to reduce fuel consumption depending on fuel moisture, e.g. by 5.3 % or 10.4 % of lignite or wood respectively for reference fuel conditions.

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