25.05.2020

Analysis and Improvement of Pressure Swing Adsorption System

Gas Plant Instrumentation

George Øvrebø
Magnus Herland
Lars Myrestølen

Bachelor’s thesis in Mechanical Engineering

ANALYSIS AND IMPROVEMENT OF PRESSURE SWING ADSORPTION SYSTEM

George Øvrebø

Lars Vegard Stakseng Myrestølen

Magnus Herland

Department of Mechanical- and Marine Engineering

Western Norway University of Applied Sciences

NO-5063 Bergen, Norway

Høgskulen på Vestlandet

Fakultet for Ingeniør- og Naturvitskap

Institutt for maskin- og marinfag

Inndalsveien 28

NO-5063 Bergen, Norge

Norsk tittel: Analyse og Forbedring av Trykk Sving Absorbsjons System

Author(s), student number: George Øvrebø, 180319

Lars Vegard Stakseng Myrestølen, 183080

Magnus Herland, 571638

Study program: Mechanical Engineering

Date: [June 2020]

Report number: IMM 2020-M20

Supervisor at HHVL: Boris Balakin

Assigned by: Equinor ASA

Contact person: Johannes Fjeldstad

Antall filer levert digitalt: 1

Foreword

This bachelor thesis is written as a mandatory project every student graduating the Department of Mechanical and Marine Engineering at Western Norway University of Applied Sciences (WNUAS) must complete. We are students in the program of mechanical engineering. The thesis is written in cooperation with Equinor Kollsnes as an industrial challenge they experienced with their air dryers. During the work on this project we had use of several our earlier subjects, such as Thermodynamics, Fluid Mechanics and Hydraulics, Hydraulic Machinery and Pumps. Our internal sensor was Professor Boris Balakin and external sensor and contact person was Johannes Fjeldstad at Equinor Kollsnes.

It could be mentioned that this thesis is written during the flu pandemic of 2020, and we tried to make it to not have any impact of the quality of this paper. It did show to be difficult to meet in person with both our internal and external sensors, but we did our best to work around this, worked separately with good dialogs over mail and web meetings.

We have received great guidance throughout the project and want to specially thank

Dr. Müller who provided useful papers and long experience in the field.

Chemical Engineer Johannes Fjeldstad who proved to be a valuable contact person that gave us important data needed to solve the task.

Professor Boris Balakin for great guidance and expertise throughout the project.

Magnus Herland	George Øvrebø	Lars Myrestølen
Magnus Herland	George Øvrebø	Lars Myrestølen

Abstract

Air drying is a process of removing unwanted amounts of steam in the air. There are several ways this is done in the industry. This paper will focus on air drying through a Pressure Swing Adsorption System, also called a PSA system. The main point here is not building a new air dryer, but improving an existing one that underperforms. But research-wise it could be used for both. In the present situation, the air dryer is not capable of handling the desired amount of flow, or exceeds the maximum amount of moisture allowed in the system.

The team will look for solutions by figuring out how the system works today, through research and calculations. And will then proceed to calculate how the system must look like to work desirable. It will also be done research on how to possibly add a Temperature Swing System (TSA), and on how that could improve todays operations.

Sammendrag

Luft tørking er en prosess for å fjerne uønskede mengder damp i luften. Det er flere måter dette gjøres i bransjen. Denne artikkelen vil fokusere på luft tørking gjennom et Pressing Swing Adsorpsjon System, også kalt et PSA-system. Hovedpoenget her er ikke å bygge en ny lufttørker, men å forbedre en eksisterende som under utfører. Men forskningsmessig kan det brukes til begge deler. I denne situasjonen er ikke luft tørkeren i stand til å håndtere den ønskede mengden strømning, eller overskrider den maksimale mengden damp som er tillatt i systemet.

Teamet vil se etter løsninger ved å finne ut hvordan systemet fungerer i dag, igjennom å se på tidligere arbeid på luft tørkere og beregninger. Og vil deretter fortsette med å beregne hvordan systemet må se ut for å fungere ønskelig. Det vil også bli undersøkt hvordan en eventuelt kan legge til et temperatursvingsystem (TSA), og hvordan dette kan forbedre dagens drift ytterligere.

Contents

Nomenclature

Symbols

A	$= cross\ section$	$[m^2]$
C	$= coefficient\ of\ disgharge$	
c_p	$= isobaric\ specific\ heat$	$[kJ/kg \cdot K]$
c_l	$=\ latent\ heat$	$[kJ/kg]$
d	$= diameter$	$[m]$
D	$= pipe\ diameter$	$[m]$
E	$= electric\ power$	$[kW]$
G	$= gas\ feed\ rate$	$[kg/s \cdot m^2]$
H_m	$= molal\ humidity$	$[m_v/m_a]$
h	$= enthalphy$	$[J/kg]$
k	$= permeability$	$[m^2]$
L	$= lenght$	$[m]$
M	$= molar\ mass$	$[g/mol]$
m	$= mass$	$[kg]$
$\dot{m}$	$= mass\ flow$	$[kg/s]$
P	$= pressure$	$[Pa]$
R	$= ideal\ gas\ constant$	$[J/mol \cdot K]$
RH	$= relative\ humidity$	$[\%]$
t	$= time$	$[s]$
T	$= absolute\ temperature$	$[K]$
V	$= volume$	$[m^3]$
v	$= velocity$	$[m/s]$
W	$= power$	$[kW]$
Q	$= volume\ flow\ rate$	$[m^3/s]$
Z	$= compressibillity\ factor$	

Greek symbols

β	$= diameter\ ratio$	
Δp	$= differential\ pressure$	$[pa]$
ΔT	$= differential\ temperature$	$[K]$
Δx	$= length\ og\ bed$	$[m]$
ε	$= expansibility\ factor$	
ε_p	$= external\ particle\ void\ fraction$	
λ	$= regeneration\ period$	$[s]$
η	$= efficiency$	
μ	$= dynamic\ viscosity$	$[Pa \cdot s]$
ρ	$= density$	$[kg/m^3]$
σ	$= adsorption\ capacity$	$[m_w/m_d \cdot s]$
ω	$= regeneration\ fraction$	

$Subscripts$

$1,2,3 \quad = stages$
$a \quad = air$
$c \quad = collumn$
$cr \quad = critical$
$COMP \quad = compressor$
$d \quad = desiccant$
$e \quad = heating\ coil$
$f \quad = fraction$
$FILT \quad = filter$
$FO \quad = flow\ orifice$
$INST \quad = instrument$
$P \quad = pipe$
$R \quad = relative$
$r \quad = residence$
$s \quad = standardized$
$v \quad = vapor$
$w \quad = water$
$\varepsilon \quad = patricle\ void$
$\lambda \quad = regeneration\ period$

Introduction

Equinor at Kollsnes is a natural gas processing plant that started up in 1996 and has since increased their size and production capability. They receive wet gas, or LNG (liquefied natural gas), from offshore platforms. This wet gas is separated into dry gas, then compressed and delivered. The processing plant at Kollsnes is accountable for about 40% of all the dry gas exported from Norway [1].

At Kollsnes there are two sets of air dryers, the purpose of these air dryers is to make sure the compressed air supplying the utility system and instrumentation is free of moisture. The consumers consists of working air and instrument air. Moisture in the utility system can lead to corrosion, wear and tear on valves and when the outside temperature drops to below zero it can also lead to freezing in the pipes [2]. The air dryers were installed in 1996, and due to the lower flow compared to today, one dryer set was capable of drying the entire air flow. That meant that one set could do all the drying while the other one was in standby. In this thesis this setup will be noted as 2x100%, and where there is need for both driers to work constantly, 2x50% will be used. The dewpoint requirement at startup was -40°C (we will come back to dewpoint later). While one dryer was active the other was in standby, then after 8 hours they switched. Running all the flow through one dryer was practical when they had to change the desiccants, meaning they could change the one in standby without interfering with the flow. Compared to today's setup, where they have to run a much larger flow through one dryer, which it is not design for, making the drying less efficient, and air with a higher humidity is delivered to the consumers.

The PSA (pressure swing adsorption) system was invented in the mid 50's, and patented by its inventor Dr. Charles W. Skarstrøm in 1960 [3]. A PSA system has at least two columns in parallel, the columns are filled with a desiccant. The desiccant is a porous material with a small diameter and a large surface area. Desiccants can be found in different shapes and of different material, and in this system there is active alumina ($\gamma - Al_2O_3$) in the shapes of beads. Wet pressurised air flows into one column and the water vapor is adsorbed by the desiccant, and the result is dry pressurised air. A part of this dry air is used to regenerate (dry) the other bed, that previously was the adsorbing one. This changes back and forth, so one bed is always adsorbing and one regenerating. This cycle is called a NEMA cycle. In this situation each column has 5 minutes of adsorbing then 5 minutes of regenerating, so a total 10-minute NEMA cycle. For regeneration dry air flows through the top (downstream) of the depressurized column, and dries the desiccant before it is exhausted to atmosphere at the bottom of the column.[4]

As a result of the increased size of the processing plant the need for dry compressed working and instrument air has also increased. When the air dryers where designed they were designed for a flow of $1220 sm^3/h$ at a dewpoint of -40°C (see Appendix 1). Now the flow is twice as high, with a dewpoint requirement of -70°C. This is a dewpoint standard class 1 for instrument air [5]. As a result of the increased air the air dryers must be run at 2x50%, opposite to the original setup of 2x100%. This made it no longer possible to have one unit in standby. The loss of the possibility of having a standby unit leads to difficulties when the desiccant is changed, then all the air must flow through one air dryer and as a result of this the dewpoint can drop to as low as -30°C.

To fix the problem at Kollsnes there be made changes to the current setup. There have several options that could be considered:

- Search for more suitable desiccant that would have a higher adsorbent capability
- Install an additional two sets of air dryers, so that the total number of columns is eight opposed to today's four.
- Increase the size of the current columns, making it possible for more desiccant in each.

A choice must be made that suits the plant at Kollsnes best. And then figure out how to best implement it into the system.

Researching desiccants has shown no suitable candidate that would improve the air dryers to the desires that Equinor has. Thus, removing the first option of changing the desiccant. Installing an additional set of air dryers would solve the problem. It would make it possible for two air dryers working and drying air simultaneously, while there could be two in standby. This could be done at the original cycle of eight hours of drying and eight in standby. While this seems like a valid solution to the problem and could be considered an "easy fix" it has its downsides. First of, it would be costly to buy, install and run two sets of additional dryers. It would also require a lot of extra space, space that after visiting the plant at Kollsnes they do not possess. Thus, removing option number two. That leaves option number three. Increasing the size of the columns to make room for more desiccant has shown to be the best solution for the problem. So that is what the work is done towards in this paper.

First action that must be taken is to calculate how todays operation works. Then find the new mass of desiccant which can dry the new doubled mass flow, to the requirements Equinor has. When the new mass of desiccant is decided, we can find the new length of the beds. There is also much to take in consideration when doing this. It must be made sure of that the pressure drop over the beds does not exceed the requirements Equinor has, making the pressure in the utility system lower than the minimum pressure allowed. If this happens, there must be done research on solutions to minimize pressure drop in packed beds.

Working on a way of solving this project there must be done extensive research on air drying, air flow through packed beds and understanding on how pressure swing adsorption works. For drawing the design of the new beds and system Creo parametric will be used. Frequent meetings with both intern and extern supervisor will also be important. This has been shown difficult as a consequence of the current COVID-19 outbreak that have greatly impacted our physical research at Kollsnes.

Analysis

The biggest part of this thesis is the theoretical analysis of the system. It is mostly based on specifications and behaviour of the process. There is no practical approach to the problem, except for some methods and equations based on previous experiments.

Very few PSA systems are built straight from the drawing board without any experiment. Having test data and/or small-scale experimental figures are essential for good results. It is key to analyse and see the behaviour of the active alumina that is being used and how the system itself behaves with various flow rates of air. Therefore, some of the numbers/factors that we are implying in this thesis are based on actual behavioural difference that has been noticed by us or/and researchers such as Dr. Skarstrøm, who has this as their profession. Since the 1950's when Dr. Skarstrøm developed the PSA system as we know it today, there has been great technological enhancements. Such as ASPEN tech. This is a program where you can simulate as close as possible to the actual PSA system you are intending on building. In our case we did not have the resources or availability to utilise this program even though it would have been great help. We decided on taking a different route and chose a more analytic approach for a greater understanding of the PSA system. Since we already had a running system at Kollsnes we have enough data to complete this analysis to build a new and improved PSA system, suiting Equinor's needs now and for the foreseeable future.

System components

Compressors
Equinor had chosen three two-stage screw air compressors in parallel to supply air at Kollsnes. They are supplied by Atlas Copco. The compressors are controlled by PCDA (Process Control Data Acquisition) to run in certain modes. One compressor is in load, one in lead and the last in standby. The PCDA is periodically changing the selection of which compressor that operates in which mode to limit wear and fatigue. With greater demand for air, two compressors may be set to load mode. The combined air flow is 14 100 Sm^3/h, design pressure of 10,5 barg and a total shaft power of 2160 kW.
Table 1. Compressor components:

Component	Tag
Compressor train A	KA-301A
Compressor train B	KA-301B
Compressor train C	KA-301C

More information can be found in appendix 2.

Air inter- and after coolers

The compressed air is cooled using nine heat exchangers, three for each compressor line. The intercoolers are water-cooled, located within the compressor housing and cooling the fluid between the low- and high-pressure stages. After the high-pressure compression, six heat exchangers cool the air further; three water cooled and three air cooled. All heat exchangers are equipped with ball float condensate drains.

Table 2. Rated exchanged heat:

Component	Cooling medium	Power	Tag
Intercooler	Water	240 kW	HA-302A/B/C
Aftercooler	Water	360 kW	HA-307A/B/C
Aftercooler	Air	85 kW	HC-301A/B/C
Total		750 kW	

More information and charts can be found in appendix 2.

Filters

Before and after the instrument air dryers, there are located four filters. All filters are cartridge type filters with borosilicate filtering medium where the pre-filter has a 1-micron particle filtering and the after-filter have 50 microns. Only two filters are operative at any given moment, one upstream and one downstream of the dryer columns. Shut-off valves are located upstream and downstream of each filter, allowing operators to engage the standby filter and disengage the operating filter during filter change. Both pre- and after filters are equipped with differential pressure transmitters to ensure the PCDA is alarmed when the filters are needed to be changed. The same applies to compressor inlet filters. The dryer columns are protected from dust, debris and water slugs by the pre-filters and the after-filter protects the instrument air system from desiccant dust and particles.

More information can be found in appendix 3.

Table 3. Filtering components:

Component	Tag
Dryer pre-filter, train A	CB-305A-1/305A-2
Dryer after-filter, train A	CB-302A-1/302A-2
Dryer pre-filter, train B	CB-305B-1/305B-2
Dryer after-filter, train B	CB-302A-1/302A-2
Compressor inlet filter	CA-303A/B/C

Dryer columns

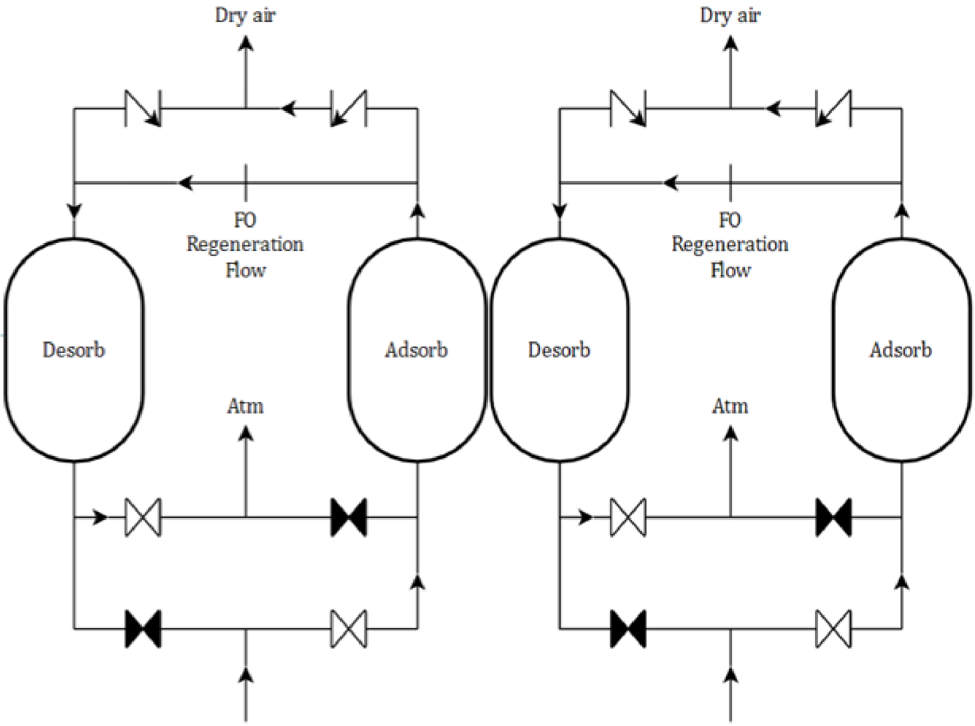


FIGURE 1.

In total there are four dryer columns at site, built like pressure vessels. Two to adsorb and two to desorb (regenerate). The way this work is the active adsorbing columns sends a percentage of the dried air into the columns in regenerating stage, this dries the wet bed and sends the moist air to atmosphere Their original setup was 2 x 100% capacity with one train standby. At this time they are performing as 2 x 50%, due to increased air consumption in the instrumentation. The dryers are partially filled with a desiccant, in this instance, $\gamma - Al_2O_3$ or commonly known as activated alumina. The desiccant mass was originally 300 kg but was increased to 365 kg. The desiccant bed is supported with three layers of mesh, and 7, 40 x 10 mm flat stock bars.
The vessel is made out of 24" schedule 30 pipe with standard pipe caps and the inlet and outlet flanges are 4" and 3" respectively, with schedule 40 connecting pipes. A hand-hole is welded to the column wall with a dimension of 4".
A detailed drawing and more information is available in appendix 1 and 3.
Table 4. Dryer components:

Component	Tag
Dryer column, train A	VK-304A-1/304A-2
Dryer column, train B	VK-304B-1/304B-2

Flow orifice
There is a flow orifice within the dryer unit. The orifice controls the amount of dry airflow from the adsorbing column and into the regenerating column. The bore, also known as orifice in the plate is accurately calculated to suit the desired air flow.
Table 5. Flow orifice components:

Component	Tag	Oriface diam.
Orifice, train A	FO-8098	7,3 mm
Orifice, train B	FO-8118	7,3 mm

Process controls

When the drying process is half-way through its NEMA-cycle, the flow direction must be redirected in order to switch the bed operations. This is performed by pneumatic valves. Solenoids are activated by the PCDA, and they in turn actuates the valves. Solenoids are fed with dry air from the downstream side of the dryers and feed line is pressure relieved. More details on operation is found in appendix 4.

Table 6. Process control components:

Components	Tag
Solenoids, train A	HV-7060/7061
Directing valves, train A	HV-8101/8102/8103/8104
Solenoids, train B	HV-7080/7081
Directing valves, train A	HV-8106/8107/8108/8109

Air receivers

Throughout the instrument air system there are placed eight compressed air receivers. Three of them are receiving wet air whilst the rest are receiving sufficiently dried air. They are placed before and after the dryers, respectively. Their role is to prevent larger fluctuations in air pressure and air accumulation. As the instrumentation rapidly and inconsistently makes changes to the process, it consumes more air. The PCDA cannot activate compressors fast enough, so the receivers accumulate to compensate for the excess air. The large combined volume of the receivers accumulates sufficient amount of air. In case of an emergency, if the compressors trips or shuts off, the receivers can supply enough air for an emergency shutdown of the entire plant. They will last for approximately 30 minutes.

Condensate drains, and heat tracing are placed in every receiver.

Table 7. Size and volume for all air receivers.

Tag	Dimensions (mm)	Volume (m^3)
63-VL301A	1100 x 2100	1,996
63-VL301B	1100 x 2100	1,996
63-VL303	2000 x 7600	23,88
63-VL304	2000 x 6000	18,85
63-VL305	2000 x 6000	18,85
63-VL306	2000 x 6000	18,85
63-VL307	2000 x 7600	23,88
63-VL308	2000 x 4200	13,19
Total		121,5

Piping

All pipes between compressor high pressure outlet and instrumentation are classified in ANSI schedule 40, and all flanges are classed to ANSI 16.5 B with raised faces and graphite gaskets.

Table 8. Size and routing for system piping.

Origin	Destination	Size
Compressor HP outlet	Manifold collector	6” – DN150
Manifold collector	Pipe branch, N2/wet air receivers	10” – DN250
Pipe branch, N2/wet air receivers	N2 reciever	8” – DN200
Pipe branch, N2/wet air receivers	Wet air receivers	4” – DN100
Wet air receivers	Pre-filters	3” – DN80
Prefilters	Dryer columns	3” – DN80
Dryer columns	After-filters	3” – DN80
After filters	Instrumentation	3” – DN80

Transmitters

The gas plant at Kollsnes is controlled and monitored by a number of transmitters and actuators, feeding the PCDA with process information. Equinor provided measurement logs from the transmitters of the instrument air process. This has helped us to accurately determine mass flows, volume flows and densities at various stages of the process.
Table 9. Process transmitters:

Tag	Unit	Description
64FT7008	Sm3/h	Air mass flow rate, Nitrogen generation unit
63FT7029	Sm3/h	Air mass flow rate, Compressor A
63FT7059	Sm3/h	Air mass flow rate, Compressor B
63FT7089	Sm3/h	Air mass flow rate, Compressor C
63KA301A	A	Amperage, compressor motor A
63KA301B	A	Amperage, compressor motor B
63KA301C	A	Amperage, compressor motor C
63MT7106	Deg C	Temperature dewpoint, instrument air
63PDT7094	barg	Differential pressure, prefilter
63PDT7095	barg	Differential pressure, after-filter
63PIC7105	barg	Gauge pressure, instrument air
63PT7101	barg	Gauge pressure, compressor collector manifold
63TIC7100	Deg C	Temperature, compressor collector manifold
63TT7003	Deg C	Temperature, to HP-compressor A
63TT7004	Deg C	Temperature, from HP-compressor A
63TT7005	Deg C	Temperature, to aftercooler, compressor A
63TT7009	Deg C	Temperature, from LP-compressor A
63TT7033	Deg C	Temperature, to HP-compressor B
63TT7034	Deg C	Temperature, from HP-compressor B
63TT7035	Deg C	Temperature, to aftercooler, compressor B
63TT7039	Deg C	Temperature, from LP-compressor B
63TT7063	Deg C	Temperature, to HP-compressor B
63TT7064	Deg C	Temperature, from HP-compressor B
63TT7064	Deg C	Temperature, to aftercooler compressor B
63TT7069	Deg C	Temperature, from LP-compressor B

The letter designations is listed for explanation:

Designation	Description
FT	Flow transmitter
KA	Electric current transmitter
MT	Moisture transmitter
PDT	Differential pressure transmitter
PIC	Instrumentation pressure transmitter
PT	Pressure transmitter
TIC	Instrument temperature transmitter
TT	Temperature transmitter

Table 10. Overview of timelines, operation and parameters.

When?	Flow	Dryer status	Dewpoint
Start up	1220	One operative, one standby	-40 C
Now	2178.1	Both operative	-64 C
In the future	2178.1	One operative, one standby	-70 C

Original operation

Starting with the initial numbers stated in P&IDs and charts that were acquired, it is possible to calculate the properties of air. The air compressor packages where designed to satisfy the following parameters:

Service to:	Flow	Pressure	Temperature	Dewpoint
Nitrogen generation	8000	11	35	
Instrument air	1060	9	35	-40
Plant air	160	9	35	-40

The dryers only supply dry air to the instrument air system and plant air system and all flow rates are logged in Standard Cubic Meters. This unit converts the actual flow rate to standard ambient temperatures with temperature and pressure equal to: $P_1 = 101\ kPa$ and $T_1 = 288K$. To convert them back, the ideal gas formula is applied to create a volume fraction of the standard measurement. In addition to this, there is a necessity to investigate compressibility factor of the fluid. It is a function of the critical values of air and can be estimated from charts or calculated using fluid properties. In this instance, Nelson-Obert generalized compressibility chart have been used [6]:

$$Z = f(T_R\,, P_R)$$

$$T_R = \frac{T}{T_{cr}} = 2.2$$

$$P_R = \frac{P}{P_{cr}} = 0.267$$

$$Z \approx 1$$

The result concluded above states that there are no alternations in the compressibility. Furthermore, there are no measurements throughout this paper that exceeds the critical values, nor are there measurements that has a great variance to the parameters used. Therefore, the compressibility factor is: $Z = 1$, and remains unchanged.
Actual flow is solved through equation 1:

$$Q = V_f \cdot Q_s = \frac{P_1 T_2}{P_2 T_1} \cdot Q_s = 0.043\ \frac{m^3}{s} \tag{1}$$

According to flow charts, the air density was listed at $\rho_6 = 12.60$ and $\rho_7 = 10.31$
Therefore, the air mass flow can be calculated using equation 2:

$$\dot{m}_a = Q \cdot \rho = 0.542 \frac{kg}{s} \tag{2}$$

Current operation

To set a starting point of the calculations, there was found a short and stabile logging period in the PCDA, with consistent numbers. This log became the basis of the project.
Equinor was also in possession of ambient temperature and humidity. This data was measured about the same time as the process logging, only two minutes in advance. There is a reason to believe that the ambient temperature and humidity remained the same within this short period of time.
Table 11. Ambient air conditions:

Unit	Description	Measurement
Deg C	Dewpoint	-2.70
Deg C	Temperature	2.80
%	Relative humidity	66

Table 12. List of all transmitters used to analyse the drying process with their average values over the logging period and their placement:

Tag	Description	Measurement	Unit	Stage
FT-7008	Air flow rate, N2 generation	5730.37	Sm^3/h	5
FT-7029	Air flow rate, compressor A	4851.15	Sm^3/h	5
FT-7059	Air flow rate, compressor B	2852.92	Sm^3/h	5
FT-7089	Air flow rate, compressor C	194.4	Sm^3/h	5
KA-301A	Amperage, motor A	43	A	
KA-301B	Amperage, motor B	31	A	
KA-301C	Amperage, motor C	0	A	
MT-7106	Dewpoint, instrumentation	-64	℃	7
PDT-7094	Differential pressure, prefilter, train A	0.123	$barg$	
PDT-7095	Differential pressure, after-filter train A	0.015	$barg$	
PIC-7105	Pressure, instrumentation	8.03	$barg$	7
PT-7101	Pressure, compressors	9.05	$barg$	6
TIC-7100	Temperature, instrumentation	18.56	℃	7
TT-7003	Inlet temp. high pressure element A	39.9	℃	3
TT-7004	Outlet temp. high pressure element A	177.6	℃	4
TT-7005	Outlet temp. water cooled aftercooler A	33	℃	5
TT-7009	Outlet temp. low pressure element A	178.2	℃	2
TT-7033	Inlet temp. high pressure element B	36.9	℃	3
TT-7034	Outlet temp. high pressure element B	166.5	℃	4
TT-7035	Outlet temp. water cooled aftercooler B	31.6	℃	5
TT-7039	Outlet temp. low pressure element B	173.6	℃	2
TT-7063	Inlet temp. high pressure element C	36.5	℃	3
TT-7064	Outlet temp. high pressure element C	41.1	℃	4
TT-7065	Outlet temp. water cooled aftercooler C	9.5	℃	5
TT-7069	Outlet temp. low pressure element C	37.5	℃	2

Compressor power.
To estimate the electric power invested in the compressors and the exchanged heat of the coolers, Equinor supplied the sufficient information to solve the problem. PCDA logs provides the temperature between every compression stage and every cooling stage in the compressor packages as well as standardized flow rate.
The reason to obtain this information is to determine the electric energy that can be preserved in the future. This is done in through heating while the desiccant is regeneration. Investing a small amount of electric energy to heat the regeneration air can justify a reduction in regeneration air flow. More information about this can be found in the section of Future operation, Heating.

Table 13. Temperatures logged by the transmitters at each stage:

Component	Stage	Compressor A		Compressor B		Compressor C	
		°C	K	°C	K	°C	K
Compressor inlet	1	2.8	275.8	2.8	275.8	2.8	275.8
Low pressure compressor	2	178.2	451.2	173.6	446.6	37.5	310.5
Intercooler	3	39.9	312.9	36.9	309.9	36.5	309.5
High pressure compressor	4	177.6	450.6	166.5	439.5	41.1	314.1
Water cooled aftercooler	5	33	306	31.6	304.6	9.5	282.5
Air cooled aftercooler	6	19	292	19	292	19	292

Table 14. Differential temperatures over the components:

Component	Compressor A	Compressor B	Compressor C
	K	K	K
Low pressure compressor	175.4	172.8	34.7
Intercooler	138.2	136.7	1
High pressure compressor	137.6	129.6	4.7
Water cooled aftercooler	144.6	134.9	31.6
Air cooled aftercooler	14	12.6	-9.5

Table 15. Volume flow rate discharged from the compressors logged by the transmitters:

Compressor	Standardized air flow (Sm^3/h)
A	4851.15
B	2852.42
C	194.4

For a complete power analysis of the fluid flow, air mass flow rate and fluid enthalpy are required. Air volume flow rate is given in a standardized form. The conditions of this specified volume are: $P = 1\ atm$ and $T = 288\ K$.
In a pressurized air system the specific volume of the air changes throughout the piping due to heat- and pressure losses. To obtain a good basis for equilibrium, air mass flow rate is preferable to use in this problem because it is consistent.
Air mass flow can be calculated using equation 3. But first, the specific density and actual air flow rate must be determined, preferably at the location of the flow transmitters. Equation 3 states:

$$\rho_5 = \frac{MP_5}{RT_5} \tag{3}$$

This combined with equation 1, the parameters can be inserted into equation 2:

Compressor	Mass air flow (kg/s)
A	1.65
B	0.97
C	0.07

To determine the enthalpy of the flow medium can be a tricky task. Fluid enthalpy is only dependent on temperature, and fortunately, tables for air enthalpy is highly available[6]. But the tables do not include very specific fluid temperatures. To solve this problem, interpolation of the tables is used to determine enthalpy more accurately. Equation 4 illustrates the principle of interpolation:

$$h = h_1 + \frac{T - T_1}{T_2 - T_1} \cdot (h_2 - h_1) \tag{4}$$

Where h and T is the known values and values with prefixes 1 and 2 are the closest lower and higher table values, respectively.
Table 16. Enthalpies after all stages in the compressor packages, after inserting all parameters into equation 4:

Component	Stage	Compressor A	Compressor B	Compressor C
		kJ/KgK	kJ/kgK	kJ/kgK
Compressor inlet	1	275.92	275.92	275.92
Low pressure compressor	2	453	455.08	310.74
Intercooler	3	313.25	310.13	309.74
High pressure compressor	4	455.89	441.1	312.3
Water cooled aftercooler	5	306.22	304.82	282.6
Air cooled aftercooler	6	292.16	292.16	292.16

All low- and high-pressure compressors are cooled by tempered water. Unfortunately, the Covid-19 outbreak forced Equinor to cut the number of operators at the plant, including engineers. Therefore, it was not possible to perform temperature measurements of the process that are not covered by the transmitters. With a lack of parameters, the heat that would be transferred from the compressors to the fluid is in this case neglected.
Shaft power invested in compressed air and exchanged heat can be solved using equation 5:

$$W = \dot{m} \cdot \Delta h \tag{5}$$

Table 17. Compressor shaft power invested in each stage of each compressor, after inserting all parameters into equation 5:

Compressor	Power, LP (kW)	Power, HP (kW)	Power Total (kW)
A	286.84	235.34	412.18
B	173.79	127	300.79
C	2.44	0.18	2.62

The similar approach can be applied to calculate heat exchanged by the heat exchangers.
Table 18. Exchanged heat after inserting all parameters into equation 5:

Heat exchanger	Compressor A	Compressor B	Compressor C
	kW	kW	kW
Intercooler	225.23	140.6	0.07
Water-cooled aftercooler	246.96	132.2	2.1
Air-cooled aftercooler	23.2	12.28	-0.67
Total	495.39	286.08	1.5

The air discharged from the compressors is divided to two consumers: nitrogen generation unit and instrumentation. The PCDA logs air flow from each individual compressor and flow to nitrogen generation. The differential flow between them is assumed to be the exact flow to instrumentation.
Equation 6 illustrates flow equilibrium:

$$Q_{COMP} = Q + Q_{N2} \tag{6}$$

Therefore:

$$Q = Q_{COMP} - Q_{N2} = 2178.1 Sm^3/h$$

The instrumentation share percentile can be calculated using equation 7:

$$\frac{Q}{Q_{COMP}} = 0.276 = 27.6\% \tag{7}$$

Power invested to deliver air to instrumentation can be estimated using the results from equation 4 and 7:

$$W_a = W_{COMP} \cdot 0.276 = 197.5 kW$$

The electric motors powering the compressors are operating with a voltage: $U = 11000V$. Within the logging timeframe, the motors average amperage was: $I = 73.98A$. The motor power and total efficiency of the motor/compressor assembly can be determined using equation 8 and 9:

$$W_M = IU\sqrt{3} = 1409.5 kW \tag{8}$$

$$\eta_{tot} = \frac{W_{COMP}}{W_M} = 0.508 = 50.8\% \tag{9}$$

Using the compressed air share percentile from equation 6 and electric power from equation 8, there is possible to estimate electric power invested into instrument air:

$$E_{DRY} = 1409.5kW \cdot 0.276 = 389.02kW$$

The figure concluded above is the estimated power invested into the air entering the dryers. A PSA dryer system is known for having a low efficiency if there are no heating involved. In the following regeneration section, there will also be an estimation of how much of this invested energy will be exhausted to atmosphere and how to reduce it.

Adsorption Dryers

Specifications:

All specification was provided by Equinor through P&IDs, machine drawings and charts.

It is essential to calculate the exact volume of each part of the dryer columns.

The columns are built with a section of pipe and two pipe caps in both ends.

A fine mesh is located 77 mm below the bottom of the pipe to support the mass of the desiccant**.**

The vessel volume was listed to: $V = 0.75m^3$.

Equation 10 solves for cylindrical volumes:

$$V_{cyl} = 0.25 \cdot \pi \cdot d^2 \cdot \Delta x \tag{10}$$

Through equation 10 the volume of the pipe caps can be estimated:

$$V_{cap} = \frac{V - 0.25 \cdot \pi \cdot d_c^2 \cdot L_{cyl} - 0.077m}{2} = 0.124\ m^3$$

The desiccant volume can then be solved using equation 11:

$$V_d = \frac{m_d}{\rho_{b_d}} = 0.48m^3 \tag{11}$$

Furthermore, with the use of equation 10, the length of bed:

$$\Delta x = \frac{V}{0{,}25 \cdot \pi \cdot d_c^2} = 1.64m$$

To estimate the void volume, external porosity must be known. According to UOP the desiccant bulk density is $\rho_d = 753\ kg/m^3$ (see appendix 6). Furthermore, the density of Al_2O_3 is $\rho = 3950\ kg/m^3$.

Equation 12 can be rearranged to solve for external void fraction:

$$\rho_d = (1 - \varepsilon_p) \cdot \rho \tag{12}$$

$$\varepsilon_p = 0.81$$

However, the alumina desiccant does not consist of solid particles, the particles' surface is consistent mostly of micropores. The external void fraction estimated above is too high because of these pores and the fact that the desiccant is consistent of only 95% alumina. To solve this, a generic average void fraction is used: $\varepsilon_p = 0.695$. To estimate the air residence time within the dryer, volume flow rate is necessary.

The actual volume flow rate at the dryer inlet can be solved through equation 1:

$$Q_6 = Q_s \cdot V_f => \ Q_6 = Q_s \cdot \frac{P_1 T_6}{T_1 P_6} = 0.0615 \ \frac{m^3}{s}$$

After reducing the flow rate in half, the residence time for each dryer column is calculated using equation 13:

$$t_r = \frac{V_\varepsilon}{Q} = 10.73 \ s \tag{13}$$

Desiccant

Within compressed air systems, there are a known phenomenon of airborne moisture to condensate inside piping, heat exchangers, air receivers, etc. These components are indeed drained to a common open drain. Because of this, the condensation rate cannot be monitored at site. Therefore, there is assumed to be no condensation of water within the process components. And following, the humidity measured at the compressor inlet is assumed to be the same as dryer inlet.

Dewpoint is a way of measuring the amount of moisture in the air. The dewpoint is the temperature which the air space becomes saturated and water condensates. To give an example: if the outside dewpoint is 5°C at a fixed pressure, then as soon as the temperature drops below 5°C the air becomes saturated. And therefore, vapor condensates and water droplets are formed. When we define the dewpoint temperature as low as -70°C, it does not mean that the temperature must fall below -70°C in the system, but rather if that air was in atmospheric pressure, because we must remember that the pressure inside the system is at much higher that atmospheric, leading to less space for water vapor which leads to an higher temperature where the water condensates [7].

Table 19. Parameters to determine molal humidity:

Position	Temperature	Dewpoint	Relative humidity
Compressor inlet	2.8	-2.7	66%
Process dewpoint sensor	19	-64	100%

By using a psychrometric chart[6], and dewpoint/humidity tables (see appendix 5). The molal air humidity is the following:

Position	Molal humidity
Compressor inlet	$3.06 \cdot 10^{-3} m_w/m_a$
Process dewpoint sensor	$3.79 \cdot 10^{-6} m_w/m_a$

To estimate the water mass adsorbed by the dryers, it is a necessity to perform water mass flow rate calculations through the dryer columns. Again, there is assumed to be an equilibrium, that the differential air humidity between the dryer inlet and outlet is defined as adsorbed humidity. The volume flow rate is divided in half to illustrate the water adsorption rate of each dryer. From equation 2 the results are:

$$\dot{m}_a = 0.37 \frac{kg}{s}$$

Equation 14 solves for water mass flow rate that are saturated in the compressed air:

$$\dot{m}_w = \dot{m}_a \cdot H_m \tag{14}$$

Inserting parameters into equation 14:

$$\dot{m}_{w_1} = \dot{m}_a \cdot H_{m_1} = 1.132 \cdot 10^{-3} \frac{kg}{s}$$

$$\dot{m}_{w_6} = \dot{m}_a \cdot H_{m_6} = 1.402 \cdot 10^{-6} \frac{kg}{s}$$

Results from equation 14 are used to determine the desiccant's adsorption rate through equation 15:

$$\dot{m}_{w_6} = \dot{m}_{w_1} - \dot{m}_{w_6} = 1.13 \cdot 10^{-3} \frac{kg}{s} \tag{15}$$

By multiplying the water adsorption rate with regeneration period, the mass of adsorbed water per cycle can be determined:

$$m_\lambda = \dot{m}_{ad} \cdot t = 0.339 \frac{kg}{\lambda}$$

Finally, equation 16 illustrates the desiccant adsorption capacity:

$$\sigma_{ad} = \frac{\dot{m}_{ad} \cdot \lambda}{m_d} = 9.28 \cdot 10^{-4} \frac{m_w}{m_d} \tag{16}$$

Pressure losses
Pressure losses are induced by filtering devices, flow restrictions, etc. Accurate drawings for pipes, filters, reducers, expanders or air receiver nozzle geometries were not available in this thesis. Considering the low viscosity for air, frictional pressure losses is considered to be neglectable. The three remaining major pressure loss contributors are pre-filters, after-filters and the desiccant beds. Utilizing the pressure- and differential pressure transmitters, this is possible to solve for losses over the beds. The dryer trains work in parallel and therefore, pressure losses over them are assumed equivalent.
Equation 17 solves for pressure loss over the packed beds in the columns:

$$\Delta P_c = p_{COMP} - p_{INST} - \Delta p_{FILT} \tag{17}$$

$$= P_6 - P_7 - \Delta P_{FILT} = 0.88\, bar$$

Regeneration
As the desorbing bed is regenerated, the purge air flow is determined by the flow orifice located downstream of the dryer columns. The Reader-Harris/Gallagher equation can solve for mass air flow though the orifice [8].
To solve the main equation, other parameters must be dealt with first, starting with air density at dryer outlet using equation 18:

$$\rho_7 = \frac{MP_7}{RT_7} = 10.82 kg/m^3 \tag{18}$$

And actual air flow at the dyer outlet from equation 1:

$$Q_7 = V_{f_7} \cdot Q_s => \; Q_7 = Q_s \cdot \frac{P_1 T_7}{T_1 P_7} = 0.0361 \; \frac{m^3}{s}$$

Equation 19 solves for superficial air velocity, in this instance through the dryer outlet pipe, where pipe diameter, D=40,1mm:

$$v_P = \frac{Q_7}{A_P} = 28.58 \frac{m}{s} \tag{19}$$

Density, air velocity, pipe diameter and viscosity can be used to estimate pipe turbulence using equation 20. The distance from the dryer outlet to the pipe mating to the orifice plate is short, and therefore, there is assumed the change in air turbulence is neglectable between them. Since the regeneration volume flow rate is unknown, parameters for the dryer outlet pipe are used. Air viscosity is: $\mu = 1.88 \cdot 10^{-5} Pa \cdot s$.

$$Re_P = \frac{\rho_7 v_P D}{\mu} \approx 6.6 \cdot 10^5 \tag{20}$$

Equation 21 solves the diameter ratio:

$$\beta = \frac{d_{FO}}{D} = 0.182 \tag{21}$$

This ratio two can be used to solve the equation for the expansibility factor. As illustrated in appendix 3, the air passing the orifice is exhausted to atmosphere. The pressure downstream of the orifice is then estimated to be atmospheric.

$$\varepsilon = 1 - (0.351 + 0.256\beta^4 + 0.93\beta^8)\left[1 - \left(\frac{P_1}{P_7}\right)^{\frac{1}{\gamma}}\right] \quad ,\gamma \approx 1.4 \tag{22}$$

$$= 0.718$$

Flanges connecting to the orifice plate is not equipped with any tapping. Therefore, the distance quotients are equal to: $L' = M' = 0$
A function of orifice Reynold's number can be calculated:

$$A' = \left(\frac{19000\beta}{Re_d}\right)^{0,8} = 1.37 \cdot 10^{-3} \tag{23}$$

All results will be inserted into equation 24 to solve for orifice coefficient of discharge:

$$C = 0.5961 + 0.0261\beta^2 - 0.216\beta^8 + 0.000521\left(\frac{10^6\beta}{Re_P}\right)^{0.7} +$$
$$(0.0188 + 0.0063A')\beta^{3.5}\left(\frac{10^6}{Re_P}\right)^{0.3} +$$
$$(0.043 + 0.08e^{-10L'} - 0.123e^{-7L'})(1 - 0.11A')\frac{\beta^4}{1-\beta^4} -$$
$$0.031(M_2{}' - 0.8M_1{}')\beta^{1.3} + 0.011(0.75 - \beta)\left(2.8 - \frac{D}{0.0254}\right) \tag{24}$$

After inserting all parameters into equation 24, the result is:

$$C \approx 0.6$$

Which is a common standard recommendation for coefficient of discharge.
Finally the Reader-Harris/Gallagher equation can be stated and calculated:

$$\dot{m}_{FO} = \frac{C}{\sqrt{1-\beta^4}} \cdot \varepsilon \cdot \frac{\pi}{4} \cdot d_{FO}^2 \cdot \sqrt{2 \cdot (P_7 - P_1) \cdot \rho_7} \tag{25}$$

$$\dot{m}_{FO} = 0.0757\frac{kg}{s}$$

It I common to refer to regeneration flow as a fraction or percentile. The regeneration air mass flow will be compared to the total air flow rate entering the dryers. The orifice mass air flow rate is doubled to illustrate the flow though bot orifices:

$$\omega = \frac{\dot{m}_{FO}}{\dot{m}_a} = 0.204 = 20.4\% \tag{26}$$

With this it is possible to estimate the power lost in the regeneration flow through both flow orifices:

$$E_{FO} = E_{DRY} \cdot \omega = 81.3kW$$

Future Operation

Considering the design for the system upgrades, it is assumed that the instrumentation will continue to work as it does today. With that, the flowrates remain the same, with addition that only one dryer train is to be used at a time.
From equation 2: the air mass flow rate though the columns will be doubled:

$$\dot{m}_a = Q_s \cdot \rho_s = 0.74 \frac{kg}{s}$$

Air humidity at compressor inlets will be changed, due to a different and longer logging period. Air humidity in the instrumentation are a set goal for this thesis. The goal is to attempt to keep the dewpoint temperature at -70 degrees centigrade.
Water content in the air is the following:

Location	Dewpoint	Molal humidity
Compressor inlet	0°C	$3.64 \cdot 10^{-3} m_w/m_a$
Dryer outlet	-70°C	$1.58 \cdot 10^{-6} m_w/m_a$

It is still presumed that no moisture is condensed within heat exchangers, pipes or air receivers. The NEMA cycle remains the same: 10 minutes, whereof five of them is regeneration:

$$\lambda = 300\ s$$

To determine the desiccant's water adsorption rate for the new dryer setup, equation 14 will be used:

$$\dot{m}_{ad} = \dot{m}_a \cdot \Delta H_m = 2.692 \cdot 10^{-3} \frac{kg}{s}$$

Which for one complete adsorption period, adds up to:

$$m_\lambda = \dot{m}_{ad} \cdot \lambda = 0.808 kg$$

Desiccant
To estimate the desiccant mass of the new bed, there is possible to use residence time, more advanced mass transfer equations or the desiccant adsorption capacity. All three results will be considered when the new desiccant mass is chosen. At the gas plant there is observed that the desiccant vessels cannot be extended radially. Therefore, they can only be extended vertically.

Method 1:
With the basis of the periodically adsorbed water and desiccant adsorption capacity, there is possible to estimate the desiccant mass of the new bed:

$$m_d = \frac{m_\lambda}{\sigma_{ad}} = 870.25 kg$$

Method 2:
The second approach is to extend the new desiccant bed in relation to increased air flow. It will be extended to the point where the new dryer geometry allows the new residence time to be matched with the old bed. The backside of this approach is that it does not use any parameters that include humidity or dewpoint. This will result in a dewpoint equal to the current.
Equation 11 and 12 states:

$$t_r = \frac{V_\varepsilon}{Q} \qquad \text{and} \quad V_\varepsilon = V_d \cdot \varepsilon_p$$

Therefore:

$$V_d = \frac{t_r \cdot Q}{\varepsilon} = 0.95m^3$$

Furthermore, based on equation 10, the solution for the desiccant mass based on matching the residence time is:

$$m_d = V_d \cdot \rho_d = 715.35kg$$

Method 3:
The final method to estimate the new desiccant mass is a more advanced gas purification equation. All units will be converted to imperial for the simplicity of executing the equations. The total length of the new bed is divided into two parts: mass transfer zone and length of unused bed (LUB). The majority of the adsorption takes place in the mass transfer zone, and the length of unused bed will achieve a very low water saturation.
Certain parameters must be distilled from the current bed:
Equation 27 solves for the gas feed rate through the desiccant [9]:

$$G = \frac{m_a}{\lambda \cdot A_c} = 2.64 \ \frac{kg}{s \cdot m^2} \tag{27}$$

The bed is not uniformly saturated with the adsorbed water. The bed can be divided into two major parts: mass transfer zone and unused bed. These parts have a lower and higher water saturation, respectively. Furthermore, the stochiometric front is the defined as the border between the two.
To solve the length of the mass transfer zone, equation 28 applies:

$$L(MTZ) = \left(\frac{v_c}{35}\right)^{0.3} \cdot 1.7 = 0.57 \ m \tag{28}$$

Equation 29 solves for the total length of the bed:

$$L_d = L_s + 0.5 \ MTZ \tag{29}$$

This equation is rearranged to solve for the distance to the stochiometric front:

$$L_s = L_d - 0.5 \ MTZ = 2.75 \ m$$

Equation 28 solves for the distance to the stochiometric front:

$$L_s = \frac{100G \cdot \left(\frac{\Delta Y}{\Delta X}\right) \cdot \lambda}{\rho_d} \tag{30}$$

This can be rearranged to solve the air to desiccant ratio:

$$\left(\frac{\Delta Y}{\Delta X}\right) = \frac{L_s \cdot \rho_d}{100G \cdot \lambda} = 0.7675$$

For the new bed, the gas feed rate is doubled since it increases linearly to air mass flow rate. Equations 26 through 28 is inserted with parameters for the future bed:

$$L_d = \frac{100G \cdot \left(\frac{\Delta Y}{\Delta X}\right) \cdot \lambda}{\rho_d} + 0.5 \left(\frac{v_c}{35}\right)^{0.3} = 3.1\ m$$

And finally, the desiccant mass from equation 11:

$$m_d = V_d \cdot \rho_d = 638\ kg$$

Again, equation 10, 11 and 12 can be utilized to solve for the properties of the new bed, based on the different theoretical approaches.
Table 20. Properties of the three different estimations of the new bed.

Method	Bed length	Desiccant mass	Void volume	Residence time
Adsorption capacity	4.227 m	870.25 kg	0.8 m^3	13 s
Residence time	3.47 m	715.35 kg	0.66 m^3	10.7 s
Mass transfer equations	3.1 m	638 kg	0.59 m^3	9.6 s

To conclude the results above, it is worth mentioning that all methods are based on the current operation. Therefore all methods are based on the theory that the desiccant will behave and operate the exact same way. Method three, on the other hand, is based on previous experiments by external researchers done earlier. It is reason to believe that using the adsorption capacity and residence time will not take the sufficient parameters in consideration. The third method using mass transfer equations will therefore be the best solution for the new descant mass.

Pressure Drop

Considering the new desiccant mass and fluid velocity, there will be a greater pressure drop caused by the bed. The most accurate way to estimate this is using Ergun's equation [10]:

$$\frac{\Delta P}{\Delta x} = \frac{150 \cdot \mu \cdot (1-\varepsilon)^2 \cdot v_c}{\varepsilon^2 \cdot d_p^2} + \frac{1.75(1-\varepsilon) \cdot \rho \cdot v_c^2}{\varepsilon^3 \cdot d_p} \tag{31}$$

However, certain parameters have not been obtained to use this accurately.
The estimated pressure drop in the current operation contributed by the bed neglects the frictional and local pressure drops. Due to the other various contributors, rearranging this equation in favour of external void fraction will provide an extreme result of $\varepsilon = 0.11$. In addition to this, using a known bulk and solid density gives an opposite extreme result of $\varepsilon = 0.81$. Again, the generic average porosity of $\varepsilon = 0.695$ is used to complete this problem.
In the new bed the flow is doubled relative to the current setup, resulting in a doubling of superficial velocity.
inserting all parameters for into equation 31 will result in pressure drop contributed by the bed:

Operation	Superficial velocity	Pressure drop
Current	0.11 m/s	126 Pa
Future	0.22 m/s	922 Pa

Considering these results, the differential pressure is $\Delta P = 796\ Pa$. A pressure increase of 0.008 bar caused by increased desiccant mass appears unnatural. A much greater pressure drop is expected with the change of the parameters.

Regeneration

The regeneration of the desiccant will in general work the same way as it used to. A flow orifice will allow a portion of the dried air to pass the saturated desiccant and evacuate water. In addition to this, there will be heating elements in the air passing the orifice. This will help the water to evacuate the bed and justify reducing the regeneration mass flow rate to save energy. According to Professor Donald White at Aircel, the total invested energy can be significantly reduced if the regeneration air flow is heated [11]. A 25% reduction is considered to be an obtainable goal. To lower the invested energy, regeneration air flow will be reduced. Therefore, future orifice losses and the electric heating will be equal to the current orifice losses with the savings factor:

$$E_{FO} + E_a + E_w + E_v = 81.3\ kW\ \cdot 0.75$$

Equation 26 is rearranged in favour for the future air mass flow rate passing the orifice:

$$E_{FO} = 389.02 \cdot \frac{\dot{m}_{FO}}{0.74}$$

The heat will increase the air and water temperature to $T = 100°C$. Therefore, the differential temperature is the following:

$$\Delta T = T - T_7 = 81.4K$$

Specific and latent heats for the masses are listed:

$$c_{p_a} = 0.75(kJ/kg \cdot K)$$

$$c_{p_w} = 4.18(kJ/kg \cdot K)$$

$$c_{l_v} = 2256.4\ kJ/kg$$

Equations 32, 33 and 34 solve for heat required to increase the temperatures of the air flow and adsorbed water, as well as moisture evaporation:

$$E_a = \dot{m}_{FO} \cdot c_{p_a} \cdot \Delta T \tag{32}$$

$$E_w = \frac{m_\lambda \cdot c_{p_w} \cdot \Delta T}{t} = 0.92\ kW \tag{33}$$

$$E_v = \frac{m_\lambda \cdot c_{l_v}}{t} = 6.1\ kW \tag{34}$$

At this point, two of the equations is stated with the new orifice air flow rate. Equations 26 and 32 is rearranged in favour of air mass flow rate to solve the problem:

$$\dot{m}_{FO} = 0.091 \frac{kg}{s}$$

Now as the new orifice air flow rate is determined, equations 24 and 25 can be rearranged in favour for diameter ratio. With the new parameters, the result is:

$$\beta = 0.2$$

Equation 21 is rearranged in favour for new orifice diameter:

$$d_{FO} = \beta \cdot D = 0.00802\ m \approx 8\ mm$$

Heating

As mentioned in the regeneration section, the air flow passing the flow orifice will be heated to assist the evaporation of adsorbed moisture. The heat is added using two heating elements, in both ends of the flow orifice. Heating up of the entirety of the desiccant mass with a heat equal to all the moisture being evaporated is not desirable. This is caused by the big mass and specific heat of the desiccant, and it will require an additional 168 kW to elevate the temperature within the regeneration period. In addition to this, the desiccant has its highest absorption capability at about 25 ℃ [12]. The desiccant bed itself will only be saturated up to the Stoichiometric front. Most of the time the Mass Transfer Zone and the top end of the bed also known as the unused bed will have close to no absorbing in its early cycle life. The amount of power needed to dry the entire bed will also not be cost effective and it will cut down on production time because there would be a need for a cool down step.

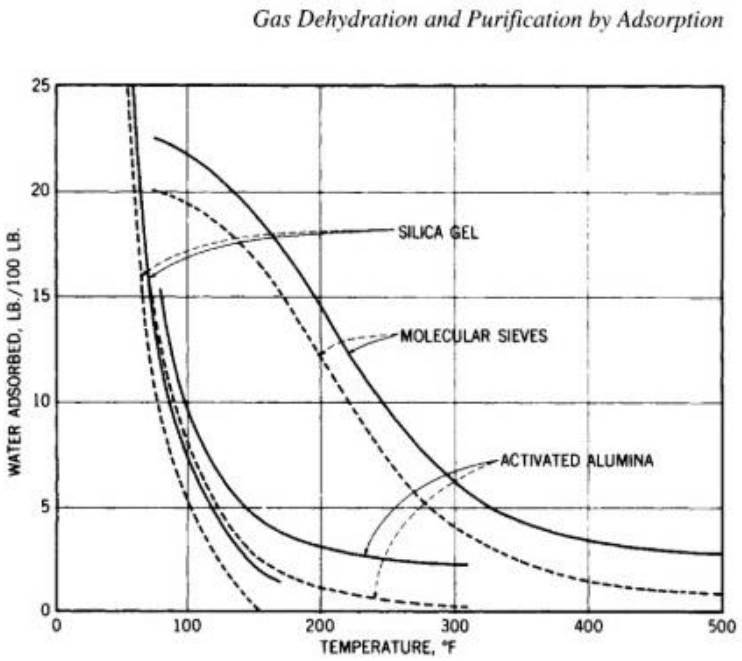


FIGURE 2. (ARTHUR L. KOHL, RICHARD NIELSEN, 1997, P. 1045) [12].

This certain picture illustrates the absorption capabilities that Activated Alumina has compared to the rise in temperature. The line referred to is the continuous line and not the dashed, since the dashed line belongs to a separate experiment done by UOP (1990).

The total heat required to assist regeneration:

$$E = E_a + E_w + E_v$$

Equations 32 and 33 is calculated in the regeneration section. Equation 34 can be solved:

$$E_a = \dot{m}_{FO} \cdot c_{p_a} \cdot \Delta T = 5.56\ kW$$

The total heat is the following:

$$E = 12.58\ kW$$

The figure that is concluded above is the total heat needed for regeneration for each dryer train. The amount of power we would need to heat the desiccant mass with a temperature change of 81 K is a staggering 168 KW, using Active Alumina specific heat of 0.955 KJ/KgK [13]. Compared to the 12.58 KW that we are contributing to the regeneration we believe that this will be sufficient to make a higher regeneration efficiency. As mentioned earlier, this heat is only for assisting the regeneration.

Conclusion

We have concluded in our thesis about improving the pressure swing absorption system at Kollsnes. The conclusion is to extend the vessel cylinder with 1191 millimetres so the new total length will be at 3200 millimetres. While increasing the height of the vessels we will also increase the amount of desiccant. The new mass will be at 638 kg and the beds will be packed compared to what they were before. With the drying state being taken care of, we must regenerate the desiccant to allow it to absorb more moisture for a new cycle of 5 min. Before a new drying cycle is allowed, we must perform a regeneration of the desiccant, as mentioned before this is also 5 min. In the regeneration part of the cycle we have introduced heat. As a result of this, the regeneration more efficient, up to 25%. The regeneration air that comes from the neighbouring vessel will first pass through a pipe that connects the two vessels at the top. This pipe will be equipped with a new orifice plate, with a orifice diameter of 8 mm. The actual heating is situated external to the bed. The external heating will run along the piping that is mentioned above and heat up the regeneration air. The power needed for this heating is 12.6 KW. The heat will allow the desiccant to desorb the moisture that it has absorbed during the drying cycle. This will also enhance the amount of moisture the desiccant that is situated in the begin of the bed can absorb. This will also prolong the risk of completely saturating the bed since the mass transfer zone will be contained within stable boundaries. Along with the purge air and regeneration air moisture will be pushed down through the vessel, counter-direction of the air being dried. The moisture will in the end exit at the bottom of vessel. The additional pressure drop caused by the new desiccant mass remains partially inconclusive. With the appropriate parameters and/or software simulations, we may have obtained a better conclusion.

References

[1] Equinor ASA. https://www.equinor.com/no/what-we-do/terminals-and-refineries/kollsnes-processing-plant.html (accessed 14.04.2020.

[2] R. K. Mobley, "Fluid Power Dynamics," ed: Elsevier, pp. 233-235.

[3] C. W. Skarstrøm, "Method and Apparatus for Fractionating Gaseous Mixtures by Adsorption," Patent US2944627,

[4] C. W. Skarstrøm, "Heatless Fractionation of Gases Over Solid Adsorbents," *Recent Developments in Separation Science,* vol. 2, p. 95, 1972.

[5] *Air Quality Classes*, ISO 8573, 2010.

[6] Y. Cengel, M. A. Boles, and M. Kanoglu, *Thermodynamics, An Engineering Approach*, 9 ed. 2 Penn Plaza, New Yor: McGraw-Hill Education, 2019.

[7] H. Struchtrup, *Thermodynamics and Energy Conversion*. Berlin, Heidelberg: Springer Berlin / Heidelberg, 2014.

[8] M. J. Reader-Harris and J. A. Sattary, "The Orifice Plate Discharge Coefficient Equation-The Equation for ISO 5167-1," *National Engineering Labratory, East Kilbride, Glaskow*.

[9] A. L. Kohl and R. Nielsen, "Gas Purification," 5. Ed., 5. ed. Houston, Texas.: Gulf Publishing Company, 1997, p. 1061.

[10] S. Ergun, "Fluid Flow Through Packed Columns," in *Chemical Engineering*, 1952, p. 48.

[11] D. White. "Regenerative Desiccant Compressed Air Dryers." https://www.airbestpractices.com/technology/air-treatmentn2/regenerative-desiccant-compressed-air-dryers (accessed 26.02.2020.

[12] A. L. Kohl and R. Nielsen, "Gas Purification," 5. ed. Houston, Texas.: Gulf Publishing Company, 1997, p. 1045.

[13] E. ToolBox. "Dehumidification." https://www.engineeringtoolbox.com/dehumidification-d_141.html?fbclid=IwAR0nEXGgPwyNpegybflUCH1QMhlxSmUdZcggW559olhjLcs60eXOk5o8_gw (accessed 02.04.2020.

Appendix

Appendix 1. Original setup

Aker Engineering/Kellogg Joint Venture
Troll Phase I - Onshore Plant
Client: A/S Norske Shell - Contract C91910

Page : 3 of 63
Date : 23.02.95
Rev. : 0

Document No. : C030-PH-O-KH-020
Document Title : OPERATIONAL MANUAL
COMPRESSED AIR - SYSTEM 63

1 SYSTEM INFORMATION

1.1 System Description

The purpose of this system is to provide dry and oilfree compressed air at right temperature to
- Nitrogen Generation Units
- Instrument Air network
- Plant Air.

It essential to have this system in operation, there is a buffer estimated to 30 min., therafter the plant will go down.

All the equipment for compressed air production is located in area A 53A.

Three parallel compressor trains provide compressed air to a manifold. From the manifold, the comprerssed air is distributed to one wet air receiver for nitrogen generation, and two parallel air drier trains. After this, the dry air is manifolded together and distributed to the sub-header for instrument- and plant air.

Each compressor train consists of one air filter with heat tracing and one electrical driven two-stage screw compressor with air cooled recycle- and after cooler. Each compressor train has its own lube oil and is supplied with cooling medium from the Tempered Cooling Water System (40).

Each air drier unit consists of one wet air receiver, one pre-filter for the air-drier, two drier units (adsorber vessels), and one after filter.
The wet air receivers are both storage- and buffer tank. The two parallel drier columns are switching between operation and regeneration, one in operation and one in regeneration, every 8 hour, the timer can be adjusted as required.

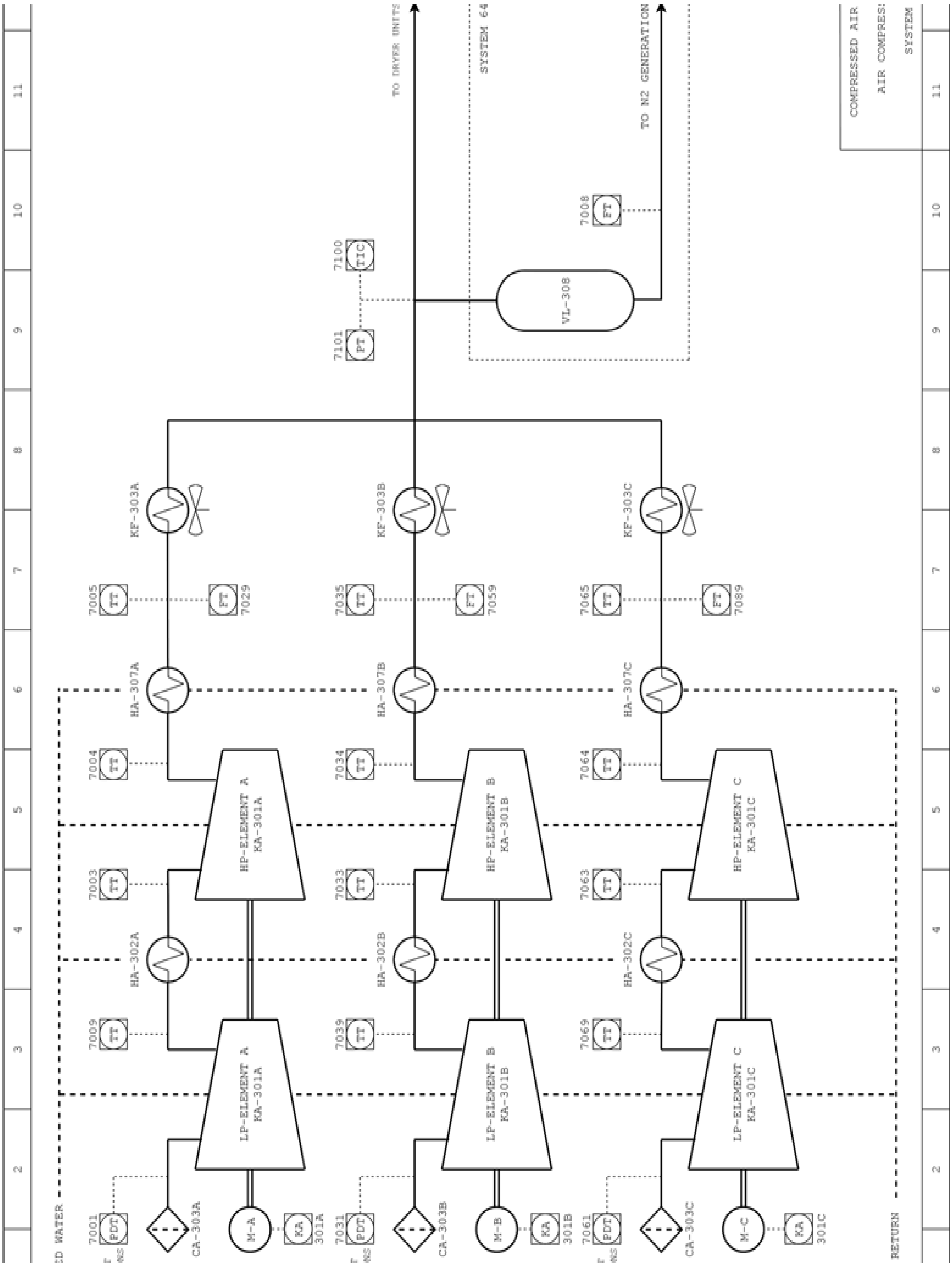
Compressed Air supply:

Service to	Flowrate (Sm^3/h)	Discharge Pressure (bara)	Temperature (°C)	Dewpoint (°C)
Nitrogen Generation Unit	8.000	11	35	
Instrument Air Network	1.060	9	35	- 40
Plant Air Distribution	160	9	35	- 40

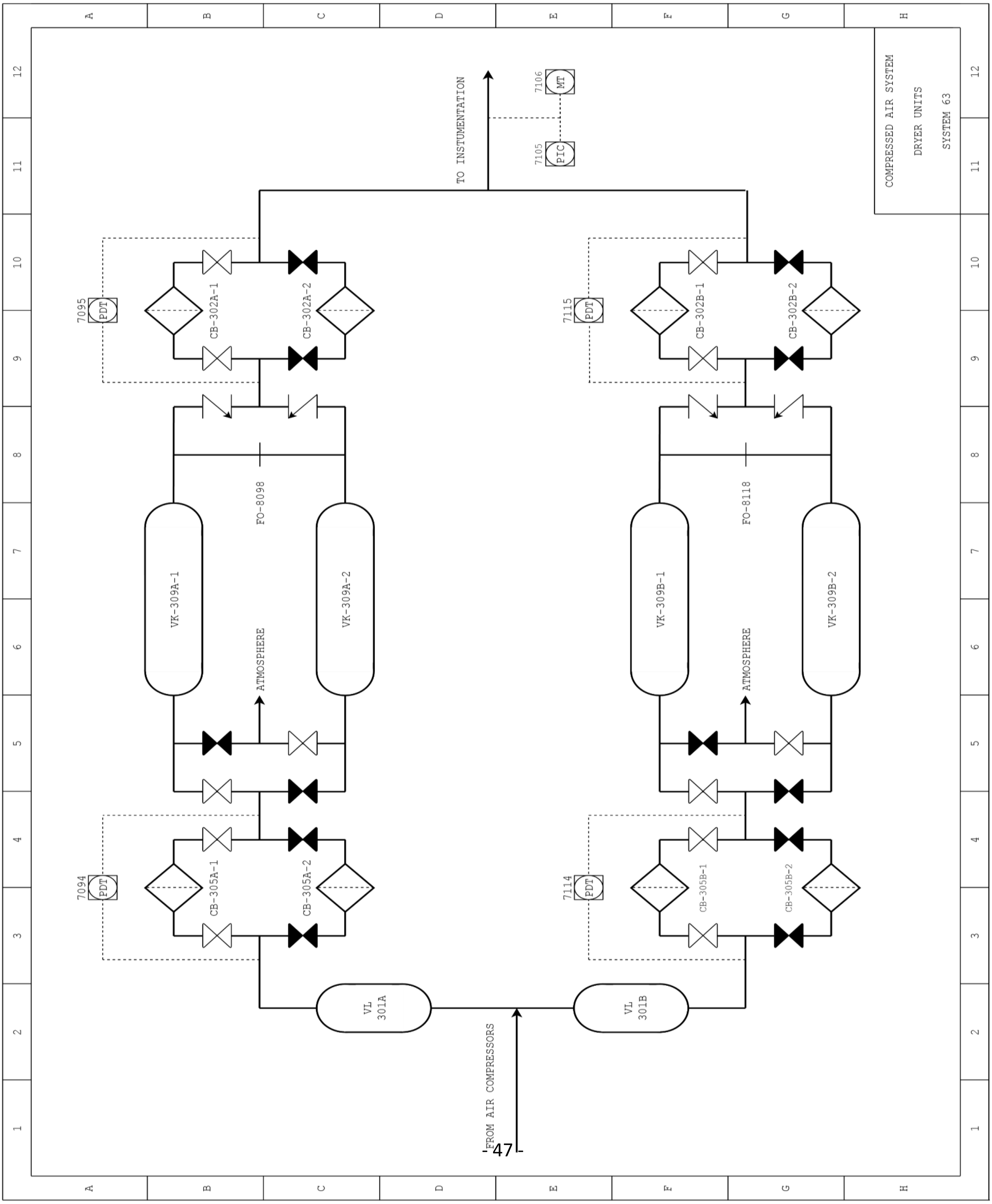
1.2 Process Description

The compressed air system comprises
- two skid mounted assemblies,
- three wall mounted air intake pre-filters

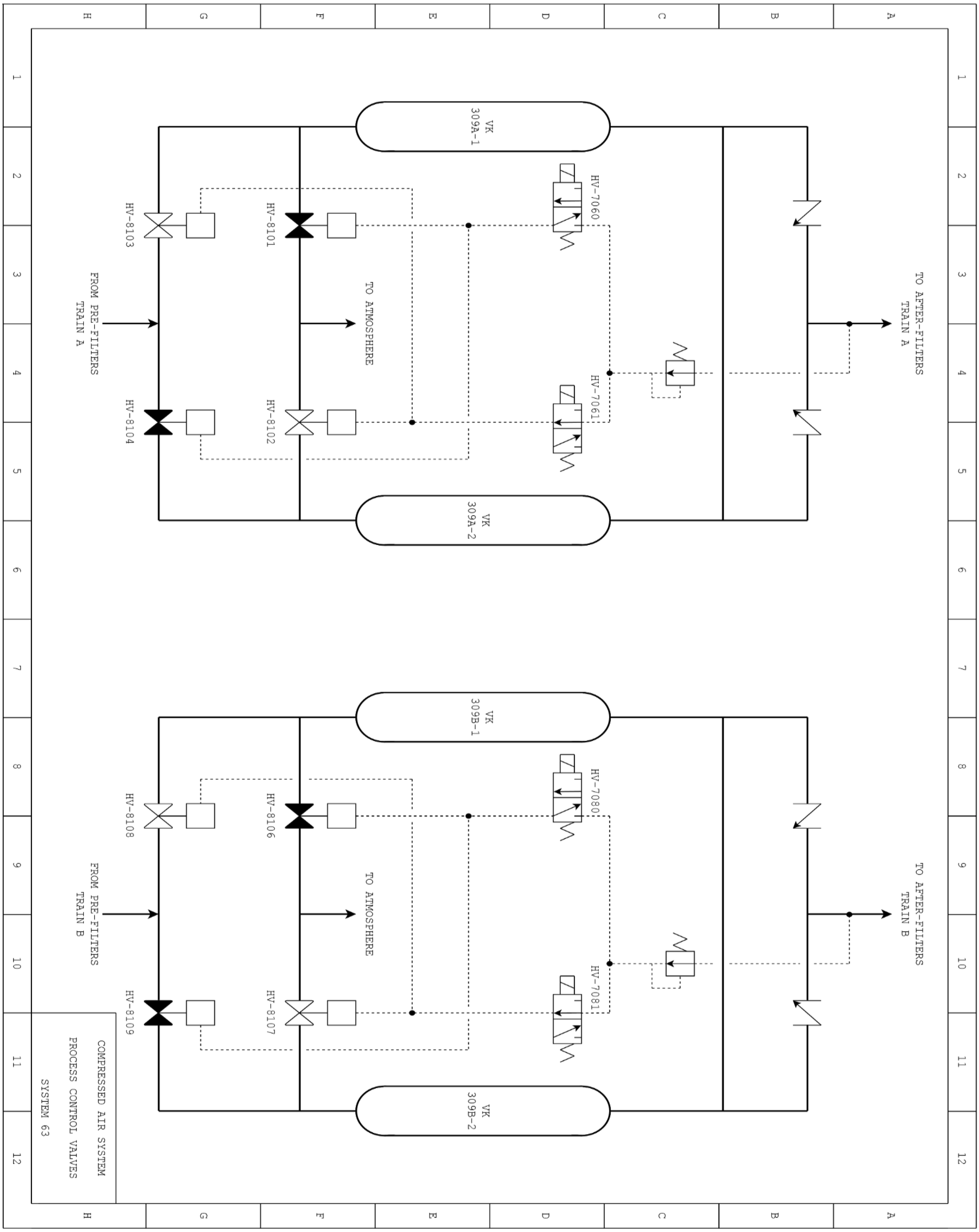
Appendix 2.


TO DRYER UNITS
SYSTEM 64
TO N2 GENERATION
COMPRESSED AIR
AIR COMPRES
SYSTEM
VL-308
7100 TIC
7101 PT
7008 FT
KF-303A
KF-303B
KF-303C
7005 TT
7029 FT
7035 TT
7059 FT
7065 TT
7089 FT
HA-307A
HA-307B
HA-307C
7004 TT
7034 TT
7064 TT
HP-ELEMENT A KA-301A
HP-ELEMENT B KA-301B
HP-ELEMENT C KA-301C
7003 TT
7033 TT
7063 TT
HA-302A
HA-302B
HA-302C
7009 TT
7039 TT
7069 TT
LP-ELEMENT A KA-301A
LP-ELEMENT B KA-301B
LP-ELEMENT C KA-301C
7001 PDT
7031 PDT
7061 PDT
CA-303A
CA-303B
CA-303C
M-A
M-B
M-C
KA 301A
KA 301B
KA 301C
RETURN

Appendix 3


COMPRESSED AIR SYSTEM
DRYER UNITS
SYSTEM 63
TO INSTUMENTATION
7106 MT
7105 PIC
7095 PDT
7115 PDT
7094 PDT
7114 PDT
CB-302A-1
CB-302A-2
CB-302B-1
CB-302B-2
FO-8098
FO-8118
VK-309A-1
VK-309A-2
VK-309B-1
VK-309B-2
ATMOSPHERE
ATMOSPHERE
CB-305A-1
CB-305A-2
CB-305B-1
CB-305B-2
VL 301A
VL 301B
FROM AIR COMPRESSORS

Appendix 4.


VK
309A-1
VK
309A-2
VK
309B-1
VK
309B-2
HV-7060
HV-7061
HV-7080
HV-7081
HV-8101
HV-8102
HV-8103
HV-8104
HV-8106
HV-8107
HV-8108
HV-8109
FROM PRE-FILTERS
TRAIN A
FROM PRE-FILTERS
TRAIN B
TO ATMOSPHERE
TO ATMOSPHERE
TO AFTER-FILTERS
TRAIN A
TO AFTER-FILTERS
TRAIN B
COMPRESSED AIR SYSTEM
PROCESS CONTROL VALVES
SYSTEM 63

Appendix 5. Moisture Conversion Table

Dew Point °C	Dew Point °F	Vapor Pressure (Water/Ice in Equilibrium) mm of Mercury	PPM on Volume Basis at 760 mm of Hg Pressure	Relative Humidity at 70°F%	PPM on Weight Basis in Air
-90	-130	0.00007	0.0921	0.00037	0.057
-88	-126	0.00010	0.132	0.00054	0.082
-86	-123	0.00014	0.184	0.00075	0.11
-84	-119	0.00020	0.263	0.00107	0.16
-82	-116	0.00029	0.382	0.00155	0.24
-80	-112	0.00040	0.562	0.00214	0.33
-78	-108	0.00056	0.737	0.00300	0.46
-76	-105	0.00077	1.01	0.00410	0.63
-74	-101	0.00105	1.38	0.00559	0.86
-72	-98	0.00143	1.88	0.00762	1.17
-70	-94	0.00194	2.55	0.0104	1.58
-68	-90	0.00261	3.43	0.0140	2.13
-66	-87	0.00349	4.59	0.0187	2.84
-64	-83	0.00464	6.11	0.0248	3.79
-62	-80	0.00614	8.08	0.0328	5.01
-60	-76	0.00808	10.6	0.0430	6.59
-58	-72	0.0106	13.9	0.0565	8.63
-56	-69	0.0138	18.2	0.0735	11.3
-54	-65	0.0178	23.4	0.0948	14.5
-52	-62	0.0230	30.3	0.123	18.8
-50	-58	0.0295	38.8	0.157	24.1
-48	-54	0.0378	49.7	0.202	30.9
-46	-51	0.0481	63.3	0.257	39.3
-44	-47	0.0609	80	0.325	49.7
-42	-44	0.0768	101	0.410	62.7
-40	-40	0.0966	127	0.516	78.9
-38	-36	0.1209	159	0.644	98.6
-36	-33	0.1507	198	0.804	122.9
-34	-29	0.1873	246	1.00	152
-32	-26	0.2318	305	1.24	189
-30	-22	0.2859	376	1.52	234
-28	-18	0.351	462	1.88	287
-26	-15	0.430	566	2.3	351
-24	-11	0.526	692	2.81	430
-22	-8	0.640	842	3.41	523
-20	-4	0.776	1020	4.13	633
-18	0	0.939	1240	5.00	770
-16	3	1.132	1490	6.03	925
-14	7	1.361	1790	7.25	1110
-12	10	1.632	2150	8.69	1335
-10	14	1.950	2570	10.4	1596
-8	18	2.326	3060	12.4	1900
-6	21	2.765	3640	14.7	2260
-4	25	3.280	4320	17.5	2680
-2	28	3.880	5100	20.7	3170
0	32	4.579	6020	24.4	3640
2	36	5.294	6970	28.2	4330
4	39	6.101	8030	32.5	4990
6	43	7.013	9230	37.4	5730
8	46	8.045	10590	42.9	6580
10	50	9.029	12120	49.1	7530
12	54	10.52	13840	56.1	8600
14	57	11.99	15780	63.9	9800
16	61	13.63	17930	72.6	11140
18	64	15.48	20370	82.5	12650
20	68	17.54	23080	93.5	14330

www.asge-online.com

Appendix 6.

ADSORBENTS

UOP™ D-201 ACTIVATED ALUMINA

DESCRIPTION

D-201 is a spherical activated alumina with high surface area, high adsorption capacity, and optimal abrasion resistance.

APPLICATIONS

D-201 activated alumina is specifically designed as a high-capacity, abrasion-resistant desiccant. Its excellent adsorption and desorption characteristics make it especially suitable for heatless (pressure swing) adsorption applications, such as dehydration and cryogenic air purification.

GRADES

1/4"	(6.4 mm)	(1/4 inch nominal)
3x6 mesh	(6.7 mm x 3.3 mm)	(3/16 inch nominal)
5x8 mesh	(4.0 mm x 2.4 mm)	(1/8 inch nominal)
7x12 mesh	(2.8 mm x 1.4 mm)	(1/16 inch nominal)

TYPICAL CHEMICAL ANALYSIS

Al_2O_3 (wt-%)	94.6
SiO_2 (wt-%)	0.02
Fe_2O_3 (wt-%)	0.02
Na_2O (wt-%)	0.35
Loss on ignition (wt-%)	5.0

TYPICAL PHYSICAL PROPERTIES

	5x8 Beads	
Surface area (m^2/gm)	350	
Bulk density (lb/ft^3)	47	(753 kg/m^3)
Crush strength[1] (lb_f)	35	(15.8 kg_f)
Abrasion loss (wt-%)	0.2	

[1] Crush strength varies with the sphere diameter. The crush strength reported is for a 5 mesh (4.0 mm) sphere.

REGENERATION

D-201 activated alumina can be regenerated for re-use by purging or evacuating at elevated temperatures.

SHIPPING INFORMATION

D-201 activated alumina is available in multiwall moisture-proof bags, steel drums, and quick load bags.

FOR MORE INFORMATION

For more information, contact your UOP representative or UOP's Adsorbents business at:

UOP M.S. S.r.l.
Viale Milanofiori,
Strada 1, Palazzo E1,
I-20090 ASSAGO MI
Milan, Italy.
Telephone: 39-02-892241
Fax: 39-02-5750-0145
E-mail: Adsorbents@uop.com

UOP 3262m-10 0401A1K-A4

UOP LLC
25 East Algonquin Road
Des Plaines, IL 60017-5017
www.uop.com