INTRODUCTION

The technological evolution of humanity has revolutionized and opened up a lot of possibilities to improve the efficiencies and feasibilities of various processes. An important development in this regard is the handling of high-pressure operations in industrial processes, which has enabled engineers and scientists to accomplish new levels of productivity. The first major high-pressure application is the transfer of a higher amount of thermal energy to steam, which has been used for steam engines since the eighteenth century. Second major contender of high-pressure applications is the gas-phase reactions whose equilibrium can be influenced externally by varying their pressure, following the “Le Chatelier’s principle.” For example, Haber’s process requires high pressure to shift the equilibrium to the product side. These requirements led to the development of high-pressure rotary equipment such as compressors, pumps and static equipment like pipes, heat exchangers fittings, and tubes.

Centrifugal compressors work on the principle of dynamic compression and give continuous flow without any pulsation. They draw the gas in between the blades of a fast-rotating impeller and accelerate it to high velocity, thereby transferring mechanical energy into the enthalpy of the gas being compressed. The gas is then discharged through a diffuser where the velocity head is transformed into pressure.

Multistage carbon dioxide compressor efficiency enhancement using waste heat powered absorption chillers

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Abstract

The performance of a multistage centrifugal compressor is highly influenced by the ambient conditions, especially during the summer seasons; their capacity shrinks and thus the power requirement for compression will increase. The prime cause of these constraints is the interstage cooling limitations. This study simulates various suction conditions of a multistage compressor on Aspen HYSYS® and suggests its debottlenecking by making the suction temperatures comparable to winter seasons. This is achieved by installing an additional exchanger at the downstream of each interstage cooler, cooling down the gas further by using absorption refrigeration chillers. These chillers are powered up by the waste heat recovered from the exhaust steam coming from the prime mover, steam turbine, of the same compressor. This modification will save a considerable amount of power (663 kW), net savings (Gross Savings – OPEX: 72 289 $/y), and reduce the carbon footprint (954 ton/y) of the overall process.

KEYWORDS

Aspen HYSYS®, centrifugal compressor, chiller, compressor performance, power saving
type of dynamic compressors is axial flow compressors. The classification is based on the direction of flow relative to the axis of compression. The centrifugal compressors offer lower maintenance and running cost along with a typical compression ratio below 3.5 per stage or a maximum compression ratio of 5. Going beyond the value of compression ratios up to 5 would result in supersonic velocities in the diffuser, which necessitates a complex design of the diffuser. On the contrary, axial flow compressors offer a compression ratio of 10 at the expense of high maintenance and running cost.

In the past, many techniques have been employed to optimize the performance of the compressor. One such effort has been done by Golden et al to optimize the performance of a compressor in a Fluidized Catalytic Cracker using two alternate approaches. The first approach primarily focused on reducing the discharge pressure of a compressor, which in turn would increase the capacity at the expense of the head, as per compressor curves. However, this reduction might not be possible due to the downstream constraints of the compressor. For example, in a urea manufacturing plant, the compressed CO₂ and ammonia are the reaction feed gases. Reducing the discharge pressure would mean lowering the pressure at which the reaction takes place, which will ultimately reduce the efficiency of the reaction and lower the production. Another approach utilized in this study is increasing the suction pressure of the compressor, which will increase the density of the gas. This was achieved by reducing the pressure drop in the equipment installed at the upstream of the compressor such as replacing the trays of a column with structured packing. In the reported case study, this modification increased the capacity of the compressor by 30%.

Albusaidi and Pilidis discussed the effects of density variation on the compressed gas. Keeping all other parameters constant, it was reported that the density reduces with a decrease in the molar mass of the gas, resulting in reduced flow rates, pressure ratio, and polytropic head across the compressor. In another study, Albusaidi and Pilidis stated that reducing the suction pressure of a compressor would reduce the flow rate and pressure ratios. Furthermore, the authors discussed the deviation of various parameters of the compressor caused by the suction temperature. It was concluded that a reduced temperature would result in higher discharge pressure, higher pressure ratio, and enhanced capacity, as discussed in some recent works.

Popli et al evaluated the use of absorption chillers to cool down the suction temperature of the compressed air, which was then used for the combustion to power up a gas turbine. The authors proposed a scheme to generate the steam in a waste heat recovery steam generator from the flue gases of the gas turbine at 550°C to run three single-effect Lithium-Bromide water (LiBr-H₂O) absorption chillers for the heat recovery. It was concluded that the inlet air cooling can enhance the power output and energy efficiency of a gas turbine by 23.2% and 13%, respectively. Furthermore, the electrically powered vapor compression type chillers were also considered for the same job, but they were economically unfeasible owing to high electrical power demand. On the other hand, the payback of LiBr-H₂O-based chillers was only a few years due to relatively low Capital Expenses (CAPEX) and Operating Expenses (OPEX). LiBr-H₂O-based chillers have another advantage of direct utilization of waste heat which could potentially save considerable power consumption if used to cool gas or air before compression. Better performance of CO₂ power cycles from waste heat recovery has been reported in a recent paper.

The study conducted by Popli et al laid a significant foundation for process optimization via heat recovery of the gas turbine. Various studies are performed to optimize the performance of the compressor, however:

- The area of centrifugal compressor optimization by inlet air cooling based on the recovery of waste heat in absorption chiller has not been explored much. This idea combines the two strategies for performance enhancement, that is, waste heat recovery and interstage cooling, each of which is usually employed exclusively.
- There has been no thermodynamic and economic evaluation of compressors based on the waste heat recovery using the absorption chillers to improve the process efficiency. These evaluations explore the possibility as well as the feasibility of the system, both of which should be studied in order to assess the usefulness of any proposed solution.

This study thoroughly investigates the variation in the performance of a compressor owing to the change in cooling water supply (CWS) temperatures, which is anticipated to significantly affect the performance of intercoolers between the stages of the compressor. It can result in high suction temperature for all stages and hence reduced capacity and lower discharge pressure will be obtained. Later on, an alternate method has been suggested to enhance the performance of the compressor by the installation of a chiller at the suction of each stage to lower the temperatures. It can make the compressor work virtually at the same suction temperatures as in winter seasons. This modification has a provision to be taken offline in winter seasons or when not needed.

2 | CONSTRAINTS OF INTERSTAGE COOLERS DURING SUMMER SEASON

Energy requirements for the process of compression go up by increasing suction temperature. It is shown in the Figure 1 which reveals the data of suction temperature along with the
power consumption of a centrifugal compressor for 2 years. As the suction temperatures increase during hot seasons, the overall power consumption also increases, with all other parameters kept constant. This rise in power is due to the increased kinetic energy of the gas molecules, which increases the volume of the gas per unit area and eventually forces the compressor to do more work to get the desired pressure, which is also evident from the equation of compressor power stated below in Equation (1):^7^ 

\[
\text{Power (kW)} = \frac{\dot{m}}{3600} \cdot 8.314 \cdot \frac{Z}{MW \times (\eta - 1) / \eta} \times T_{\text{suc}} \times P_{\text{r}}^{(\eta - 1)/\eta} - 1
\]

Achieving high discharge pressures would have been practically impossible in a single stage due to the temperature rise, increasing the required head for compression proportionally,^16,17^ thereby making it more energy intensive to compress the gas further at the same temperature.\(^\text{10}\) To get high discharge pressures, the process of compression is split into stages along with intercoolers placed in-between the subsequent stages.\(^\text{18,19}\) Intercooling usually makes the suction temperature equal at all stages, thus making the overall process approach isothermal compression. The interstage coolers remove the heat added to gas during compression via cooling with the cooling water (CW). Once the heat is removed from the process, the cooling water is returned to

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**FIGURE 1** Variation of total power consumption and suction temperature along the year

**FIGURE 2** (A) Variation of cooling water temperature and its impact on suction temperature, (B) variation of volumetric flow rate across the compressor along the year, and (C) variation in the compressibility factor of compressor discharge with a change in the suction temperature (D) variation in the density with a change in suction temperature
the cooling towers where it is cooled down by the natural phenomenon of evaporation. The process of cooling the gas in between the stages increases the economic feasibility of compression. However, the cooling water temperature depends on the ambient conditions, such as temperature and humidity. These factors affect the rate of evaporation in the cooling towers. During the summer seasons, the ambient temperatures and humidity are very high. This seasonal variation results in about a 10-12°C increase in summer seasons on average. Figure 2A explains the effect of ambient conditions on cooling water and suction temperatures of a multistage carbon dioxide (CO₂) compressor for 2 years. It is evident from Figure 2A that the overall change in both CWS and suction temperature is purely cyclic. During the winter seasons, the CWS temperature is minimum, resulting in the lowest suction temperature for the cooling CO₂ in interstage coolers of the compressor. This change in CW temperature makes the process of compression ambient temperature-dependent due to the change in the specific volume of gas. Since the centrifugal compressors are designed for a limited volumetric flow rate, it can be inferred that the capacity in terms of the mass flow rate entering the compressors reduces with an increase in the suction temperature. These changes in the capacity are also evident from the corrected volumetric flowrate of CO₂ across a multistage compressor for 2 years. It is clear that compressor gives maximum volumetric flow rate in winter seasons when the CWS is at its lowest value (see Figure 2B). The other factor is the compressibility factor (Z) which directly influences power and its demand affected by suction temperature. CO₂ shows a unique behavior when compared without incorporating the “principle of corresponding states.” It shows negative deviation at lower pressure and higher temperature and a positive deviation at very high pressures and low temperatures from ideal gases. For the scope of this study, the gas in negative deviation zone is considered, that is, the volume of the gas is lesser when compared to an ideal gas at a given temperature and pressure.

Although the compressibility factor directly affects the overall power consumption of compression, its effect gets shadowed by the change in suction temperature. Figure 2C reveals the variation in the compressibility factor of the gas at the suction and discharge streams of the third stage of compressor against the compressor suction temperature. The plot has been generated using the Aspen HYSYS® simulation model of the compressor at 118% load. Figure 2D explains the change in density for variation in suction temperature. A linear relation between these two parameters is observed within the temperature ranges relevant to this study. The increase in density results in an increase in the capacity for a centrifugal compressor as there is a higher mass flow at the same volumetric flow rate with lower temperatures.

3 | METHODOLOGY

This paper analyses the effects of seasonal variations on a 4-stage centrifugal compressor performance, compressing CO₂ gas. The study used the simulation of CO₂ compressor curves on Aspen HYSYS® V9 using UNIversal QUasiChemical (UNIQUAC) as the fluid package. Then, a modification is proposed to mitigate those seasonal variation effects and the efficacy of the proposed modification is evaluated.

3.1 | Existing and proposed schemes

The existing and proposed schemes of compressor are shown in Figure 3. The uncompressed gas at 72°C and 0.18 kg/cm² enter the existing scheme from a CO₂ separation network which is cooled down in a CW-based cooler to 41°C. After a series of compression stages, the gas is discharged at 155 kg/cm² as a final product. On the other side, this CO₂-based compressor is driven by an extraction-condensing steam turbine (rated power 8320 kW). After passing through the high-pressure (HP) and low-pressure (LP) turbines, the high-pressure steam goes into a condenser. The steam turbine consumes a total of 77 200 kg/hr of high-pressure steam (101 kg/cm²) in HP steam turbine to pass through its blades. Afterward, a fraction of 64 195 kg/hr at 24 kg/cm² is taken off from the HP turbine which is used in the process, whereas the rest of the steam, that is, 13 005 kg/hr goes to drive the LP turbine where it transfers its kinetic energy further. Finally, it comes out exhausted as saturated steam at 0.8 kg/cm² and is sent to the condenser, where it is cooled down to change its state to sub-cooled water by transferring the latent heat to the cooling water.

As per current schematic, the temperature of gas post interstage cooler or at the suction of first 3 stages is roughly 41°C, the power consumption required for compression of gas to the required pressure can be reduced. This statement is in-line with the compressor power equation, that is, Equation (1), which calculates the power consumption of a centrifugal compressor for a polytropic process. As per the equation, the increased suction temperature increases the total power requirement of compression for the polytropic compression process.

The proposed modification aims to reduce the overall power consumption of the compressor or result in an increased volumetric flow rate at the same power consumption depending on the process requirements. This is achieved by reducing the suction temperature of the gas after interstage coolers to 20°C from the current value of 41°C. This will also reduce the carbon footprint of the overall process as lesser fuel will be burned to generate the same amount of power. The additional cooling will be done with the help of newly proposed coolers in accordance with Figure 3B, whereas...
the existing interstage coolers will stay as per the existing scheme. The chiller can also be taken offline during winter seasons for energy optimization, without altering the process. Two Aspen HYSYS® steady-state simulations were built to represent the actual process and the proposed scheme. The selected fluid package is Lee–Kesler–Plöcker (LKP) which represents the gaseous systems with reasonable accuracy. Initially, the reference conditions for the 4-stage centrifugal CO₂ compressor were simulated using its compressor curves for validation. The compressor was simulated using LKP equation of state, which is a modified form of Benedict, Webb, Rubin, Starling equation of state. This is due to its higher accuracy for polar gases and particularly for CO₂ on high pressures compared to other equations of state. Cooler-1, Cooler-2, and Cooler-3 are the proposed coolers whereas, E-1, E-2, and E-3 are the existing interstage coolers as shown in the Figure 4. The process specifications for the reference state are presented in Table 1.

3.2 | Case studies

To investigate the effect of seasonal variations on compressor performance, different cases were designed, the details of which are given in Table 2. Subscript A represents that the process is operating in summer conditions, whereas subscript B represents the winter conditions.

First of all, reference conditions were simulated with the original compressor curves (Case 1A). This case serves as the validation of the developed simulation, because the simulated
and actual values may be compared to determine whether the simulation is accurately representing the system. For comparison, Average Absolute Relative Error (AARE) may be used, which is defined in Equation (2) as:

$$\text{AARE} = \frac{1}{n'} \sum_{i=1}^{n'} \left| \frac{\text{Simulated}_i - \text{Actual}_i}{\text{Actual}_i} \right| \times 100\%$$  \hspace{1cm} (2)

Then, in Case 1B, the suction temperatures of all stages, except the fourth stage were reduced to 20°C as they are in winter seasons (from Figure 2A) keeping all the other parameters constants. Stage 4 temperature was not reduced because after the third stage during CO₂ compression, the pressure crosses the threshold value of its critical pressure (75.22 kg/cm², 31°C) and its temperature must be kept significantly higher than its critical value to avoid phase change. The flow was kept as per the reference conditions, 91% of rated value.

Followed by that in Case 2A, the flow rate of the compressor was increased to the maximum value, which was about 118% of the rated value. The suction temperatures of all stages were kept identical, as in Case 1A for the summer season. Later on, in Case 2B all suction temperatures except at the 4th were reduced back to 20°C as in Case 1B and the rest of the parameters were kept constant.

### 3.3 Duty calculation and chiller design

The cooling load calculations per stage have been performed by the Kern’s heat exchanger design method. The

![Diagram](Image)

FIGURE 4 Simulations of existing (A) and proposed (B) schemes of the compressor
calculations in Table 3 show that to reduce the temperature of all suction, in the peak summer season, back to their winter values, that is, from 41°C to 20°C, 374 kW will be required for CO₂ at 118% of the rated capacity (\( \dot{m} = 20.7 \) kg/s). Thus, the accumulated cooling load for the first three stages will be 1122 kW, which would require about 320 Refrigeration tons (TOR).

The next step is the selection of source for the required cooling and heat recovery. Absorption chillers are preferred for industrial applications due to their lower operating cost. Their advantage of directly utilizing heat makes them stand out from all other cooling media types, which has made them a very optimum option for heat recovery in the industry worldwide. They usually employ demineralized water as a cooling media named as chilled water (CHW). Electrical chillers would require additional energy which will make the whole modification economically unfeasible, as found out for a similar study on gas turbines by Popli et al. On the contrary, absorption chillers for the suggested modification will recover the waste heat as their source of energy. The energy required to power up the proposed absorption chiller would come from the exhaust steam coming out from the steam turbine and going to the condenser. It is proposed to recover this energy instead of wasting the latent heat of the steam into the atmosphere. The coefficient of performance (COP) of the proposed chiller is 0.77 as found by Popli et al. The total amount of heat required for 1122 kW cooling load would be 1447 kW, as per Equation (3)

\[
\text{COP of Absorption Chiller} = \frac{Q_C}{Q_H} \tag{3}
\]

Power available from Exhaust Steam (kJ/hr) = \( \dot{m} \times \Delta \dot{H} \times 0.7 \tag{4} \)

Mass flowrate of Steam required for Chiller = \( \frac{Q_H}{\Delta \dot{H}_{\text{sat}}} \) \tag{5}

\[ Q = U \times A \times \theta_{\text{LMTD}} \tag{6} \]

A simple process flow diagram of the proposed chiller is revealed in the Figure 5. \( Q_C \) is the heat absorbed by the evaporator of the chiller and \( Q_H \) is the heat provided to the generator of the chiller in the form of steam, whereas the pump transfers the dilute solution to the generator for regeneration before exchanging its energy with the concentrated solution coming back from the generator. Therefore, to run the chillers only a handful of 3326.5 kg/hr will be required for 1447 kW of heat requirement of the proposed chiller as calculated from Equation (5) out of total 13 005 kg/hr of the total exhaust steam. The total energy this exhaust stream has is 5750 kW as per Equation (4). All calculations assume that only 30% of the steam remains uncondensed, which will also serve to accommodate the energy losses during the whole process.
4 | RESULTS AND DISCUSSION

The first part of this study was to reproduce the actual process in Aspen HYSYS® environment using the original compressor curves (Case 1A). AARE was used for the comparison of simulated and actual values. The results of this case are presented in Table 4, where it can be seen that the simulation is accurately representing the actual process. The process parameters in the simulation were found to be very close to actual values and negligible errors were obtained.

These chillers provided chilled water at 16°C to newly installed exchangers at the downstream of the existing interstage coolers. The pressure drop of these newly installed coolers was also incorporated in the simulation model to reflect the changes to the overall power consumption, it brought only minor changes in overall power consumption which got subsided in the overall reduction of power proposed modification. Next, the cases described in the methodology were simulated, the results of which are shown in Table 5. For Case 1B, the suction temperatures were decreased to the temperatures encountered in winter seasons by using the proposed scheme of downstream chillers. This reduction in suction temperatures of about 11°C for first three stages resulted in about 3.15% savings (495 kW) in the total power consumption of compressor including mechanical losses. Similar results were observed by comparing the cases 2A and 2B, operating at 118% of the rated flow, in which the gas cooling at the suction saved about 663 kW of the total power consumption of the compressor. Finally, Case 3 was simulated to evaluate the degree of increase in the corrected volumetric flow rate by gas cooling at suction if the power is kept the same as in Case 2A. It was observed that the volumetric flow of the gas in the proposed scheme was increased by 631 NMC/h of the volumetric flow in Case 2A, that is, the existing scheme in summer. The main reason behind this increase is the lower specific volume of the gas.

In short, the installation of LiBr single-effect absorption chiller to reduce the suction temperatures of a centrifugal compressor in the summer season resulted in the following benefits:

- 495 kW reduction in the total power consumption of the centrifugal compressor at 91% of the rated flow rate (63 160 kg/hr).
- 663 kW reduction in power at 118% of the rated flow rate (74 528 kg/hr).
- 1447 kW heat recovery by utilization of latent heat of the exhaust stream to power up a single-effect absorption chiller for the cooling load of inlet gas cooling on 118% of the rated flow.

4.1 | Economic evaluation

The estimated breakdown of the CAPEX for installation of chiller is stated in Table 6. The cost of single-effect LiBr-H₂O absorption chiller is 95 US$/kW, for heat exchangers, the cost is 1 US$/kW and cost of installation and piping is 11 US$/kW. The annual OPEX for electricity and water requirements of an absorption chiller is US$ 4.1/kW and 2 US$/kW. The proposed modification is suggested to be kept in service for 280 days excluding winter seasons and for that time slot, it is set to save 663 kW which is synonymous to $96 312.25 saving annually for the natural gas price of 1021/MMBTU. With these savings, the payback for the proposed modifications is 7 years and 7 months.

\[
\text{Payback} = \frac{\text{CAPEX}}{\text{Net savings}}
\]

\[
\text{Annual Power Savings} = \frac{\text{MMBTU}}{\text{yr}}
\]

\[
= \text{Savings (kW)} \times \frac{86 400 \times 280 \times 0.9478}{10^6}
\]

4.2 | Environmental impact

Environmental impact of the industry had never been this important since the evolution of mankind. Facts reveal that sea levels have risen by 8-9 inches since 1880. The cumulative rise in the temperature of land and ocean has been increasing by 0.07°C every decade since 1880; however, it has doubled since the postindustrial era. This is resulting in faster depletion of glaciers, storms across the planet, and permanent alterations to Earth’s geological, biological, and ecological systems. One of the major contributors to all these environmental deteriorations is Greenhouse Gases (GHGs). To save the planet, we must cut down the sources generating GHGs.

Industrial activities contribute to more than 50% of the total carbon dioxide in Pakistan. A total of 178 Million Metric Tons were generated in 2016 in Pakistan. A lot of heat generated during various industrial processes is wasted, which has a deteriorating effect on the environment. Waste heat recovery is the solution to this problem; it not only spares the environment from unwanted heat but also improves the economic efficiency of the industry. Recent reviews and studies have analyzed various processes for their potential of waste heat.

Chillers used for cooling also have an environmental impact, they vary in their method of operation depending on the type of cooling medium used. Chillers operating on refrigerant have significant contributions to global warming due to two main reasons: the release of refrigerants and emissions of greenhouse gases by associated energy use.
of refrigerants resulting in ozone depletion and an increase in global warming is unavoidable lifetime environmental impacts. The extent of impact of the environment is directly related to the refrigerant properties in particular to its global warming potential.\(^2\) Different modifications in terms of design, service, and manufacture are proposed by the researchers.\(^3\) One such solution lies in the adaptation of ideal refrigerants that have higher stability, safety, and compatibility with the environment. The adsorption-based system is a potential solution which reduces the environmental impact by using water as a cooling medium.\(^3\)

Absorption chillers, however, are advantageous for heat recovery. As it can operate on the heat of flue gases or waste heat to optimize the process fuel economy along with the net reduction in the heat emissions to the environment.\(^3\) Steam-based absorption chiller utilizes water and waste heat of process to produce cooling. By consuming the heat of fuel gases, it reduces the waste of heat in the environment.

Moreover, by waste heat recovery technique implemented in such chiller, the carbon footprint is also reduced. This project will not only efficiently utilize the heat being wasted in the environment but is also set to reduce the carbon emissions. A net of 663 kW of power savings power requirement will reduce 68.09 kg/hr of fuel gas to be burned as per Equation (8) having the following molar analysis CH\(_4\) 70%, N\(_2\) 18.9%, CO\(_2\) 11%, and C\(_2\)H\(_6\) 0.1%. The cumulative Lower Heating Value (LHV) of the fuel gas is 50 MJ/hr.\(^3\) The annually saved moles of the fuel are 457 Metric Tons lesser fuel consumption per year\(^*\) by Equation (9), with an assumption of 70% fuel to steam conversion to accommodate losses. This is based on the fact that during the winter the proposed modification will not be needed as the ambient temperature reduces significantly. Therefore, for the sake of calculations, a year has been considered of 280 days, as winter seasons play no part in power savings via this modification. However, a \(^*\) is shown with the word “year” to distinguish it from the normal.

As per the stoichiometry Equation (11), the moles of CO\(_2\) will also be same as of fuel gas. The mass of CO\(_2\) which is prevented from being generated due to lesser power requirement is calculated by Equation (12), which is derived from the definition of mole, that is, Equation (10), and came out to be 954 Ton per annum with the given fuel analysis, as stated in the Table 7. This is completely independent of 1447 kW of heat recovery from the condenser to power the chiller which is also an added advantage.

\[
\text{Fuel Savings (kg/hr)} = \frac{\text{Reduction in Power (kW)}}{\text{LHV (kJ/kg)} \times 0.7} \times 86 400 \times 280
\]

\[
\text{Moles of Fuel} = \frac{\text{Mass of Fuel}}{\text{Average Molar Mass}}
\]
This study investigates the adverse effects of the seasonal variance in ambient temperature as well as humidity on the performance of a four-stage CO₂ compressor. Moreover, it suggests the possible debottlenecking of the system by installing downstream chillers at each stage, which reduces the suction temperatures to the levels encountered in winter seasons. Aspen HYSYS® simulations were developed and validated using the process specifications at the reference point. The existing and proposed schemes were simulated at two different flows to assess the efficacy of the proposed scheme. The possibility of increasing the flow using the same power was also investigated. The following observations were made as follows:

- 954 tonnes of carbon dioxide per annum is prevented from being generated due to lesser power requirement.
- 1447 kW of heat recovery from the condenser to power the chiller.
- Payback for the proposed modifications is 7 years and 7 months.
- Annual savings of 663 kW which is synonymous to $72 289/year for the natural gas cost of 1021/MMBTU.

Therefore, it can be concluded that the proposed modification can result in increased performance of the compressor, which can be channeled either for power saving or increased throughput. Economic analysis and environmental analysis show that these improvements justify the costs incurred by the interstage coolers. Although these results present an optimistic overview of the benefits that may be achieved by the proposed scheme, the purpose of this study is to demonstrate the potential of the proposed scheme.

### Economic evaluation

| Parameters                                      | Power (kW) |
|------------------------------------------------|------------|
| Cases                                          | Percent of rated flow | Mass flow (kg/hr) | Stage 1 | Stage 2 | Stage 3 | Stage 4 | Total power | Power saved |
| Case 1A                                        | 91%        | 57 419             | 1912    | 2342    | 2252    | 881     | 7387        | 495         |
| Case 1B                                        | 91%        | 57 419             | 1779    | 2171    | 2061    | 881     | 6892        |             |
| Case 2A                                        | 118%       | 74 528             | 2676    | 3140    | 3000    | 1109    | 10 326      | 663         |
| Case 2B                                        | 118%       | 74 528             | 2500    | 2909    | 2744    | 1109    | 9933        |             |

### Economic evaluation of the project

| Economic evaluation                             |           |
|------------------------------------------------|-----------|
| Total power savings (118% load)                | 663 kW    |
| Annual power savings (280 d of operation)      | 15 635.11 MMBtu/y* |
| Corresponding fuel savings (70% fuel to power conversion efficiency) | 457.826 Ton/y* |
| Cost of fuel gas\(^{25}\)                      | 6.16 $/MMBtu |
| Gross savings                                  | 96 312 $/y* |
| Total operating cost of electricity per year   | 16 146.7 $ |
| Total operating cost of water per year         | 7876.4 $   |
| OPEX                                           | 24 023 $   |
| Total cost of exchangers                       | 132 363 $  |
| Cost of absorption chiller                     | 374 131 $  |
| Installation cost for chiller and exchangers   | 43 320 $   |
| CAPEX                                          | 549 814 $  |
| Net savings (gross savings - OPEX)             | 72 289 $/y* |
| Payback (Equation 7)                           | 7.6 y*     |

\(^{25}\)Based on rate of Dollar on 07 September 2020, $1 = 165.86 Pakistani Rupees (PKR).
### Nomenclature

| Symbol | Description |
|--------|-------------|
| A      | Area Required per Exchanger |
| AARE   | Average Absolute Relative Error |
| CAPEX  | Capital Expenses |
| CHW    | Chilled Water |
| COP    | Coefficient of Performance |
| CW     | Cooling Water |
| CWS    | Cooling Water Supply |
| DMW    | Demineralized Water |
| GHG    | Greenhouse Gas |
| HP     | High-Pressure |
| LHV    | Lower Heating Value |
| LKP    | Lee–Kesler–Plöcker |
| LMTD   | Log Mean Temperature Difference |
| LP     | Low-Pressure |
| ṁ      | Mass flow Rate |
| n      | Number of Moles |
| n'     | Number of Observations |
| NMC    | Normal Meter Cube |
| OPEX   | Operating Expenses |
| PKR    | Pakistani Rupee |
| Pr     | Pressure Ratio |
| QC     | Heat absorbed by the evaporator of the chiller |
| QH     | Heat provided to the generator of the chiller |
| T      | Temperature |
| T_{suc} | Suction Temperature |
| TOR    | Tons of Refrigeration |
| U      | Overall Heat Transfer Coefficient |
| UNIQUAC | UNIversal QUAsiChemical |
| US$    | United States Dollar |
| Z      | Compressibility Factor |
| ρ      | Density |
| η      | Polytropic exponent |
| θ_{LMTD} | Log Mean Temperature Difference |

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**REFERENCES**

1. Manabe R, Nakatsubo H, Gondo A, et al. Electrocatalytic synthesis of ammonia by surface proton hopping. *Chem Sci*. 2017;8(8):5434-5439.

2. Eggers R. *Industrial High Pressure Applications: Processes, Equipment, and Safety*. Germany: John Wiley & Sons; 2012.

3. Lüdtke KH. *Process Centrifugal Compressors: Basics, Function, Operation, Design, Application*. Springer Science & Business Media; 2004.

4. Boyce MP. *Gas Turbine Engineering Handbook*. UK: Elsevier; 2011.

5. Casey MV. The effects of Reynolds number on the efficiency of centrifugal compressor stages. *J Eng Gas Turbines Power*. 1985;107(2):541-548.

6. Rajadurai JS. *Thermodynamics and Thermal Engineering*. New Delhi: New Age International; 2003.

7. Golden S, Fulton SA, Hanson DW. Understanding centrifugal compressor performance in a connected process system. *Petrol Technol Quart*. 2002.

8. Sahu JN, Chava VSRK, Hussain S, Patwardhan AV, Meikap BC. Optimization of ammonia production from urea in continuous process using ASPEN Plus and computational fluid dynamics study of the reactor used for hydrolysis process. *J Ind Eng Chem*. 2010;16(4):577-586.

9. Albusaidi W, Pilidis P. An iterative method to derive the equivalent centrifugal compressor performance at various operating conditions: Part II: modeling of gas properties impact. *Energy*. 2015;8(8):8516-8536.

10. Albusaidi W, Pilidis P. An iterative method to derive the equivalent centrifugal compressor performance at various operating conditions: Part I: modelling of suction parameters impact. *Energy*. 2015;8(8):8497-8515.

11. Mohammed RH, Qasem NAA, Zabair SM. Enhancing the thermal and economic performance of supercritical CO2 plant by waste heat recovery using an ejector refrigeration cycle. *Energy Convers Manage*. 2020;224:113340.
12. Mohammed RH, Alsagri AS, Wang X. Performance improvement of supercritical carbon dioxide power cycles through its integration with bottoming heat recovery cycles and advanced heat exchanger design: a review. *Int J Energy Res.* 2020;44(9):7108-7135.

13. Popli S, Rodgers P, Eveloy V. Gas turbine efficiency enhancement using waste heat powered absorption chillers in the oil and gas industry. *Appl Therm Eng.* 2013;30(1):918-931.

14. Radchenko R, Radchenko N, Tsoy A, Forduy S, Anatoliy Z, Kalinichenko I. Utilizing the heat of gas module by an absorption lithium-bromide chiller with an ejector booster stage. *AIP Conf Proc.* 2020;2285(1):030084.

15. Njoku IH, Oko C. Comparative thermoeconomic and environmental impact analyses of different combined cycle power systems. *Int J Energy Clean Environ.* 2021;22(2):17-50.

16. Abbaspour M, Chapman KS, Krishnaswami P. Nonisothermal compressor station optimization. *J Energy Res Technol.* 2005;127(2):131-141.

17. Hiller S-J, Matzgeller R, Horn W. Stability enhancement of a multistage compressor by air injection. *J Turbomach.* 2011;133(3):7.

18. Canièrè H, Willockx A, Dick E, De Paepe M. Raising cycle efficiency by intercooling in air-cooled gas turbines. *Appl Therm Eng.* 2006;26(16):1780-1787.

19. Romeo LM, Bolea I, Lara Y, Escosa JM. Optimization of intercooling compression in CO2 capture systems. *Appl Therm Eng.* 2009;29(8-9):1744-1751.

20. Saw SZ, Nandong J. Simulation and control of water-gas shift packed bed reactor with inter-stage cooling. *IOP Conf Ser: Mater Sci Eng.* 2016;121:012022.

21. Bassily AM. Performance improvements of the intercooled reheat recuperated gas-turbine cycle using absorption inlet-cooling and evaporative after-cooling. *Appl Energy.* 2004;77(3):249-272.

22. Singh O. *Applied Thermodynamics.* New Delhi: New Age International (P) Limited Publishers; 2003.

23. Kern DQ. *Process Heat Transfer.* New Delhi: Tata McGraw-Hill Education; 1997.

24. Zabala ES. Technological and economic evaluation of district cooling with absorption cooling systems in Gävle (Sweden). Proceedings Master Thesis at University of Gävle, Sweden; 2009.

25. Industrial Consumers Sale Prices and Minimum Charges Effective July 1, 2019, Vide Ogra's Notification Dated June 29, 2019, website, July 1, 2019.

26. Lindsey R. Climate Change: Global Sea Level, August 14, 2020; 2020.

27. Dahlman RLAL. Climate Change: Global Temperature, August 14, 2020; 2020.

28. worldometer. Pakistan CO2 Emissions.

29. Su Z, Zhang M, Xu P, et al. Opportunities and strategies for multigrade waste heat utilization in various industries: a recent review. *Energy Convers Manage.* 2021;229:113769.

30. Ouyang T, Xu J, Su Z, Zhao Z, Huang G, Mo C. A novel design of low-grade waste heat utilization for coal-fired power plants with sulfuric acid recovery. *Energy Convers Manage.* 2021;227:113640.

31. Calm JM. Emissions and environmental impacts from air-conditioning and refrigeration systems. *Int J Refrig.* 2002;25(3):293-305.

32. Calm JM, Didion DA. Trade-offs in refrigerant selections: past, present, and future. *Int J Refrig.* 1998;21(4):308-321.

33. Calm J. *Comparative Global Warming Impacts of Electric Vapor-compression and Direct-fired Absorption Equipment;* Final report. Electric Power Research Inst., Palo Alto, CA (United States); Calm (James M.), Great Falls, VA (United States). EPRI-TR-103297; 1993.

34. Shmroukh AN, Ali AHH, Ookawara S. Adsorption working pairs for adsorption cooling chillers: a review based on adsorption capacity and environmental impact. *Renew Sustain Energy Rev.* 2015;50:445-456.

35. Ryan W. Driving absorption chillers using heat recovery. *ASHRAE J.* 2004;46(9):S31.

36. Sheykhi M, Chahartaghi M, Balakheli MM, Kharkeshi BA, Miri SM. Energy, exergy, environmental, and economic modeling of combined cooling, heating and power system with Stirling engine and absorption chiller. *Energy Convers Manage.* 2019;180:183-195.

37. Petela K, Manfrida G, Szlek A. Advantages of variable driving temperature in solar absorption chiller. *Renewable Energy.* 2017;114:716-724.

38. Green DW, Perry RH. *Perry's Chemical Engineers' Handbook* (no. C 660.28 P47 2008.).

How to cite this article: Ul Haq SE, Uddin F, Ammar Taqvi SA, Naqvi M, Naqvi SR. Multistage carbon dioxide compressor efficiency enhancement using waste heat powered absorption chillers. *Energy Sci Eng.* 2021;00:1–12. https://doi.org/10.1002/ese3.898