Turbomachinery group for cold energy concept application

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Abstract. The paper discusses a special attempt to improve COP and compressor capacity. Theoretical and experimental results are presented, and they show a very positive trend. Compressor and Expander Group impellers are selected among the Car Engine Turbocharger Units. Modifications are proposed to achieve the best performance, taking the peculiar characteristics of the refrigerant into consideration. Tests have confirmed the positive saving trend, which can reach up to 22%-24% for an optimized bleed vapour generator and turbomachinery-based booster unit.

1. Introduction

Energy saving, climate changes and environmental pollution call for a decrease in worldwide electricity consumption without reducing the quality level of life and industrial production. One of the products that is widely used is the cooling power produced by Vapour Compression Refrigeration Plants (VCRPs) powered by electricity.

Reducing electricity consumption as well as the related primary energy and pollutant emissions are important problems, because the electricity consumed worldwide by VCRPs is on the order of 15% - 17%, which contributes some 4% – 5% of global pollutant emissions. In industrialized countries, this consumption reaches 23% – 25%, and in warm countries, it rises to about 40% [1-3]. It can be easily understood that numerous attempts have been investigated and applied toward reducing the electric power consumed per unit of cooling power. These attempts are mainly based on thermal regeneration, which in vapour compression cycles leads to increasing the condensate subcooling through cycle internal processes. Attempts have also been made to reduce the main compressor pressure ratio by power regeneration by means of ejectors or positive displacement machinery [4,5]. Due to the low efficiency of these types of machinery, special attention has not been given to the field.

The Power Regenerated Cycle presented in [2] was investigated in the COLD ENERGY project. This cycle uses turbocompressor and turboexpander groups to increase the VCRP main compressor inlet pressure. The interesting cold energy results have shown improvements of up to 15% in COP for constant cooling power, which has stimulated the industrial production of such units for installation in VCRPs.

This paper deals with the analysis of the cycles and their optimisation to fulfil the minimum electricity consumption per unit of cooling power, which means the best use of the main compressor in terms of increased main compressor cooling power, together with the Coefficient of Performance (COP). The development of the components in pursuing the goal is discussed along with the main results. An industrial solution is presented.
2. Scientific and technical background
The scientific and technical aspects of this work regard the following:

- Improvements that can be achieved, in terms of cooling power and COP, by modifying a vapour compression refrigeration simple cycle (SC) as a reference, adding some devices;
- Optimum sizing of the innovative devices, using economically viable technologies;

2.1. Simple cycle as a reference
A simple cycle has been taken as a reference. Sketches of the plant and of the p-H Chart thermodynamic cycle are shown in Figure 1.

![Figure 1. Simple cycle arrangement](image)

Four main components characterize this type of plant: the Main Compressor (MC); the Condenser (K); the Main Expansion Valve (MEV) and the Main Evaporator (ME).

2.1.1. Component models. The SC model is established by a modular approach. Each module corresponds to one component. Details of the models can be found in [3, 6-12].

- The working fluid (WF) is one of the components whose properties are defined by a set of nonlinear equations (WFNLES):

  \[ F_{WF} (T, p, H, s, v, c_p, c_v, c_s, other) = 0 \] (1)

  There is one independent variable for boiling point and dew point lines and two independent variables for liquid and vapour-gas zones.

- Main compressors (MC) are usually utilized in semi-hermetic arrangement as shown in Figure 2a. Internal processes are rather complex: sketches are given in Figures 2b and 2c. The MC model is represented by a MCNLES that establishes correlations between variables:

  \[ F_{MC} (m_{MC}, p_1, T_1, p_2, T_2, \eta_{MC}, \eta_v, P_{MC}, q_d, \eta_{em}) = 0 \] (2)

  \( \eta_{MC}, \eta_v, q_d \) and \( \eta_{em} \) are expressed by empirical functions. \( V_d, n_0, n_N, P_N, P_{MX} \) are data associated with the compressor and electric motor. \( V_d \) can be replaced by pressure ratio and efficiency versus mass flow rate map curves for constant speed. \( T_1, p_1, p_2 \) are MC boundary conditions that are exchanged in the matching between components.

- The condenser is a three-zone heat exchanger as shown in Figure 3. The model establishes correlations among the variables by the KNLES:

  \[ F_k (m_k, p_2, T_2, p_3, p_4, p_5, \Delta p_{sh}, \Delta p_d, \Delta p_g, T_3, T_4, T_5) = 0 \] (3)

  \( m_k, T_2, p_2, \Delta p_d \) as well as the condensing temperature \( T_k \) are given. One of the constraints is:

  \[ T_3 - T_k = 0 \] (4)

- The main expansion valve ensures an adiabatic fully dissipative process. The model [6,8] is described by the MEVNLES correlations:

  \[ F_{MEV} (p_5, T_5, H_5, p_6, T_6, H_6) = 0 \] (5)
The evaporator is a two-zone heat exchanger as shown in Figure 4. The evaporator steady state model establishes EVNLES correlations between the variables involved in:

\[ F_{ev}(n_e, T_e, T_0, H_6, T_7, p_7, H_7, T_1, p_1, H_1, \Delta p_{67}, \Delta p_{71}) = 0 \]  

(6)

Figure 2. Main compressor model, p-H chart processes, torque vs speed of an induction motor

Figure 3. Condenser TQ chart

Figure 4. Evaporator TQ chart

2.1.2. The overall SC model. The SC is described by a set of independent nonlinear equations (SCNLES) that is the sum of the (1) - (6) NLES. The number of equations is equal to the number of unknown variables. \( T_k, T_{ev}, \Delta T_{sc}, \Delta T_{sh}, V_{sh}, n_{ev}, n_{sc}, P_{ev}, P_{MX} \) are the data. There is no DoF. The SCNLES also contains all the empirically defined relationships.

2.2. Power Regenerated VCRP

The most interesting Power Regenerated Cycles (PRCs) that have been investigated in the COLD ENERGY project are shown in Figure 5. Power regeneration is based on internal power cycles (IPCs) that drive compressors to increase the pressure at the main compressor suction. Each IPC uses as working fluid a fraction of the condensate mass flow that is bled and reduced in pressure by the bleed valve (BV). The Bleed Mass Flow (BMF) vaporises, receiving the heat power from the condensate stream, which increases the subcooling in the bleed vapour generator (BVG). Figure 6 shows a diagram of the BVG processes. As shown in Figure 5, the superheated bleed stream (SBS) is expanded in a turbine that drives a compressor where the stream from the main evaporator is compressed before moving towards the main compressor inlet together with the bleed expander exhaust stream.

The model of each Power Regeneration Internal Cycle (PRIC) (index j) is made from \( j \)th IPCNLES that can be added to the SCNLES and acquiring \( s_{BVG}, p_{BVG} \) and \( \mu_{I_0} \) as degrees of freedom. The domain of definition of variables and the degree of freedom are bounded by the Approach and Pinch Point Temperature Differences (DTA and DTPP) in the \( j \)th BVG and by the data for each compressor expander group, diagrammed in Figure 7. The BVG model establishes a BVGNLES:
\[ F_{\text{BVG}}(\mu_j, H_j, T_j, T_{b,j}, H_{b,j}, P_{b,j}, T_{h,b,j}, P_{h,b,j}, DTA_j, DTTP_j, u_{j}, \eta_j, \sigma_{cs}, \sigma_{sh}, S_{BVG_j}, \text{Architecture, Geometry}) = 0 \]  
\( \mu_j \) and \( \mu_{b,j} \) being the fractions of the overall condensate mass flow that gives the heat power being subcooled and the fraction of the bleed cold stream receiving the heat power. \( \sigma_{cs} \) and \( \sigma_{sh} \) are the fractions of the overall BGV heat transfer surface area \( S_{BVG_j} \).

Figure 5. Layouts and p-H chart cycles for three options

Figure 6. Bleed vapour generator TQ chart

Figure 7. Compressor and expander processes

An additional PRIC submodel is for the Compressor Expander Group (CEG). A sketch of the CEG and TS chart processes is given in Figure 7. CEGNLES establishes the following correlations

\[ F_{\text{CEG}}(m_j, m_{b,j}, T_{c,j}, p_{c,j}, H_{c,j}, p_{c1}, T_{c2}, H_{c2}, T_{c3}, H_{c3}, p_{c3}, T_{c4}, H_{c4}, p_{c4}, T_{c5}, H_{c5}, p_{c5}, T_{c6}, H_{c6}) = 0 \]  
\( \eta_{ce}, \eta_{cs} \), and \( \eta_m \) being specified as data or functions.

2.2.1. The overall PRVCRC. The overall Power Regenerated Vapour Compression Refrigerated Cycle (PRVCRC) is represented by an Overall NLES (ONLES) which consists of the sum

\[ \text{ONLES} = \text{SCNLES} \cup \text{BVGNLES} \cup \text{CEGNLES} \]

\( \text{BVGNLES} \) and \( \text{CEGNLES} \) being the sum of all the \( \text{BVGNLES}(j) \) and the \( \text{CEGNLES}(j) \). The degrees of freedom for each \( j \)-th PRIC are \( S_{BVG_j}, p_{b,j} \) and \( m_{b,j} \). If these \( 3 \cdot N_b \), DoFs are inside the
feasibility domain, a solution can be calculated. Of course, an optimization process can be established:

\[ \text{Search } s_{\text{BVGj}}, \overline{p}_{bj} \text{ and } \overline{m}_{bj} \quad \forall \ j = 1, N_b \]

\[ \text{maximize } \text{COP} = \text{CP} / P_{MC} \quad (10) \]

subject to the ONLES.

Some inequality constraints are restrictions on temperatures and pressures.

Once the compressor and expander impellers have been selected, it is necessary to provide the inlet ducts, the compressor diffuser, the expander nozzle and the expander exhaust diffuser. These subcomponents must be sized and the impeller profiles must be modified to achieve the best performance.

2.2.2. Turbomachinery Selection. The Turbomachinery Selection (TS) model is based on similitude concepts using the Car Engine Turbocharger Technology (CETT) Data Base. Details are given in [3]. This model leads to a TSNLES.

\[ F_{TS}(n, m, T_{i1}, p_{i1}, H_{c1}, H_{e1}, \eta_{e1}, \omega_{e1}, D_{e1}, m_{e1}, T_{e1}, p_{e1}, p_{e4}, D_{e4}, \eta_{e4}, \omega_{e4}, u, C_{pjo}) = 0 \quad (11) \]

This additional model introduces the compressor specific angular speed \( \omega_{e1} \) as one more degree of freedom eliminating \( \eta_{e1} \) and \( \eta_{e4} \) as inputs. Compressor inducer tip speed is bounded by the choking. The modified ONLES problem (equation (10)) that also contains \( F_{TS} \) leads to the optimizing of the PRVCRC performance with the more suitable turbomachinery impellers being selected.

2.2.3. Turbomachinery Sizing. Once the nominal speed, the compressor exducer and expander inducer dimensions have been selected together with the impeller types, the hub and shroud profiles are known. The problem is that of modifying the shroud profiles of the compressor and expander to achieve the best performance, minimizing the losses.

2.2.3.1. Turbomachinery 1D MLM. The shroud profiling of the compressor and expander CETT selected impellers is performed using a 1D Mean Line Modelling approach. The compressor and expander through flow sections are shown in Figure 8. The 1D ML is based on the use of channel sections that are quasi-orthogonal to the channel centreline, see Figure 9. Using the mass and rothalpy conservation, it is possible to establish a set of NLES for any impeller that, for each radius \( r \), correlates:

\[ F_i(m, r, h, p, W, b \Delta h_i) = 0 \quad (12) \]

If \( r \) and \( b(r) \) are given, the \( h \), \( p \) and \( W \) can be found. Due to complex flow in the channels (an example is given in Figure 10), entropy production is associated with the following losses: distributed friction along the channels; incidence; loading; shock; clearance; mixing; recirculation; windage; leakages; diffuser and collector. Disc windage loss can play an important role when the impeller input power is not high. Loss \( \sum \Delta h_l \) can be calculated as an objective function, thus the following problem can be stated:

\[ \text{Search } \overline{b}(r_j) \quad \forall \ j = 1, N_r \]

\[ \text{minimize } \overline{f}_{ob} = \sum \Delta h_l \quad (13) \]

subject to TSNLES.

The solution of this problem provides the best \( W(r) \), \( p(r) \) and \( h(r) \) distributions along the channel centreline and to establish the shroud profile:

\[ \text{SP}(r, \varepsilon_j) = 0 \quad (14) \]

then the analysis problem can be solved.
2.2.3.2. Compressor Model Characteristics. Modelling the centrifugal compressor impeller flow is a complex task; Figure 10 shows a typical picture. As concerns the exit triangle of velocity, the deviation connected with the shift factor has to be taken into account. The flow blockage has to take the recirculation (wake) zone into consideration.

2.2.3.3. Expander Characteristics. The incidence is the difference between the angle of the flow approaching the rotor channel inlet and the optimal flow angle associated with the impeller and flow field features. The deviation of the flow in respect to the direction of the channel axis must be calculated taking the channel characteristics and the flow features into account. High Mach number and post expansion conditions produce a substantial flow turning. The nozzle-rotor interspace size also plays an important role in the flow direction and losses.

2.2.3.4. Turbomachinery sizing outcomes. Application of the 1D MLM provides the profiles of the compressor and expander impellers as shown in Figure 11. A typical compressor off-design analysis result is given in Figure 12.
3. Theoretical and experimental results

The PRVCRC has been optimized taking into consideration the use of a $V_d = 3 \text{ dm}^3$/turn screw compressor running at a synchronous speed of 3000 rpm. For 2B1G layout with R404a WF having $T_i = +40^\circ\text{C}$ and $T_{ev} = -40^\circ\text{C}$, $\Delta T_{mc} = 5^\circ\text{C}$ and $\Delta T_{sh} = 5^\circ\text{C}$. Optimum performance versus $p_{b1}$ is shown in Figure 13, which for constant Cooling Power (CP) shows how the main compressor power decreases, achieving a minimum of about 650–700 kPa bleed pressure. For $V_d \cdot n_o = \text{const}$ (i.e. for the specified main compressor) it can be remarked that by increasing the 1st bleed pressure, COP and CP rise until they reach the maximum value, close to 650–700 kPa. The optimum compressor pressure increase is about $DPC = 80 \text{ kPa}$, thus the MC suction pressure becomes about 210 kPa, while for the SC it is about 133 kPa. MC pressure ratios are 8.7 and 14 respectively for the optimum PRVCRC and for the SC. For the optimum PRVCRC, the selected CEG speed is about 4 krad/s and impeller tip velocities are 143 m/s and 137 m/s respectively for compressor and expander. The selected shroud impellers have been profiled for the R404a optimum CEG compression and expander processes. Figure 14 shows the machined compressor wheel and Figure 15 shows the original and the machined expander impellers.

![Figure 11. Modified impeller shroud profiles](image1)

![Figure 12. Simulated pressure and temperature distributions from the compressor inlet to the exit](image2)

![Figure 13. Optimum CP, COP, $m_{MC}$ and $m_{ex}$ versus the 1st bleed pressure](image3)

![Figure 14. Two views of the machined compressor impeller](image4)

![Figure 15. Original and machined expander impellers](image5)
Figure 16 (a) gives the volute and nozzle arrangements. Figures 16 (b) and (c) show the variable geometry mechanism. The compressor diffuser has been selected as a vaneless one, owing to the de-rated working conditions that could occur. The outer diffuser diameter is 2.8 times larger than the impeller exducer diameter.

To verify the Power Regeneration Concept by means of IPCs, a Test Bench (TB) for the 2B1G scheme shown in Figure 5 was designed and built. Figure 17 shows the TB built and the CEG inside the hermetic container. The TB consists of four essential elements:

1. Condenser unit including the controlled the condensing water loop;
2. Calorimeters section, in which cooling power produced by the plant is measured, also called the cooling load system;
3. Compressor-expander group (CEG);
4. Cold liquid R404a piping for hybrid ball bearing lubrication purposes.

The TB is provided with special software that ensures the automatic control, measured data acquisition and storage of all parameters related to the overall operation of the refrigerating system, and more specifically of compressor-expander group.

The CEG was installed in this test bench, and some results are summarized in following. Tests have been carried out for different expander nozzle openings using the bypass valve (BPV) to adapt the expander flow rate to what is needed. Figure 18 shows the CEG rotational speed and Compressor Pressure Difference (CPD) versus time (normalised) elapsed during the test. Up to about 4000 times the tests were carried out with no condenser bypass flow rate; the deficiency of the bleed vapour mass flow rate does not allow running over 25 krpm, and consequently the pressure increase through the compressor was not higher than 20 kPa. After the normalised 4000th time the condenser bypass valve was opened, increasing the mass flow rate through the expander and reducing it through the compressor. Thus, the CEG reached an equilibrium at a higher rotational speed. The lack of the bleed vapour is connected to the insufficient heat transfer in the BVG; work is now being done to fix this problem.

Figure 19 gives the expander pressure difference, the compressor pressure difference and the CEG speed versus expander mass flow rate; the measured points and tendency lines are shown. According to theory, Expander Pressure Difference (EPD) and krpm vs Expander Mass Flow (EMF) seem to have a linear behaviour, while for the CPD vs EMF, the figure refers to 40% of the nozzle opening, which corresponds to the design condition for this type of main compressor. The corrected mass flow shows a linear behaviour.

As shown in Figure 20, the EPD vs EMF slope increases for low nozzle openings, and EPD greater than 500kPa is obtained for large mass flow rates that can be obtained with BVG having a large Number of heat Transfer Units (NTU), i.e. high effectiveness. The corresponding rpm are in the range of 38-40 krpm with about 90 kPa of compressor pressure increase, with the pressure ratio being around 1.7.
Figure 17. Cold energy test bench photo

Figure 18. CPD and krpm versus normalised time for 40% nozzle opening

Figure 19. Expander and compressor pressure differences and CEG rotational speed versus expander mass flow rate

Figure 20. EPD versus Expander mass flow rate for constant nozzle openings

Figure 21 (a) & (b). Expander pressure difference (a) and power saving (b) versus CEG speed
Figure 21 (a) shows the expander and compressor pressure differences versus rpm. The CEG without expander flow is pushed by the main compressor suction action. Increasing the expander mass flow, the velocity increases up to 13 krpm, with no compressor positive pressure difference. For higher rpm, both compressor and expander pressure differences increase up to the design conditions, indicated by yellow triangles.

The analysis of power saving, using 100 kW cooling power as a reference, is given in Figure 21 (b). The measured power saving in respect to the simple cycle is about 12%-14% with the plant running outside of the optimized conditions and with no bypass flow rate. The predicted power saving, using the measured data to deduce the results under optimized conditions corresponding to about 80 kPa of compressor pressure increase, is about 22%-24% and higher than expected by the theoretical analysis.

4. Conclusions

The utilization of an internal power cycle for power regeneration purposes brings about a lowering of the power consumption per unit of cooling power produced by a VCRP. It has been demonstrated that the cycle and components can be optimized up to achieve about 22%-24 % of power saving, with the reference being a simple cycle.

Turbomachinery for the compressor expander group can be selected from the units available in the Car Engine Turbocharger Technology. Modifications to the impeller profiles must be taken into consideration. Tests have confirmed the theoretical approach.

The results obtained by the COLD ENERGY project provided the motivation to continue both in performance improvement and product industrialization. An industrial application is now being investigated and some initial forecasts have been studied and will be demonstrated in dedicated industrial VCRPs.

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**Nomenclature**

**Symbols**

| Symbol | Description |
|--------|-------------|
| b      | Channel height |
| cs     | Sound Speed   |
| $c_p$, $c_v$ | Specific Heat Capacity |
| $C_{spo}$ | Spouting Velocity |
| D      | Diameter      |
| F      | Function      |
| H, h   | Enthalpy      |
| m      | Mass Flow Rate|
| n      | Rotational Speed (rpm) |
| $n_o$  | Synchronous Speed |
| N_r    | Number of Radial Nodes |
| P      | Power         |
| p      | Pressure      |
| Q      | Heat          |
| r      | Radius        |
| s      | Entropy       |
| S      | Surface       |
| T      | Temperature   |
| u      | Heat Transfer Coefficient |
| v      | Specific Volume|
| $V_d$  | Delivery Volume |
| W      | Work          |
| z      | Cartesian Coordinate |
| $\Delta$ | Difference   |
| $\eta$ | Efficiency   |
| $\mu$  | Mass Flow Fraction|
| $\sigma$ | Heat Transfer Surface Fraction |
| $\omega$ | Angular Speed |

**Subscripts**

| Symbol | Description |
|--------|-------------|
| b      | blade, bleed |
| c      | compressor, cold |
| d      | dispersed, lost |
| e      | expander |
| em     | electric motor |
| ev     | evaporator |
| h      | hot |
| i, j, k | order number |
| k      | condenser |
| m      | mechanical |
| MC, mc | main compressor |
| ml     | mechanical loss |
| MX     | maximum |
| N      | nominal |
| s      | specific, isentropic, shroud |
| sc     | simple cycle |
| sh     | superheated |
| v      | volumetric |