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The Mathematical Method and Thermodynamic Approaches to Design Multi-Component Refrigeration used in Cryogenic Process Part II: Optimal Arrangement

M. Mafi[1*], B. Ghorbani[2] , G. R. Salehi[3], M. Amidpour[4] and S. M. Mousavi Nayenian[5]

[1] Department of Mechanical Engineering, Imam Khomeini International University, Qazvin, Iran
[2,3,4,5] Energy System Engineering Department, K. N. Toosi University of Technology, Iran

Abstract: A refrigeration cycle is a chain of various pieces of equipment including compressors, condensers, evaporators, and expansion devices each of which takes on a particular thermodynamic process duty in the cycle that eventually results in the production of the required refrigeration. The results of optimized mixed refrigerant cycles (MRCs) in Part I show that the configuration of MRC is an effective parameter in power consumption. In spite of simplicity of MRCs machinery configuration in comparison with a conventional cascade cycle, it is possible to imagine different configurations, providing the process required refrigeration, for these cycles. But the question here is how to find a configuration taking advantage of the existing complicated interactions between low temperature process and refrigeration cycle so that the maximum efficiency and the best functionality will be guaranteed. In this paper, based on the success of the systematic method proposed in part I for the optimal selection of refrigerant composition and operating pressures, the method is extended to give optimal arrangement of the cycle components. An illustrative example is presented to show how the proposed method can be utilized in order to achieve the optimal MRC configuration.

Keywords: Mixed Refrigerant Cycle, Low Temperature Process, Optimal Arrangement

1. Theoretical Background

In part I of this work, two MRCs were developed for a typical olefin plant utilizing a mixture of methane, ethane, propane and nitrogen as cycle working fluid to replace the pure ethylene refrigeration cycle (Ghorbani, Mafi, Amidpour, Mousavi Nayenian, & Salehi, 2013). Also, a systematic design method to meet the objective of minimum shaftwork in the compressor was suggested to optimize the major parameters of each MRC including: high

[*] Corresponding Author.

Authors' Email Address:
[1] **Mostafa Mafi** (m.mafi@eng.ikiu.ac.ir), [2]**Bahram Ghorbani** (bahram330ghorbani@gmail.com),[3] **Gholam Reza Salehi** (salehi_kntu@yahoo.com), [4] **Majid Amidpour** (amidpour@kntu.ac.ir), [5] **Seyed Mojtaba Mousavi Nayenian** (mousavi@kntu.ac.ir)

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and low operating pressures and refrigerant mixture composition. The strength of suggested method is that it combines mathematical programming and thermodynamics methods. This characteristic of the proposed method can generate optimal design solutions and good understanding of design problems. While mathematical programming produces optimal solution for the cycle, thermodynamics programming draws the composite curves of the MRC processes, and therefore expresses the evaluation of the solution procedures in a visual way so that the designer will have understanding and confidence in the solution.

For instance, Figures 1 and 2 illustrate the Grand Composite Curves (GCC) for the above mentioned MRCs after optimizing the key parameters using enumerative method. Dashed lines in these figures represent condensation of MR in the cooling water and liquid propylene refrigerant condensers. It can be found out from these figures that the hot and cold composite curves in configuration B are matched better than those in configuration A.

The main reason for the differences between GCC of MRCs with various configurations lies in diverse possible configuration and arrangements of these cycles and their dedicated equipment. In other words, it can be stated that having optimized the design variables of MRCs, their GCCs are suitable criteria for comparing various configurations of these cycles. These curves can be regarded as a qualitative criterion to measure the cycles deviation from the desired condition (close matching between hot and cold composite curves).

The composite curves can be redrawn by replacing the temperature with Carnot factor ($\eta = 1 - [T_0 / T]$), resulting in the ECC (Exergy Composite Curves) as shown in Figures 3 and 4.

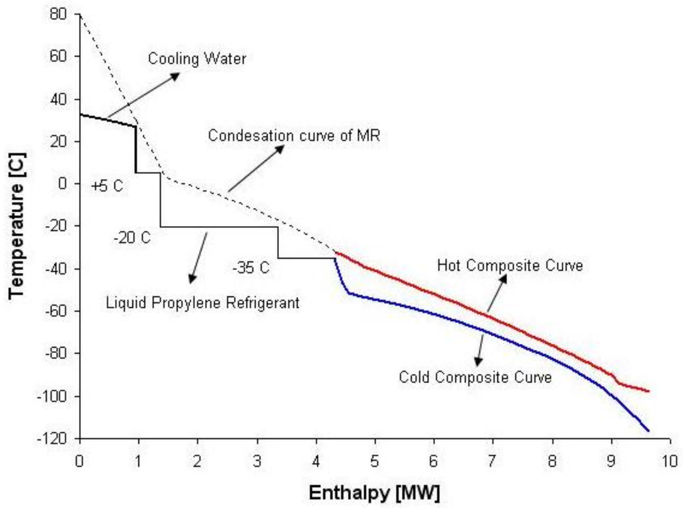

Figure 1. The GCC of configuration A mixed refrigerant cycle

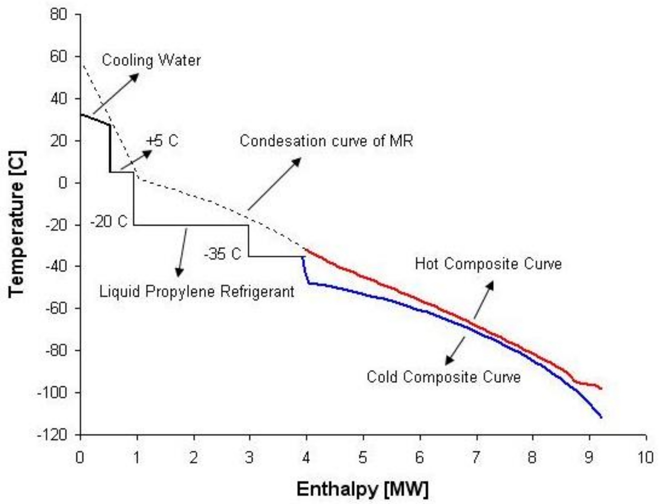

Figure 2. The GCC of configuration B mixed refrigerant cycle

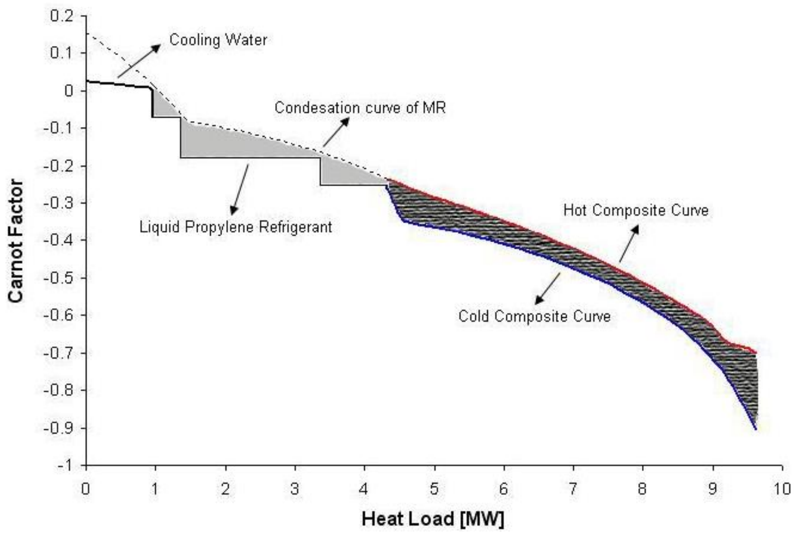

Figure 3. Exergy composite curves of configuration A mixed refrigerant cycle

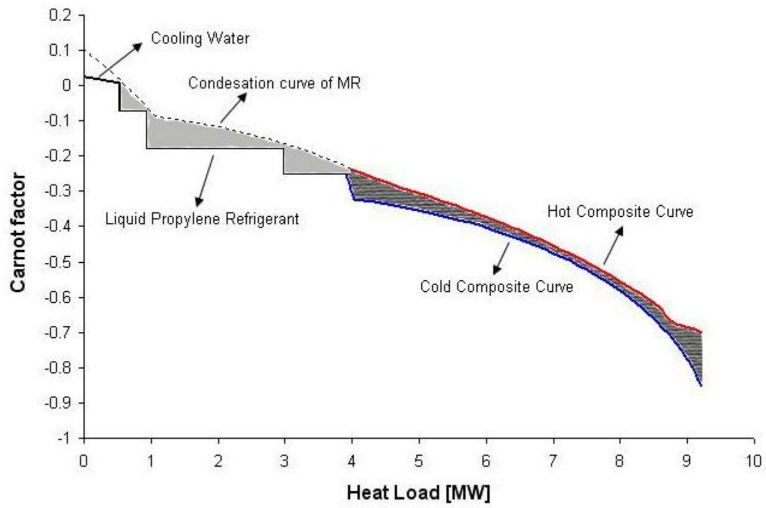

Figure 4. Exergy composite curves of configuration B mixed refrigerant cycle

The area between the curves in these diagrams represents the exergy loss in the utility exchangers (Linnhoff & Dhole, 1992) . The question, here, is whether the changes in the area enclosed in ECC given for MRCs due to change in the cycle arrangement equal the change in power consumption of cycles.

In response to the above question, consider the Tables 1 and 2. The data obtained in these tables compare the difference between the values of enclosed area between MR condensation and liquid propylene refrigerant curves (light grey colored region) with the difference between power consumption of propylene cycle (precooling cycle) for configurations A and B optimized using Enumerative method. Also, the obtained data present the difference between the values of enclosed area between hot and cold composite curves (dark grey colored region) in Figures 3 and 4 with the difference between power consumption of MRCs.

Table 1. Comparison of the difference between the area of light grey colored regions in figures 1 and 2 with the difference between power consumption of propylene cycles

Row No.	MRC configuration	Power consumption of Propylene cycle compressor (kW)	Area of light grey colored region (kW)
1	Configuration A	4089	1061
2	Configuration B	4118	1093
Difference between the values of rows No. 1 & 2 (kW)		29	32

Table 2. Comparison of the difference between the area of dark grey colored regions in figures 1 and 2 with the difference between power consumption of MRCs

Row No.	MRC configuration	Power consumption of MRC compressor (kW)	Area of dark grey colored region (kW)
1	Configuration A	1876	5044
2	Configuration B	1528	4841
Difference between the values of rows No. 1 & 2 (kW)		203	348

From the data analysis of Tables 1 and 2, it is evident that values of the difference between power consumptions and the one between the area of enclosed regions for configurations A and B are close to each other. This difference can be minimized computing the exergy losses in throttling valves and compressors along with adding them to exergy loss in heat exchangers (the area of enclosed region). Therefore, ECC can be used as a tool to establish proportionality between the changes in the configuration of MRC and the change in power consumption of the cycle.

In low temperature processes with pure refrigerant cycle, ECC can be used as a shaftwork targeting tool to directly estimate the change in power consumption of the cycle (owing to modifications and improvements in the cycle configuration) without going through detailed refrigeration calculations (Linnhoff & Dhole, 1992). In other words, ECC can assist the designer in finding the best refrigeration system configuration.

Unlike pure refrigerant cycle, graphical targeting approaches such as ECC and GCC cannot be used directly to optimize the MRC configuration; because the optimization requires adjustments to the refrigerant composition. But an MRC features a simpler machinery configuration in comparison with a pure refrigerant multiple-stage cycle. Therefore, ECC and GCC charts can be used as a graphical tool for feeling and understanding MRC behavior. These curves tend to provide “suggestive elements” for improving the cycle configuration.

Let us consider the ECC charts of the MRCs developed in part I. As it can be seen from Figures 3 and 4, the gap between the hot and cold composite curves for configuration A is larger in the temperature range of -60°C to -90°C in comparison with configuration B. It is caused by passing MR immediately after expansion valve into the subcooler of configuration A that imposes a great temperature difference along with thermal exchange consequently leading to a rise in exergy loss (Figure 3 of Part I). Also, configuration B cycle has been equipped to multistream heat exchangers which result in a better matching between the hot and cold composite curves (Figure 4 of Part II).

2. Proposed Method for Optimizing MRC Configuration

The objective of this paper is to present a methodology for finding the optimal MRC configuration for providing refrigeration in a certain low temperature process. Due to the power and capabilities of GCC and ECC charts in indicating room for further improvement in MRC process, the systematic design method proposed for optimal selection of refrigerant composition and operating pressures in part I has been extended to cover the cycle’s configuration optimization.

Figure 5 depicts the methodology. It comprises two main parts: Basic design phase and Detail design phase. The procedure commences from an initial guess for MRC configuration.

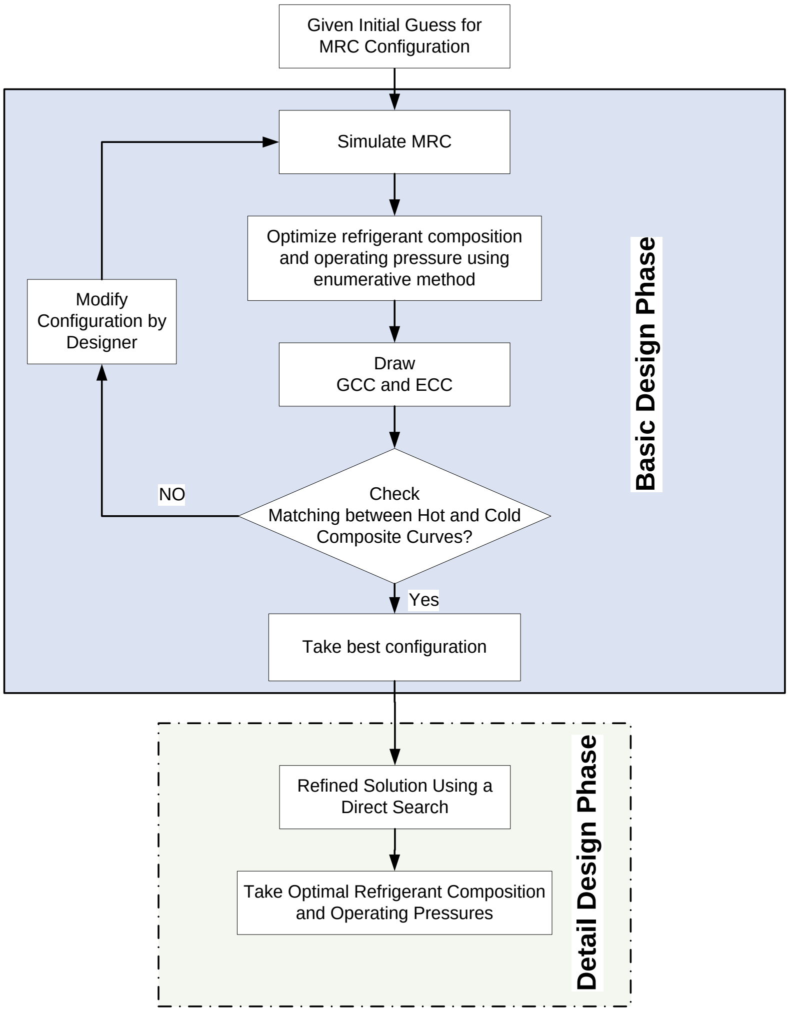

Figure 5. Proposed design method to give optimal arrangement of MRC

In Basic design phase, by employing the Enumerative method, the best refrigerant composition and operating pressures of the cycle are found under conditions imposed by low temperature process. Drawing the GCC and ECC charts is the next step. By considering the match between hot and cold composite curves of initial guess, room for further improvement is indicated. Then, the designer can modify the cycle configuration based on understanding MRC behavior, heuristics and judgment. The procedure stops when no further improvement is possible.

In Detail design phase, the MRC found by the Basic phase is fed directly to the search stage to explore further possibility of reducing the power consumption. To cut down the computational time of the explorative stage in the basic design phase, the search space firstly is specified using a rough step, and then in the next iterations, by being able to guess the cycle behavior as well as restricting the domain of search space, the discrete steps can be postulated. An illustrative example of how to utilize the proposed method to achieve the optimal MRC configuration has been presented in the next section.

3. Case Study

In part I, two MRCs have been designed on the basis of the characteristics of the olefin plant cryogenic section in place of pure ethylene refrigeration. By considering the ECC and GCC charts of these cycles, it is concluded that using multistream heat exchangers in the cycle configuration will lead to better matching between hot and cold composite curves, resulting in a lower power consumption of the cycle. Indeed, by using suitable multistream heat exchangers, the distribution pattern of hot and cold streams will be then out of the designer's scope of work, and the burden will be on the optimization algorithm of the systematic design method. Knowing this idea significantly aids the cycle configuration optimization. For an example, in MRC with configuration B, the designer has proposed a certain arrangement for heat exchangers that imposes distribution plan of hot and cold streams. Consequently, the systematic design method will be able only to optimize the refrigerant composition and operating pressures. To overcome this problem, the configuration depicted in Figure 6 has been suggested. It can obtain a better match between hot and cold streams.

The results show that by altering the cycle configuration to take a better advantage of heat integration between MRC and process streams in multistream heat exchanger, a drastic reduction in cycle power consumption (239 kW) has occurred.

Figure 7 illustrates the ECC chart of configuration C mixed refrigerant cycle. By comparing the ECC in configurations B and C (Figures 7 and 4), a considerable improvement between the hot and cold composite curves can be observed in configuration C.

Figure 7 indicates that although the proposed design method has caused the hot and cold streams in heat exchanger to close each other, there is still considerable temperature difference between hot and cold streams around pinch point at the heat exchanger network. In order to cut down the temperature difference along the heat exchanger, it is possible to distribute the thermal exchange existing between cold and hot streams among several cascade heat exchangers.

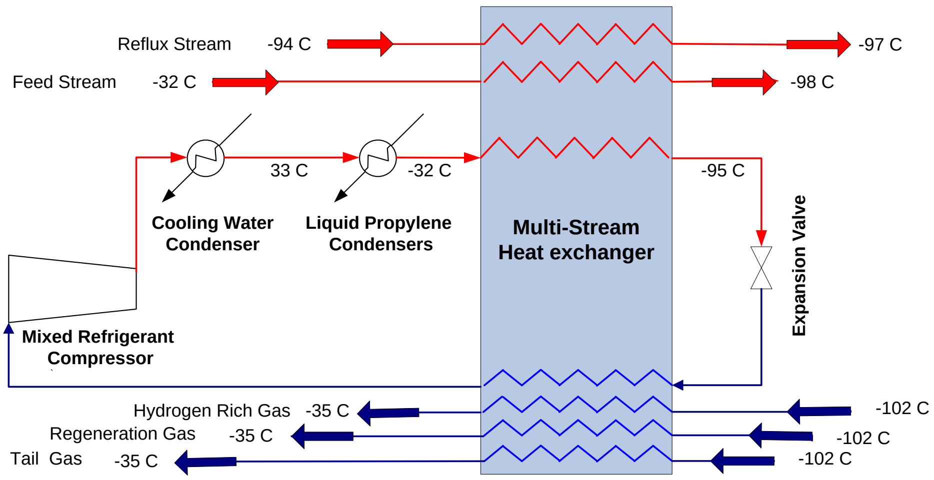

Figure 6. A mixed refrigerant cycle suggested to obtain a better match between hot and cold streams in multistream heat exchanger (Configuration C)

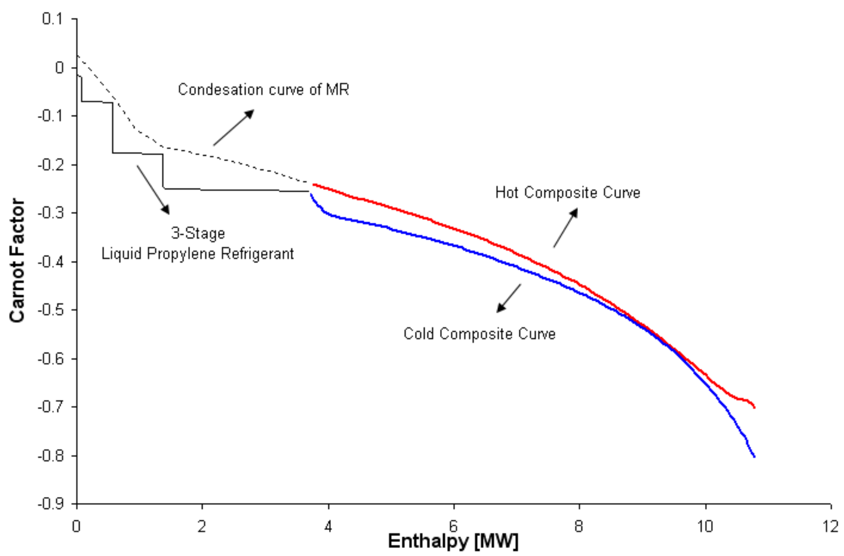

Figure 7. The ECC of configuration C mixed refrigerant cycle

Figure 8 illustrates a two-stage MRC proposed in current work instead of a single-stage cycle (configuration C) in order to decrease the power consumption. In this cycle, the amount of refrigerant flow in each heat exchanger is different, adding one more degree of freedom in the design of MR systems. This extra degree of freedom creates opportunities to achieve a more efficient design, but it also causes more complexities in the modeling of MR systems. The process conditions are set as they are in two MRCs developed in the previous section.

It is reasonable to assume that all outlet streams from the cold end multistream have equal temperatures (Lee, 2001). This temperature playing a significant role in refrigeration cycle is called the cycle intermediate temperature. Intermediate temperature can alter the shape of the hot and cold composite curves. Figure 9 shows the GCC chart for the MRC shown in Figure 8.

In this study, the change to the intermediate temperature has been done manually, following observations and heuristics. Tables 3 and 4 compare the results derived from optimization of parameters of two-stage refrigeration cycle with other cycles.

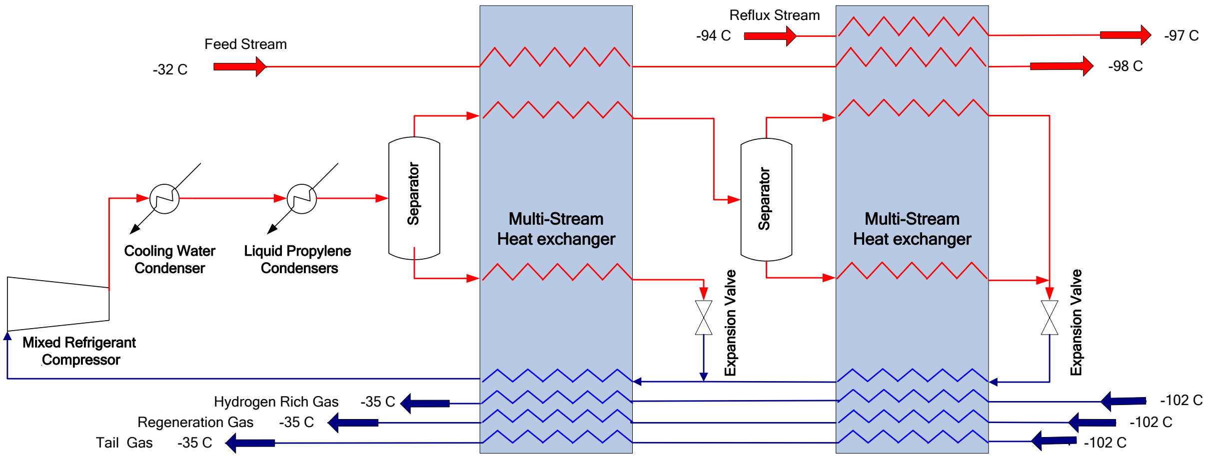

Figure 8. A two-stage MRC proposed in current work instead of a single-stage cycle (configuration C) in order to decrease the power consumption

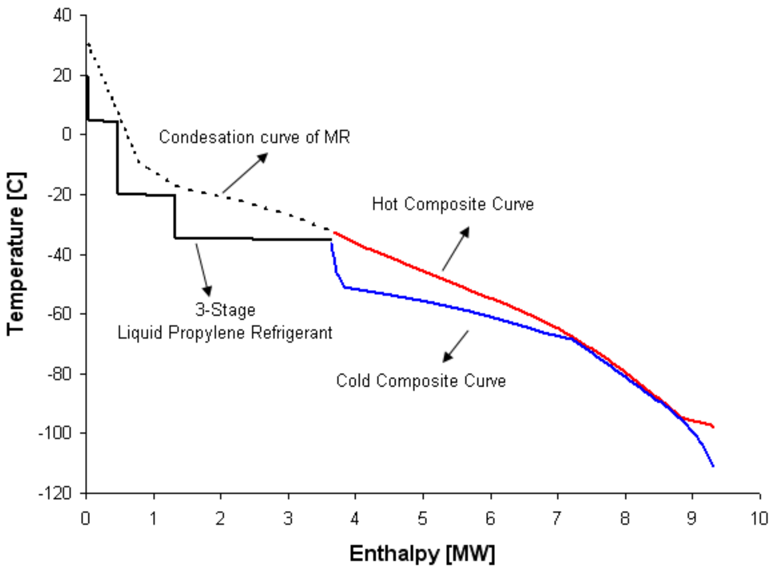

Figure 9. The GCC of two-stage MRC, intermediate temperature is equal to -65°C

Table 3. The key parameters of various optimized MRCs using enumerative method

Configuration	Operating Pressure (kPa)		Mixed Refrigerant Composition (mol%)			
	Pmin	Pmax	Propane	Ethane	Methane	Nitrogen
B	200	1250	36	34	29	1
C	260	800	30	45	24	1
two-stage	140	600	43	41	15	1

Table 4. Comparison between the compressors power consumption of various optimized MRCs using enumerative method

Configuration	Power consumption of MRC (kW)
B	1528
C	1239
two-stage	1230

It can be found from this table that in spite of the more complex two-stage refrigeration cycle in comparison with configuration C, there is no drastic reduction in cycle power consumption. Thus, it can be deduced that configuration C cycle is the best substitute for refrigeration cycle of the pure ethylene refrigerant in olefin plant analyzed in this study.

It should be noted that lower power consumption can be expected if the number of stages is increased, but inevitably results in greater complexity and difficulty in control. Moreover, the effect of reducing power consumption by increasing the number of stages is progressively diminishing.

4. Conclusion

In this paper, based on the success of the systematic method proposed in Part I for designing MRC with a given configuration, the method is extended to give optimal arrangement of the cycle components. The essence of the extended method is the proper combination of pinch and exergy analyses in a visual way. Thus, the causes of inefficiency in mixed refrigerant cycle configuration can be quickly identified. Based on the insights, the designer can confidently evolve a better design and introduce ideas for improving the cycle arrangement.

The solution that gives the lowest shaftwork requirement may incur extra large heat transfer area and thus capital costs. Combining the work presented in this paper with a design method for the synthesis of multistream heat exchanger and to give better guidelines for picking the most economic solution to minimize total cost may be considered at future work.

Nomenclature

MRC	mixed refrigerant cycle
ECC	exergy composite curve
GCC	grand composite curve
MR	mixed refrigerant
Pmin	low operating pressure of refrigeration cycle
Pmax	high operating pressure of refrigeration cycle

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