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Comparative Analysis of Heat Exchangers for High-Temperature High-Pressure Helium Component Testing

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INTRODUCTION

Helium gas loops have been designed and built to gain a better understanding of gas thermohydraulic phenomena that take place in a helium system. Some of these loops are used for validation and testing of components for high-temperature gas-cooled reactors (HTGRs). However, most of them operate at pressures and temperatures lower than prototypic HTGR conditions. While these loops can provide valuable information about gas-cooled reactor components, the operating envelope of the experiment is constrained by the maximum operating conditions of the helium loop.

In response to the lack of an experimental facility that can provide the infrastructure needed to validate and test components at nominal pressures and temperatures of HTGRS, the HElium Component Testing Out-of-pile Research (HECTOR) facility was designed at Idaho National Laboratory with the assistance of University of Idaho and Walsh Engineering. With capability to test at temperatures up to 800°C and pressures of 8 MPa, HECTOR serves as a critical tool for the advancement of HTGR technology. The facility's primary role is to provide a controlled, high-fidelity environment for the assessment of component resilience and efficiency under nominal HTGR conditions.

In the quest to enhance the efficiency and performance of HECTOR, a comparative analysis of three distinct types of heat exchangers—shell and tube, offset strip fin, and printed-circuit—was conducted, focusing primarily on two critical metrics: the required surface area and pressure-drop characteristics. The shell-and-tube heat exchanger, renowned for its robust design and widespread industrial application, was evaluated against the offset strip fin and the cutting-edge printed-circuit heat exchangers, both of which are lauded for their compactness and thermal effectiveness. This comparative study aims to provide detailed insights into the thermal management capabilities of each heat exchanger type under the conditions inherent to HECTOR, thereby facilitating an informed selection for systems demanding high operational integrity and efficiency.

DESIGN OVERVIEW

The HECTOR system was designed to operate at a maximum pressure of 8 MPa and a maximum operating temperature of 800°C. A three-dimensional (3D) computer-aided design (CAD) model of the system was created and is shown in Fig. 1. HECTOR's primary loop is a single closed loop, supplied with helium from a gas-supply system. Helium gas is circulated through the primary system using a booster compressor capable of flowing the gas at a mass-flow rate of 0.15 kg/s. The gas initially flows through the first of two recuperative heat exchangers into the first section of heaters. Following the heaters, the helium flows into the second heat exchanger to the final set of heaters to ramp to the maximum temperature of 800°C. At this point, the gas flows into the environmental chamber where the individual experiments will be held. The gas flows back through both recuperative heat exchangers into the final heat exchanger coupled to a chiller subloop to complete the system [1].

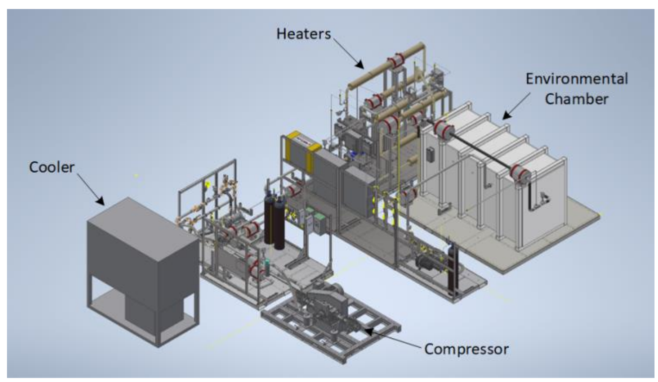

Fig. 1. 3D CAD of HECTOR facility.

RESULTS

The baseline for the comparative analysis was a shell-and-tube heat exchanger. A shell-and-tube design was considered for the baseline due to the robustness of the correlations available in literature. The design process for the HECTOR shell-and-tube analysis was the Delaware method, in which a series of correction factors are found to calculate the overall heat-transfer coefficient [2]. This was done for both the cold and hot side of each heat exchanger. The overall heat-transfer coefficient was then used to find the required heat-transfer area for the heat exchanger. This was done using the following equation:

$$Q = UA\Delta T_{LMTD} \quad (1)$$

The required heat transfer rate, Q, was defined by the system requirements. The system requirements also defined the log mean-temperature difference, ΔT_{LMTD}. Using the calculated overall heat-transfer coefficient, $\underline{U}$, the required heat-transfer area, A, was found. Once the required area was found, the geometry of the heat exchanger was changed to obtain a geometry that had a higher heat-transfer area. For the shell-and-tube design, this involved manipulating the shell-side diameter, tube diameter, number of tubes, and tube length. This process was repeated for each heat exchanger designed.

The working fluids and operating temperatures were consistent for each heat exchanger analyzed. The two recuperative heat exchangers use helium gas for both the hot and cold sides; however, the chiller heat exchanger uses helium for the hot side and a 50% glycol 50% water mixture for the cold side. The inlet and outlet temperatures were consistent for each design calculation. The mass-flow rates of the working fluids, inlet and outlet temperatures for the hot and cold sides as well as the thermal-load requirements can be found in Table I and Table II respectively.

Table I. Heat exchanger inlet and outlet temperatures.

	Hot Inlet (°C)	Hot Outlet (°C)	Cold Inlet (°C)	Cold Outlet (°C)	Thermal Load (kW)
Chiller Heat Exchanger	166	107	6.67	15	46.0
Recuperator #1	518	166	120	480	32.6
Recuperator #2	800	518	490	770	22.56

Table II. Heat exchanger working fluid and mass flow.

	Hot Side Fluid	Hot Side Mass Flowrate (kg/s)	Cold Side Fluid	Cold Side Mass Flowrate (kg/s)
Chiller Heat Exchanger	Helium	0.15	50/50 Glycol and Water	1.602
Recuperator #1	Helium	0.15	Helium	0.15
Recuperator #2	Helium	0.15	Helium	0.15

The pressure drops were calculated using both the Delaware method for the shell side and traditional friction-factor correlations for the tube side. The results from these calculations are found in Table III below.

Table III. Analysis of shell-and-tube heat exchangers.

	Required Heat Transfer Area (m^2)	Designed Heat Transfer Area (m^2)	Pressure Drop (kPa)
Chiller Heat Exchanger	13.65	13.84	61.56
Recuperator #1	29.43	29.92	134.67
Recuperator #2	26.00	26.18	86.31

The primary variables for increasing this area were to increase the number of tubes in the bundle, increase the tube diameter, as well as the length of the tubes. The results from manipulating these factors were tube bundles on the order of 150 individual tubes and a tube length of between 1.85 and 4 m. To allow the tube bundle to fit, the shell diameter also had to increase. These constraints were consistent for all three heat exchangers within the HECTOR loop. The conclusion drawn from this geometry is that shell and tubes serve as a good baseline for this heat exchanger analysis, but could have issues with manufacturability given the loop constraints.

The pressure drops found were low with reference to system pressure, approximately 3.5%. This is due, in part, to the size requirements of the heat exchangers. Due to their large cross-sectional areas, the Reynolds number calculated was in the laminar regime, resulting in small friction coefficients and, therefore, smaller pressure drops in the system.

The second heat-exchanger design analyzed was printed-circuit heat exchangers (PCHEs). These are prized for their robustness at high temperatures and pressures, as well as their large heat-transfer area-to-volume ratios. To complete this analysis, a known geometry for the plate design was selected to allow for use of known experimental correlations [3]. N. Bartel et al. found that varying the pitch angle on the channels of the PCHEs changed the required heat-transfer area and the resulting pressure drop. Because of this, the analysis was done on both ends of the experimental data, 10- and 20-degree pitch angles. It was found that the 10-degree pitch angle required more area, but results in smaller pressure drop while the 20-degree pitch angle was the opposite. The results from this analysis are found in Table IV.

Table IV. Analysis of printed-circuit heat exchangers.

	Required Heat Transfer Area (m^2)	Designed Heat Transfer Area (m^2)	Pressure Drop (kPa)
20 Degree Pitch Angle			
Chiller Heat Exchanger	2.23	2.42	10.25
Recuperator #1	4.06	4.44	100.44
Recuperator #2	4.21	4.22	161.82
10 Degree Pitch Angle			
Chiller Heat Exchanger	2.66	2.77	11.29
Recuperator #1	4.00	4.00	96.09
Recuperator #2	3.62	3.77	142.06

The results from the PCHE analysis show that the required heat-transfer area is much lower than that of the shell-and-tube heat exchanger for both the 10- and 20-degree pitch angles. The resulting pressure drops are similar to the shell and tube baseline case, still results in approximately 3.5% system-pressure drop across all three heat exchangers. Because the required area is much lower, the geometry is also smaller. The PCHEs are on the order of 1 m in length and 0.23 m in height.

The final heat-exchanger design to be analyzed was the offset-strip-fin design. This analysis followed a similar approach to that of the PCHEs. The correlations and design considerations were found by Manglik [4]. Manglik found correlations for both the required heat-transfer area as well as pressure drop for rectangular offset-strip-fin designs. The geometries for the heat exchangers were found in London and Kays [5]. The selected geometry was 1/8-20.06(D). This layout had a large heat-transfer area-to-volume ratio, which was the purpose for the selection. The results from this analysis can be found in Table V.

Table V. Analysis of offset-strip-fin heat exchangers.

	Required Heat Transfer Area (m^2)	Designed Heat Transfer Area (m^2)	Pressure Drop (kPa)
Chiller Heat Exchanger	11.34	11.36	1.92
Recuperator #1	44.70	45.44	0.396
Recuperator #2	35.17	35.19	1.79

The resulting required heat-transfer areas were between those of the PCHE and shell-and-tube designs while the pressure drops were much lower than the PCHE design. It is important to note that the primary variable to manipulate to reach the area requirement is the overall size of the heat exchangers. Manipulating this variable resulted in heat exchangers as large as 15.625 m^3 cubes. This result directly influenced the low pressure drops calculated. Because of the scale of these heat exchangers, Reynolds numbers show very low correlation of low fluid velocity to friction factors.

CONCLUSION

In conclusion, the comparative analysis of the shell and tube, printed-circuit, and offset-strip-fin heat exchangers has yielded valuable insights into the optimal application scenarios for each design. The shell-and-tube design served as a baseline, with its large required heat-transfer area being a significant drawback for applications where space efficiency is crucial and large heat-removal requirements exist. Despite this, its low pressure drop remains an advantage.

On the other hand, the offset-strip-fin heat exchanger, despite its relatively large required heat-transfer area, also maintains a low pressure drop, similar to the shell-and-tube heat exchanger. While this might seem advantageous, the excessive area requirement is still a limiting factor.

The printed-circuit heat exchanger emerged as the superior choice for the HECTOR facility, which demands high thermal efficiency within a constrained volume. Its minimal required heat-transfer area, in conjunction with normal pressure drop levels, offers a balanced solution. The PCHE's advanced design and manufacturing techniques allow for exceptional performance in high-temperature and high-pressure applications, making it the preferred option in this comparative analysis.

Ultimately, the choice of heat exchanger must be based on a combination of factors, including thermal efficiency, space constraints, pressure-drop considerations, and cost. The PCHE, with its optimal balance of a small required heat-transfer area and acceptable pressure-drop levels, stands out as the best option within the context of this analysis. However, it is essential to consider the specific operational demands and environmental conditions of each application to ensure that the selected heat exchanger is indeed the most suitable for the intended use. Further exploration into hybrid or improved designs could potentially offer better performance characteristics and cater to an even wider range of industrial applications.

NOMENCLATURE

Q = heat load
A = required heat transfer area
U = overall heat transfer area
T_{LMTD} = Log mean temperature difference
Δ = delta

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