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DECARBONISATION THROUGH INNOVATION

Development of a hydrocarbon cascade high-temperature heat pump for industrial decarbonization

N. Ahlbrink[a], F.D. Holdt[a], J.L. Poulsen[a], B. Zühlsdorf[a]

[a]Danish Technological Institute, Kongsvang Alle 29, 8000 Aarhus C, Denmark

Abstract

High-temperature heat pumps (HTHPs), capable of supplying heat above 100 °C, are emerging as a crucial technology for decarbonizing industrial heat by efficiently utilizing waste heat and clean electricity. These systems are increasingly recognized in climate policy roadmaps, with significant targets set for their deployment in the coming decades. HTHPs can be integrated at either the process or utility level, each offering trade-offs between efficiency, complexity, and ease of implementation. Despite clear environmental and economic benefits, market penetration of HTHPs at temperatures exceeding 100 °C remains limited due to current gaps in compressor technology among others.

Addressing this, the presented research describes the development and laboratory testing of a cascade HTHP system using isobutane and isopentane as refrigerants, designed for industrial applications requiring heat up to 170 °C. The system, with a heating capacity of approximately 500 kW and a coefficient of performance around 2.5, demonstrates the potential for high efficiency and operational flexibility through advanced integration and control strategies. Testing revealed necessary optimizations to control systems and heat exchanger configuration, ensuring stable, effective operation under varying laboratory conditions. Temperatures of up to 168°C have been demonstrated. The system will be transferred to an industrial site for real-world demonstration. This work underscores the promise of HTHPs for industrial energy transition, while highlighting ongoing challenges in system integration, control, and real-world deployment.

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Keywords: High temperature heat pump; hydrocarbons; cascade system;

1. Introduction

High-temperature heat pumps (HTHPs), capable of supplying heat above 100°C, offer efficient, low-emission solutions for industrial heating by recovering excess heat and utilizing clean electricity. With their high performance, HTHPs are expected to deliver optimal techno-economic results for a wide variety of industrial processes operating above 100°C.

Recognized as a crucial technology in the transition to sustainable industrial heating, HTHPs play a significant role in decarbonization strategies. For example, Denmark's Climate Council suggests that a quarter of the country's industrial CO2 reduction target should come from industrial heat pumps delivering 15 PJ/year—a demand echoed by the International Energy Agency's Net Zero by 2050 Roadmap, which calls for HTHPs to cover 15% of low/medium-temperature industrial heat demand by 2030 and 30% by 2050 [1].

HTHPs can be integrated either decentral into the production processes on process or unit level or placed centrally on utility or sector level. Process-level integration minimizes distribution losses and maximizes efficiency but requires advanced process integration, flexible control, and can face tough investment criteria. On a utility or sector level, while distribution losses are higher, advantages include load balancing, larger and more cost-effective equipment, and easier acceptance by industry due to minimal process modifications. Sector integration with district heating networks offers further opportunities for load optimization, industrial symbiosis, and local energy balancing.

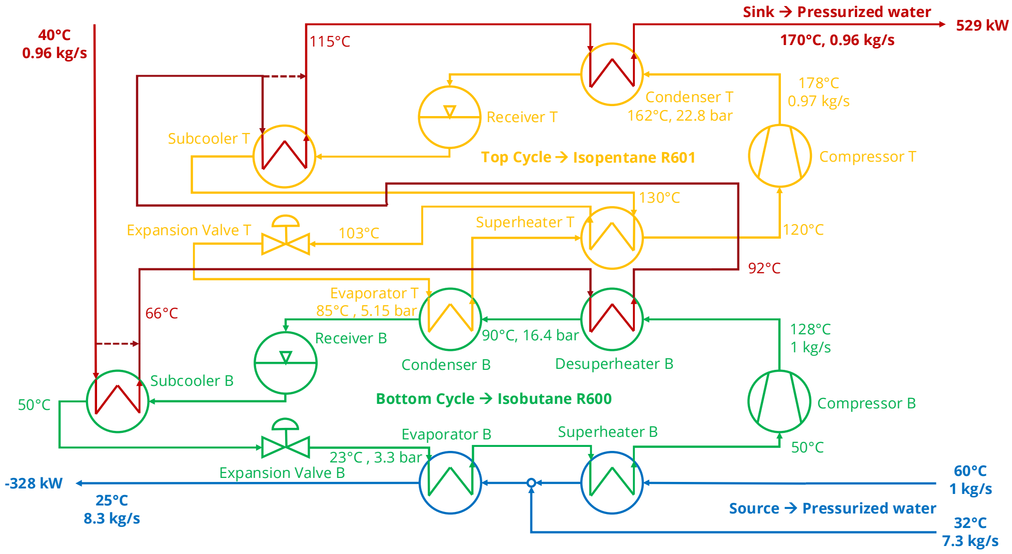

Figure 1: HTHP system - Main loops and expected operational data at design point

Widespread adoption necessitates diverse, competitive technical solutions and robust supply chains. Key to this is the development of standard equipment for various integration levels, business models to accelerate market uptake, and cross-sector collaboration.

Industrial heat pumps work by recovering heat from a low-temperature source and upgrading it with electricity—efficiency described by the Coefficient of Performance (COP), which is higher for smaller temperature lifts. While most existing systems serve applications with moderate temperature lifts (particularly ammonia systems below 90°C), factors such as declining electricity-to-fuel-price ratios, increasing environmental awareness, and the drive for greater industrial energy efficiency are making HTHPs attractive even at higher temperatures and/or higher temperature lifts.

However, the current market lacks sufficient technologies for supply temperatures above 100°C, mainly due to limitations in compressor technology. Three development paths are being pursued:

- Process industry compressors, suitable for large capacities and most competitive for large capacities due to economy of scale.
- Modified refrigeration compressors, which offer the best prospects for high-performance, cost-effective systems produced in large volumes (50 kW to 15 MW range).
- Novel or adapted compressors (e.g., automotive, spindle compressors), which have promising potential for high efficiency and cost-effectiveness but are mostly in prototype stages.

For successful HTHP deployment, interdisciplinary development, demonstration, and sector integration are essential. This paper addresses two of these needs. It presents the development of a HTHP system (by Multi Køl & Energi) using a cascade configuration using butane and pentane with semi-hermetic compressors from Frascold, delivering approximately 500 kW heating capacity. Testing and optimization of the HTHP system in laboratory settings in combination with lessons learned are described. These recent findings are part of an R&D project named "Demonstrating high-temperature heat pumps at different integration levels" led by the Danish Technological Institute (DTI) and is funded in part by the Danish Energy Agency.

2. System Description

The HTHP system presented in this paper is a cascade high-temperature heat pump utilizing hydrocarbons as refrigerants. The bottom cycle employs isobutane (R600a), while the top cycle uses isopentane (R601a). Two pressurized water loops serve as the heat source and heat sink, respectively. Key temperatures, pressures, and flow rates are provided in Fig. 1, while photographs and schematic diagrams of the system are presented in Fig. 2.

The system is arranged as a cascade heat pump incorporating plate heat exchangers. On the low-pressure side, waste heat is captured from water streams at approximately 30°C to 40°C (evaporator) and 50°C to 60°C (superheater) and transferred to isobutane in the bottom cycle. The isobutane vapor is then compressed by a

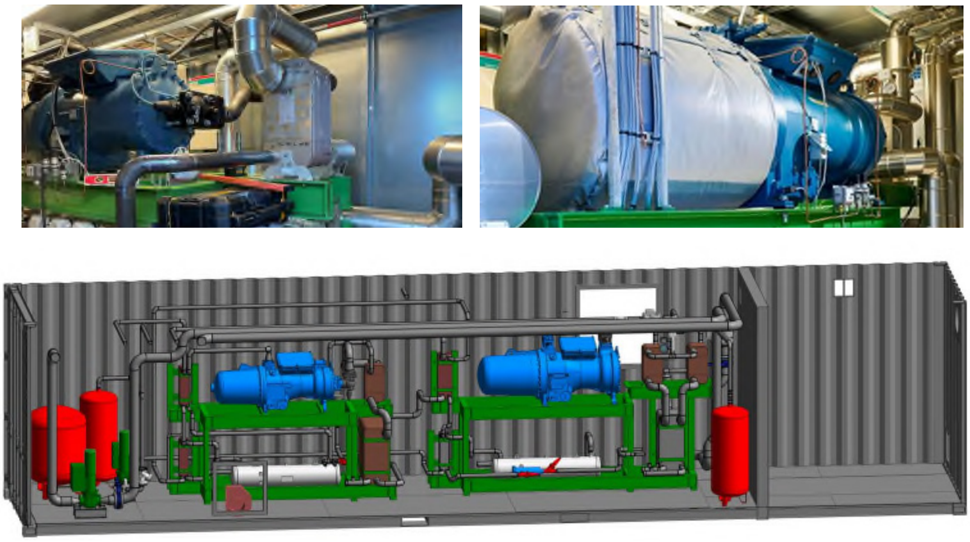

Figure 2: Pictures and 3-D arrangement drawing of the HTHP system

semi-hermetic screw compressor to the condenser/cascade heat exchanger, operating at a condensing temperature of about 90°C.

On the high-pressure side, heat is absorbed by the top cycle (isopentane) in the cascade exchanger/evaporator at approximately 85°C, then compressed by a semi-hermetic screw compressor to a condensing temperature of around 162°C. This process produces an outgoing water temperature of about 170°C, providing a capacity of approximately 500 kW. The hot water on the high-pressure side passes through several plate heat exchangers, maximizing heat recovery throughout the system before undergoing final heating from 120°C to 170°C. This approach enhances system efficiency by lowering both vapor and liquid temperatures at different stages, thereby facilitating more effective heat transfer. The process is illustrated in the T-Qdot diagram in Fig. 3. The system is highly integrated and specifically optimized for its target industrial process, which is heat recovery from process air and producing hot air up to 150°C.

The system design is a result of a design study facing multiple system requirements:

- Targeted process needs a heat pump capable of producing hot air up to 150°C, which is then further heated to about 280–300°C.
- The high temperature lift (difference between source and sink temperatures) and capacity ($\approx$500 kW) place strong demands on refrigerant choice.

Several factors guided the selection:

- Thermodynamic Performance: High COP (Coefficient of Performance) is preferred to keep energy use low.
- Operating Pressure Limits: Compressors are limited to maximum pressures ($\leqslant$30 bar is a practical limit).
- Safe Operating Range: Minimum pressure should be above atmospheric to prevent ingress of air/moisture in case of leaks.
- Critical Temperature: Needs to be high enough to allow operation at the required process temperatures without reaching excessive pressures.
- Natural/Low-GWP Refrigerants: Preference for hydrocarbons (HCs) or HFOs due to environmental considerations.

The outcome of the screening process resulted into Isobutane, which was chosen for the bottom stage because it provides a high COP, works above atmospheric pressure, and has a manageable maximum pressure for available compressors. Isopentane was chosen for the top stage because it offers a good COP, stays within safe pressure limits, and is suitable for higher temperature operation, while also being a natural, low-GWP refrigerant. Together, they provide efficient, safe, and environmentally responsible operation for the high-temperature heat pump in this specific industrial process. Some refrigerants like CO_2 could provide a good temperature glide but require excessively high pressures not achievable by available compressors easily.

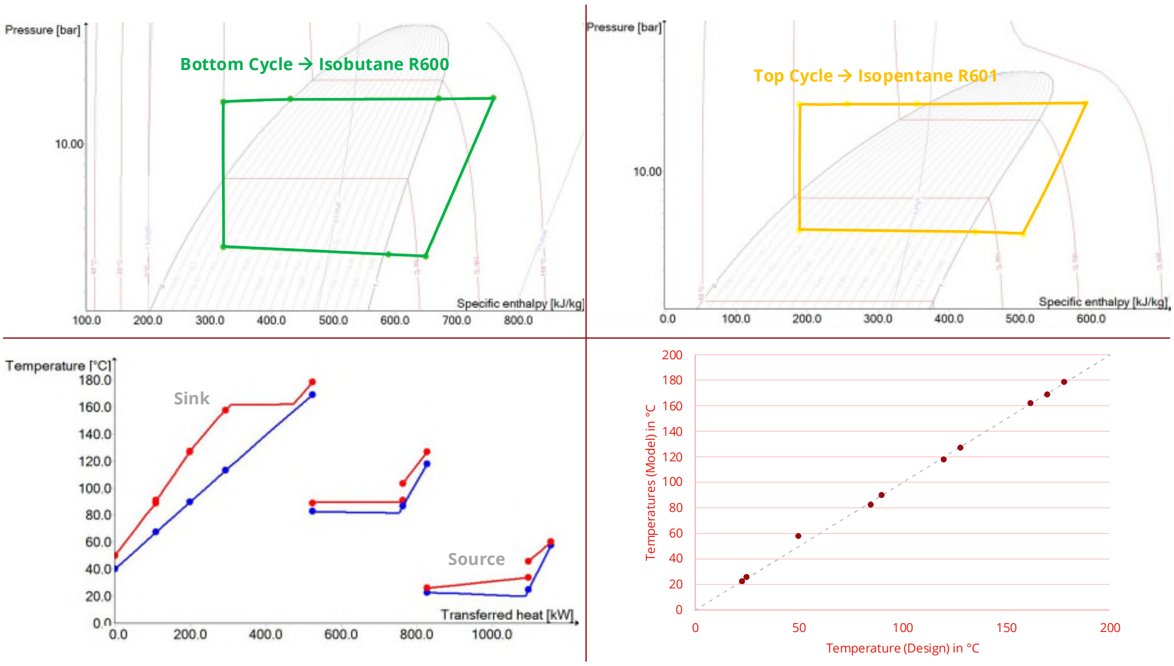

Figure 3: Visualization of the design point in log p-h diagrams for bottom (top left) and top (top right) cycle, T-Qdot diagram (bottom left) and a comparison of model results with design point data (bottom right)

The intermediate water loops are employed between the heat pump refrigerant circuits and the actual process air, to reduce the risk of flammable refrigerant leaks into the process environment.

An additional oil return system was implemented to ensure the high-pressure compressor consistently receives adequate lubrication. This measure was necessitated by the unique behaviour of the novel oil type compared to conventional refrigeration compressor oils.

The complete system is housed within a soundproofed, ATEX-certified 40-foot container, enabling easy relocation as required.

To allow for adequate analysis of operational data obtained during laboratory testing, a steady-state model has been developed in EBSILON Professional (version 17.02), incorporating as-built heat exchanger characteristics and compressor design parameters. A comparison with the specified design conditions is presented in Fig. 3. Water inlet temperatures, water flow rates, and refrigerant flow rates in the model are set to match the design specifications. The simulation results display good agreement with expected system performance under design conditions. In addition, the model can evaluate off-design scenarios by utilizing the actual performance characteristics of the system components. During laboratory testing, the model shall be employed to estimate expected performance values and facilitate direct comparison with experimental results.

3. Laboratory testing

Laboratory testing prior to on-site demonstration is used to commission, evaluate, and optimize the system before its final installation. The system is designed to increase the water temperature from 40°C to 170°C with a COP of approximately 2.5, delivering a heating capacity of 500 kW.

The HTHP was tested at the DTI laboratory across a range of evaporator and condenser temperatures, with test conditions incrementally adjusted to achieve the target water outlet temperature of 170°C. Over 100 hours of testing were conducted, resulting in significant improvements to the control system and overall system performance.

In the initial control strategy, each of the two compressors was set to achieve its own condensing pressure setpoint, with one regulator responding faster than the other. This caused instability and slow system starts, and the compressors negatively influenced each other's operation. The revised control strategy uses table-based operational set points. Both compressors’ speeds are synchronized (their VFD frequencies are locked), so they ramp up and down together. The isobutane compressor always starts first; the isopentane compressor starts once the isobutane condenser reaches a suitable pressure.

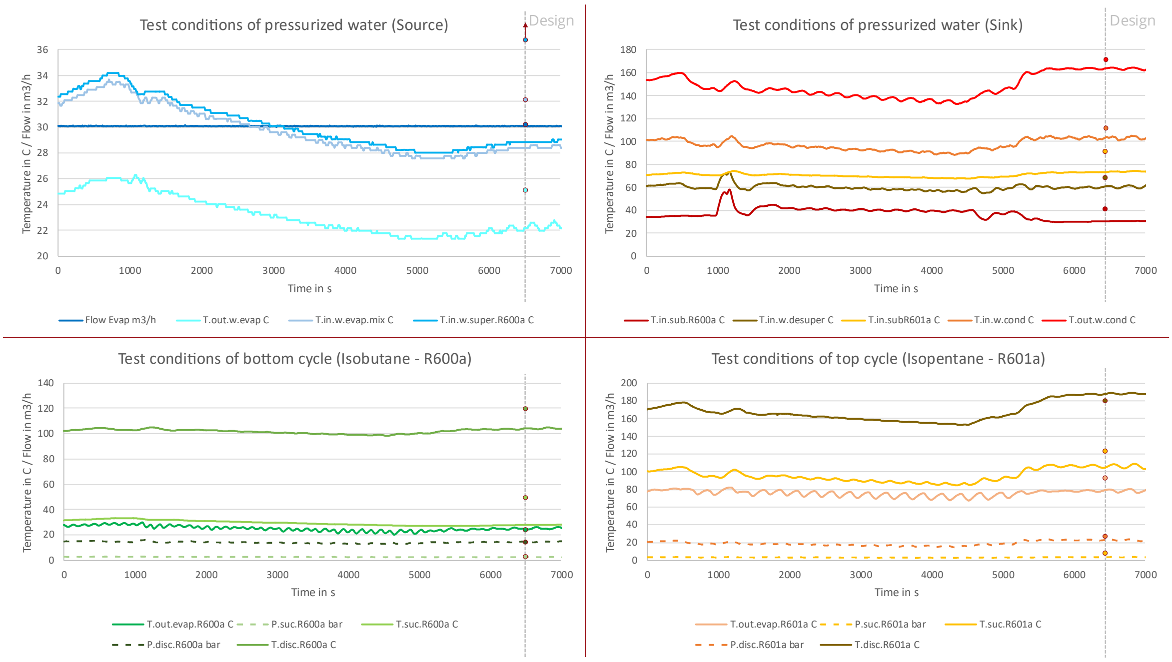

Figure 4: Visualization of operational test data for source water loop (top left) and sink water loop (top right), R600a bottom cycle (bottom left) and R601a top cycle (bottom right)

During initial testing it was also identified that one heat exchanger (Subcooler T – top cycle) was found to be oversized due to an overly strict pressure drop requirement. This led Subcooler T to remove more heat than intended, affecting isopentane superheat in top cycle. A bypass was fitted around Subcooler T to address this.

The HTHP system has undergone testing with varying inlet and outlet conditions on both the sink and source sides, resulting in a range of operational scenarios for both refrigeration cycles. However, due to the highly integrated system design, the flexibility to accommodate such variations is inherently limited. To address this, an additional bypass line for the Subcooler B in the bottom cycle has been installed (as shown in Fig. 1). Fundamentally, the bottom cycle is responsible for the initial temperature lift of the sink water via integrated heat exchangers (subcooler and superheater), and it also supplies heat to the top cycle evaporator. As the sink water inlet temperature increases, the demand on the bottom cycle's integrated heat exchangers decreases. To maintain the bottom cycle within an optimal operating range under such conditions, the subcooler bypass can be employed, providing additional operational flexibility.

Unlike the final application, where heat sources at two distinct temperature levels will be available, the laboratory configuration supplies water to both the superheater and evaporator at the same temperature due to setup constraints of the test facility. In the laboratory setup, water at 30-70°C serves as the heat source but also as input to the sink side. Thus, crossing temperature profiles between the sink and source sides, as in the design case, cannot be tested. Test conditions are modified to allow meaningful comparison close to design conditions.

Fig. 4 presents exemplary test data from approximately two hours of operation. During this test, source water was supplied at the design flow rate with temperatures ranging from 28°C to 34°C. The sink water inlet temperature was maintained between 30°C and 40°C, also at design flow rate. Stable system operation was achieved, with sink water outlet temperatures reaching up to 168°C like nominal operating point. The test demonstrates high sink water temperature lift as well as high outlet temperature.

A comparison of the test data with the design conditions is presented in Fig. 4, where the expected design temperatures for all four loops are indicated in addition to time series data. The test data demonstrates that the cycle operation achieves high outlet temperatures close to the design targets; however, intermediate temperature levels deviate—sometimes significantly—from the design conditions. These deviations are attributable to the laboratory test setup: source water is supplied to both the evaporator and superheater at the same temperature of ~30°C (instead of supplying 60°C to the Superheater B) and design flow rate, unlike the final application where two distinct temperature levels will be available. Water at the same temperature is also supplied to the sink inlet. In the presented test, the overall water inlet temperature was reduced to approximately 30 °C, leading to significantly different operating conditions, particularly in the bottom cycle. The lower inlet temperature to the superheater results in lower superheat, as well as lower compressor inlet and outlet temperatures, which in turn affects the evaporator pressures in the top cycle. HTHP presented can cope with the conditions at the cost of lower system performance different than design conditions.

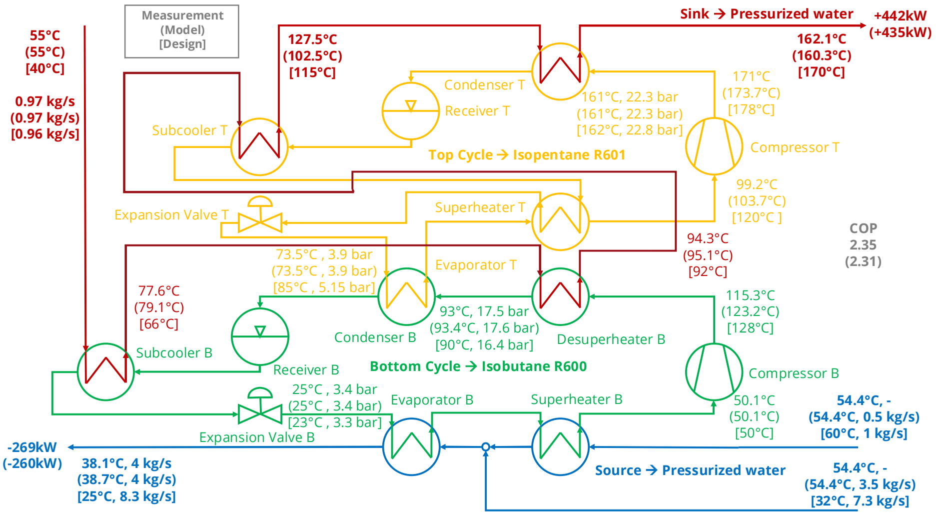

Figure 5: Modified test conditions – Superheater B is supplied with source water closer to its design temperature, while the Evaporator B receives source water at the same temperature but at a significantly lower flow rate than in the original design.

To address these discrepancies, a second test phase was carried out under modified conditions; one representative data point is shown in Figure 5. In this configuration, the Superheater B is supplied with source water closer to its design temperature, while the Evaporator B receives source water at the same temperature but at a significantly lower flow rate to compensate for the increased temperature relative to the original design. The objective is to operate the compressors as close as possible to their design conditions despite the increased sink water inlet temperature. There is generally good agreement between the model and the measured results, with the main deviations observed in the Compressor B outlet temperature and the water temperature at the Subcooler T outlet. These discrepancies are currently under further evaluation. The system was operated with the Subcooler T bypass closed, resulting in increased heat transfer compared to both the design and the model.

4. Summary and Outlook

This paper presents the design and laboratory evaluation of a high-temperature cascade heat pump system developed for industrial applications requiring heat output up to 170 °C. The system employs a cascade configuration using isobutane (R600a) for the lower stage and isopentane (R601a) for the upper stage, with semi-hermetic screw compressors and two pressurized water loops serving as heat source and sink. The design prioritizes a high coefficient of performance, safe and manageable operating pressures, and the use of environmentally friendly, low-GWP refrigerants. Intermediate water loops are used to reduce the risk of flammable refrigerant leaks into the process environment, and the entire setup is contained within a portable, ATEX-certified container. The system delivers a heating capacity of approximately 500 kW.

Laboratory testing, conducted at the Danish Technological Institute, validated system performance across a range of operating conditions. Key findings include the successful achievement of targeted water temperature lifts and output up to 170 °C. The testing process involved iterative optimizations, particularly of the control strategy, where synchronization of compressor operation improved stability and startup behavior. Issues such as heat exchanger sizing and system flexibility were addressed by introducing bypasses to accommodate varying source and sink conditions. Although some deviations from design conditions were observed due to laboratory constraints, the system demonstrated reliable, high-temperature operation and offered valuable insights for further optimization and real-world deployment.

Acknowledgements

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References

[1] International Energy Agency, Net Zero by 2050; A roadmap for the Global Energy Sector.