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Thermal system design for battery-electric agricultural machinery

Benjamin Wilk, Jan Wieckhorst, Philipp Heymann, Ludger Frerichs

The electrification of agricultural machinery, which is currently primarily progressing towards battery-electric drive systems in vehicles of lower power classes, requires solutions to optimize energy efficiency. This poses complex challenges for energy management and thermal regulation of the vehicles. In addition to increasing overall efficiency through the further development of the electric-mechanical drivetrain, the thermal system is gaining particular importance. By utilizing waste heat, there is potential to extend operating time by reducing the electrical energy demand for climate control of the vehicle cabin and conditioning of the traction battery.

This paper presents an approach for the design and implementation of thermo-hydraulic system architectures for battery-electric agricultural machinery. A focus of the investigation lies on the integration of standard vehicle cabins, as used in machines powered by internal combustion engines. This avoids the need for additional cabin variants in the product portfolio. Since these cabins are designed for high water inlet temperatures of around 90 °C, their use in battery-electric vehicles due to the lower temperature level of the component cooling medium presents specific challenges for thermal management, requiring the cabin to be conditioned quickly, sufficiently, and efficiently even from a low starting temperature level.

To address these challenges, hybrid operating modes are developed from a system architecture that combine various heating methods, thus enabling flexible and energy efficient conditioning of both cabin and traction battery. Consequently, an experimental system architecture is presented that allows conditioning of the cabin and traction battery with three essential heating methods.

Keywords
Thermal management, heat pump, battery-electric tractor

The increasing electrification of agricultural and construction machinery intensifies the need for efficient heating systems to extend the operating time of the machines. Previous approaches to heating battery-electric vehicles, which primarily rely on electric resistance heaters, result in a significant reduction of operating time (Konz et al. 2023). Heat pumps offer a promising alternative by utilizing waste heat from vehicle components such as the electric motor, power electronics, and the mechanical drivetrain. (Bach et al. 1966, Remsburg 1998, Joos 2004, Konz et al. 2023). In addition, by intelligently networking the thermal system through the use of valves, waste heat can be efficiently utilized for heating (Pichlmaier and Ehrl 2025). In this context, Fendt has presented a schematic layout of the thermal management system on the test tractor model e100 Vario, where a thermal management system was developed and tested using a heat pump along with corresponding secondary circuits equipped with valves (Breu and Pichlmaier 2017). However, no information was provided regard-

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ing the potential for efficiency improvements through an integrated thermal management system. Therefore, the concept study included a heat pump system, whereas such a system has not yet been installed in the currently available production vehicles.

In the broader context of agricultural technology, Rigitrac has introduced the SKE 40 e-direct Electric tractor as the first production machine on the market capable of heating the cabin using a heat pump (Rigitrac Traktorenbau AG 2025). As part of a pilot and demonstration project funded by the Swiss Federal Office of Energy, further information was published regarding the heat pump system used. The system employs a brine-to-brine heat pump with two switchable hydraulic circuits that utilizes waste heat from the electric motor, inverter, and hydraulic oil cooling to condition both the vehicle cabin and the battery. The studies published within the project show that the system achieves an average COP of approximately 2.9 during heating operation and enables an extension of the operating time by about 6 to 10% compared to a purely electric resistance heater, depending on the usage profile. It is also emphasized that, due to the use of internal waste heat sources, an external air source can be omitted, thereby avoiding icing issues on the evaporator. The system architecture is also based on switchable hydraulic circuits and valves, enabling combined heating and cooling operation (Vogel 2026).

The presented studies aim at the development of an integrated thermal system to utilize waste heat flows for the conditioning of the battery and vehicle cabin. There are various thermal system architectures available to meet the conditioning requirements of a wide range of components within a liquid cooling circuit (Menken 2016). The most prominent method in the automotive sector is demonstrated by Tesla with the Oktovalve concept, which enables holistic thermal management including waste heat utilization using with a single valve (TESLA Inc 2024).

Previous studies have already demonstrated the fundamental potential for utilizing heat pumps and waste heat in electrified vehicles. However, these approaches predominantly focus on individual subsystems or specific vehicle applications, without providing a systematic consideration of a fully integrated thermal management system for battery-electric agricultural machinery. In particular, the flexible coupling of different heat sources and heat sinks, the representation of various operating states within a unified system architecture, as well as the integration of existing standard vehicle cabins from internal combustion engine powered vehicles have so far been addressed only to a limited extent. The present paper addresses this research gap by developing and analysing a multi circuit, valve controlled thermal management system that enables various heating and cooling strategies within an unchanged cabin hardware architecture.

System Analysis

In the following, the system configuration is considered with exclusively liquid-cooled components. To gain a fundamental overview of the possible system states, a variant analysis is conducted, as shown in Table 1. This allows for determining the maximum number of system states and excluding any non-logical states. Non-logical states include, for example, simultaneously heating and cooling the battery, as illustrated in Table 1, column 63 (highlighted in red).

Table 1: Variant analysis for determining system states in thermal management. 1: System active, 0: System inactive, green highlighted: logical, red highlighted: non-logical

Case:	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	...	63
Battery cooling	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	...	1
Battery heating	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	...	1
Cabin cooling	0	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	...	1
Cabin heating	0	0	0	0	1	1	1	1	0	0	0	0	1	1	1	1	...	1
Oil cooling	0	0	1	1	0	0	1	1	0	0	1	1	0	0	1	1	...	1
Electronics cooling	0	1	0	1	0	1	0	1	0	1	0	1	0	1	0	1	...	1

After reducing the matrix to 48 logical system states, a further filtering of the variant analysis is carried out to identify the system states relevant for real-world vehicle operation.

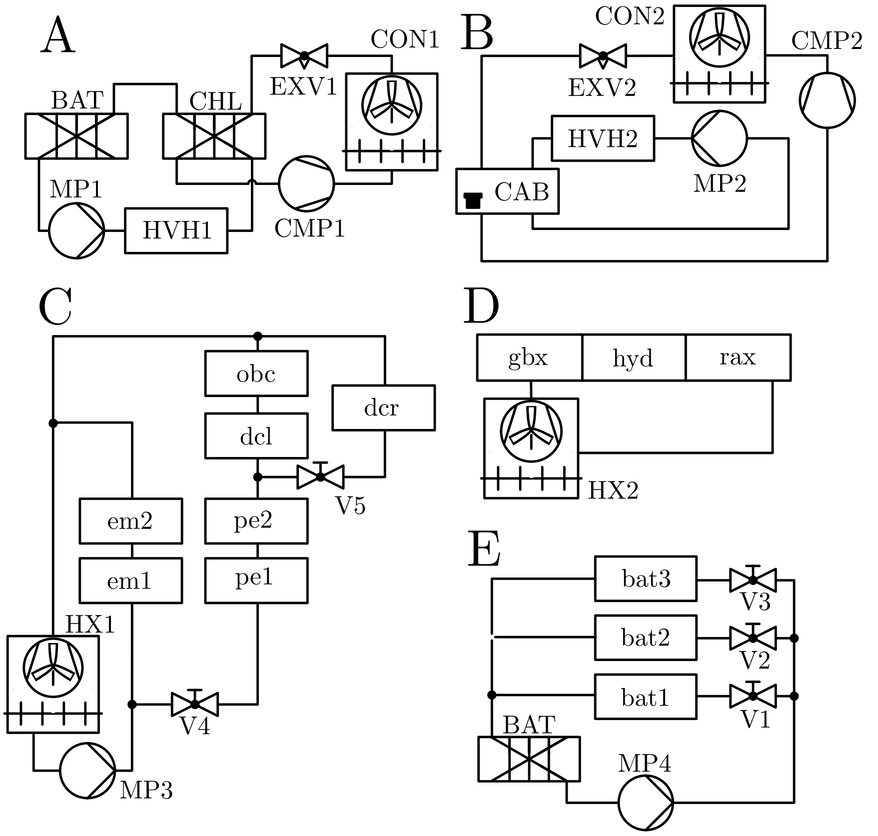
The method presented in Table 2 analyses various operating scenarios of the vehicle under different temperature conditions, starting from defined initial temperatures and including different phases such as charging or driving. The aim is to determine the system states that are necessary for operation, thereby reducing the complexity of software development.

These states are shown in the column „Filtered Var.“ (0 = not required, 1 = required) and can be assigned to the corresponding system states.

Table 2: Variant filter depending on the start temperature and the vehicle system state; L: low, M: medium, H: high; 0 = not required, 1 = required

	Case:	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	...	47
L. Temp.	Before: Charging	1																...	
	During: Charging	1	1			1	1							1	1			...	
	Before: Driving	1				1								1				...	
	During: Driving	1	1		1	1	1		1					1	1		1	...	1
M. Temp.	Before: Charging	1																...	
	During: Charging	1	1											1	1			...	
	Before: Driving	1												1				...	
	During: Driving	1	1		1									1	1		1	...	1
H. Temp.	Before: Charging	1																...	
	During: Charging																	...	
	Before: Driving	1								1				1				...	
	During: Driving																	...	1
	Filtered Var.	1	1		1	1	1		1	1				1	1		1	...	1

After the variant matrix has been reduced to the system states required in real operation, development can begin on a schematic system that maps the essential states. At this stage, the detailed system design is not yet considered, and components such as a refrigerant dryer or other auxiliary equipment are omitted. Figure 1 shows a thermal system architecture for an electric agricultural machine that fulfils all system states previously determined through the variant analysis. It is important to note that the standard vehicle cabin from the internal combustion engine powered vehicle is used unmodified. Each component can thus be fundamentally cooled or heated according to the specified requirements.



BAT	Heat exchanger	EXV	Electric expansion valve
CAB	Cabin	GBX	Heat exchanger
CHL	Chiller	HVH	High voltage heater
CMP	Compressor	HX	Water/air heat exchanger
CON	Condenser	MP	Media pump
ELE	Heat exchanger	V	Valve

bat	Battery	hyd	Hydraulic
dcl	DC-DC converter	obc	On board charger
dcr	DC-DC converter	pe	Inverter
em	E-Machine	rax	Rear axle

Figure 1: Basic system setup of a thermal management system for a battery-electric agricultural machine; circuit A: battery conditioning, circuit B: cabin climate control, circuit C: electronics component cooling, circuit D: drivetrain cooling, circuit E: battery conditioning liquid circuit

Circuit A is responsible for conditioning the battery circuit and provides both heating and cooling functions. The connection to Circuit E is made via the plate heat exchanger BAT, which enables a further liquid circuit for the immersion cooling of battery cells using a dielectric fluid. This fluid must not encounter with conductive media. A coolant pump (MP1) circulates a water-glycol mixture through the high voltage heater (HVH1) and the evaporator (CHL) of the refrigerant circuit (CMP1, CON1, EXV1) to the heat exchanger (BAT). In heating mode, the high voltage heater heats the water-glycol mixture, which then transfers heat to the battery via the primary side of the heat exchanger (Circuit E). Another coolant pump (MP4) in Circuit E transports the heated medium to hydraulic balancing valves (V1-3), which ensure the distribution of the volume flow to the parallel arranged battery modules (bat1-3). These valves compensate for flow losses that may occur due to asymmetric hose routing.

Circuit B describes the system setup for cabin climate control, incorporating a reheat operation where the air is cooled below the dew point to remove moisture and subsequently reheated to regulate interior humidity. A standard vehicle cabin with water/air and refrigerant/air heat exchangers arranged in series is used. Cabin cooling is carried out via a separate refrigerant circuit (CMP2, CON2, EXV2) with direct air cooling of the condenser. Heating is realized by a high voltage heater (HVH2) supplied with a water-glycol mixture, which transfers heat to the interior air via a heat exchanger.

The liquid cooled Circuit C supplies electric components such as DC converters, inverters, and machines with the water-glycol mixture. The captured waste heat is dissipated to the ambient via the heat exchanger (HX1). Volume flow distribution is controlled by hydraulic balancing valves (V4-5) according to the respective cooling requirements.

Finally, Circuit D controls the oil cooling for the shared reservoir of the transmission, hydraulics, and rear axle. The oil releases heat to the ambient air via a heat exchanger (HX2), with the media pump integrated into the rear axle.

For the design of such a system, a comprehensive system analysis is required in advance, considering the achievable volume flows in the individual subsystems as well as the maximum permissible inlet temperatures. Water pumps (designed as flow machines) are subject to characteristic performance curves, which mean that due to the resulting pressure losses, it is not possible to flow the required volume through an unlimited number of components. Furthermore, the maximum inlet temperature of the coolant under specified load conditions must not be exceeded, as otherwise the required system performance would be compromised.

Electrical components such as power electronics and electric machines must be operated within their specified temperature limits because otherwise, due to so-called derating, the electrical output power is reduced to prevent thermal overload and damage to the electronics. In addition, it is aimed to keep the coolant temperature as high as possible to maximize the efficiency of the heat exchanger to the environment. When the temperature difference between the coolant and the ambient air is low, either the air volume flow must be increased or the heat exchanger surface area enlarged. However, the latter is severely limited by the restricted installation space, for example, in standard tractors. At the same time, the electrical power demand for fan operation should be minimized since agricultural machines usually require active cooling air supply, which significantly influences energy consumption. Therefore, the following outlines the analysis of the hydraulic-thermal system using an example.

Analysis of the steady-state thermal system curve of a cooling circuit

Figure 2 shows a temperature-power diagram of an exemplary cooling circuit for electronic components. This diagram allows the determination of the inlet temperature, the minimum required cooling capacity, and the total volume flow needed for the air-water heat exchanger (HX).

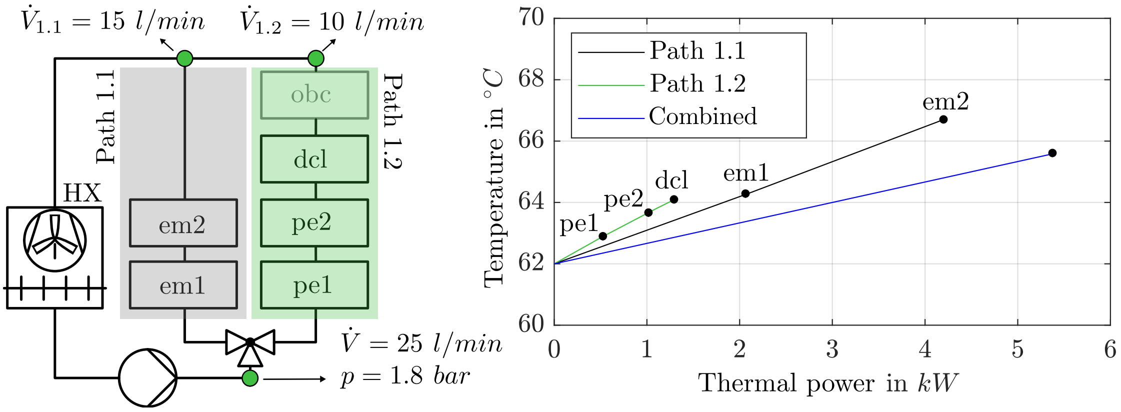


Figure 2: Temperature-power diagram of an exemplary cooling circuit for electronic components; HX: heat exchanger, em: electric machine, pe: inverter, dcl: DC/DC converter, obc: on-board charger

The arrangement of the components is carried out according to their maximum allowable inlet temperatures and thermal sensitivities. Subsequently, taking into account the volume flow requirements, they are systematically assigned to the respective cooling circuits (Reiter 2020). For this purpose, the required cooling water volume flow specified by the manufacturer for cooling the components is assumed. To determine the outlet temperatures, the temperature-power diagram according to Brunner et al. (2017) is used. This is an established method for the thermal evaluation of coupled processes and enables the systematic analysis of temperature and power levels in both continuous and batch processes. The application is carried out according to the principles described in the manual for the analysis of continuous and batch processes.

Using this method, the system's starting temperature is iteratively increased until the maximum allowable inlet temperature of the serially connected components is reached as a thermal boundary condition. This ensures that all components operate within their specified temperature limits. For the path ELE 1.1, the water inlet temperature for the component EM2 is 67 °C. Furthermore, the required thermal capacity of the heat exchanger can be determined. For this purpose, the total volume flow of the coolant from paths ELE 1.1 and ELE 1.2 is calculated, and then the temperature change resulting from the mixing of fluids with different volume flows is computed. By summing the heat outputs from paths ELE 1.1 and ELE 1.2, the total heat load to be dissipated at this operating point (Combined) is determined. In the illustrated example, the heat exchanger (HX) must be capable of dissipating a heat load of 5.4 kW at a water inlet temperature of 66 °C and a volume flow of 25 l/min.

Analysis of the hydraulic system characteristic curve of a cooling circuit

Figure 3 shows the system and pump characteristic curves and illustrates the pressure behaviour as a function of the volume flow rate. The black line represents the pressure losses caused by all components integrated into the circuit, while the green line depicts the characteristic curve of the water pump.

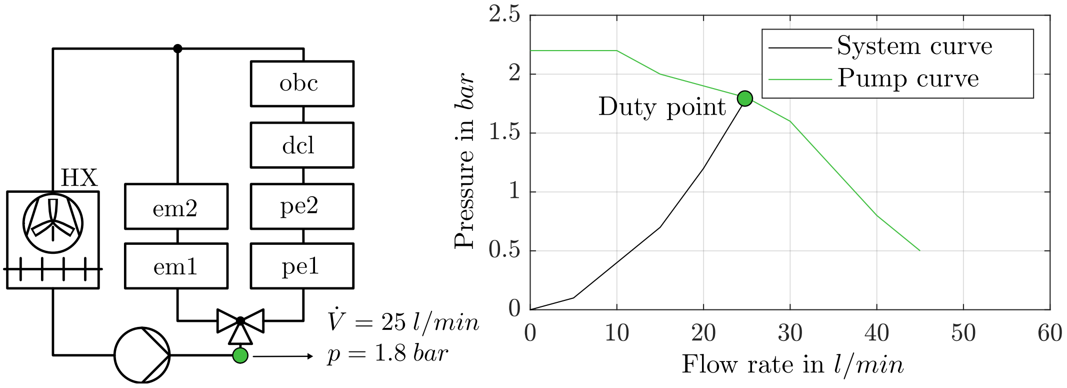


Figure 3: Identification of the hydraulic system characteristic curve; the green point represents the measurement point of the system characteristic curve at the pump outlet; HX: heat exchanger, em: electric machine, pe: inverter, dcl: DC/DC converter, obc: on-board charger

The black line shows a continuous increase in pressure loss with rising volume flow. This behaviour is typical for hydraulic systems, as pressure loss in turbulent flow increases disproportionately with increasing volume flow (Sigloch 2022). This results from friction losses in the piping and additional losses in components such as heat exchangers and valves. At low volume flows, pressure losses are marginal but increase significantly with higher flow rates. The green line represents the delivery pressure of the coolant pump as a function of volume flow. Initially, at low volume flows, the pump generates a high delivery pressure. With increasing volume flow, the delivery pressure decreases, which is characteristic of flow machines (Bohl and Elmendorf 2013). his decrease is due to rising dynamic losses that occur at higher flow rates. The intersection of the black and green lines defines the duty point of the system. At this point, the pressure generated by the pump equals the pressure loss in the circuit. The duty point indicates the actual working condition of the system and quantifies the delivered volume flow in the system. The analysis of the diagram highlights the necessity for precise matching between pump performance and system requirements. By optimizing the duty point, the energy consumption of the system can be reduced (Allelein et al. 2010).

During development, it is important to verify the cooling circuit's performance when system requirements change, to ensure the minimum required coolant volume flow is not undershot. Moreover, the number of components arranged in series within a cooling circuit is limited. This limitation is caused by the increasing pressure loss with each additional component, which leads to a reduction in volume flow for a given pump design. If the required minimum volume flow rate is not maintained, the thermal performance of the cooling circuit can no longer be guaranteed. Therefore, for a given pump, the cooling circuit is divided into parallel cooling paths. Individual cooling paths are hydraulically separated by heat exchangers once the minimum required volume flow rate can no longer be maintained due to the cumulative pressure losses (Konz et al. 2023).

Heat pumps in battery-electric agricultural machines

Heat pumps represent an effective solution for optimizing the thermal management of battery-electric agricultural machinery, particularly when grid connected preconditioning is unavailable. A model-based sensitivity analysis indicates that utilizing a heat pump for waste heat recovery during cold starts extends the operating time of a farm tractor with a 100 kW powertrain and a 100 kWh battery by approximately 7–11% under a moderate mixed-usage profile, compared to a standard resistance heater. Given a coefficient of performance (COP) between 2 and 4, this translates to an additional 14 to 22 minutes of operation (Wilk et al. 2026). To experimentally validate these modelled savings, a vehicle integrated thermal system will be developed.

Functioning of the Heat Pump

Heat pumps transport thermal energy by using mechanical work from a lower to a higher temperature level. They extract heat from the environment and transfer it to a higher temperature level (Böckh and Stripf 2015). The process comprises four main phases: evaporation, compression, condensation, and expansion (Dohmann 2016). Within the vehicle, a vapor-compression refrigeration cycle can be utilized both as a heat pump for heating and as a refrigeration system for cooling (Weustenfeld 2018). Figure 4 illustrates the operation of a vapor compression heat pump cycle, where heat is absorbed from the environment via the evaporator and rejected at a higher temperature level at the condenser.

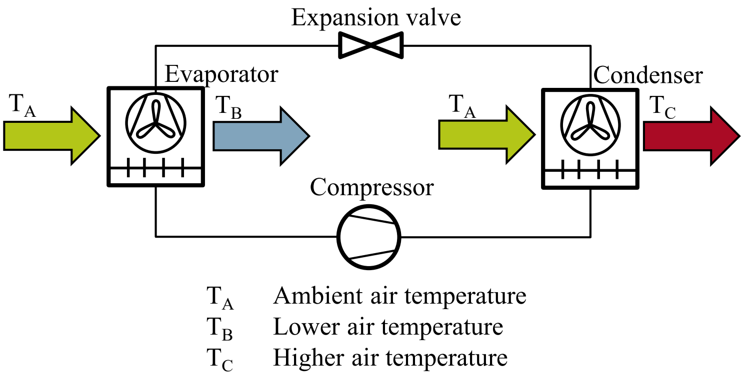


Figure 4: Schematic of the heat pump principle concerning the process temperature

Secondary circuits for waste heat utilization

A secondary circuit can be integrated to recover waste heat from the transmission and hydraulic systems. By using a heat exchanger, the primary and secondary loops remain hydraulically decoupled, enabling heat transfer without any fluid exchange.

Direct waste heat utilization

In this case, the hydraulics could be coupled with a water-glycol circuit. This waste heat can then be introduced into the heating process, which transfers the heat directly to a heat sink, such as the vehicle cabin and/or the battery (Figure 5).

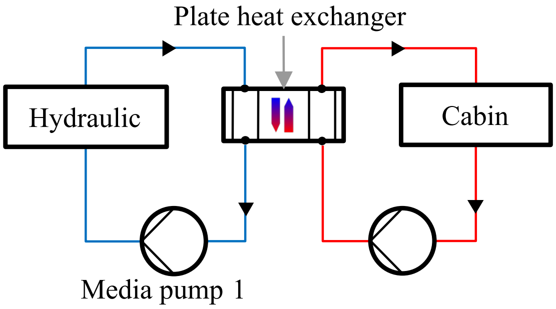


Figure 5: Schematic for a simple secondary circuit for direct waste heat utilization from the hydraulics

The heat flow could also be supplied to an evaporator of a vapor-compression refrigeration cycle, which is then discharged to another secondary circuit on the condenser side of the vapor-compression refrigeration cycle at a higher temperature level. This has the advantage that the supplied waste heat can already be utilized at temperatures below 60 °C in the hydraulic oil circuit. This is particularly the case when a standard vehicle cabin is used without modification of the heat exchanger, as lower temperature differences result in reduced heating performance of the heat exchanger.

Heat pump operation with ambient air

Figure 6 shows a heat pump in which heat is extracted from the environment with the help of the primary circuit (green connecting lines) and the heat exchanger HX. This heat is then raised to a higher temperature level in the vapor-compression refrigeration cycle and transferred to the secondary circuit via the plate heat exchanger CON. The heat transferred there can then be used to heat the cabin with the help of the heat exchanger located inside the cabin. In addition, the heating capacity increases due to the compressor work added in the vapor-compression refrigeration cycle, which is reflected in the condenser heat flow.

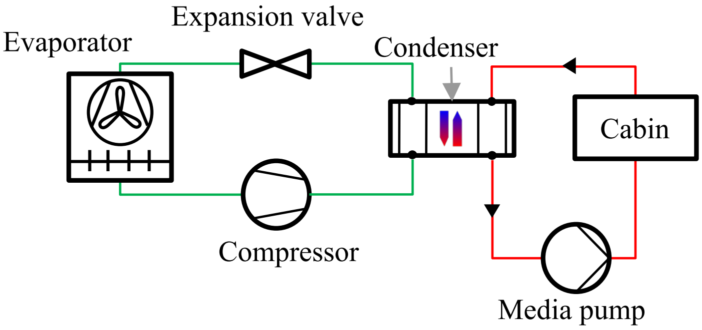


Figure 6: Schematic of a vapor-compression refrigeration cycle with a secondary circuit in heat pump operation using ambient air

Heat pump with waste heat source

Figure 7 illustrates a multistage heating system designed to transfer heat from a hydraulic oil circuit to a water-glycol heating circuit for the vehicle cabin. This system essentially consists of a central primary circuit and two secondary circuits, each performing different functions and thermally coupled via heat exchangers while ensuring media and pressure side decoupling.

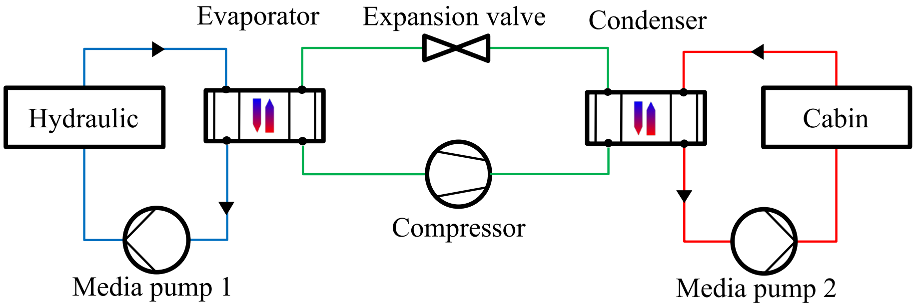


Figure 7: Vapor-compression refrigeration cycle with two secondary circuits for the utilization of waste heat from the hydraulic oil circuit using the heat pump principle

The primary circuit, shown in green, operates in the vapor-compression refrigeration cycle serves as a thermal intermediary between the secondary circuits. It absorbs the heat flow from the hydraulic system, compresses the refrigerant to increase pressure and temperature, and then transports the heat to the heating circuit. This function enables flexible control of heat transfer without directly mixing the media.

The left secondary circuit, depicted in blue, represents the hydraulic oil circuit and functions as a heat source for the thermal management system. Waste heat can be extracted here at a lower temperature level. It is essential to ensure that a sufficient amount of usable waste heat is available in the energy balance, particularly considering the specific required operating temperature of the hydraulic system. Excessive cooling of the hydraulic oil leads to increased viscosity, which, due to the nonlinear temperature-viscosity relationship (e.g., shown in the Ubbelohde diagram in Bauer and Niebergall (2020), can cause disproportionately higher hydraulic-mechanical losses and thus efficiency losses. The actual available waste heat depends on the application as well as the operating times and cycles of the hydraulic system. However, certain thermal and operational boundary conditions must be met to use a cooling or oil circuit as a heat source for a heat pump system.

Crucial factors include a sufficiently high and as continuous as possible temperature level, as well as adequate thermal power of the respective circuit. Additionally, heat extraction from the circuit must not cause inadmissible cooling of components or operating media, since in hydraulic systems, for example, increased viscosity due to cooling can lead to higher flow and friction losses. Moreover, the temporal availability of waste heat is important because, especially under dynamic load profiles, only low waste heat flows may be available at times. Therefore, the suitability of a circuit as a heat source must be evaluated considering the specific usage profile, thermal power, and permissible temperature limits of the involved components.

However, the hydraulic oil circuit does not necessarily have to be used as a waste heat source. Fundamentally, waste heat from electrical components can also be used if this is energetically and system-technically advantageous. In general, the combination of a central vapor-compression refrigeration cycle primary circuit with two media separated secondary circuits allows for the utilization of waste heat from the hydraulic system while simultaneously ensuring media and pressure side decoupling (based on previous investigations by the first author).

Switchable multi circuit systems

For the thermal conditioning of vehicle components, particularly the vehicle cabin, a switchable heat pump system with an attached secondary circuit is used. The schematic layout of the system is shown in Figure 8. The system is characterized by its ability to operate in both heating and cooling modes within the refrigeration system without hydraulic modifications, with the switching of the secondary circuits carried out using four 4/2-way valves.

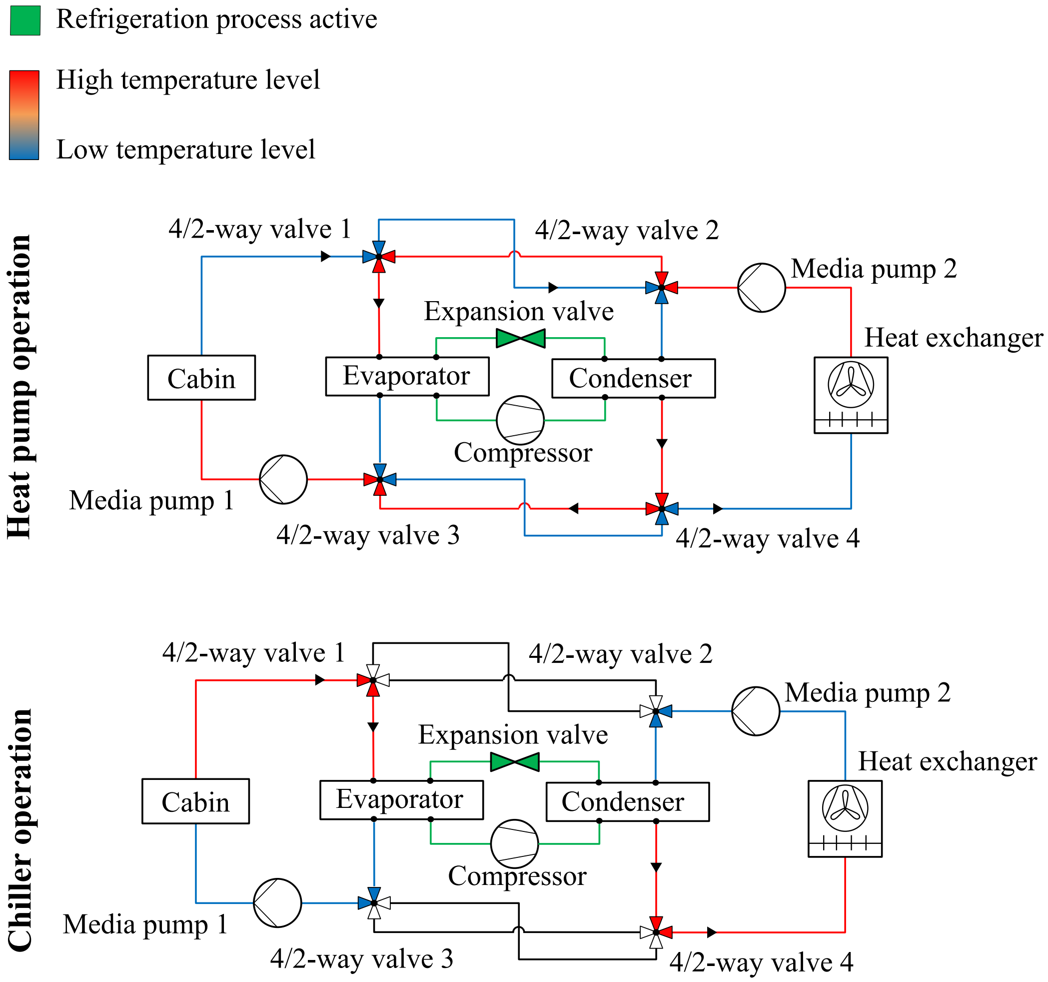


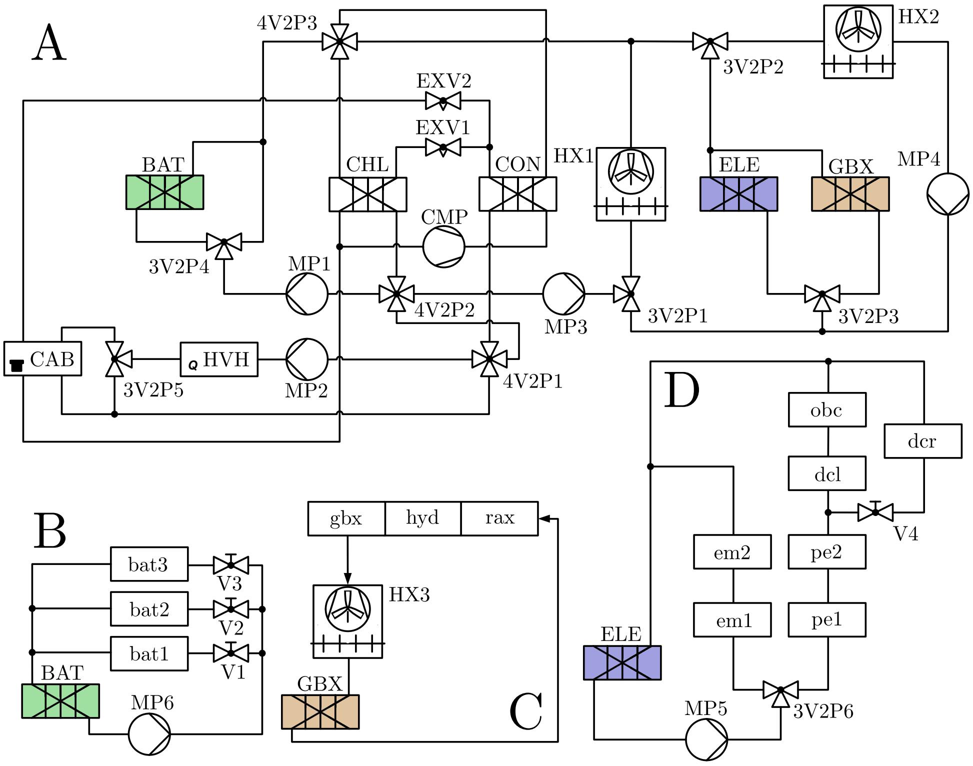
Figure 8: Representation of the refrigeration process with integrated secondary circuit and switching valves for flexible system switching between heat pump and chiller operation

The primary circuit is designed as a closed refrigerant loop and remains hydraulically unchanged throughout all operating states. Key components such as the refrigerant compressor, expansion valve, evaporator, and condenser are permanently integrated and remain mechanically unaltered. The secondary circuit is hydraulically coupled to the primary circuit via a plate heat exchanger (evaporator and condenser). This enables bidirectional energy exchange between the refrigerant in the primary circuit and the secondary medium, for example, a water-glycol mixture. The medium circulates through the targeted vehicle component or space (e.g., the cabin) to regulate its temperature by absorbing or releasing thermal energy.

The heat exchanger for ambient air is designed for bidirectional operation. Depending on the selected operating mode, it serves either as the condenser, which releases heat to the secondary circuit (Figure 8, heat pump operation), or as the evaporator, which absorbs heat from the secondary circuit (Figure 8, cooling operation). Switching between heating and cooling mode is performed solely by targeted control of the 4/2-way valves 1 through 4. This procedure enables, as shown in Figure 8, both cooling and heating of the vehicle cabin. The low external temperature level is raised to a higher temperature level via the refrigeration process. The presented concept offers several advantages: it enables reversible use of the same hardware for heating and cooling, reduces system complexity in the refrigerant circuit by eliminating mechanical switching devices within it, and allows flexible and demand-oriented thermal conditioning of various vehicle components.

In heat pump operation, the heat exchanger exposed to ambient air must be frost-resistant. When using ambient air as the heat source, especially at low temperatures, there is a risk of icing, since the heat exchanger can reach temperatures below 0 °C during operation. Therefore, an appropriate defrosting strategy is required, for example, cyclic switching between heat pump and cooling operation, to prevent icing of the heat exchanger at ambient air and ensure normal operation. Depending on the water-glycol filling volume or thermal capacity in the secondary circuit, a defrosting process can reduce system efficiency and additionally complicate control of the warm side, which decreases interior comfort. Using the water-glycol secondary circuit, a different heat exchanger (water/water) can be employed instead of the heat exchanger (water/air), which supplies the system with waste heat from components and makes it usable at a low temperature level.

Figure 9 illustrates a thermal management system that conditions both the battery and the vehicle cabin using a heat pump and a secondary circuit. This setup is a hybrid solution (heat pump and resistance heater) that enables the heating process of the battery and cabin.



BAT	Heat exchanger	GBX	Heat exchanger
CAB	Cabin	HVH	High voltage heater
CHL	Chiller	HX	Water/air heat exchanger
CMP	Compressor	MP	Media pump
CON	Condenser	4V2	4/2-way valve
ELE	Heat exchanger	3V2	3/2-way valve
EXV	Electric expansion valve	V	Valve

bat	Battery	hyd	Hydraulic
dcl	DC-DC converter	obc	On board charger
dcr	DC-DC converter	pe	Inverter
em	E-Machine	rax	Rear axle
gbx	Gearbox		

Figure 9: Fully integrated thermal management system with heat pump; A: main circuit, B: battery circuit, C: hydraulic circuit, D: electronics circuit; coupling of the circuits via the shown plate heat exchangers BAT (green), GBX (brown), and ELE (blue)

The number of associated system states is listed in Table 3. By adding a heat pump, the theoretically possible number of system states increases to 256. However, this state matrix is limited to logical states, which can significantly reduce the number of required states (approx. 30).

Table 3: Determination of system states for an advanced thermal management system, HVH: high voltage heater, HP: heat pump. 1: system active, 0: system inactive, highlighted in green: logical, highlighted in red: illogical, according to Wilk (2026)

Case:	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	...	255
Battery cooling	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	...	1
Battery heating - HVH	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	...	1
Battery heating - WP	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	...	1
Cabin cooling	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	...	1
Cabin heating - HVH	0	0	0	0	0	0	0	0	1	1	1	1	1	1	1	1	...	1
Cabin heating - WP	0	0	0	0	1	1	1	1	0	0	0	0	1	1	1	1	...	1
Oil cooling	0	0	1	1	0	0	1	1	0	0	1	1	0	0	1	1	...	1
Electronics cooling	0	1	0	1	0	1	0	1	0	1	0	1	0	1	0	1	...	1

In the illustrated thermal management system, the thermal heat flow generated by the components waste heat can, among other options, be supplied to the heat pump circuit. The primary focus for utilizing waste heat lies in the hydraulic oil circuit and the circuit responsible for cooling the electrical components. Additionally, all other components can be cooled according to their respective operating temperature range.

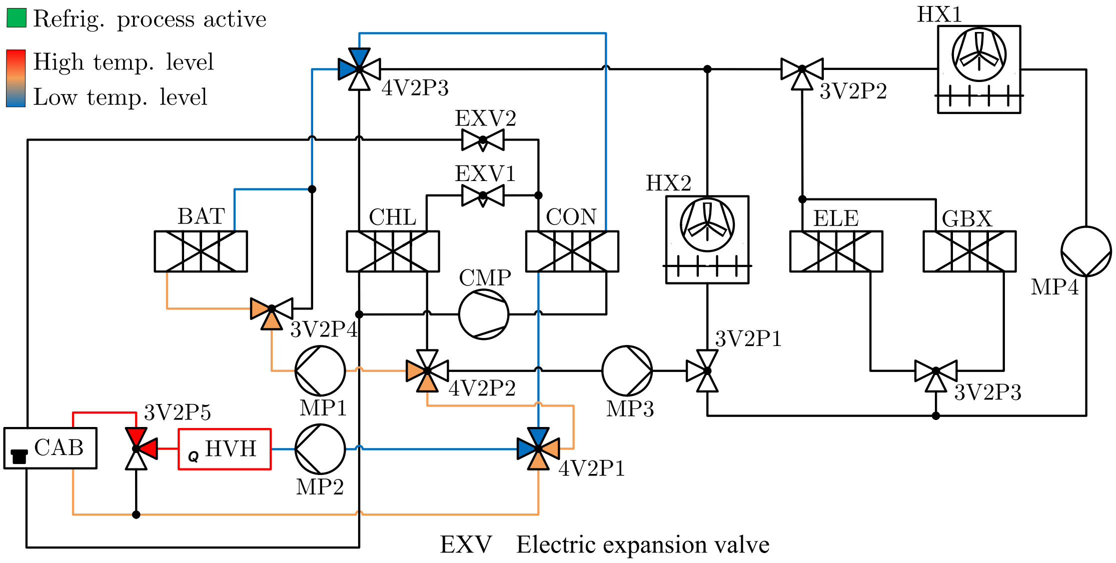
At the core of the setup (section A) lies the thermodynamic cycle process. This can, for example, utilize waste heat flows from subsystems C and/or D, i.e., from the perspective of thermal management system development, from the cooling circuit of the electrical components and/or from the lubrication oil circuit of the drivetrain. The coupling of the various systems used as heat sources is achieved via two plate heat exchangers designated ELE and GBX. Furthermore, as previously explained, the battery conditioning circuit is located in a separate subsystem labelled B.

The constructed system can assume a variety of operating modes. As shown earlier in the variant matrix, the complexity increases exponentially, making the control of such a system demanding. Nevertheless, this system design is used to map various operating modes through a thermal management system that enables different system states exclusively through software control, while the underlying hardware architecture remains unchanged.

In total, 31 different requirement variants are investigated with the shown setup, which can be adjusted depending on vehicle status and system temperature. Moreover, the battery and cabin can be heated via a high voltage heater, heat pump, or directly coupled with the heat source.

The thermal management system presented below is designed as a multi circuit, valve-controlled heat management concept that enables demand-oriented distribution, absorption, and release of thermal energy between various vehicle components. Switching between the individual operating modes is carried out exclusively by targeted control of the integrated valves, while the hydraulic basic structure of the circuits remains unchanged.

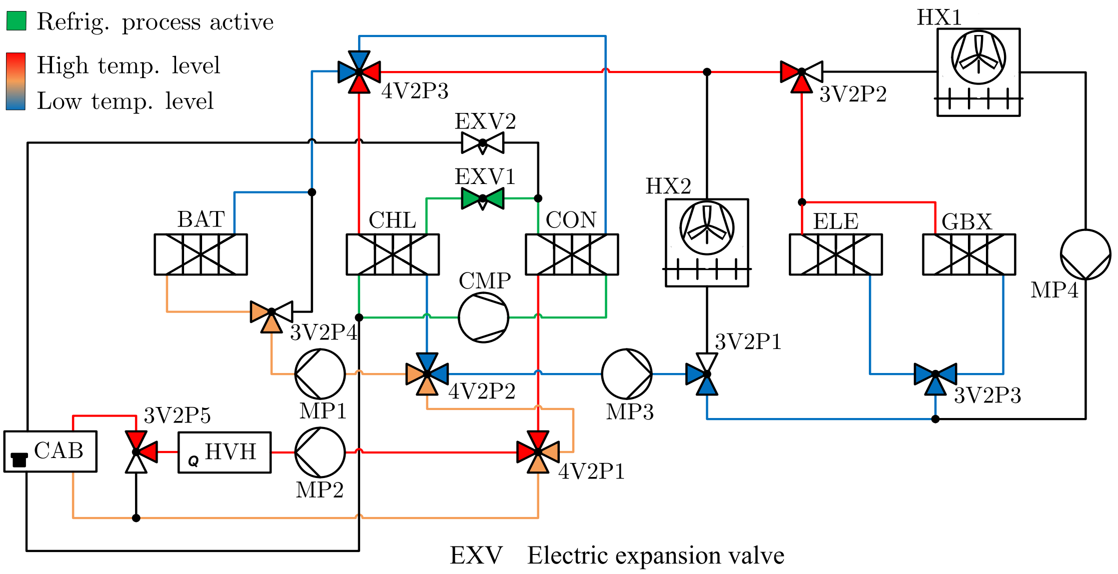
The thermal system state when using the high voltage heater is illustrated in Figure 10. In this configuration, the cabin heat flow is primarily controlled via the high-voltage heater's power supply, with valves 3V2P_5 and 3V2P_4 providing selective regulation for both the cabin and battery circuits.



BAT	Heat exchanger
CAB	Cabin
CHL	Chiller
CMP	Compressor
CON	Condenser
ELE	Heat exchanger
EXV	Electric expansion valve
GBX	Heat exchanger
HVH	High voltage heater
HX	Water/air heat exchanger
MP	Media pump
4V2	4/2-way valve
3V2	3/2-way valve

Figure 10: System configuration for heating with the high voltage heater

The use of the heat pump with the aid of waste heat is illustrated in Figure 11. The heat flow for the cabin is primarily determined by the output of the thermal machine, which essentially consists of the refrigerant compressor speed and the heat flow at the inlet of the refrigerant evaporator. In this case, the input heat flow for the evaporator is provided by the thermal waste heat from the electronic components and the transmission.



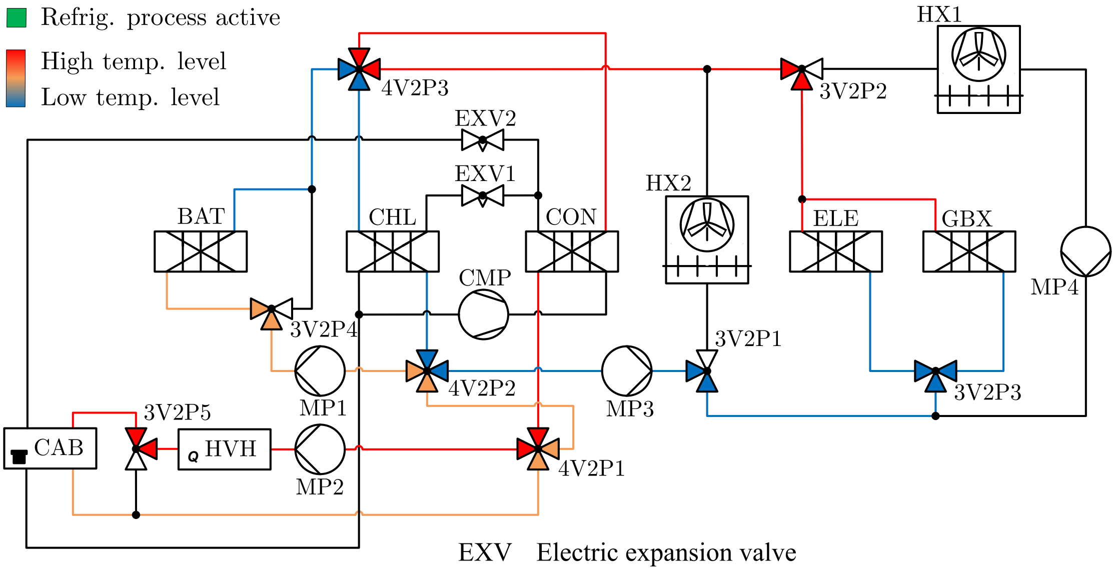
		EXV	Electric expansion valve
BAT	Heat exchanger	GBX	Heat exchanger
CAB	Cabin	HVH	High voltage heater
CHL	Chiller	HX	Water/air heat exchanger
CMP	Compressor	MP	Media pump
CON	Condenser	4V2	4/2-way valve
ELE	Heat exchanger	3V2	3/2-way valve

Figure 11: System configuration for heating using the heat pump principle

Alternatively, valve 3V2P1 can be switched to the heat exchanger HX2. In this operating mode, the heat pump extracts the required heat from the ambient air. Therefore, the heat exchanger must be explicitly designed for operation under icing conditions. This includes structural measures to increase frost resistance (e.g., appropriate fin geometry and spacing, material and coating optimizations) as well as the integration of an effective defrosting strategy that reliably drains the melt water produced during defrosting.

In addition, if needed, an additional heat flow can be introduced into the system via the high voltage heater. The distribution of heat flow to the cabin and battery is regulated by valves 3V2P_5 and 3V2P_4 and can be specifically adjusted to the respective heat demand of the cabin and/or battery.

The direct utilization of thermal waste heat is illustrated in Figure 12. In this system configuration, the heat flow from the drivetrain (GBX) and the electronic components (ELE) is used directly to heat the battery and the vehicle cabin. The vapor-compression refrigeration cycle is not active in this operating mode. Instead, waste heat is used conventionally, mirroring the engine coolant heating found in standard combustion vehicles. This example serves as a reference to highlight the difference compared to modern thermal management strategies in electric vehicles, which, for instance, employ heat pumps.



		EXV	Electric expansion valve
BAT	Heat exchanger	GBX	Heat exchanger
CAB	Cabin	HVH	High voltage heater
CHL	Chiller	HX	Water/air heat exchanger
CMP	Compressor	MP	Media pump
CON	Condenser	4V2	4/2-way valve
ELE	Heat exchanger	3V2	3/2-way valve

Figure 12: System configuration for heating with direct waste heat utilization

Conclusions

Integrating a heat pump with secondary circuits into the vehicle presents technical challenges, particularly regarding space requirements. Innovative vehicle design approaches, especially for standard tractors, are necessary to overcome these challenges. One example is the radiators, which must be larger due to the lower temperatures and relatively small volumetric flow rates (water/air). This holds true even though the heat flows to be dissipated are significantly lower compared to an internal combustion engine vehicle of the same power rating. Additionally, the radiators increase further in size if they are to be supplied with cooling air as quietly and efficiently as possible by an electric fan. A subjective comparison of noise emissions from today's tractors shows that a major portion of the noise is caused by the cooling fan. This must be resolved to avoid a contradiction with the concept of a low-noise, battery-electric tractor.

Combining different heating methods enables a flexible adaptation to varying operating conditions while utilizing the standard vehicle cabin without modifications. Potential modifications would otherwise include adjusting the heat exchanger surfaces, airflow volume, or transfer media (such as using water-glycol for interior cooling), installing a windshield heater, or integrating additional radiant and contact heating surfaces. Furthermore, any such adaptation would necessitate a reassessment of cabin thermal comfort, particularly if the heat transfer method alters the occupant's perception of warmth.

The presented schematic configuration additionally facilitates the targeted examination and assessment of different operational states under realistic conditions, laying the groundwork for developing efficient thermal management strategies. Future studies should aim to optimise the system architecture with a focus on thermal efficiency and noise reduction, while also enabling the integration of these systems into cost-effective serial machinery. These modular thermal management solutions could play a significant role in prolonging the service life of machines and enhancing the energy efficiency of electrified farm equipment.

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Authors

Benjamin Wilk, M.Sc., is an employee in Electrical Drive Technology, and Dipl.-Ing. (FH) Philipp Heymann is the Head of the Electrical Drive Technology Department at Claas Industrietechnik GmbH, Halberstädter Straße 15-19, 33106 Paderborn. **Dr.-Ing. Jan Wieckhorst** is the Head of Tractor Advanced Development at Claas Tractor SAS, Halberstädter Straße 15-19, 33106 Paderborn.

Prof. Dr. Ludger Frerichs is the Director of the Institute for Mobile Machines and Commercial Vehicles at TU Braunschweig, Langer Kamp 19a, 38106 Braunschweig.