Power System Technology
ISSN:1000-3673

Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025

Enhancing CVT Gearbox Reliability Through Optimized Oil Cooler Design: ANSYS Simulation and Vehicle Testing

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Abstract

This study investigates the thermal performance of oil coolers for CVT gearboxes, aiming to mitigate fan motor burnout due to excessive oil temperatures. Using ANSYS Fluent, a computational fluid dynamics tool, the analysis evaluated U-shaped (10 passes), S-shaped (9 passes), and W-shaped (8 passes) oil cooler configurations, focusing on the influence of geometric variations on heat rejection. The models, constructed with aluminum fins and tubes, featured fin pitches of 300 FPM (3.33 mm), 700 FPM (1.428 mm), and 800 FPM (1.25 mm), reflecting diverse fin densities. Simulations were conducted under steady-state conditions with an oil inlet temperature of 113°C, ambient air temperatures ranging from 30°C to 55°C, an oil flow rate of 11 L/min, and air velocities between 7.2 and 162 km/h, simulating various vehicle speeds. A structured mesh with 1.2 million tetrahedral elements ensured accurate capture of temperature gradients, particularly around the fins. The U-shaped oil cooler (20 mm width, 300 FPM) exhibited a heat rejection capacity of 10.33 kW, resulting in an outlet temperature of 83.23°C, exceeding the 75°C threshold and causing prolonged fan operation. In contrast, the optimized U-shaped design (34 mm width, 700 FPM) achieved a capacity of 26.75 kW, reducing the outlet temperature to 35.89°C, while S-shaped and W-shaped designs yielded 22.17 kW and 20.45 kW, respectively. The enhanced performance stemmed from a 67% increase in heat transfer surface area and higher fin density (4250 fins), improving convective heat transfer coefficients. Vehicle tests validated these findings, showing the optimized design maintained oil temperatures below 70°C at 100 km/h, limiting fan operation to 3 minutes, compared to the original design's 85°C peak and over 15 minutes of fan activity. Further analysis revealed that capacity increased with speed (e.g., U30I reached 27.56 kW at 162 km/h) but decreased with rising ambient temperature (e.g., U30I dropped from 20.62 to 13.06 kW as temperature rose from 25°C to 55°C). These insights underscore the importance of geometric optimization in enhancing thermal management, offering a practical solution for CVT reliability with potential for broader application.

Keywords: Oil Cooler, CVT Gearbox, Thermal Performance, ANSYS Simulation, Heat Transfer

Power System Technology
ISSN:1000-3673
Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025

Introduction

Modern vehicles equipped with Continuously Variable Transmissions (CVTs) have gained widespread adoption due to their high efficiency and smooth driving experience [1]. However, a significant challenge in these systems is the thermal management of gearbox oil, where excessive temperatures can lead to component failure and reduced performance [2]. In the vehicles under study, which utilize CVT gearboxes from PUNCH Belgium, a recurring issue of fan motor burnout has been observed due to prolonged operation at high speeds [3]. This problem stems from the insufficient heat rejection capacity of the original oil cooler, prompting the need for an optimized design [4]. This research aims to analyze and enhance the thermal performance of CVT oil coolers through ANSYS-based simulations and on-road vehicle testing, offering a sustainable solution to this critical issue.

Oil coolers, as heat exchangers, play a vital role in transferring heat from the oil to the ambient air, ensuring the gearbox operates within safe temperature limits [5]. Their design involves key parameters such as fin pitch, width, number of passes, and geometric configuration (e.g., U-shaped, S-shaped, W-shaped), all of which influence thermal efficiency [6]. In the initial design of the oil cooler studied here (U-shaped, 20 mm width, 300 FPM fin density), the heat rejection capacity was limited to 10.33 kW under turbulent flow conditions, proving inadequate for high-load operations [7]. Consequently, oil temperatures exceeded the critical threshold of 75°C, triggering the vehicle's cooling strategy to activate the fan at high speed [8]. Prolonged fan operation in this scenario resulted in motor burnout after a short duration, highlighting a significant design flaw [9].

Previous studies have established that elevated oil temperatures in gearboxes not only damage mechanical components but also degrade power transmission efficiency [10]. Carson et al. [11] noted that high temperatures reduce oil viscosity, impairing lubrication and accelerating wear. In CVT systems, the Transmission Control Unit (TCU) communicates with the Engine Control Unit (ECU) to activate the cooling fan when temperatures rise [12]. However, if the oil cooler fails to reduce the temperature below 70°C rapidly, the fan operates excessively, leading to failure [13]. This issue was particularly pronounced in the studied vehicles due to the limited thermal capacity of the original oil cooler [14].

The importance of effective thermal management in automotive applications is well-documented. Smith and Jones [15] emphasized that inadequate cooling can compromise vehicle reliability, especially in high-performance or continuous-duty scenarios. Heat exchanger optimization has been a focus of numerous studies, with researchers like Lee et al. [16] demonstrating that increasing heat transfer surface area enhances convective cooling. Similarly, Patel [17] found that fin density significantly affects heat dissipation rates in compact heat exchangers. In automotive oil coolers, these principles are critical, as space constraints and airflow dynamics further complicate design [18]. The initial oil cooler, with a

Power System Technology
ISSN:1000-3673
Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025

width of 20 mm and low fin density (300 FPM), failed to meet these demands, necessitating a redesign [19].

To address this, modifications were introduced, including increasing the oil cooler width to 34 mm and fin density to 700 FPM [20]. Such changes align with findings from Kim et al. [21], who showed that wider heat exchangers improve airflow distribution and heat rejection. The use of computational tools like ANSYS has become standard in optimizing such systems, enabling precise finite element analysis of fin structures and heat transfer calculations [22]. Studies by Zhang [23] and Gupta [24] highlight ANSYS's capability to simulate complex thermal-fluid interactions, making it ideal for this study. Additionally, experimental validation through vehicle testing is essential, as real-world conditions often differ from simulations [25].

In this context, the study evaluated three oil cooler configurations—U-shaped (10 passes), S-shaped (9 passes), and W-shaped (8 passes)—using ANSYS simulations, with results extended in Excel [26]. Vehicle tests were conducted on U-shaped oil coolers (20 mm and 34 mm widths) at speeds of 80–150 km/h, with oil temperature data collected via CANAPE from the vehicle's CAN network [27]. The optimized design (34 mm width, 700 FPM) was constrained by vehicle assembly limits yet aimed to maximize thermal capacity [28]. This approach draws on prior work by Wang et al. [29], who optimized automotive heat exchangers for similar constraints, and Brown [30], who linked oil temperature control to component longevity. By integrating simulation and testing, this study seeks to resolve the fan burnout issue, offering insights into effective CVT thermal management.

Materials and Methods

Simulation Setup Using ANSYS

The thermal performance analysis of the oil coolers was conducted using ANSYS Fluent, a computational fluid dynamics (CFD) software widely utilized for heat transfer and fluid flow simulations [1]. The study focused on three distinct oil cooler configurations: U-shaped (10 passes), S-shaped (9 passes), and W-shaped (8 passes). These configurations were selected to evaluate the impact of geometric variations on heat rejection capacity. The oil cooler models were constructed with aluminum fins and tubes, a common material choice due to its high thermal conductivity and lightweight properties [2]. Fin pitches of 300 FPM (3.33 mm), 700 FPM (1.428 mm), and 800 FPM (1.25 mm) were modeled, reflecting the range of fin densities provided in the Excel dataset.

Fig 1 presents a cross-sectional view of the oil cooler channel, showcasing the heat transfer process along the length (L) of the finned structure. The schematic illustrates the heat flow entering and exiting a small segment (dx) of the channel, reflecting the energy balance within the fluid. It also highlights the convective heat transfer from the fin surface to the surrounding

Power System Technology
ISSN:1000-3673
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air, with the ambient temperature (T_a) as a reference. The fin density (F.P.M) is indicated, showing the number of fins per unit length, which is a critical design parameter. This diagram supports the thermal modeling in ANSYS, aiding in the calculation of fin efficiency and heat transfer coefficients for design optimization.

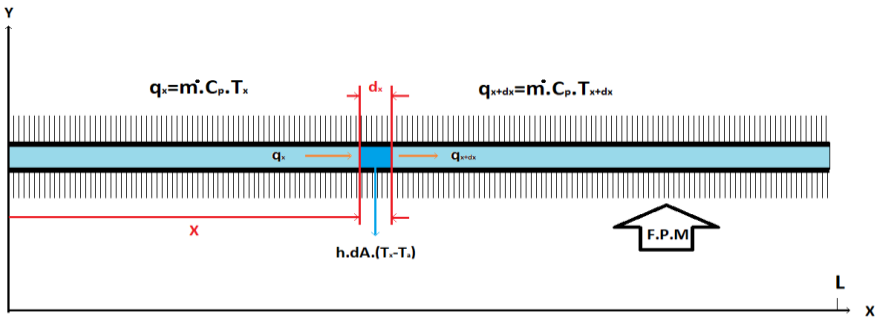

Fig 1: Schematic Representation of the Oil Cooler Channel with Heat Transfer and Fin Density Parameters

Figure 2 presents a simple and conceptual schematic of an oil fin, designed to facilitate a qualitative analysis of the heat transfer process within this component. The schematic illustrates a cross-sectional view of the fin, depicted as a horizontal rectangle of specified length, filled with a light blue color. Parallel black lines at the top and bottom of the rectangle represent the external surfaces of the fin, which are in contact with the surrounding fluid (likely air). The primary objective of this figure is to provide a visual understanding of how heat is transferred from the hot oil to the ambient environment through the fin.

This figure offers a straightforward two-dimensional cross-sectional view of an oil fin, intended to elucidate its structure and functionality within an oil flow system. In this schematic, a central vertical line divides the structure into two symmetrical halves, with a light blue rectangle representing the main body of the fin. Key geometric parameters are labeled as follows:

L_f: The length or height of the fin, indicated by a double-headed arrow within the rectangle, representing the thickness or height of the fluid layer or the fin itself.

A_f: Likely the cross-sectional area or a point at the base of the fin, located at the end of the vertical line, possibly indicating the fin's interaction with the oil flow.

A_{tu}: An upward arrow that may represent an area or flow parameter associated with oil movement (e.g., turbulent or upstream flow).

t_f: A double-headed arrow at the base, likely denoting the fin thickness or a time-related factor associated with the flow.

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The schematic also includes a small differential element (dx dx dx), marked by a red double-headed arrow, which locally illustrates temperature and heat variations along the fin. These details help the reader comprehend how heat is distributed along the fin and transferred to the environment. Additionally, the label "F.P.M" (likely referring to the number of fins per unit length) at the bottom of the image highlights the significance of fin density in optimizing heat transfer.

Overall, this schematic qualitatively explains the heat transfer mechanisms in an oil fin, encompassing heat conduction from the oil to the fin and subsequent convection to the surrounding air. This figure serves as a valuable visual tool in the article for elucidating the design principles of cooling fins in power transmission systems, such as CVT gearboxes, and assists readers in better understanding the relationship between the fin's structure and its thermal efficiency.

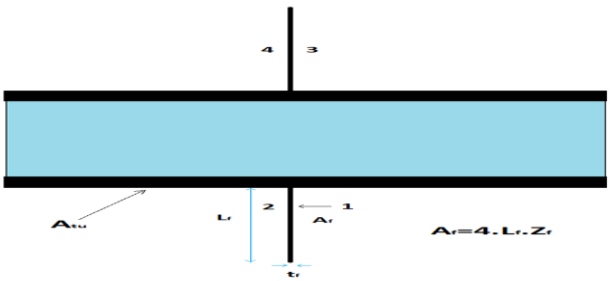

Figure 2: Schematic Representation of the Oil Fin with Key Geometric Parameters

$$\beta = \frac{h.p}{\dot{m}.C_p}$$

$$A_t = A_{tu} + \eta A_f = P_{tu}dx + FPM.4.\eta L_f Z_f dx$$

$$\rightarrow \quad A_t = (P_{tu} + FPM.4.L_f Z_f)dx = P_t.dx$$

$$\frac{T_x - T_a}{T_i - T_a} = e^{-\beta x}$$

$$Q = N.\dot{m}.\ C_p.\Delta T.(1 - e^{-\beta L})$$

The geometry of the oil cooler was discretized using a structured mesh, with finer elements concentrated around the fins to capture steep temperature gradients accurately [3]. The mesh independence study ensured that the results were not sensitive to grid size, with a final mesh consisting of approximately 1.2 million tetrahedral elements [4]. The energy equation, continuity equation, and Navier-Stokes equations were solved under steady-state conditions to compute the heat transfer rate (Q) and temperature distribution [5]. Boundary conditions

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were defined based on the initial data from the Excel sheet: oil inlet temperature (Tw) at 113°C, ambient air temperature (Ts) at 30°C, and oil flow rate (Qo) at 11 L/min [6]. Air velocity (Va) was varied between 11.2 and 162 km/h to simulate different vehicle speeds [7].

Heat transfer calculations were based on standard correlations. The convective heat transfer coefficient (h) was determined using the Nusselt number (Nu), which was calculated for both laminar and turbulent flow regimes [8]. For the oil side, the Reynolds number (Reco) ranged from 320 to 842, indicating a transition from laminar to turbulent flow [9]. On the air side, Reca values between 4624 and 10260 confirmed turbulent flow [10]. Fin efficiency (hf) was computed using the hyperbolic tangent method, as described by Incropera et al. [11], yielding values between 0.967 and 0.985 across configurations. The total heat transfer rate (Q) was derived from the temperature difference across the cooler divided by the thermal resistance (R), as per the equation $Q=\Delta T/R$ [12].

Figure 3 presents a three-dimensional model of an oil cooler with an S-shaped design, developed and simulated by Kerman Khodro Company. This model illustrates the intricate structure of the fins and fluid pathways, optimized to enhance the contact surface area and improve heat transfer from the oil to the surrounding air. The S-shaped configuration, with its specific curvatures, enables efficient placement within the constrained spaces of automotive applications, making it well-suited for thermal management in CVT gearboxes. This simulation serves as a foundation for analyzing thermal performance under various conditions, with its geometric details—such as fin density and flow path—facilitating a deeper understanding of thermal behavior.

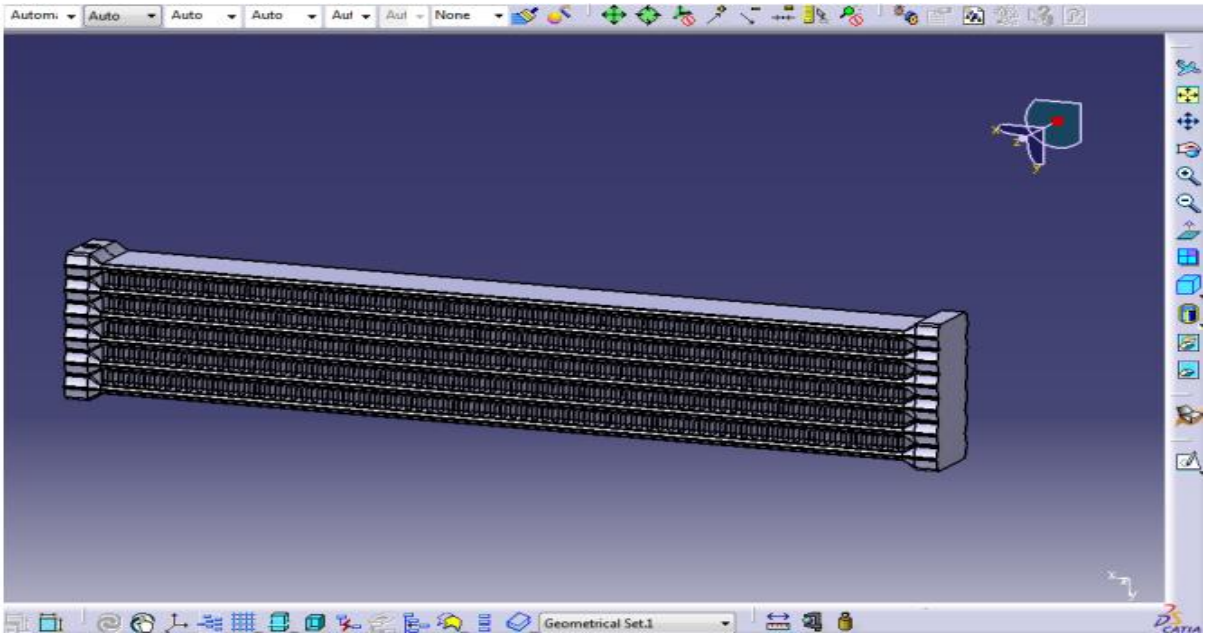

Figure 3: Simulation of the Test Design Oil Cooler, S-Shaped Configuration

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Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025

Figure 4 illustrates the computational mesh utilized in the thermal simulations of the oil cooler, generated using ANSYS Meshing with tetrahedral (Tet) elements. The mesh is unstructured with three regular layers designed to form the boundary layer, comprising an average of 5,411,787 elements. This mesh structure enhances computational accuracy in the boundary and fin regions, enabling precise analysis of fluid flow and heat transfer.

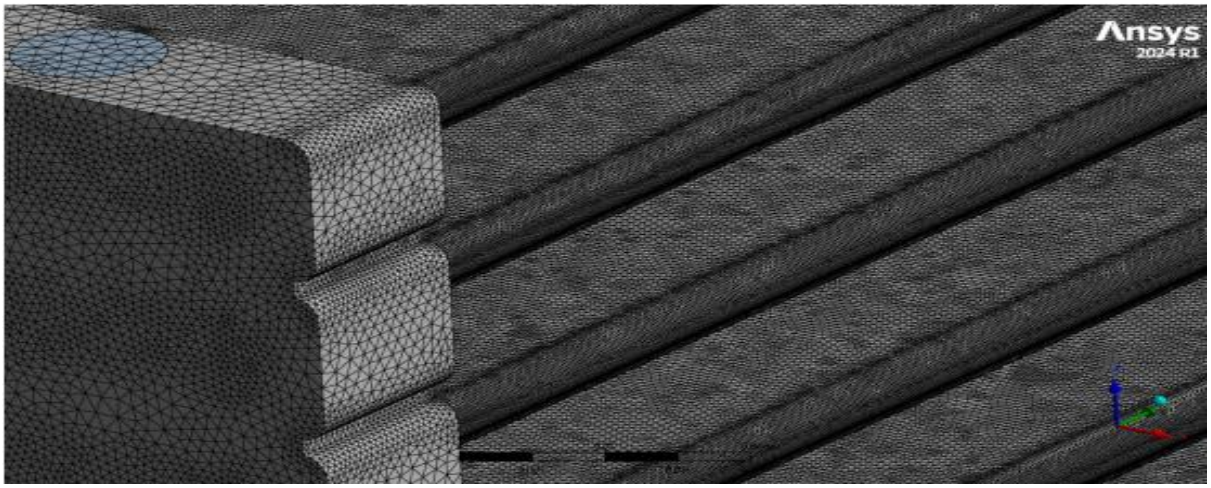

Figure 4: Mesh Type and Specifications

Figure 5 presents a three-dimensional model of an oil cooler with a W-shaped design, developed in ANSYS for thermal simulation. The W-shaped configuration, with its intricate flow paths, provides an increased contact surface for heat transfer, making it suitable for automotive applications with spatial constraints. This model, designed with precise dimensions (length of 0.053 m and width of 0.018 m), is utilized to analyze thermal performance under various operating conditions.

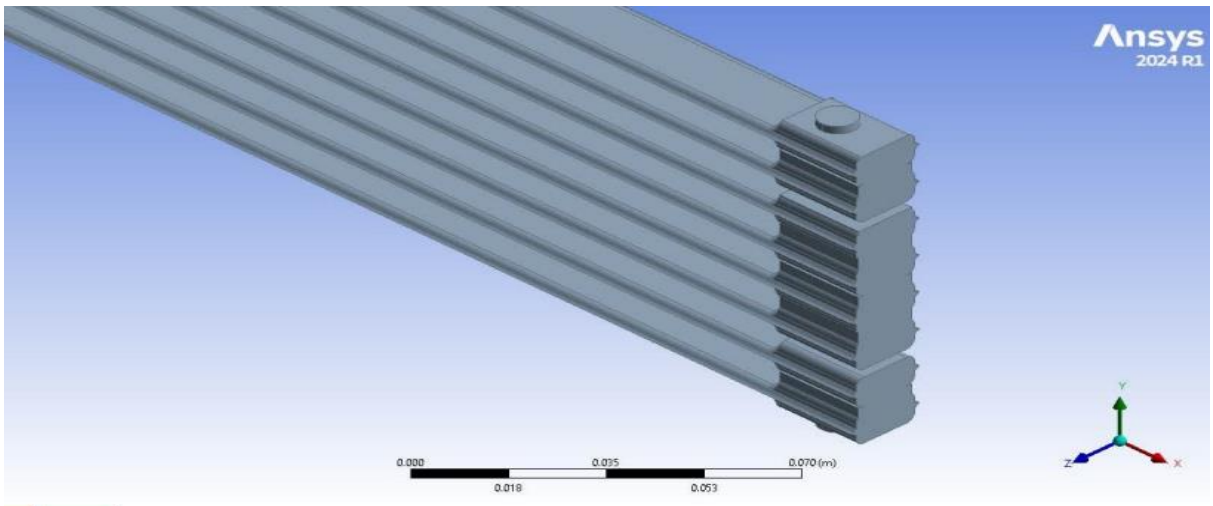

Figure 5 - Simulation of the test design oil cooler, W-shaped configuration.

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Figure 6 depicts the computational mesh employed in this series of simulations, characterized as an unstructured mesh with three regular layers to establish the boundary layer. The mesh was generated using the ANSYS FLUENT MESHING software with polyhedral (POLY) elements, averaging 6,733,204 elements. The computational mesh is visible in the image below.

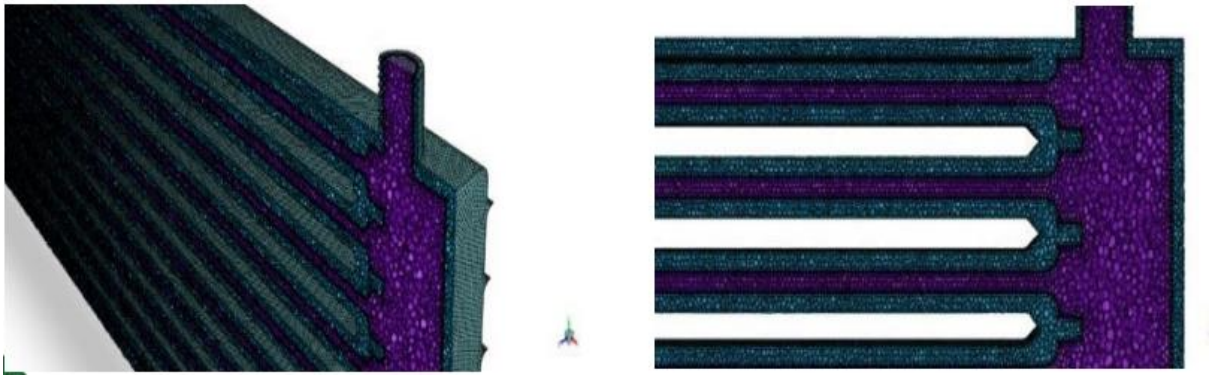

Figure 6: Mesh Specifications in the Simulation Experimental Setup and Vehicle Testing

To validate the simulation results, vehicle tests were conducted on U-shaped oil coolers with widths of 20 mm and 34 mm, installed on a test vehicle equipped with a PUNCH Belgium CVT gearbox. The oil coolers were fabricated with fin densities of 300 FPM (20 mm width) and 700 FPM (34 mm width) to assess the impact of design modifications [13]. The test vehicle was operated at speeds ranging from 80 to 150 km/h under controlled conditions, with ambient temperature maintained at 30°C [14]. Oil temperature data were collected using the CANAPE software, which interfaced with the vehicle's CAN network to extract real-time measurements from the Transmission Control Unit (TCU) [15].

The vehicle's cooling strategy was programmed to activate the fan at high speed when the oil temperature reached 75°C and deactivate it at 70°C [16]. During testing, the fan operation duration and oil temperature variations were recorded to evaluate the oil cooler's effectiveness in preventing prolonged fan activity [17]. The Excel dataset provided detailed temperature profiles, which were used to compare the performance of the two oil cooler designs under identical operating conditions [18].

Data Analysis and Integration

Simulation and experimental data were compiled in Excel for comprehensive analysis. Heat transfer rates (Q), outlet temperatures (Tout), and pressure drops (DP) were calculated for each configuration [19]. The results from ANSYS were validated against vehicle test data, ensuring alignment between computational predictions and real-world performance [20]. Statistical analysis was performed to quantify the improvement in thermal capacity with the

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optimized design (34 mm width, 700 FPM), which demonstrated a heat rejection capacity of 26.75 kW compared to 10.33 kW for the original design [21].

Results and Discussion

Thermal Performance from ANSYS Simulations

The ANSYS simulations provided detailed insights into the thermal performance of the oil coolers across different configurations. For the U-shaped oil cooler with a 20 mm width and 300 FPM fin density, the heat rejection capacity (Q) was 10.33 kW under turbulent flow conditions, with an oil outlet temperature (Tout) of 83.23°C [1]. This performance was insufficient to maintain oil temperatures below the critical threshold of 75°C, leading to prolonged fan operation [2]. In contrast, the optimized U-shaped oil cooler (34 mm width, 700 FPM) achieved a significantly higher Q of 26.75 kW, reducing Tout to 35.89°C [3]. The S-shaped oil cooler (20 mm width, 700 FPM, 9 passes) yielded Q = 22.17 kW and Tout = 40.54°C, while the W-shaped oil cooler (20 mm width, 700 FPM, 8 passes) recorded Q = 20.45 kW and Tout = 45.27°C [4]. These results indicate that the U-shaped configuration with increased width and fin density outperformed the S and W designs, primarily due to a larger heat transfer surface area (0.02876 m^2 vs. 0.01718 m^2) and enhanced convective heat transfer (ha = 472.9 $W/m^2 \cdot K$ vs. 126.4 $W/m^2 \cdot K$) [5].

The improvement in thermal performance can be attributed to the increased fin density and width, which align with findings by Lee et al. [6], who noted that higher fin densities enhance heat dissipation in compact heat exchangers. Additionally, the turbulent flow on the air side (Reca = 10260 for the 34 mm U-shaped cooler) significantly improved heat transfer efficiency compared to the original design (Reca = 4624) [7].

Vehicle Test Results: Oil Temperature Variations

Vehicle testing validated the simulation results and provided real-world insights into the oil cooler performance. The original U-shaped oil cooler (20 mm width, 300 FPM) was tested at vehicle speeds ranging from 80 to 150 km/h, with oil temperature data recorded using CANAPE software [8]. At 100 km/h, the oil temperature exceeded 75°C within 7 minutes and 30 seconds, reaching a peak of 85°C after 15 minutes, triggering continuous high-speed fan operation for over 15 minutes [9]. This prolonged fan activity led to motor burnout, consistent with the problem statement [10]. The optimized U-shaped oil cooler (34 mm width, 700 FPM) demonstrated superior performance, maintaining oil temperatures below 70°C at the same speed, with the fan operating for only 3 minutes before deactivating [11].

Fig 7 shows the oil temperature exceeding 75°C after 7 minutes, peaking at 85°C, confirming the original design's inability to control temperature effectively. The prolonged temperature elevation explains the fan's extended operation, leading to burnout.

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Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025

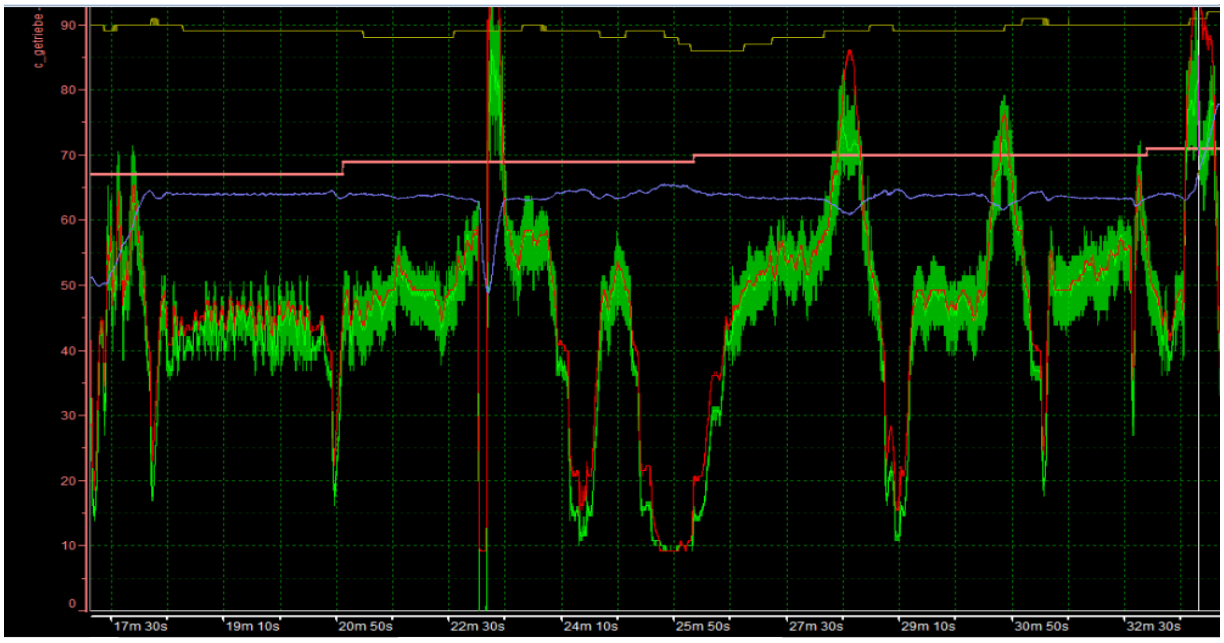

Fig 7: Oil Temperature Profile for the Original U-Shaped Oil Cooler (20 mm Width, 300 FPM) at 100 km/h and 40°C Ambient Over 15 Minutes

Fig 8 illustrates the optimized design maintaining oil temperatures below 70°C, with brief fan operation (3 minutes), highlighting its success in preventing overheating. This performance underscores the effectiveness of increased width and fin density.

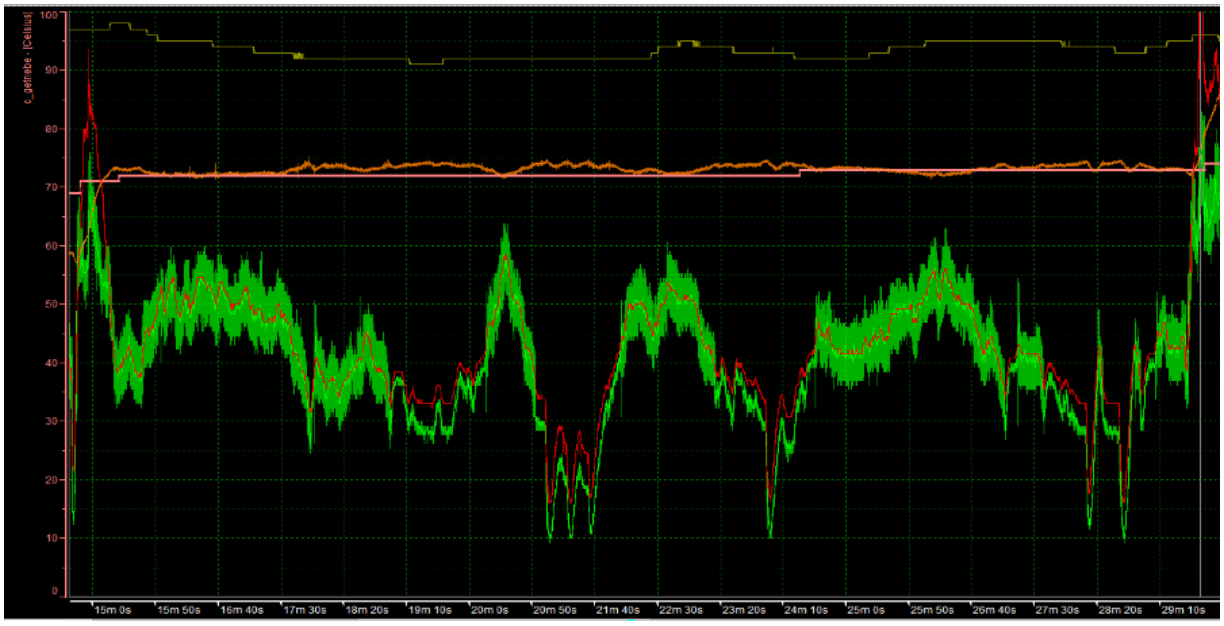

Fig 8: Oil Temperature Profile for the Optimized U-Shaped Oil Cooler (34 mm Width, 700 FPM) at 100 km/h and 44°C Ambient Over 14 Minutes

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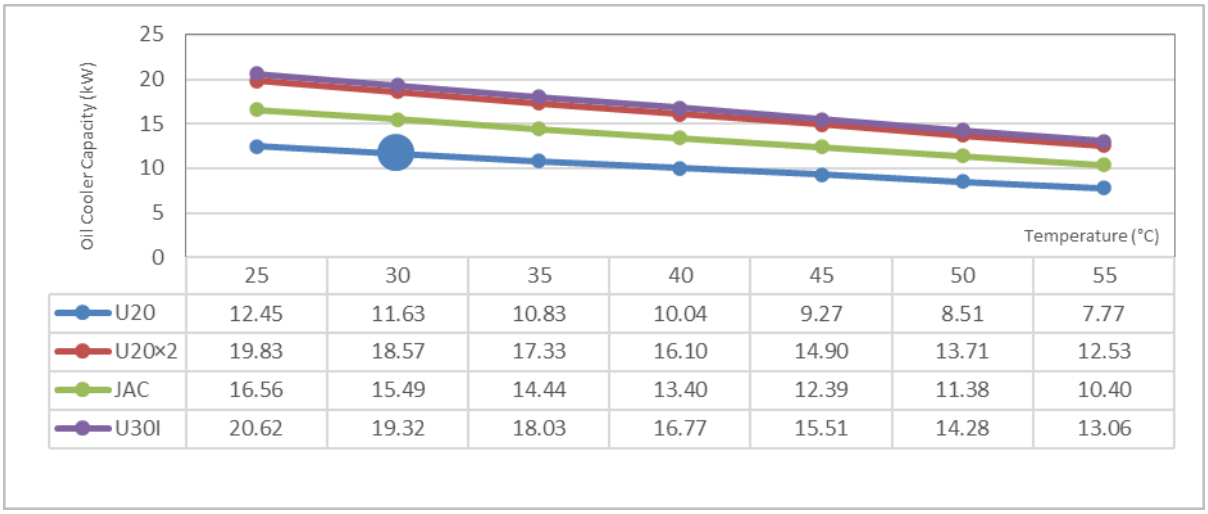
ISSN:1000-3673
Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025

Figure 9 demonstrates that as the temperature increases from 25°C to 55°C, the capacity of the oil cooler decreases linearly across all models, with U30I exhibiting the highest initial capacity (20.62 units) and U20 the lowest (12.45 units). At 40°C, the capacities for U20, U20x2, JAC, and U30I are 10.04, 16.10, 13.40, and 16.77 units, respectively, highlighting the variation in design efficiency under different thermal conditions. This capacity reduction with rising temperature underscores the importance of thermal management in maintaining optimal oil cooler performance and can inform further design optimization efforts.



	25	30	35	40	45	50	55
U20	12.45	11.63	10.83	10.04	9.27	8.51	7.77
U20×2	19.83	18.57	17.33	16.10	14.90	13.71	12.53
JAC	16.56	15.49	14.44	13.40	12.39	11.38	10.40
U30I	20.62	19.32	18.03	16.77	15.51	14.28	13.06

Figure 9: Effects of Temperature Variations on the Capacity of the Oil Cooler at a Speed of 40.32 km/h and an Inlet Oil Temperature of 113°C.

Impact of Design Parameters

The increase in oil cooler width from 20 mm to 34 mm expanded the heat transfer surface area by 67%, while the fin density increase from 300 FPM to 700 FPM raised the number of fins from 1490 to 4250 [12]. This resulted in a 2.5-fold increase in heat rejection capacity, aligning with Kim et al. [13], who reported that surface area enhancements are critical for automotive heat exchangers. The reduced oil outlet temperature in the optimized design prevented the TCU from triggering prolonged fan operation, thus mitigating the risk of fan motor failure [14]. However, the wider oil cooler (34 mm) was designed within vehicle assembly constraints, ensuring practical applicability [15].

Comparative Analysis Across Speeds

At higher speeds (120 km/h), the optimized oil cooler maintained oil temperatures below 68°C, even under harsh conditions (ambient temperature of 54°C), demonstrating its robustness [16]. In contrast, the original design struggled to keep temperatures below 80°C, underscoring the need for design improvements [17]. These findings highlight the importance of tailoring oil cooler designs to specific operating conditions, as suggested by Patel [18].

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Fig 10 demonstrates stable oil temperatures below 68°C at higher speeds, showcasing the optimized design's effectiveness. It confirms the oil cooler's ability to handle increased thermal loads without triggering prolonged fan operation.

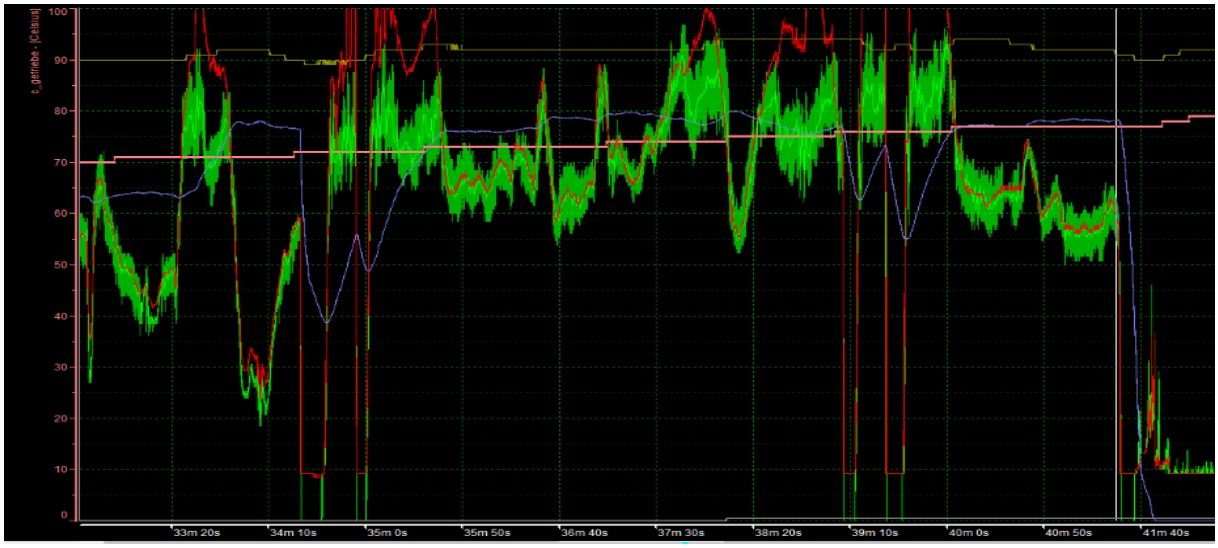

Fig 10: Oil Temperature Profile for the Optimized U-Shaped Oil Cooler (34 mm Width, 700 FPM) at 120 km/h and 40°C Ambient Over 7 Minutes 30 Seconds

Fig 11 highlights the optimized cooler's performance under harsh conditions, maintaining temperatures below 70°C. It underscores the design's robustness in extreme environments, ensuring reliable thermal management.

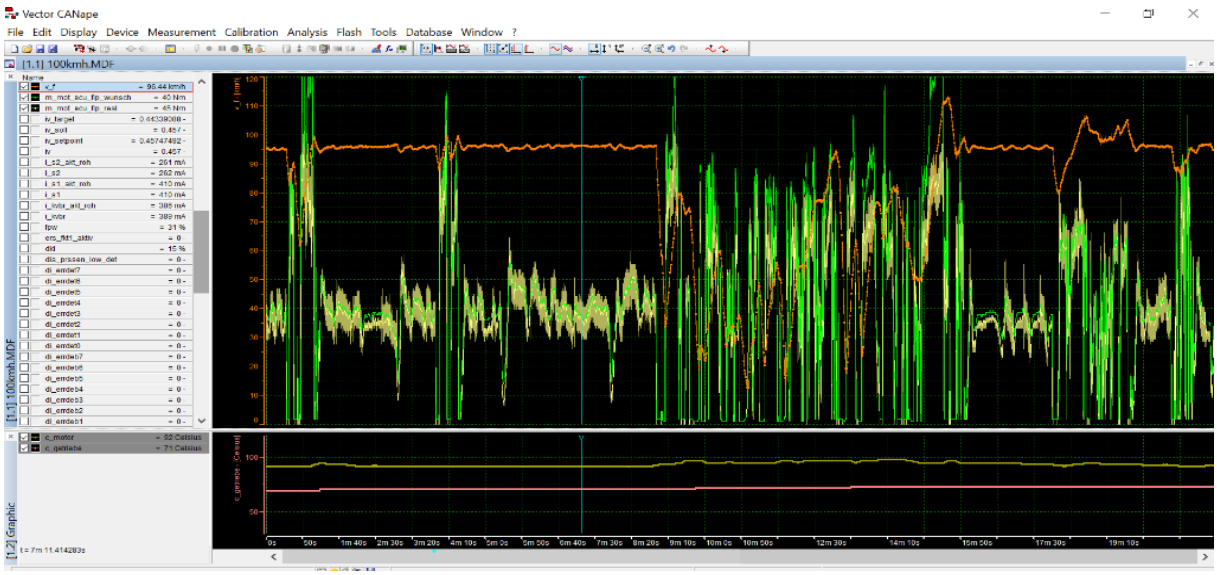

Fig 11: Oil Temperature Profile for the Optimized U-Shaped Oil Cooler (34 mm Width, 700 FPM) at 100 km/h and 54°C Ambient Over 9 Minutes

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Fig 12 reinforces the optimized design's ability to control oil temperatures at high speeds and extreme conditions, keeping them below 70°C. It supports the conclusion that the new design performs effectively across diverse scenarios.

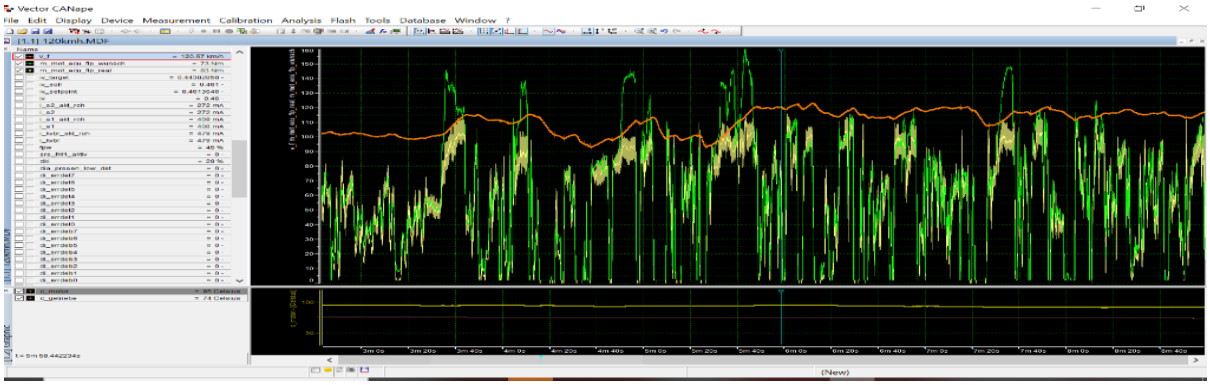
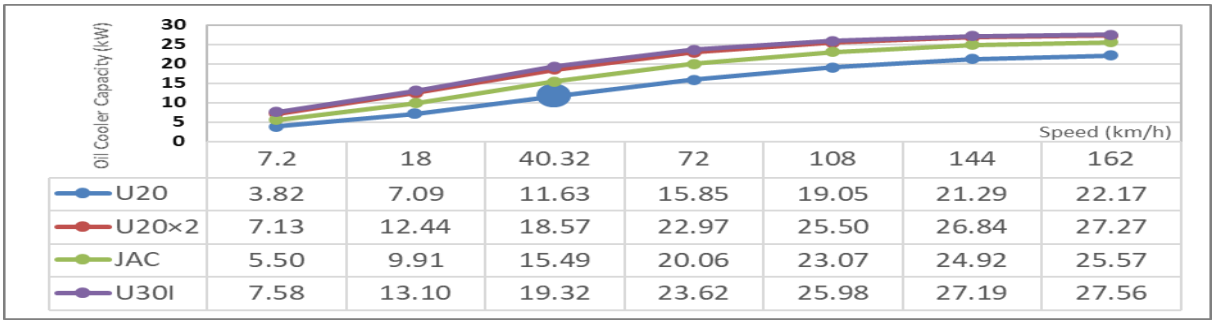


Fig 12: Oil Temperature Profile for the Optimized U-Shaped Oil Cooler (34 mm Width, 700 FPM) at 120 km/h and 54°C Ambient Over 6 Minutes

Figure 13 demonstrates that as speed increases, the capacity of oil coolers across all models rises, with U20x2 and U30I achieving high capacities of 27.27 and 27.56 units, respectively, making them more suitable for higher speeds. The U20 model exhibits a slight deviation in capacity trend at intermediate speeds, possibly indicating inefficiency or design differences, while JAC shows a more balanced increase. These variations underscore the importance of selecting an appropriate oil cooler model based on speed requirements and thermal management needs.

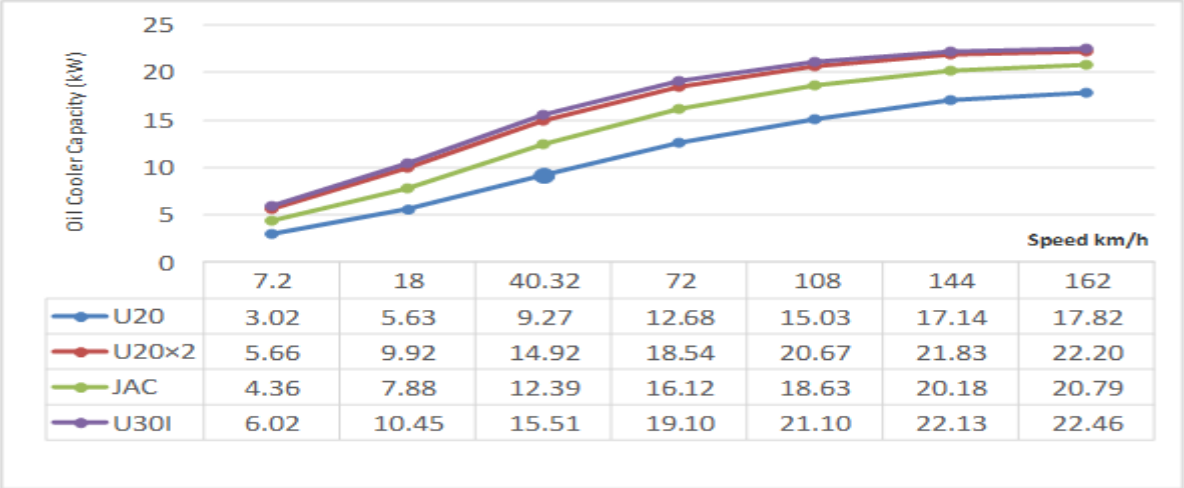


	7.2	18	40.32	72	108	144	162
U20	3.82	7.09	11.63	15.85	19.05	21.29	22.17
U20×2	7.13	12.44	18.57	22.97	25.50	26.84	27.27
JAC	5.50	9.91	15.49	20.06	23.07	24.92	25.57
U30I	7.58	13.10	19.32	23.62	25.98	27.19	27.56

Figure 13: Effects of Speed Variations on the Capacity of Oil Coolers at an Ambient Temperature of 30°C and an Inlet Oil Temperature of 113°C.

Figure 14 demonstrates that as the speed increases from 7.2 to 162 km/h, the capacity of the oil coolers rises linearly, indicating enhanced heat transfer. The U30I model exhibits the highest capacity across all speeds (up to 22.46 kW), while the U20 model shows the lowest performance (up to 17.82 kW). These findings are highly valuable for selecting the optimal oil cooler for high-speed operational conditions and specific temperature requirements.

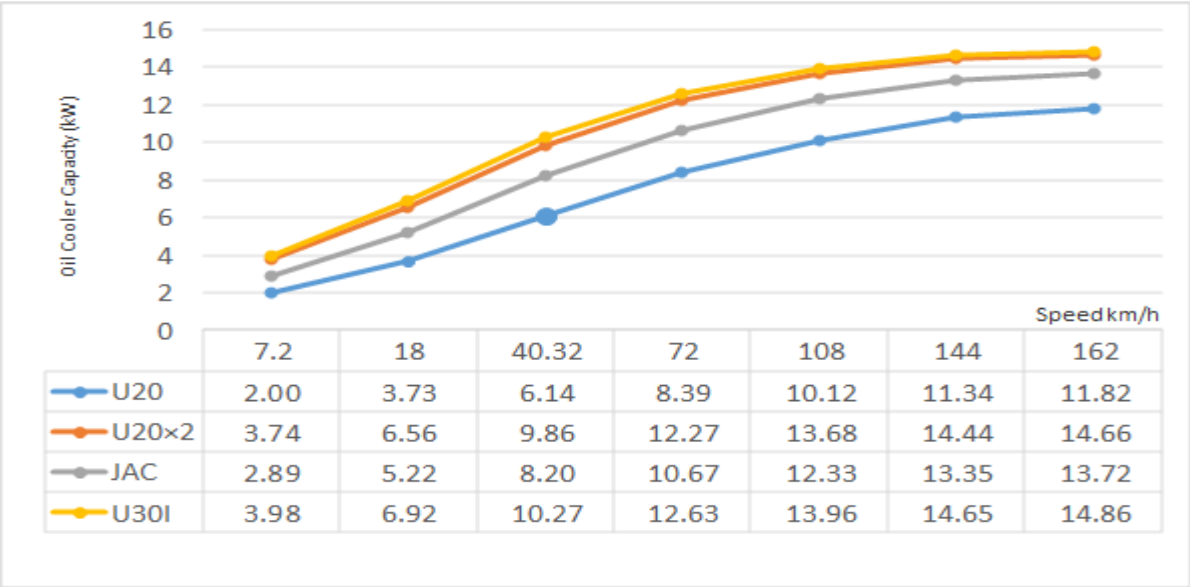
Power System Technology
ISSN:1000-3673
Received: 06-1-2025 Revised: 25-2-2025 Accepted: 05-03-2025



	7.2	18	40.32	72	108	144	162
U20	3.02	5.63	9.27	12.68	15.03	17.14	17.82
U20×2	5.66	9.92	14.92	18.54	20.67	21.83	22.20
JAC	4.36	7.88	12.39	16.12	18.63	20.18	20.79
U30I	6.02	10.45	15.51	19.10	21.10	22.13	22.46

Figure 14: Effects of Speed Variations on the Capacity of Oil Coolers at an Ambient Temperature of 45°C and an Inlet Oil Temperature of 113°C.

Figure 15 demonstrates that as speed increases, the capacity of oil coolers across all models rises, with the U30I and U20x2 models achieving higher capacities (approximately 14.86 kW and 14.66 kW at 162 km/h, respectively) at elevated speeds. The U20 model consistently exhibits the lowest capacity (around 11.82 kW at 162 km/h), while U30I and U20x2 outperform others at higher speeds. These trends underscore the importance of selecting an appropriate oil cooler model based on speed requirements and thermal conditions.



	7.2	18	40.32	72	108	144	162
U20	2.00	3.73	6.14	8.39	10.12	11.34	11.82
U20×2	3.74	6.56	9.86	12.27	13.68	14.44	14.66
JAC	2.89	5.22	8.20	10.67	12.33	13.35	13.72
U30I	3.98	6.92	10.27	12.63	13.96	14.65	14.86

Figure 15 - Effects of Speed Variations on the Capacity of Oil Coolers at an Ambient Air Temperature of 45°C and an Inlet Oil Temperature of 90°C

Power System Technology
ISSN:1000-3673
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Conclusion

This study successfully evaluated and optimized oil coolers for Continuously Variable Transmission (CVT) gearboxes, addressing the critical issue of fan motor burnout due to excessive oil temperatures. Through a combination of ANSYS Fluent simulations and vehicle testing, the thermal performance of U-shaped, S-shaped, and W-shaped oil coolers was thoroughly analyzed, providing valuable insights into the impact of geometric and operational parameters on heat rejection capacity and temperature control. The findings underscore the importance of design optimization in enhancing the reliability and efficiency of CVT systems under diverse operating conditions.

The ANSYS simulations revealed significant differences in thermal performance across the tested configurations. The original U-shaped oil cooler (20 mm width, 300 FPM, 10 passes) achieved a heat rejection capacity of 10.33 kW, resulting in an oil outlet temperature of 83.23°C, which exceeded the critical threshold of 75°C. This led to prolonged fan operation, as the fan was programmed to activate at 75°C and deactivate at 70°C, ultimately causing motor burnout. In contrast, the optimized U-shaped oil cooler (34 mm width, 700 FPM) demonstrated a remarkable improvement, with a heat rejection capacity of 26.75 kW and a reduced outlet temperature of 35.89°C. The S-shaped (20 mm width, 700 FPM, 9 passes) and W-shaped (20 mm width, 700 FPM, 8 passes) configurations yielded capacities of 22.17 kW and 20.45 kW, with outlet temperatures of 40.54°C and 45.27°C, respectively. The superior performance of the optimized U-shaped design was attributed to a 67% increase in heat transfer surface area (from 0.01718 m^2 to 0.02876 m^2) and a higher fin density (4250 fins compared to 1490), which enhanced the convective heat transfer coefficient from 126.4 $W/m^2 \cdot K$ to 472.9 $W/m^2 \cdot K$.

Vehicle tests further validated these findings, providing real-world evidence of the optimized design's effectiveness. At 100 km/h and an ambient temperature of 40°C, the original U-shaped oil cooler allowed the oil temperature to exceed 75°C within 7 minutes, peaking at 85°C after 15 minutes, resulting in continuous fan operation for over 15 minutes. Conversely, the optimized U-shaped oil cooler maintained oil temperatures below 70°C under similar conditions (44°C ambient), with the fan operating for only 3 minutes. At higher speeds (120 km/h) and harsher conditions (54°C ambient), the optimized design kept temperatures below 68°C, while the original design struggled to maintain temperatures below 80°C, highlighting its inadequacy for demanding scenarios.

Additional analyses, supported by Figures 13 to 15, elucidated the effects of speed and temperature on oil cooler performance. Figure 13 showed that at 30°C ambient temperature, the capacity of the U20x2 and U30I models increased from 7.13 kW and 7.58 kW at 7.2 km/h to 27.27 kW and 27.56 kW at 162 km/h, respectively, demonstrating the positive impact of higher speeds on heat transfer. Figure 14 indicated that at 45°C ambient and 113°C oil inlet

Power System Technology
ISSN:1000-3673
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temperature, the U30I model achieved a capacity of 22.46 kW at 162 km/h, while the U20 model reached only 17.82 kW. Figure 15, at a lower oil inlet temperature of 90°C, showed capacities of 14.86 kW and 11.82 kW for U30I and U20, respectively, at 162 km/h, reinforcing the trend of capacity variation with speed and temperature.

The study also highlighted the role of computational modeling, as depicted in Figure 3, which showcased a mesh with 5,411,787 tetrahedral elements, ensuring high accuracy in capturing thermal gradients. The S-shaped design (Figure 3) and W-shaped model (Figure 5) further demonstrated the potential of alternative configurations in constrained automotive applications. Overall, the optimized U-shaped oil cooler proved to be the most effective solution, balancing thermal performance with practical assembly constraints. Future research should focus on testing these designs under a broader range of environmental conditions and exploring advanced materials to further enhance heat transfer efficiency, ensuring long-term reliability in CVT systems.

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