Cabin Recirculation Optimization for Heating and Dehumidifying

By

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Abstract

Minimizing the power consumption of an automotive HVAC system is one of the main focuses of the automotive industry in the current day, as a reduction HVAC in power consumption can improve vehicle efficiency and provide customers with a higher quality product. During defogging operations in cold weather, the HVAC is programmed to maintain a condensation-free windshield but the question of "is the defogging operation as efficient as possible?" remains. This research develops a 1-dimensional mathematical methodology to predict the thermodynamics and psychometrics of steady-state heat exchange through the front windshield. This methodology is implemented in Siemens AmeSim software and applied for multiple possible environmental scenarios to develop a predictive simulation tool that can be used by Stellantis engineers for the purpose of improving energy efficiency of the HVAC system during vehicle development and improve the defogging strategy. Experiments conducted in a thermal wind tunnel show a strong behavioural trend between the simulation and experimental results but there is not enough data to perform an accurate correlation. Further simulations and experiments are required to fully validate the model, though the simulation could reasonably predict defogging and minimize power consumption with further development. This simulation can be expanded in the future to cover more scenarios, all window glass of the cabin, and be compatible with other HVAC control strategies to potentially accurately predict defogging controls during the development phase, or even be implemented on-board for continuous proactive steady-state defogging.

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List of Abbreviations/Symbols

1-D	1-Dimensional
AH	Absolute Humidity
COP	Coefficient of Performance
HVAC	Heating, Ventilating, and Air Conditioning Unit
RH	Relative Humidity
RAR	Recirculation Air Ratio
SC	Supercomponent

Nomenclature

Symbol	Description	Units
$\boldsymbol{\rho}$	Density	kg/m^3
$\boldsymbol{k}$	Thermal conductivity	$W/m*K$
$\boldsymbol{A}$	Area	m^2
$\boldsymbol{\mu}$	Dynamic viscosity	$kg/m*s$
$\boldsymbol{L}$	Effective length	m
$\boldsymbol{v}$	Velocity	m/s
$\boldsymbol{T}$	Temperature	℃
$\boldsymbol{R}$	Thermal resistance	K/W
$\boldsymbol{t}$	Thickness	m
$\boldsymbol{c_p}$	Specific heat capacity	$kJ/kg*K$
$\dot{\boldsymbol{Q}}$	Heat flux	W
$\dot{\boldsymbol{q}}$	Solar irradiance	W/m^2
$\boldsymbol{\varepsilon}$	Transmittance	%
$\dot{\boldsymbol{V}}$	Volumetric flow rate	m^3/s
$\boldsymbol{h}$	Heat transfer coefficient	W/m^2*K
$\boldsymbol{Re}$	Reynolds number	
$\boldsymbol{Pr}$	Prandtl number	
$\boldsymbol{Nu}$	Nusselt number	

Subscript	Description
Cond	Conduction
Int Conv	Internal convection
Ext Conv	External convection
Cabin	Cabin
Rad	Radiation
Solar	Solar radiation
Dew	Dew point
Surface	Windshield inner surface
Evap	Evaporator
Exit	Top of the windshield; designated exit of the system

Air Curtain	Internal air stream
Ext	External; convection
Int	Internal; conduction and convection
1	Windshield outer surface
Windshield	Property relating to the entire windshield
Amb	Ambient
Total	Sum; total

Chapter 1: Introduction

1.1 Context

This research explores the development and results of a 1-D simulation that mathematically predicts the optimal defogging strategy for given external conditions. This simulation is for Stellantis to aid in the reduction of power consumption of HVAC components in cold weather conditions.

Air recirculation is used in all modern vehicles to improve thermal efficiency and reduce energy required to heat the air. The automotive industry is seeking to improve thermal efficiency of vehicles to improve fuel economy of combustion engine vehicles and extend the range of battery electric vehicles. The automotive industry is seeking methods to mathematically predict defogging and develop software strategies to reduce energy consumption during defogging to improve the quality of their products.

This research progresses Stellantis' goal of providing consumers vehicles that are more efficient, improving performance, reducing operational costs, and improving the perceived quality of their vehicle.

This research is provided to Stellantis engineers at the design and development stages and is not known or a concern to the consumer. Engineers can use the simulation developed in this research as a predictive tool to design more efficient HVAC strategies. This proactive approach reduces the reliance on reactive strategies and avoid fogging from occurring before it becomes an issue for the driver, improving the quality of life for the driver due to not needing to interface with the HVAC controls to initiate defogging.

Research into predictive simulations is becoming increasingly prevalent in the modern automotive industry as more vehicle manufacturers seek simulation tools to cut back on expenses related to development, testing, and prototyping. Through potential cutbacks this research has the potential to save Stellantis and other manufacturers money compared to experimental methodologies.

The 1-D nature of this simulation reduces the time required by Stellantis engineers to perform simulations which can further save money compared to 2-D and 3-D simulations.

This simulation can be applied to all vehicles in Stellantis' current and future vehicle line-up, regardless of drivetrain, body shape, HVAC hardware, or class. An improvement in efficiency across all vehicles offered can result in a tangible benefit in reduction in emissions emitted by combustion vehicles or longer ranges by electric vehicles if they expend less energy on the HVAC in cold-weather conditions.

1.2 Objectives

1.2.1 Desired outcome

The desired outcome of this research is to find the optimal settings to defog the windshield for a set of given external conditions. Repeating this process for multiple conditions provides information to engineers about the theoretical optimal settings for defog control software and design. Being able to mathematically predict defogging with minimum energy consumption can lead to an increase in vehicle thermal efficiency.

The final objective is a 1-D simulation that can provide useful information to Stellantis' engineers for the purpose of reducing HVAC power consumption in defogging conditions.

1.2.2 Constraints

Since there are many variables that can be adjusted, some have been fixed as constants to simplify the analysis and fit within the time constraints of the research.

External temperatures considered range from 5 °C to -30 °C, the range defogging is likely to be experienced.

The study is only being done on steady state conditions, where the defogging system has already removed initial condensation from cold-start and has been running for at least ten minutes. The ten-minute target is a commonly adopted industry standard that is based on experimental studies [1], observation, and conservative defogging times for the 5-minute Canadian government-regulated warm-up period [2]. The study is not to look at removing condensation but to maintain a condensation-free state.

The research must make use of existing hardware, as this should be a software control system.

1.3 Benefit of research to Stellantis and Canada

Stellantis benefits from this arrangement since it is able to directly implement this defog control strategy into their future vehicles upon completion, and it is in a modelling software the company has access to and has many members currently using it. Stellantis has a cabin humidity model developed with a defog strategy attached, however the current strategy is a pre-programmed reactive model and Stellantis requires a predictive model for their future vehicles.

The Canadian community benefits from this research since the objective is to reduce power consumption by the HVAC system during the defogging strategy which only occurs during cold weather. The reduced power consumption improves range of electric vehicles which should increase electric vehicle adoption and improve the range of electric vehicles in Canada. This control strategy is not independent to electric vehicles, as it can be implemented on combustion and hybrid vehicles as well, which slightly reduces their fuel consumption.

1.4 Thesis Outline

Chapter 2 covers the technical background needed to understand how defogging works, how an automotive HVAC system operates, and an investigation into existing relevant research in this field.

Chapter 3 covers the methodology used to mathematically calculate the optimal defogging settings and how it minimizes power consumption.

Chapter 4 covers the software used in the simulation, and how the methodology is implemented into the software.

Chapter 5 covers the experimental procedures and how the experiments contribute to the simulation.

Chapter 6 covers the results of both the simulation and experimental data, interpretation of the results, and important information discerned from the results.

Chapter 7 covers the recommendations for future work and briefly concludes the results of the research.

Chapter 2: Background

2.1 Defogging process

In cold regions the issue of condensation forming on vehicle window glass is a well-known problem that results from cold air or cold rain chilling the glass causing the high humidity of the internal air to condensate on the glass and is more commonly known as "fogging" [3]. Vehicle manufacturers implement "defogging" controls to prevent this issue from occurring.

Cabin air recirculation is a standard operation used in all automobiles to provide some fresh oxygen to vehicle occupants while also retaining some thermal energy, thus reducing the amount of thermal energy required to maintain a constant temperature inside the vehicle [4]. Cabin air recirculation also retains some of the humidity and if not controlled causes humidity and carbon dioxide build-up leading to uncomfortable experiences for the passengers and condensation forming inside the vehicle on cold surfaces [5].

Since the recirculated air retains its thermal energy, it is preferable to use as much of it as possible, and although with it comes the higher humidity it still saves energy and as such is rarely reduced unless more fresh air is necessary. Control of humidity is therefore crucial for preventing fogging, performed by either the recirculation door or the evaporator.

Defogging is categorized into two scenarios: cold-start and steady-state. Cold-start defogging is when the vehicle is initially turned on, and the HVAC system removes sitting condensation in the cabin and heats the windshield. Steady-state defogging begins approximately ten minutes afterwards when the windshield has reached a warm state and the cabin humidity and temperature targets have been reached [1].

2.2 HVAC system operation

HVAC layout is standard across the automotive industry due to years of experimentation, trial and error, and technical improvements that have led to a conventional design. First, fresh air and recirculated air mix where the ratio is controlled by a flap called the recirculation door, then passed through the blower motor which forces the air to move through a condenser. The condenser cools the air and removes water vapour (thus drying the air) then passing through an evaporator which warms the air back up. Finally, vent doors open and close to dictate where the treated air enters the cabin. This results in effectively five methods of controlling the humidity on the front windshield: blower fan, condenser, evaporator, recirculation door, and vent doors. The controls and their correlations are as follows: opening the recirculation door decreases humidity and temperature, increasing blower motor load increases mass flow rate of air, increasing condenser load decreases humidity, increasing evaporator load increases temperature, and opening the windshield vent door increases mass flow rate of air but having other vent doors open decreases mass flow rate of air.

It should be noted that in this research the condenser is the scientific name for the component that cools the air to condense out moisture, but in industry it is called the 'evaporator'. Similarly, the evaporator is the scientific name for the component that heat the air, but in industry it is called the ‘heater core’. Mixing up scientific and industry terminology causes confusion, so this research uses the scientific naming convention under the assumption that the air is the working fluid.

2.3 Existing defogging control methodology

Control of fog on the windshield is often implemented as a reactive system with a sensor that detects when the risk of fog is high and if the window is beginning to fog up then a defog program initiates to prevent the fog from occurring [4]. There are many variables involved in defogging programs, and with combustion cars having heat engines there has historically been a low-waste heat source for the heater core, however with electric vehicles having much less by-product heat generation, energy must be ‘wasted’ to generate the heat bringing down efficiency, often up to 34% of their capacity in freezing temperatures [6]. As such, minimizing energy use in the defogging system has often been a low priority, as the priority has often been to ensure no fogging occurs, however there are methods to reduce energy consumption such as vent door control, vent air flow patterns, and better balance of energy-consuming components [5, 7].

2.4 Existing literature

From multiple experiments performed on windshield defogging it has been found that increasing the blower load (thus increasing mass flow rate) has the largest positive effect on reducing humidity, followed by the evaporator then the heater core [5, 8]. Developing a strategy around components that have the largest performance effect on defogging may be a good start, but there should also be a consideration about the power consumption of each component and if they have some interaction between each other.

Opening the windshield vent door and keeping all other vent doors closed also ensures all the treated air is directed onto the windshield. Controlling the recirculation door is also a control method, however it allows in more cold dry air, which is beneficial for removing condensation but requires more thermal energy to heat up, so it is reserved as a last resort [5, 9, 10].

Maximum thermal energy is conserved when the recirculation door is as closed as possible while still permitting sufficient oxygen into the cabin [4, 11]. Allowing in cold, fresh air requires dumping heated air out the rear of the cabin to keep a constant mass flow rate and constant cabin pressure.

Air flow leaving the vents is a well-researched topic, as it directly impacts what regions of the windshield that receive the largest flow of dry air, speed at which defogging occurs, and efficiency. Vents are often designed as straight slits which gives the airflow a laminar flow across the windshield [7, 12].

The temperature profile across the windshield is relatively uniform in the area of the driver’s vision, with lower temperatures in the corners and outer edges, especially the top which is the farthest from the vents [13]. Designing a defog control to prevent fogging at the coldest part of the windshield means the rest of the windshield is guaranteed to not be fogged.

As a vehicle increases in speed, it is essentially punching a hole in the air as it travels, and this creates pressure that acts on the vehicle and is commonly known as the ‘ram effect’. Consequences of the ram effect include air leakage through door gaps, wind noise, and most importantly for this research it increases the air flow rate travelling through the HVAC system [14, 15]. Experimental testing of this effect is required for every vehicle platform in order to include it in a simulation, making it difficult to implement into a predictive simulation without an established correlation.

Since fogging is occurring on the surface of the windshield it is important to understand the properties of glass and why condensation is only occurring on the glass surface. Glass has relatively poor thermal conductivity compared to the insulation-lined aluminium vehicle frame, and is very thin, so a lot of heat is transferred through the glass resulting in the largest heat loss in the cabin [16].

Vehicle windshields are not purely glass but two layers of treated glass with a thin membrane sandwiched between to improve strength, but this membrane is too thin to influence thermal conductivity. Windshield glass is only at a thickness of about 6.5 millimeters, and with a large temperature difference on either side it is prone to large temperature fluctuations through the windshield. The windshield is transparent, as such it allows almost all solar radiation to be transmitted through, with only a small fraction to be absorbed or reflected [10].

Vehicle dashes are typically a high-absorption black, which greatly reduces the amount of solar radiation reflected through the glass. Since glass absorbs little solar energy and has low thermal radiation emission, solar radiation may only have a small effect on the heat transfer through the glass, although it is still be considered to have a realistic model.

Other similar research involves using energy balance equations and experimental methods to determine optimal defog parameters for given situations [1]. The small number of experiments in this literature is limited likely due to time constraints for experimental methods.

Research has been done on initial warm-up, which is during the first ten minutes of vehicle operation, but this has a different set of conditions from steady-state because it actively needs to remove condensation instead of preventing condensation during steady state, thus requiring more power [7, 1].

The thickness of the condensation on the windshield is referred to as the “film thickness” and is typically thickest at the edges of the windshield, especially the bottom corners since the vents are not designed to direct airflow to the bottom corner crevices [17].

Vehicle velocity has an impact on blower fan mass flow rate, since higher velocities cause a ‘ram effect’ where air is forced through the HVAC system [4, 14]. This effect is not be considered in this research since it requires vehicle model-specific experimentation outside the scope of this research.

A closed-loop feedback system is used to continuously monitor cabin comfort and adjust the HVAC controls accordingly, such as the blower fan, the condenser, and the evaporator [18, 19]. This type of control system provides continuous feedback to a computer about the conditions in the cabin and based on a control strategy it adjusts the levels of the powered components until a desired solution is reached. This type of control system is often used for transient systems, where there is constant changes occurring. This is used in all recirculation systems, but if the system is assumed to already be steady-state the feedback system can be neglected.

The pattern of the internal airstream, which is the path of the air taken after it exits the vent onto the windshield, is crucial for distributing the thermal energy evenly across the surface of the windshield. The pattern of the internal airstream is dictated by the internal air mass flow rate, the placement of the vents, and their design [20, 21]. For simplification, the center of the windshield is approximately uniform in temperature, since that is where the focus of the internal airstream is directed, however the edges of the windshield has a non-uniform temperature because it may be out of the direct path of the internal airstream.

The windshield glass is very thin, with modern windshield glass being comprised of two thin glass panels placed parallel with a thin layer of lamination between. Due to the windshield's material properties and thinness, it is a poor thermal resistor, easily allowing heat to transfer through [16, 22]. Thermal conduction of the glass could potentially be neglected, but it is still included at the request of Stellantis for investigative purposes.

Windshield glass also has the property of being transparent, which means thermal radiation can pass through, described as having a high transmittance in scientific terms. Windshield glass has a low absorbance since most of the thermal energy is transmitted through, and it also has a low emissivity [10, 23]. Combined, the windshield glass allows most thermal radiation from the sun to pass through into the cabin and its occupants, though it does still absorb some solar energy and the effect of radiation cannot be neglected in the thermodynamic system.

The internal airflow is found to be laminar according to literature and simulations [7, 24]; however, it is possible to design the air vents to be vortex generators to produce turbulent flows which have the benefit of better air distribution at the expense of a higher heat transfer coefficient which facilitates more heat transfer near the windshield [25]. The turbulent air is beneficial for improving the performance of the defogging system but increases heat transfer through the glass increasing the thermal loss. Stellantis' current vents are designed in such a manner that the flow is intended to be laminar, so the internal air flow may be assumed to be laminar and can be validated mathematically during the research.

Several decades of continuous research and development has led to a conventional layout using a closed-loop feedback control system, and the individual effects of each component is well understood within the automotive industry [26, 27, 28].

2.5 Gap in literature

Research has been extensively performed on defogging for cold-start defogging [1] and each individual variables' impact on the thermodynamic system [5]. Research has been done on the effect of individual variables and powered components' effect on defogging performance to determine what has the most profound effect [10]. There have been many experimental studies on defogging patterns of the windshield [13] and 3-D simulations on airstreams through the cabin [7].

The gap in literature that is valuable to research is a mathematical approach to steady-state defogging for the purpose of finding the lowest power consumption possible by the system, then implemented into a simulation to consider multiple possible conditions, then perform experiments to compare the simulation with.

The address the gap, in this research three aspects of well-researched literature are combined into one investigative approach. Similar research was performed on cold-start defogging complete with mathematical and experimental analysis but an investigation into steady-state behaviour was not performed [1]. The key difference is this research focuses on the steady-state analysis, and an emphasis on minimizing power consumption instead of maximizing defogging performance.

Chapter 3: Methodology

3.0 The thermodynamic behaviour of defogging

The heat transfer through the windshield is assumed to all come from the heat lost from the internal airstream to the external environment which can be illustrated in Fig.1. This internal air is heated in the evaporator and ejected onto the bottom of the windshield. As the internal air rises along the inside of the windshield it loses heat to the outside environment, $-\dot{Q}$. When the internal air has reached the top of the windshield it becomes cold because it has lost most of its thermal energy and exits to mix with air in the cabin. The heat lost by the internal air is transferred through the windshield, heating the external air with $+\dot{Q}$.

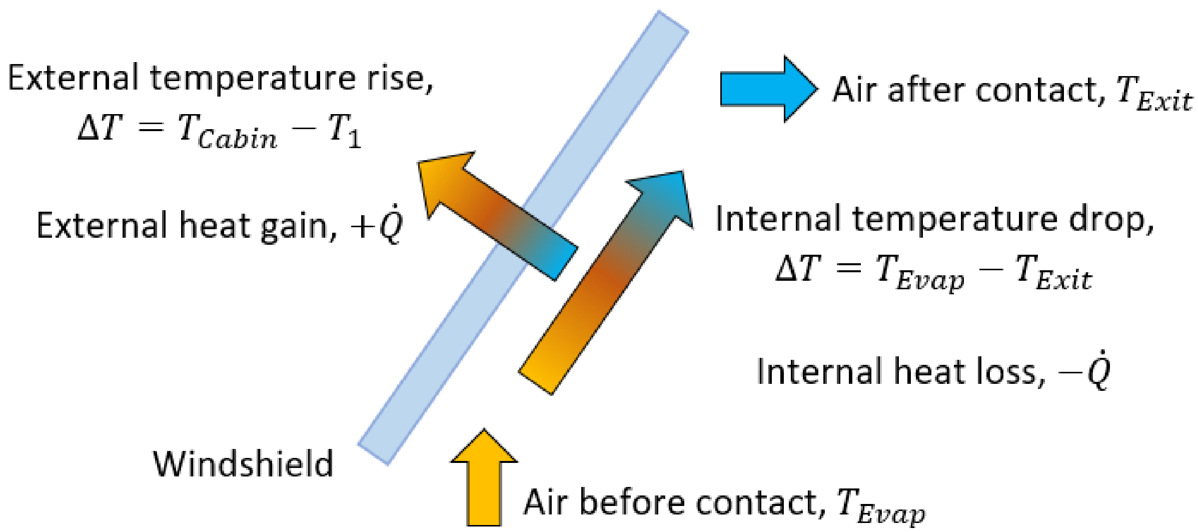

Fig. 1: Thermal equilibrium of heat transfer through the windshield

Fogging occurs when the cold internal air drops to a temperature below the dew point temperature (T_{Dew}) and 100% RH is reached. This state should always be avoided, and the T_{Exit} is the first location of fogging.

Beyond the convective and conductive heat transfers of the air and glass respectively, there exists also solar radiation. The sun heats the windshield and the internal cabin to raise all temperatures of the system, keeping T_{Exit} further from T_{Dew}.

3.1 Assumptions and simplifications

To simplify the mathematics, several assumptions have been made, listed as follows:

- The internal cabin temperature remains at a constant 25 °C. This value changes depending on user's desired cabin temperature and can be adjusted in future research. 25 °C was chosen specifically because it is easy to find data on air properties at this temperature.
- The external air flow is turbulent, which is shown when finding the external convection's Reynolds number (Re) in Chapter 3.3. External air is subject to wind, turbulent wakes from

preceding vehicles, and the design of the vehicle's front fascia, which can all cause turbulence to occur on the windshield.

- The internal air flow is laminar, which is shown when finding the internal convection's Re number in Chapter 3.3. The vents of the vehicle are designed to make the air laminar because this reduces noise from the HVAC system, as turbulence creates excessive noise at high blower fan speeds.
- Assuming all the internal airflow is produced by the blower fan, neglect the effect of air pressure on air flow through the vents due to change in vehicle velocity. More research and experiments are needed to include the 'ram effect' so for this simulation it is not included.
- Air flow exiting the vents follow a simple linear path, and do not have vortices. This assumption is what allows for a 1-D simulation, otherwise a more complex airstream requires 2-D or 3-D simulations.
- There is no debris on the windshield nor blockage of the air vents. Blocking the windshield air vents makes defogging impossible as it restricts or prevents proper air flow.
- All pressures are at atmospheric pressure, and there are no local fluctuations in pressure. Pressure could change the dew point temperature and RH of the internal air; however, to ensure a proper relationship requires experiments not conducted in this research and as a result the pressure is assumed to be constant for simplicity.
- There is no rain or other precipitation. Rain has a much higher density than air and as such has a much higher Re number and leads to higher heat transfer. An investigation into the effect of rain is not done in this research due to time restrictions.
- Neglect the effect of gravity on the air stream. Both the internal and external air streams are moving under forced convection which dominates the behaviour of the flow.
- Assume a no-slip condition on the windshield. This is a standard fluid dynamic assumption that the layer of air in contact with the windshield glass is the same relative velocity as the windshield. The no-slip condition means velocity of the internal and external air streams does not need to be adjusted for slip.
- Neglect internal radiation and solar reflection off the dash. To include this in the simulation the complex geometry of the cabin and dash needs to be known and experimentally determined, which is outside the scope of this research.
- Assume the boundary layer of the internal airflow does not allow for the mixing of the internal airstream leaving the vent and internal cabin air. The cabin air is cooler than the air exiting the evaporator and cools down the internal airstream. This assumption is valid only on the condition that the internal airstream remains laminar.
- Assume the worst-case scenario of fog occurring is at the top of the windshield, as it is the last area to receive the internal airflow. The internal airstream rejects heat to the windshield as it travels, so the final point of the farthest distance it travels must therefore be the coldest.
- The entire windshield's thermal properties are treated as glass, neglecting thermal resistance of the thin lamination. The conduction of glass is already known by Stellantis to have a very small impact on heat transfer, and the omission of a thin layer of lamination has a negligible impact on results.

- All heat input to the airflow is lost through convection with the windshield. This assumption is valid only when considering the windshield alone, but more heat may be needed to be input if considering other useful effects of the heat once it enters the cabin, such as passenger comfort.
- The windshield is a flat inclined surface to use existing empirical correlations. The curvature of the windshield is not large, and assuming the windshield to be flat allows for the use of existing correlations found in literature.
- Neglect the venturi effect of cabin air being pulled up near the windshield vent due to the low pressure of the high velocity air being pushed out the vent. This effect cools down the airstream and results in the same effect as cabin air mixing with the internal airstream. This is not included in this research since not enough experimentation or 3-D simulations have been performed to determine the magnitude of the effect this has on the internal airstream.

To further simplify the mathematics, several variables have been fixed, listed as follows:

- External relative humidity (RH) is constant at 80%. This means the only rise in humidity comes from internal sources such as passengers. All simulations use this 80% RH except for a simulation where only the external relative humidity changes are performed.
- The vents that dictate where the air is delivered are only directing airflow to the windshield. Having other vent doors open reduces the effectiveness of the internal airstream. For example, if two vent doors are open then the mass flow rate to each is halved compared to if only one vent door was open, and getting the same mass flow rate to the windshield with two vent doors open requires doubling the blower fan's mass flow rate, requiring more power, and reducing efficiency.
- Solar radiation is one of two states: 0 (nighttime) or some value to approximate the average daytime solar energy on the windshield. Solar radiation is a very complex variable that requires extensive research and current Stellantis vehicles are not equipped with sufficient sensors to determine the effect of solar radiation. An average must be used for simplification, at some expense of some accuracy for any weather conditions other than the average.

3.2 Psychrometric analysis

The thermal process is illustrated in Fig.2, beginning from the initial state (light blue circle) which is a combination of fresh and recirculated air, with a fraction determined by the recirculation air ratio (RAR) needed to maintain acceptable oxygen levels in the cabin. The condenser (dark blue arrow) cools the air to condense out some of the moisture, dehumidifying the air. The evaporator (orange arrow) reheats the air until it reaches the heated state (yellow circle) in anticipation that some heat is lost through the windshield (green arrow). An 80% RH target (red circle) is the final objective point that should be reached once the internal air has finished travelling across the windshield interior.

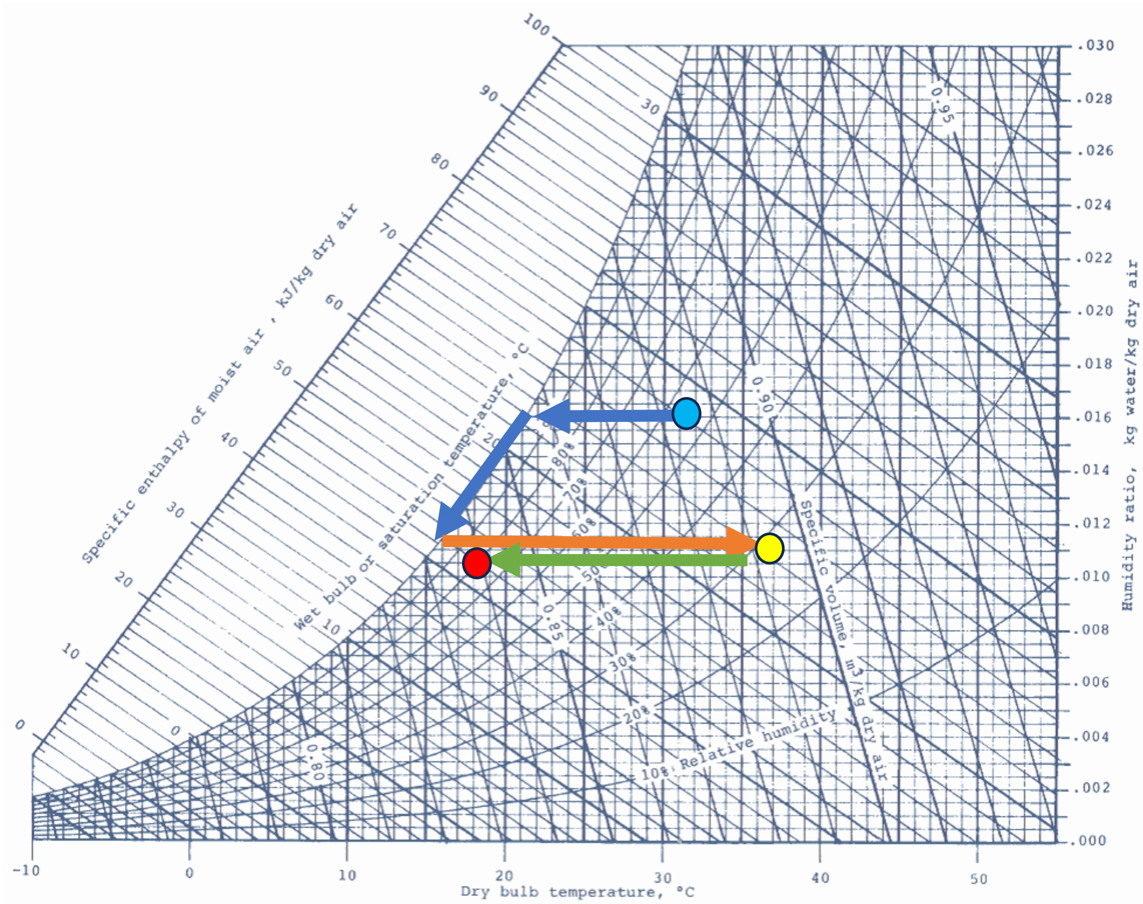

Fig. 2: Demonstration of the thermal process on a psychrometric chart

Figure 1 is merely an exaggerated example of the process and does not reflect the actual values used in the research. The values are exaggerated because the actual process is too small to clearly visualize on a graph.

When the air is at 100% RH it condenses, so the air must always remain below this level while in the cabin to prevent fogging. A factor of safety is used to avoid condensation, set at 80% RH. The target 80% RH is to be reached at the top of the windshield, where the air is coldest after losing its heat to the windshield. Operating above the 80% target runs the risk of fogging, while operating below the target means there is excessive heat being input to the air flow.

The fan, condenser, evaporator, and RAR are the only controllable variables in this simplified system, which can be adjusted to change the state of the end point.

Note that the condenser does not operate below 0 °C since the condensed water freezes in the pipes causing damage. When below 0 °C the condenser is no longer a variable, and the remaining variables must be used to control humidity. Additionally, when the condenser can no longer be used its contribution is zero, and the initial state begins at the start of the evaporator, and the humidity is controlled by the RAR.

3.3 Thermal analysis

The system is in steady state and thermal equilibrium. In equilibrium, all of the thermal energy added to the system is equal to the energy flowing out of the system, with the system being defined as the windshield, and can be simplified as a single point at the center of the windshield.

Any properties of air used in this chapter of methodology are merely an example of the values that can be used in one possible solution, but the values used for the air can be changed for a different internal or external temperature. These properties are provided in AmeSim software and do not need to be tabulated in methodology.

The equation for heat transfer is given in Eq.1 [29].

$$\dot{Q} = \frac{\Delta T}{R} \qquad \text{Eq. 1}$$

To solve the system, the resistances must be determined.

For the internal side which has two resistances, the total internal resistance can be written as Eq.2, where 'Cond' is short-form for 'Conduction' and 'Int Conv' is short-form for 'Internal Convection'.

$$R_{Int} = R_{Cond} + R_{Int\ Conv} \qquad \text{Eq. 2}$$

The resistances can be summed in this way because they act in series, demonstrated in Fig.3. Here, 'Ext Conv' is short-form for 'External Convection', 'Rad' is short-form for 'Radiation', and 'Amb' is short-form for 'Ambient'. The red arrow indicates the direction of the flow of heat through the system.

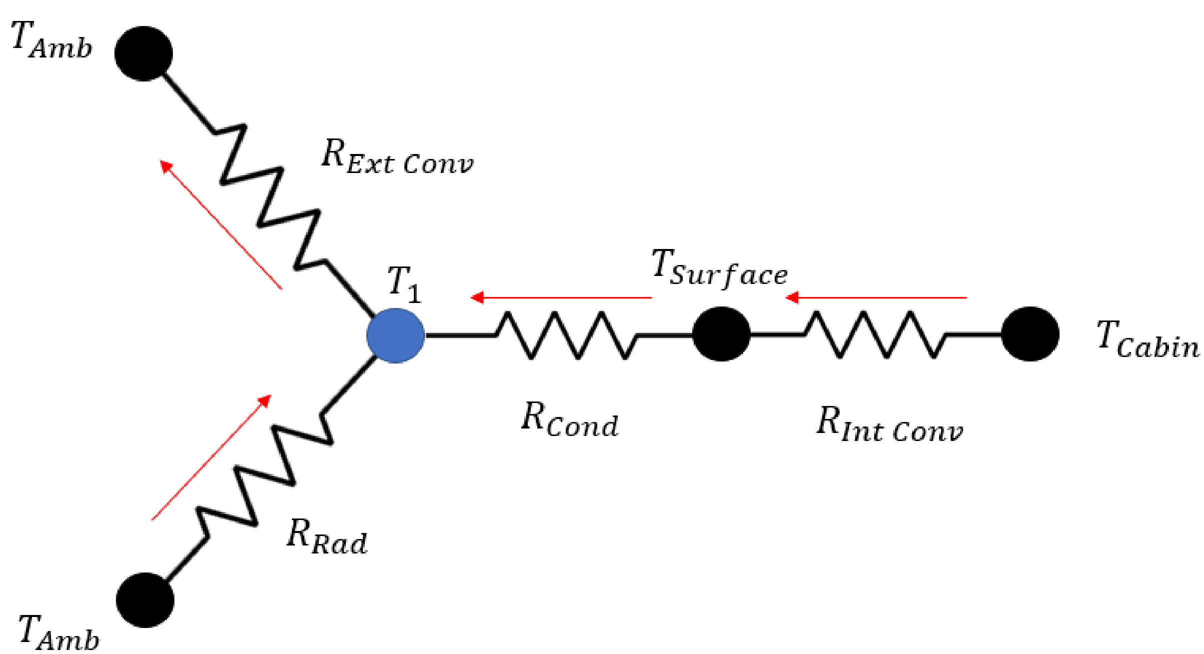

Fig. 3: Thermal resistance circuit of heat transfer through the windshield

The equation for R_{Cond} is given in Eq.3 [29].

$$R_{Cond} = \frac{t}{kA} \qquad \text{Eq. 3}$$

Where the values used in Eq.3 are provided in Table 1. These values are always constant for a given windshield model. The windshield used in this research measures approximately 0.8m high and 1.5m wide.

Table 1: Constant values used for conduction

Property	Value	Units
Windshield Thickness, t	6.35	mm
Thermal Conductivity (Glass), k	1.4 [22]	$W/m * K$
Windshield Area, A	1.2	m^2

Solving for $R_{Int\ Conv}$ using Eq.4 [29] requires a heat transfer coefficient solved for in Eq.5 [29], but it needs a Nusselt number (Nu) correlation solved for in Eq.6 [29], which also in turn needs the Reynolds number found in Eq.7 [29] and Prandtl number (Pr) found in Eq.8 [29]. Once the values from Table 2 are substituted in, the final equation for $R_{Int\ Conv}$ given in Eq.9 is only dependent on internal air velocity.

$$R_{Int\ Conv} = \frac{1}{hA} \qquad \text{Eq. 4}$$

$$h = \frac{Nu * k}{L} \qquad \text{Eq. 5}$$

$$Nu = 0.664 Re^{1/2} Pr^{1/3} \qquad \text{Eq. 6}$$

$$Re = \frac{\rho v L}{\mu} \qquad \text{Eq. 7}$$

$$Pr = \frac{c_p \mu}{k} \qquad \text{Eq. 8}$$

$$R_{Int\ Conv} = 0.1895 * v^{-1/2} \qquad \text{Eq. 9}$$

Values used in Table 2 are at a constant 25 °C and atmospheric pressure for simplicity and avoid iteration, which is necessary for more precise results. The effective length is the height of the windshield. The value 0.1895 used in Eq.9 can be replaced by a constant ‘C’ for a generalized equation.

Table 2: Constant values used for internal convection

Property	Value	Units
Density (air, 25 °C), ρ	1.184 [30]	kg/m^3
Effective Length, L	0.8	m
Dynamic Viscosity (air, 25 °C), μ	$18.37 * 10^{-6}$ [31]	$kg/m * s$
Thermal Conductivity (air, 25 °C), k	$26.24 * 10^{-3}$ [32]	$W/m * K$
Specific Heat Capacity (air, 25 °C), c_p	1.006 [33]	$kJ/kg * K$
Windshield Area, A	1.2	m^2

Air vent speeds can range from approximately 1 to 6 m/s. To maintain steady-state defogging there must always be internal airflow over the windshield.

Even at maximum airflow the Re number is below $5 * 10^5$ and therefore the flow is laminar.

The Nu number correlation used is for isothermal laminar flow on a flat plate [29], applicable for $Re \leq 5 * 10^5$ and $0.6 \leq Pr \leq 60$, where the Pr number of air is 0.707 [34].

Even with the lowest possible resistance the internal convection thermal resistance is still far more dominant than the conduction thermal resistance. The effect of conduction could be neglected, but for completeness the conduction is still be considered for this study.

When the vehicle is stationary the external convection could be natural convection with no airflow, or forced convection if there is wind, and since the forced convection is the worse case of the two for cooling the glass, only forced convection is considered. Additionally, the vehicle is not equipped with a wind speed sensor, making it impossible to determine the external airflow condition when stationary.

For forced external convection the process is very similar to internal convection but with the properties of air taken at 0 °C and a different Nu number correlation is used due to the turbulent nature of high-velocity environmental air. The properties of air slightly change depending on the external air's properties, and this is performed by AmeSim, and the values used in these calculations illustrate an example case.

Using the same Eq.7 and Eq.8 for Re number and Pr number but with new values for external air provided in Table 3 the same process is repeated with the new Nu number correlation given in Eq.10 [29]. With the same Eq.5 for heat transfer coefficient the result is Eq.11, which shows $R_{Ext\ Conv}$ is only dependent on the external air velocity.

$$Nu = 0.037 Re^{4/5} Pr^{1/3} \quad \text{Eq. 10}$$

$$R_{Ext\ Conv} = \frac{1}{hA} = 0.1244 * v^{-4/5} \quad \text{Eq. 11}$$

Values used in Table 3 are at a constant 0 °C and atmospheric pressure for simplicity and avoid iteration, which is necessary for more precise results. The value 0.1244 used in Eq.11 can be replaced by a constant 'C' for a generalized equation.

Table 3: Constant values used for external convection

Property	Value	Units
Density (air, 0 °C), ρ	1.292 [30]	kg/m^3
Effective Length, L	0.8	m
Dynamic Viscosity (air, 0 °C), μ	$17.15 * 10^{-6}$ [31]	$kg/m * s$
Thermal Conductivity (air, 0 °C), k	$24.36 * 10^{-3}$ [32]	$W/m * K$
Specific Heat Capacity (air, 0 °C), c_p	1.006 [33]	$kJ/kg * K$
Windshield Area, A	1.2	m^2

Vehicle speeds for forced convection range from 5 to 65 Mph, which converts to 5 to 104.6 km/h, equal to 2.24 to 29 m/s.

For most airflow velocities the Re number is above $5 * 10^5$ and is therefore transition or turbulent. In addition, since external air may be subject to wind or other vehicle's turbulent wakes it is assumed that the external air condition is always turbulent.

The Nu number correlation used is for isothermal turbulent flow on a flat plate [29], applicable for $5 * 10^5 \leq Re \leq 10^7$ and $0.6 \leq Pr \leq 60$, where the Pr number of air is 0.711 [34].

Solar radiation can be mathematically very complex, as it depends on variables such as the angle of the sun relative to the windshield, weather conditions, geographic position, and time of year. To simplify the solar radiation, it is equal to 0 to simulate nighttime conditions, and equal to some value for an approximate average daytime solar radiation. This average value can be found using Eq.12.

$$\dot{Q}_{Solar} = \varepsilon \dot{q} A \quad \text{Eq. 12}$$

Due to the variability of solar radiation, it is difficult to find consistent values for solar irradiance, the amount of heat transfer per unit area from the sun. The solar irradiance used is $\dot{q} = 342\ W/m^2$ [35], although further studies need to be done for a more accurate solar radiation model. All values used in Eq.12 are provided in Table 4 and are constant, with the only potential variable being solar irradiance. The chosen solar irradiance value of $342\ W/m^2$ is the global average and represents the upper limit of possible values for solar irradiance.

Table 4: Constant values used for radiation

Property	Value	Units
Solar irradiance, $\dot{q}$	342 [35]	W/m^2
Transmittance, ε	68 [36]	%
Windshield Area, A	1.2	m^2

3.4 Determination of operating points

To get the heat transfer from the resistances, Eq.13 and Eq.14 are employed.

$$\dot{Q}_{Int} = \frac{T_{Cabin} - T_1}{R_{Int}} \quad \text{Eq. 13}$$

$$\dot{Q}_{Ext} = \frac{T_1 - T_{Amb}}{R_{Ext\ Conv}} \quad \text{Eq. 14}$$

T_1 is not known, but it can be found. T_1 is the temperature on the outside of the glass, which is used instead of the inside glass temperature because the inside glass temperature can be solved for by accounting for the conduction in Eq.15 (derived from Eq.1). re-arranged into Eq.16.

$$Q_{Cond} = \frac{T_{Surface} - T_1}{R_{Cond}} \quad \text{Eq. 15}$$

$$T_{Surface} = Q_{Cond} R_{Cond} + T_1 \quad \text{Eq.16}$$

In equilibrium, the heat transfer in is equal to the heat transfer out, demonstrated in Eq.17 and expanded in Eq.18 [29].

$$\dot{Q}_{In} = \dot{Q}_{Out} \quad \text{Eq. 17}$$

$$\dot{Q}_{Int} + \dot{Q}_{Solar} = \dot{Q}_{Ext} \quad \text{Eq. 18}$$

Equation 18 is rearranged to get Eq.19, which mathematically demonstrates that the thermal energy needed for defogging is equal to the heat lost due to external air but reduced by heat from solar radiation.

$$\dot{Q}_{Int} = \dot{Q}_{Ext} - \dot{Q}_{Solar} \quad \text{Eq. 19}$$

Substituting equations Eq.13 and Eq.14 into Eq.19, then solving for T_1 gives Eq.20, with the full derivation in Appendix 1.

$$T_1 = \frac{R_{Ext\ Conv} T_{Cabin} + R_{Int} T_{Amb} + R_{Int} R_{Ext\ Conv} \dot{Q}_{Solar}}{R_{Ext\ Conv} + R_{Int}} \quad \text{Eq. 20}$$

Substituting in all the equations for heat transfer, the only remaining unsolved variable is internal velocity which can now be solved.

The next variable to solve for is the temperature the evaporator needs to achieve.

The critical temperature is T_{Exit}, which must remain above the dew point temperature while achieving the 80% RH target demonstrated in Eq.21 [37].

$$T_{Dew} = T_{Exit} - \frac{100 - RH}{5} = T_{Exit} - 4 \quad \text{Eq. 21}$$

The heat transfer through the windshield is assumed to all come from the heat lost from the internal airstream to the external environment and is assumed to be in thermal equilibrium, which can be illustrated in Fig.1 and mathematically expressed as Eq.22 [29].

$$\dot{Q} = \dot{Q}_{Windshield} = \dot{Q}_{Air\ Curtain} = hA\Delta T \quad \text{Eq. 22}$$

Equation 21 can also be expanded to be expressed as Eq.23.

$$h_{Int\ Conv}A(T_{Cabin} - T_1) = h_{Int\ Conv}A(T_{Exit} - T_{Evap}) \quad \text{Eq. 23}$$

Cancelling out the heat transfer coefficients and windshield area in Eq.23 creates Eq.24. Rearranging Eq.24 for the evaporator gives Eq. 25 which can substitute in Eq.21 to get Eq.26 which is the equation used to find the temperature the evaporator needs to raise the internal air to in order to meet the defogging requirements. A more practical version of Eq.26 is Eq.27 which eliminates the need to know the temperature of the glass and instead how much heat loss is predicted from the other existing vehicle sensors.

$$T_{Cabin} - T_1 = T_{Evap} - T_{Exit} \quad \text{Eq. 24}$$

$$T_{Evap} = T_{Exit} + (T_{Cabin} - T_1) \quad \text{Eq. 25}$$

$$T_{Evap} = T_{Dew} + 4 + (T_{Cabin} - T_1) \quad \text{Eq. 26}$$

$$T_{Evap} = T_{Dew} + 4 + \frac{\dot{Q}}{h_{Int\ Conv}A} \quad \text{Eq. 27}$$

Finally, to achieve the 80% RH target, the condenser works to achieve an absolute humidity (AH) that achieves an 80% RH at T_{Exit}. When the condenser is not operable below 0 °C, then instead the recirculation door needs to be adjusted to achieve this target.

AH is the target for the condenser because this value does not change due to the temperature increase from the evaporator or temperature loss from the windshield.

Note that although T_1 is not used in Eq.27, it is still a very useful value for experimental testing, as it is very easy to use a thermocouple to measure the temperature of the glass directly for comparison with the simulation.

3.5 Powered components' efficiencies

Each powered component needs to consume electrical power to achieve their targeted function, which can be connected using each device's efficiency curves.

The blower fan's performance curve is shown in Fig.4.

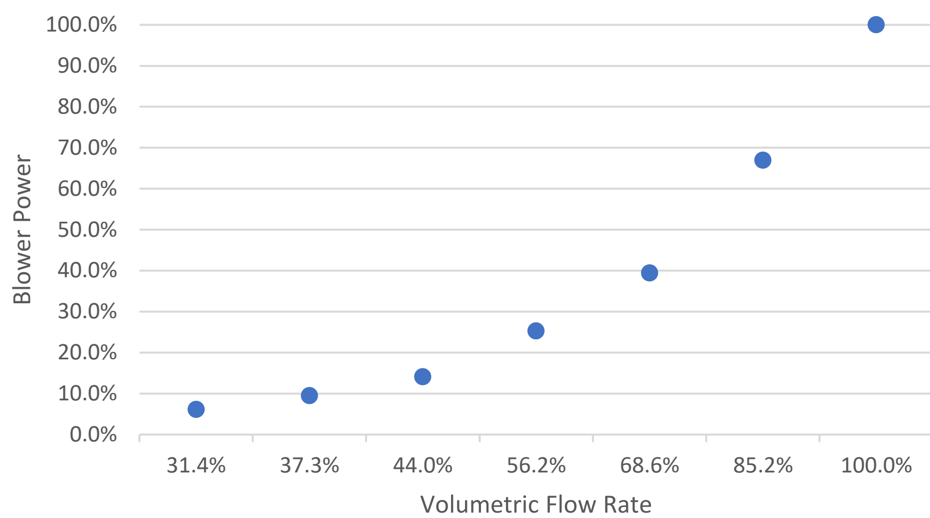

Fig. 4: Blower fan performance curve

The volumetric flow rate can be converted into velocity by dividing the mass flow rate by the area of the vents, which were measured to have a total opening area of approximately 0.025 m^2, shown in Eq. 28.

$$v = \frac{\dot{V} * \rho}{A} \quad \text{Eq. 28}$$

The coefficient of performance (COP) of the condenser pump is approximately 1.2 and the COP of the evaporator pump is approximately 1. The power can be found for the condenser from Eq.29 and for the evaporator from Eq.30:

$$P_{Condenser} = \frac{\dot{Q}_{Cond}}{COP_{Cond}} \quad \text{Eq. 29}$$

$$P_{Evaporator} = \frac{\dot{Q}_{Evap}}{COP_{Evap}} \quad \text{Eq. 30}$$

3.6 Power consumption optimization

The process for determining the overall power consumption is to set an internal air velocity and perform the steps necessary to get each component's power consumption, then sum them together in Eq.30 to get a total power consumption for that specific internal air velocity.

$$P_{Total} = P_{Blower} + P_{Condenser} + P_{Evaporator} \quad \text{Eq. 31}$$

This process is repeated multiple times for varying internal air velocities and RAR to get multiple power consumption levels. The scenario with the lowest power consumption that can defog the windshield is the optimal internal air velocity and its corresponding condenser and evaporator levels.

The optimization function for minimizing power consumption is mathematically expressed in Eq.32.

$$P_{Optimal} = min(P_{Total}) \qquad \text{Eq.32}$$

The algorithm used in the optimization function is an initial set of guesses for evaporator values followed by a binary algorithm.

Evaporator values are guessed in intervals of 100 watts until there is an RH value greater than 80% and an RH value less than 80%.

The binary algorithm guesses a value at the 50-watt halfway mark between the evaporator with corresponding RH values greater and lesser than the 80% RH target, then checks if the result is higher or lower than 80%. This process repeats for halving the value and checking if the result is higher or lower until the 1.5625 wattage mark, then takes the nearest whole number of the evaporator and condenser values that have an RH closest to the 80% target as the result.

This algorithm is simple but time consuming, though can be improved in future development of this simulation. Due to time constraints a successful automatic loop was not developed in the simulation and instead this algorithm is executed manually by the user, though this algorithm is to be made automatic for future use by Stellantis' engineers.

Chapter 4: Modelling

4.1 Siemens AmeSim 1-D software

The software used to model the mathematics and perform the calculations is Siemens AmeSim, a 1-D modelling software. The AmeSim environment is ideal for this simulation since it is a software specifically for 1-D simulations and has a 'batch simulation' feature that allows for multiple simulations to be run in parallel.

Each simulation iteration takes approximately 20 seconds to run when the internal air is below 100% RH but may take 5 minutes to run if an initial estimate results in a 100% RH. For a convergence the total number of iterations is usually around 10 and can take a total of around 30 minutes due to manual convergence.

4.2 Simulation model schematic

Figure 5 shows the procedure used by the software to perform the calculations. The hardware performance and vehicle parameters are put into the software, and external variables of interest are input. For most users the internal code should be a 'black box', where the user does not need to know how the code operates and is only interested in the result.

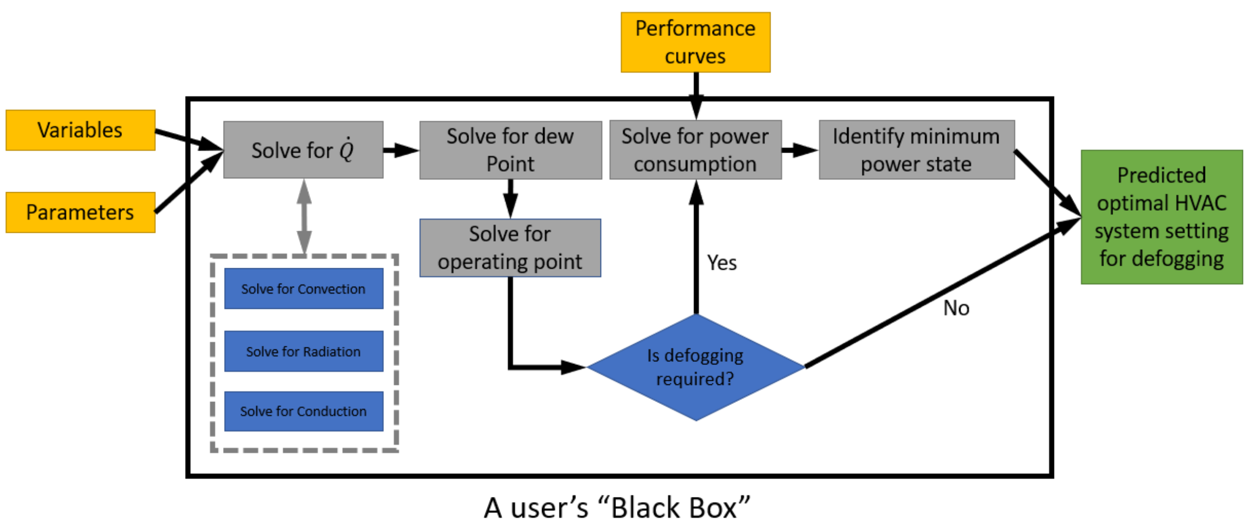

Fig. 5: Logic flow schematic of the defogging simulation

4.3 Input and output data

The input data consists of four types: internal variables, external variables, hardware performance, and vehicle parameters.

The internal variables are the blower fan flow rate which gives airflow velocity, the condenser work which gives humidity, evaporator work which gives airflow temperature, and the RAR which can be adjusted to analyze its effect. These variables are all controllable by the software.

The external variables are the external air temperature, humidity, and speed, the solar radiation effect (on/off in this simplified model), recirculation humidity depending on number of occupants. These variables are outside the control of the software.

The hardware performance is the performance curves for the blower fan, condenser, and evaporator so that their useful effect correlates to the amount of electrical power consumed to achieve the effect. These performance curves are often vehicle model specific.

Vehicle parameters are vehicle-specific measurements, depending on parameters such as the size of the windshield, thickness of the windshield glass, and its thermal conductivity.

4.4 Simulation Overview

The model is quite extensive and difficult to fit in a single figure coherently, though it can be broken down into detailed sections. AmeSim uses vector graphics to improve readability, if the image is difficult to read on a screen, enlarging the image or zooming in may help. Refer to the entire model overview in Fig.6 for how different sections connect.

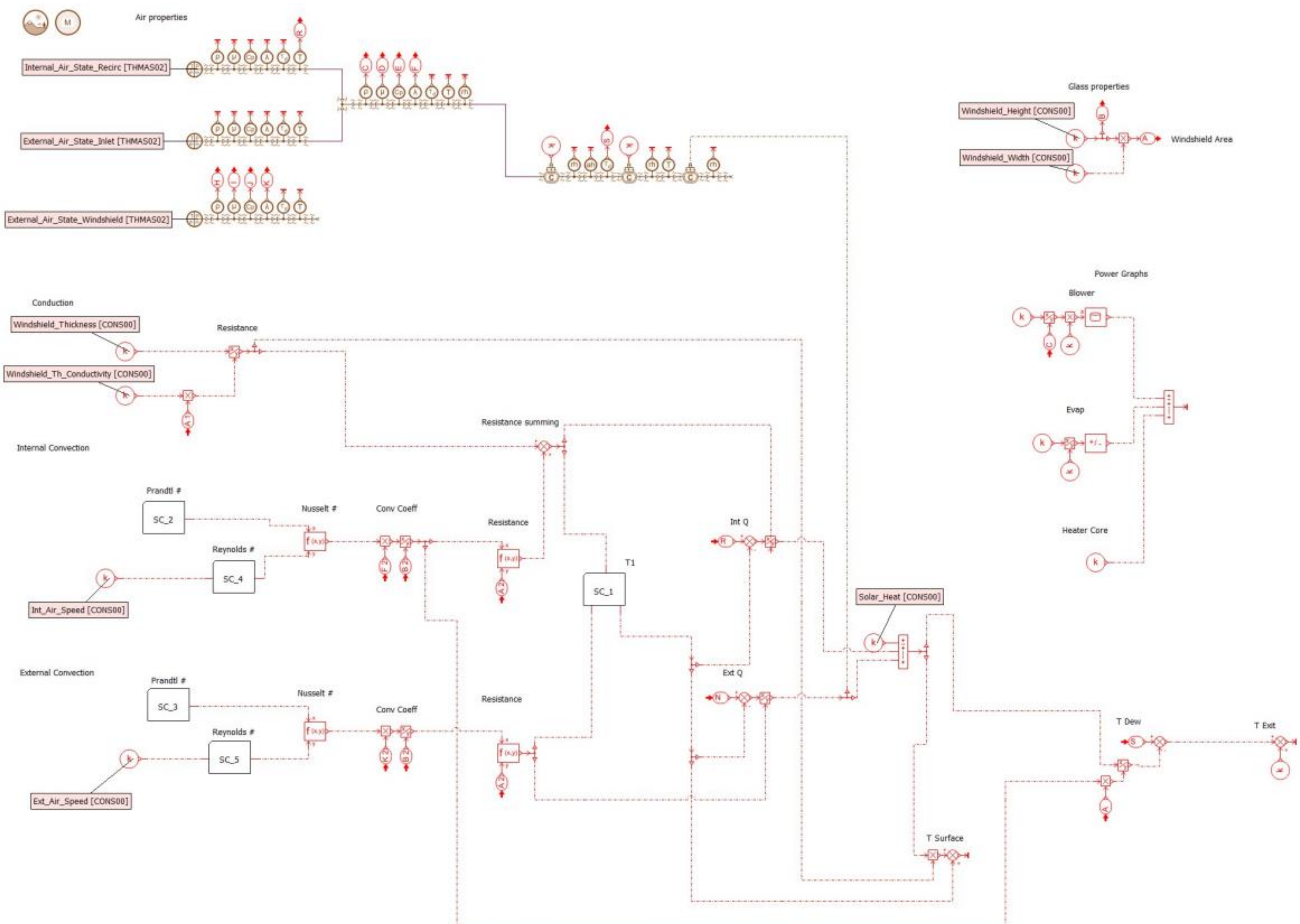

Fig. 6: Complete model overview

The properties of glass are fixed and defined in Fig.7. The bubbles with arrows and a letter indicate a value that is used elsewhere in the simulation with the same corresponding letter. The box with an 'X' is a multiplication.

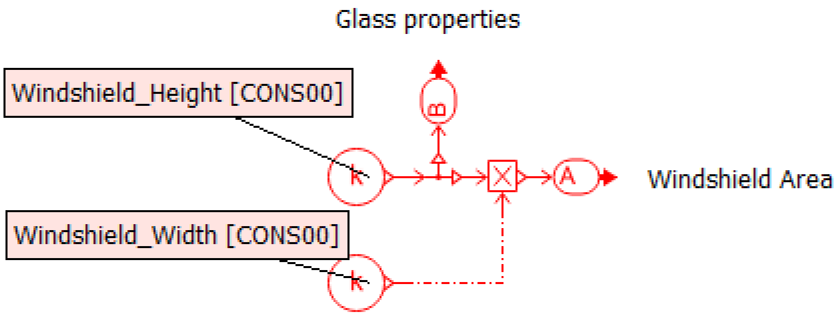

Fig. 7: Model display of the windshield parameters

The properties of air are a crucial component of the simulation, shown in Fig.8. The initial internal and external states are created on the left side, shown with labels. The circular symbols in the path are property callouts of interest, with either letters where the property is used elsewhere in the model or a stopper. The stoppers are used to allow the simulation to compile, and the properties called out may be of interest to the user for graphing purposes.

The boxes indicated with "C" are thermal chambers where heat is either added or removed from the system with the first being the condenser, the second the evaporator, and the final chamber is where heat is removed due to heat loss with the windshield. The properties of the exit of the final chamber are the properties of the air at the top of the windshield where the air must meet the 80% RH target. The far-right callout is where iteration could be added in the future to loop the simulation until it converges to an 80% RH target, as currently the convergence is done manually.

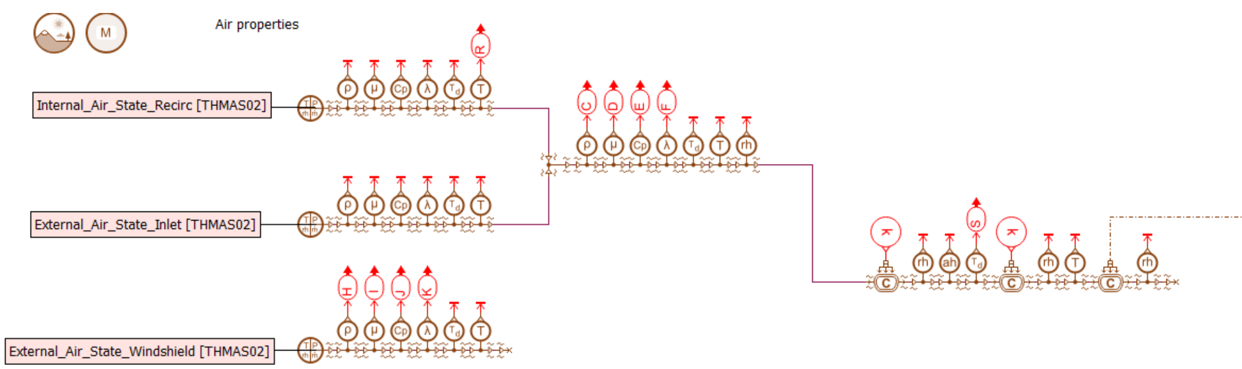

Fig. 8: Model display of the air properties

Conduction is based on the windshield and the resistance is therefore also fixed and defined, shown in Fig.9. The box with an 'X/Y' is a division.

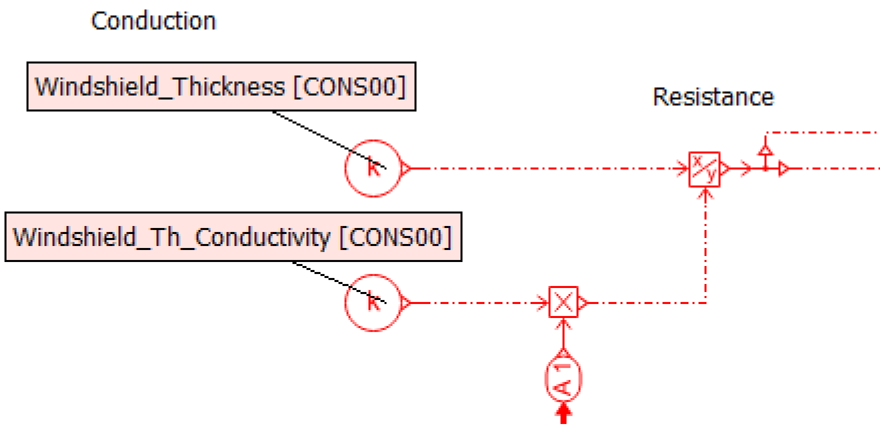

Fig. 9: Model display of conduction resistance

Convection for internal and external air is calculated the same way, with only the values input being different. Their resistances are calculated using the model in Fig.10. The large boxes with ‘SC’ are ‘Supercomponents’ that contain multiple operations within them, and their internals are shown in Fig.14, Fig.15, Fig.16, and Fig.17. The boxes with ‘f(x,y)’ are mathematical functions.

The internal air speed is determined from the input blower fan setting and external air speed is determined from vehicle velocity.

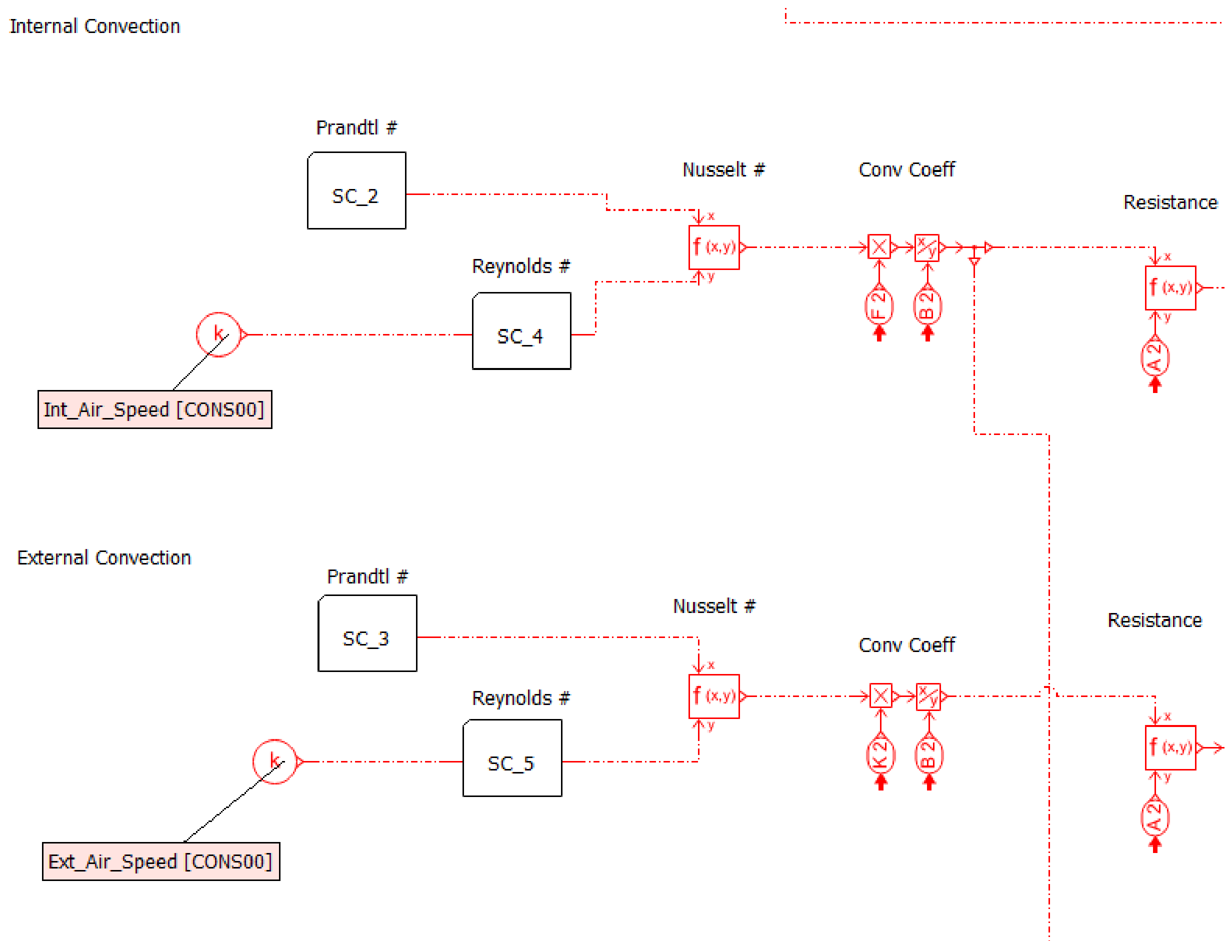

Fig. 10: Model display of convective resistance

Figure 11 shows the SC that contains T_1, which is expanded in Fig.18. The internal heat transfer and external heat transfers are solved with T_1 to find the total heat transfer through the windshield. The total heat transfer is sent to the final thermal chamber in Fig.8.

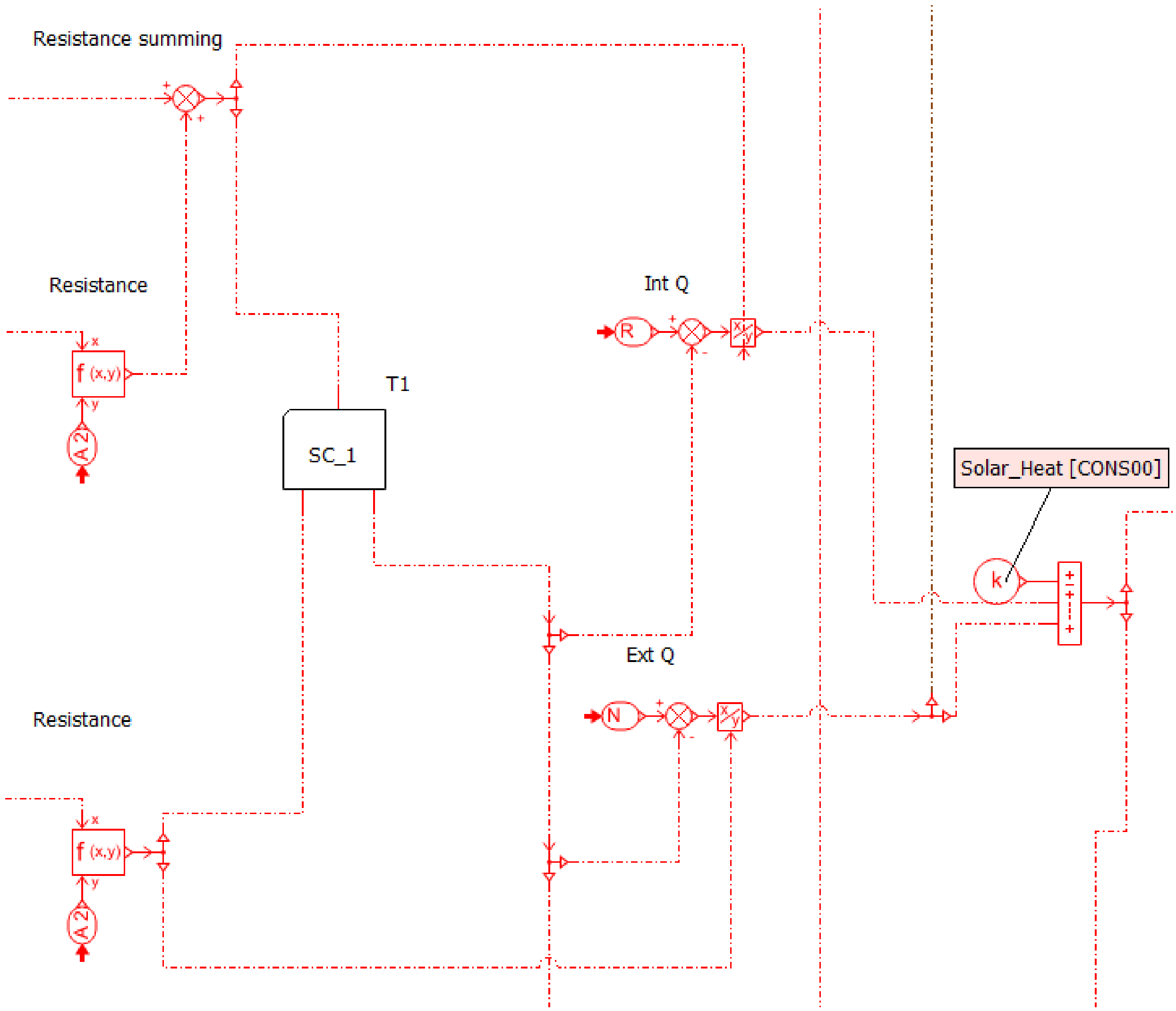

Fig. 11: Model display of solving for total heat transfer

The total heat transfer is also used with the conduction of the windshield to find the inner surface temperature for comparison in the experimental phase. In Fig.12 the T_{Dew} and T_{Exit} are solved for to complete the thermodynamic analysis of the system, where T_{Exit} is a necessary target to hit in experimental testing.

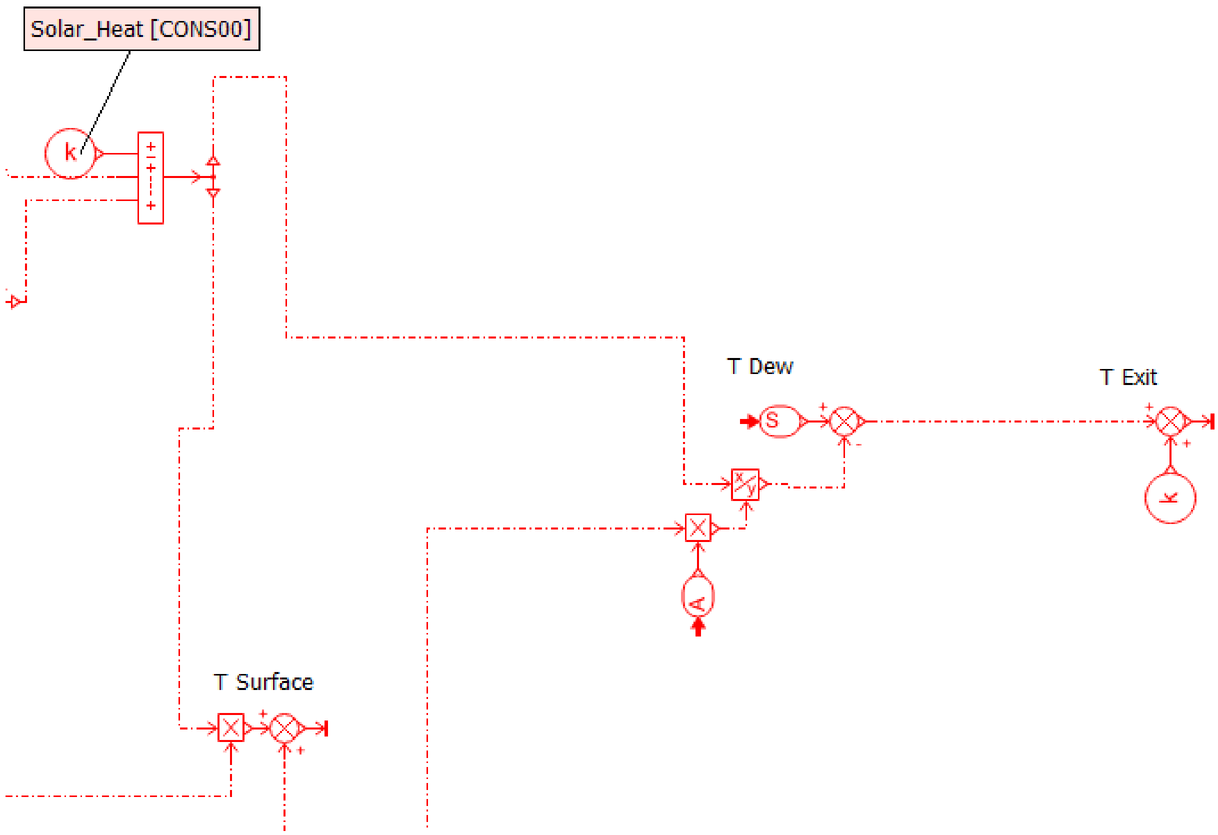

Fig. 12: Model display of the exit temperature

The final step of the simulation is to solve for power consumption, shown in Fig.13. The power consumption for each component is solved and subsequently summed together. The box with a cylinder on it is a table of values used for the blower fan's performance curve. The box with a '+/- ' on it is a sign reversal because the condenser removes energy from the air but consumes a positive amount of power.

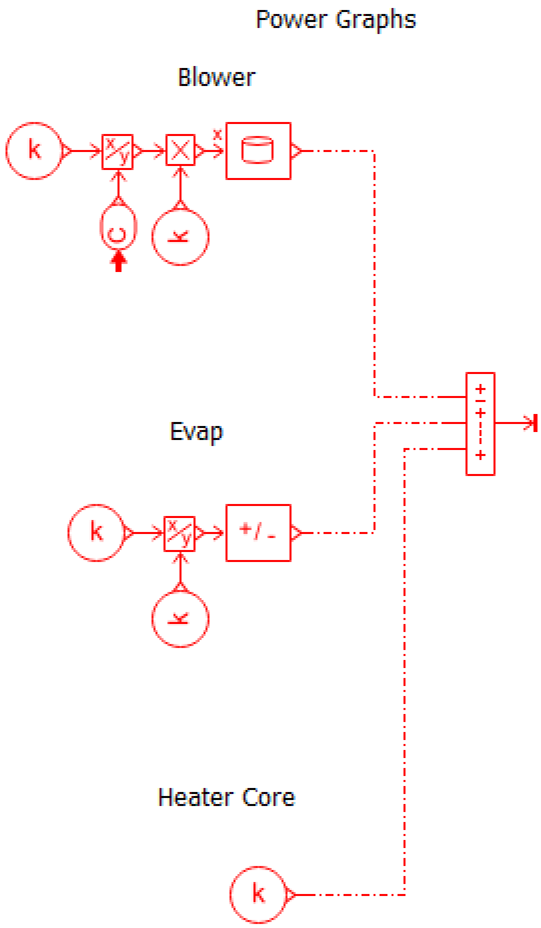

Fig. 13: Model display of the power consumption summation

The Pr number calculated using Eq.7 is solved for in the SC in Fig.10. The Pr number SC are expanded in Fig.14 and Fig.15. The SC are the same internally except for different values input. The larger gray ports are where connections to the SC enter or exit to the rest of the model.

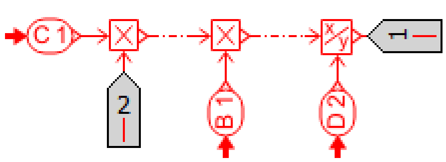

Fig. 14: Sub-model display of solving the internal Reynolds number using Eq. 7

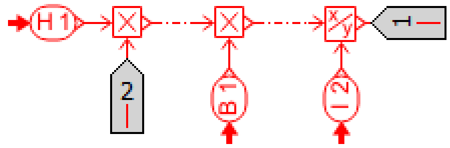

Fig. 15: Sub-model display of solving the internal Reynolds number using Eq. 7

The Re number calculated using Eq.8 is solved for in the SC in Fig.10. The Re number SC are expanded in Fig.16 and Fig.17. The SC are the same internally except for different values input. The larger gray ports are where connections to the SC enter or exit to the rest of the model.

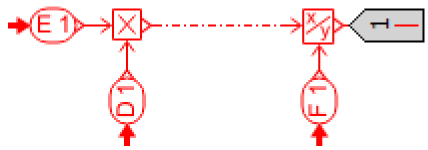

Fig. 16: Sub-model display of solving the internal Prandtl number using Eq. 8

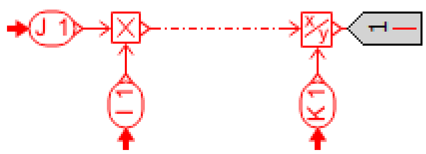

Fig. 17: Sub-model display of solving the internal Prandtl number using Eq. 8

The equation of T_1 calculated in Eq.19 is solved for in the SC in Fig.11. The SC for T_1 is expanded in Fig.18.

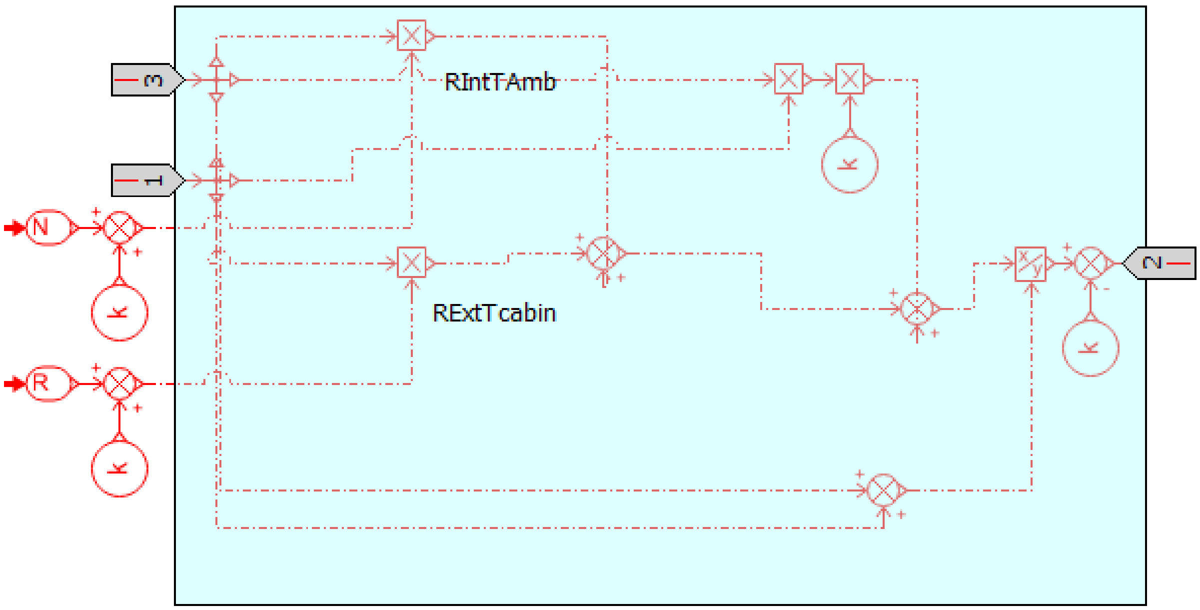

Fig. 18: Sub-model display of solving T1 using Eq.20

Chapter 5: Experimental Methods

5.1 Purpose of an experimental aspect

The experiment complements the simulation to produce values to validate simulation results and draw comparisons. Although the purpose of this research is to develop a 1-D mathematical simulation for defogging, the experiment is used to determine the accuracy and validity of the simulation, and possible recommendations for future development.

Results from the experiment are the potential beginning of further experiments performed by Stellantis to develop a correlation between the simulation and reality, understand future changes that may be important simulation, understand what further experiments may be necessary, and provide some degree of confidence that it is in fact possible to mathematically predict defogging and prepare defogging strategies in advance of prototype testing.

Determining the minimum heat loss and minimum power consumption experimentally further improves understanding of the thermodynamic behaviours of the system and potentially allow Stellantis engineers to gain insight into new best practices or develop new strategies.

The experiment also serves the purpose of finding the minimum blower fan speed setting that defogs the entire windshield, which although could be solved with a separate 3-D simulation, it may be more reliable to perform physical experiments. This minimum blower fan speed indicating the minimum power consumption, once found, may serve to be a cornerstone of future defogging control strategies.

5.2 Experimental procedures

The temperature of the windshield is the critical information needed to determine the risk of fogging when the humidity is known, and as such the windshield temperature is investigated experimentally and compared with the simulation to determine if the assumed physics and mathematical correlations are accurate.

The experimental procedure is the attachment of 16 thermocouples to the inside surface of the vehicle's windshield in a symmetrical 4 column by 4 row pattern located directly in the flow of the internal airstream in the center of the windshield. The rows are staggered to ensure the flow is not interrupted for subsequent thermocouples in the column.

All data obtained is averaged amongst the 16 thermocouples to obtain a statistically reasonable estimate of the true value of the glass temperature in the center of the windshield.

The vehicle is placed in a thermal wind tunnel where external air with known wind speed and temperature is blown onto the windshield, and the blower fan speed is adjusted to each of its pre-defined levels to find the lowest velocity that can blow air the full height of the windshield. The blower fan, condenser, and evaporator are kept constant to determine the effect of vehicle speed and external air temperature. This is repeated for multiple external air temperatures and wind speeds.

Solar radiation is mimicked in the thermal wind tunnel with the use of heat lamps. The heat lamps of the thermal wind tunnel are off for this experiment, though there was not enough time to perform further experiments with the heat lamps on to compare results. The solar heat transfer values used in the simulation are unlikely to be the same as the heat transfer from the heat lamps which could have cause a source of large data offset. More experiments need to be conducted to verify the solar radiation value chosen is reasonable.

The tests performed are as follows: test 1 is 25mph and 20 °F, test 2 is 65mph and 20 °F, test 3 is 25mph and -10 °F, and test 4 is 65mph and -10 °F. These four tests allow for an analysis in both mild and extreme conditions, as well as the individual effects of change in ambient temperature and vehicle speed.

Some values are reported in degrees Fahrenheit due to the testing being done in the United States which uses the imperial system, but the research is based in Canada which used the metric system. All data has been converted to Celsius except for information directly related to the tests themselves.

5.3 Data collection

Data of the thermocouples are collected by an on-board computer that sends the data to a server, where it is joined by data containing the wind tunnel parameters. The data itself is subject to the non-disclosure agreement, however some graphs and analytics can be shown in Chapter 6 with sensitive data removed.

Data on the HVAC's temperatures, RH, AH, volumetric flow rate, and power consumption were also recorded using existing sensors in the vehicle, though are not used in the results and are for Stellantis' purposes.

Data was collected in 5 minute intervals up to 30 minutes, and only the 30 minute data points were used in the results to ensure the system reached steady-state.

5.4 Prediction of results

The expected trends of the effect of individual variables should be a similar relationship to that seen in literature, with changes in external temperature being more impactful than vehicle speed.

The experimental results are likely be a different slope from the simulation and have some offset, but the trends and behaviour should be the same so that an empirical correlation is possible and present some discussion on why the simulation does or does not align with experimental results.

5.5 Comparison of Results

Comparison of the difference between simulation and experimental results determines how far the simulation deviates from the observed condition, and if it follows the same pattern. 4 data points are not enough to form a solid basis for an empirical correlation, but if more tests are performed in the future a correction factor can be proposed to adjust the simulation to align with experimental results. Since there are many assumptions and simplifications needed to develop this simulation, it is likely that any correction factor is relatively large.

The comparison is performed to find the difference between simulated and experimental results' offsets and the difference in slope of their individual tests.

The accuracy of the thermocouple should also be considered when comparing the results. The thermocouple has an accuracy of +/- 0.5 °C, equal to +/- 0.9 °F.

Chapter 6: Results and Discussion

6.1 Recorded results

6.1.1 Simulation results

The simulations used to obtain simulation trends in chapter 6.1.1 were all performed with a vehicle velocity 25 mph and external temperature of 20 °F. The trends for other vehicle velocities and external temperatures may differ in slope and magnitude but the overall behaviours remain the same.

Due to the non-disclosure agreement with Stellantis all values are non-dimensionalized by dividing by the property's largest value.

The relationship of system power to evaporator power is shown in Fig.19, and system power to volumetric flow shown in Fig.20 which captures the blower fan's behaviour. The significance of these figures is to show the exponential nature of the powered components, such that an increase in their useful effect has an exponential increase in power consumption.

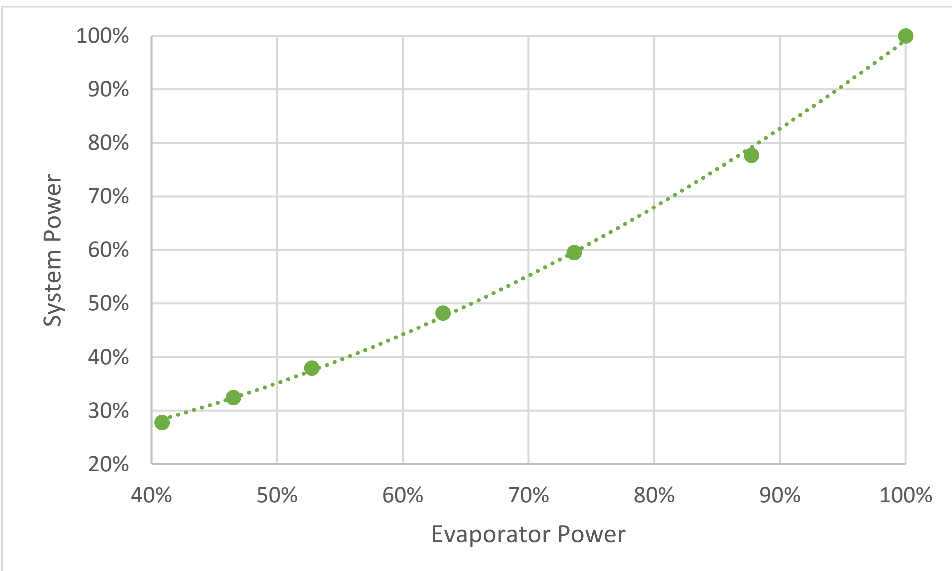

Fig. 19: System power vs. evaporator power

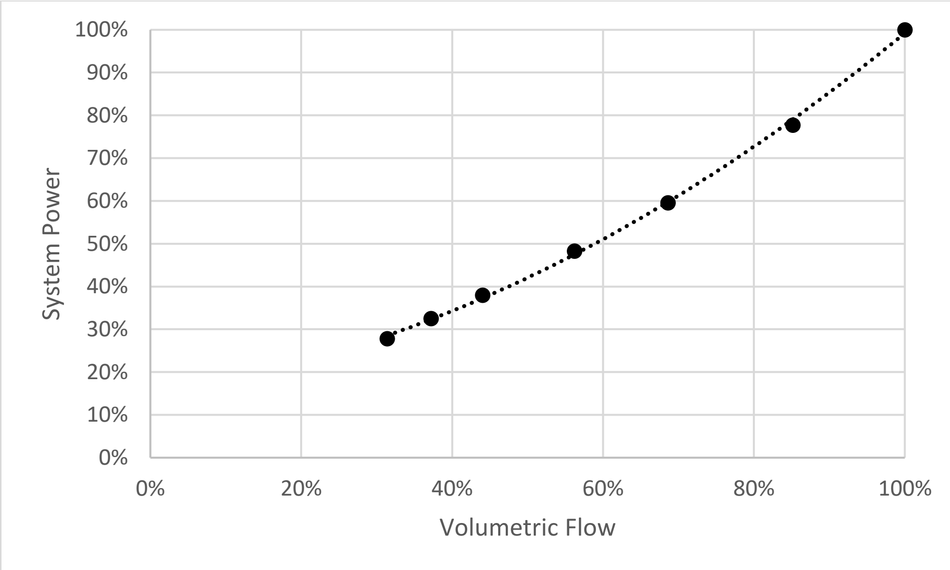

Fig. 20: System power vs. volumetric flow

The condenser's input is approximately linearly inverse to the evaporator input depending on the RH of the mixed fresh and recirculated air, as seen in Fig.21 This suggests that using the proposed strategy below 80% input RH requires a lot of condenser power to reach the 80% RH target, and above 80% input RH requires a lot of evaporator power to reach the 80% RH target.

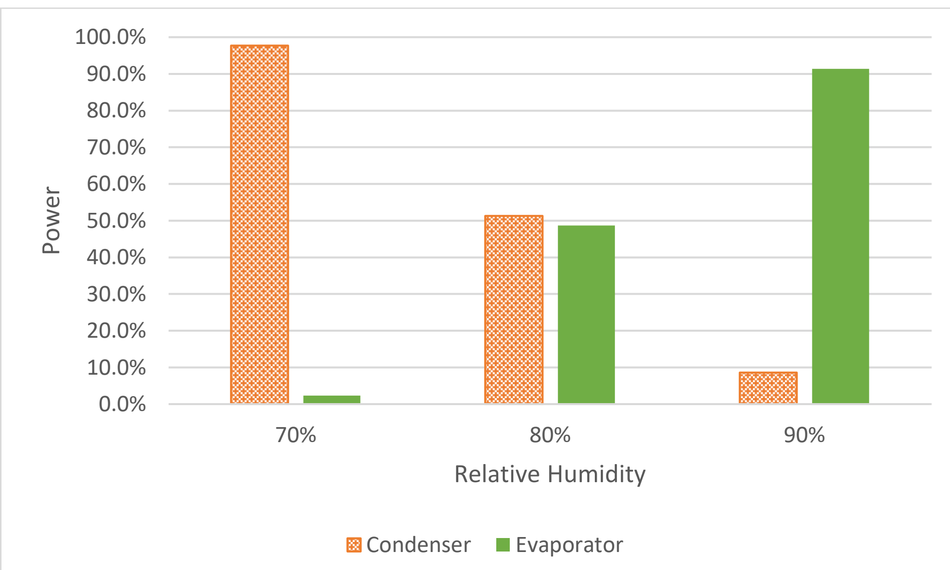

Fig. 21: Fraction of total power of condenser and evaporator vs. input relative humidity

A proposed alternative strategy is if the mixed RH is high, and the external RH is low, to allow in more fresh air to bring the humidity levels down to have less evaporator input. For low mixed RH, it is ideal to allow in enough fresh air to meet the AH goal, then only use the evaporator to raise the temperature. The only purpose of the condenser in defogging is to remove moisture, so it is more energy efficient in low mixed RH situations to let in fresh air and heat it than to spend lots of energy to cool the air then reheat it.

The relationship of blower fan power to evaporator power in Fig.22 is an important relationship because of the information that can be obtained from it. For any given point on the trendline of Fig.22, defogging only occurs when operating with higher blower fan input or evaporator input.

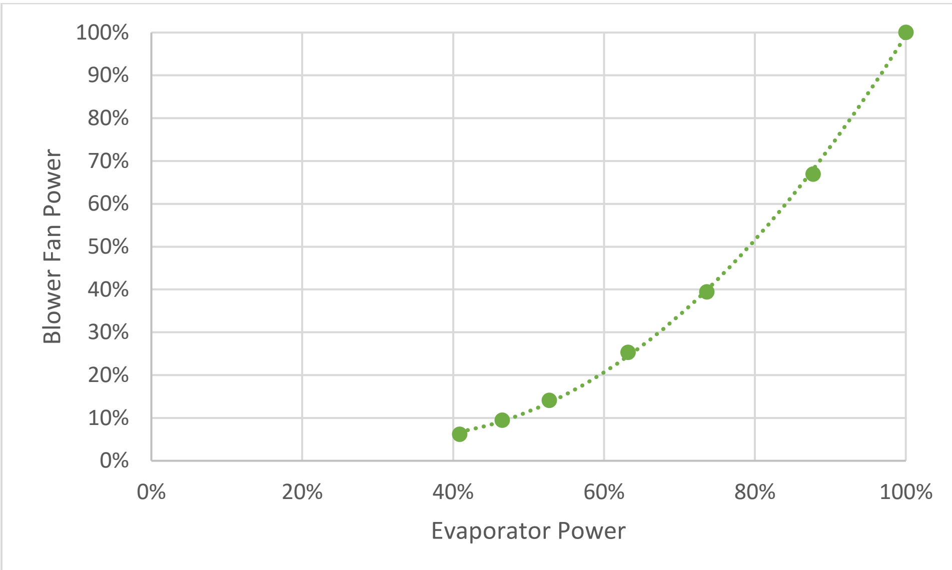

Fig. 22: Blower fan power vs. evaporator power

Operating along the trendline of Fig.23 requires the minimum power consumption with the constraint that the internal air velocity produced by the blower fan is sufficient to cover the windshield, which is determined in experiments.

Below the inflection point defogging is not possible, as not enough thermal energy is being supplied to defog the windshield. Above the inflection point, defogging is possible; however, for low blower fan speeds the air velocity is insufficient to defog the entire windshield. The minimum blower fan setting that can cover the entire windshield with its air flow is the optimal setting.

Operating below the optimal blower fan setting is unacceptable in all conditions because it cannot fully defog the windshield. Operating with a higher blower fan speed or with a higher evaporator power safely defogs the windshield, but operating with minimum evaporator power at the optimal blower fan speed consumes the minimum power.

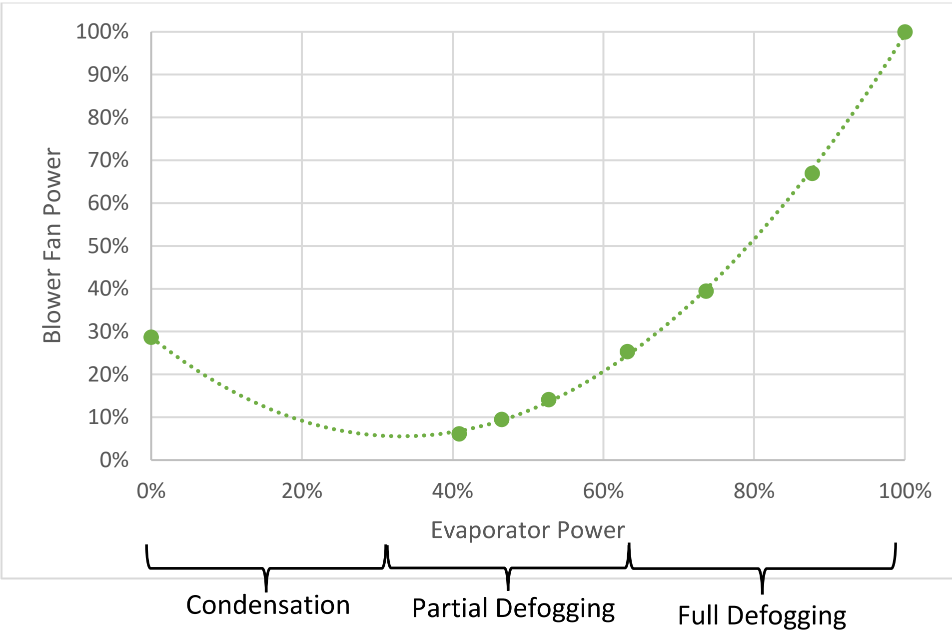

Fig. 23: Blower fan power vs. evaporator power, shown with possible operating regions

Figure 24 demonstrates the optimal condition is the point at the bottom-left of the green box, where both evaporator and blower fan are minimized while still meeting the defogging criteria. Following the curve to the right are the most energy efficient settings, which may be used in the event stronger defogging is required than what the optimal point is capable of. Operating anywhere inside the green box defogs the windshield, but off of the curve it is not the most energy efficient solution.

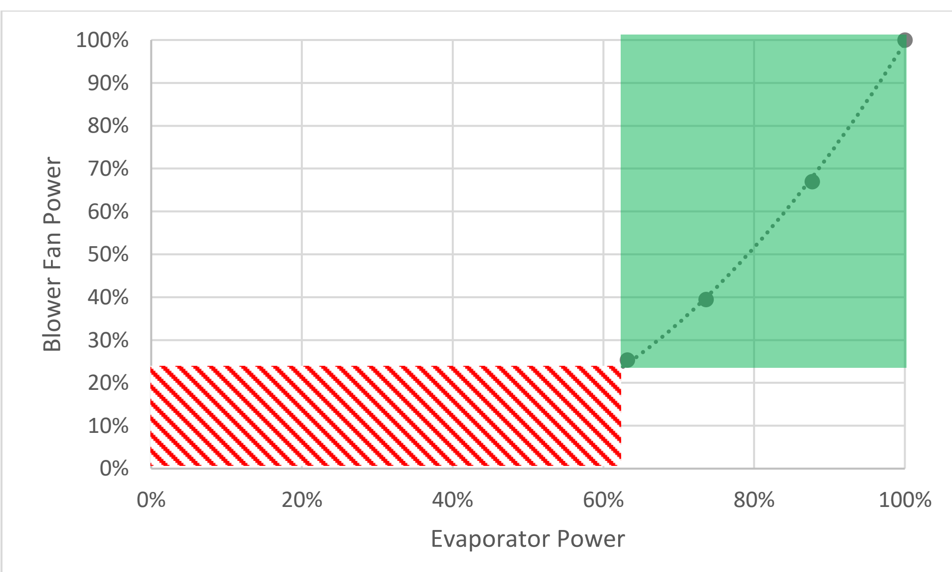

Fig. 24: Blower fan power vs. evaporator power, shown with allowable operating regions

The primary use of the condenser in defogging is to reduce the humidity of the air before being reheated. Evaluating the benefit of using the condenser over decreasing the RAR and evaluating the balance of condenser and evaporator is explored later in results. AH decreases linearly with condenser power, shown in Fig.25. Although the relationship is linear, all the thermal energy removed by the condenser must be re-added by the evaporator, requiring the minimum condenser load to meet the AH requirements without over-dehumidifying the air.

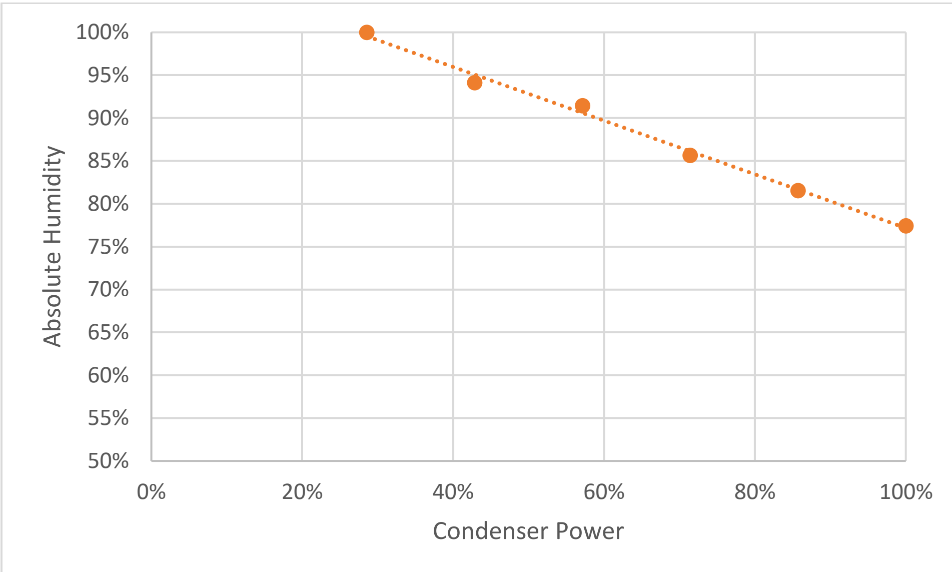

Fig. 25: Absolute humidity vs. condenser power

During simulations the difference between T_1 and $T_{Surface}$ is consistently approximately 0.3 °C. This is very small temperature difference relative to the changes in glass temperature, and therefore conduction could be neglected at the expense of a slightly larger offset to experimental values and result in a slightly reduced accuracy in defogging prediction.

6.1.2 Experimental results

Experiments were performed to determine if changing the RAR has a significant effect on defogging and what the behaviour of the change is. Experiments were conducted for RARs of 0%, 25%, 50%, 75%, and 100% to determine the RAR's effect.

For the low-speed tests of 25 mph fogging began to occur at around 90% RH for ambient temperatures of 20 °F and at around 60% RH for ambient temperatures of -10 °F. It was experimentally observed that for low-speed tests fogging began when the delta between the dew point and glass temperatures dropped to or below 5 °C. Fig.26 shows the low-speed results along with the fogging crossover point.

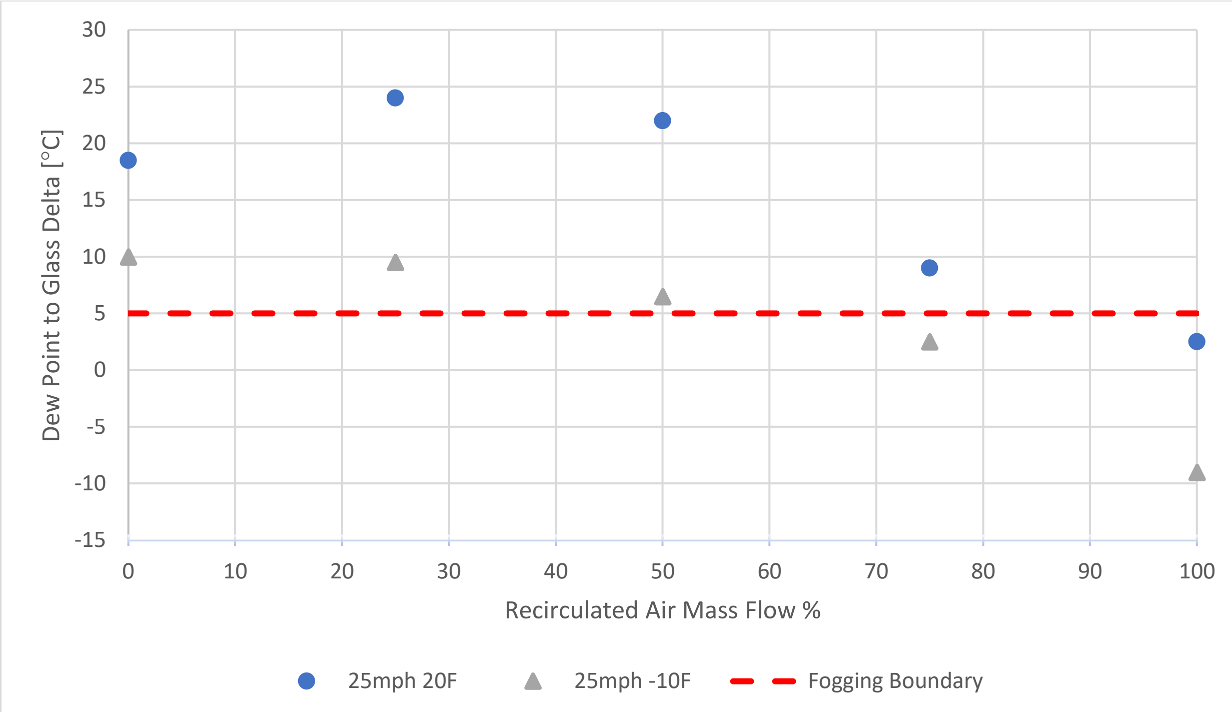

Fig. 26: Dew point to glass temperature delta vs. recirculated air mass flowrate, 25 mph

For the high-speed tests of 65 mph fogging began to occur at around 90% RH for ambient temperatures of 20 °F and at around 70% RH for ambient temperatures of -10 °F; however, it was intermittently re-fogging even at low RARs. It was experimentally observed that for high-speed tests fogging began when the delta between the dew point and glass temperatures dropped to or below 10 °C. Fig.27 shows the high-speed results along with the fogging crossover point.

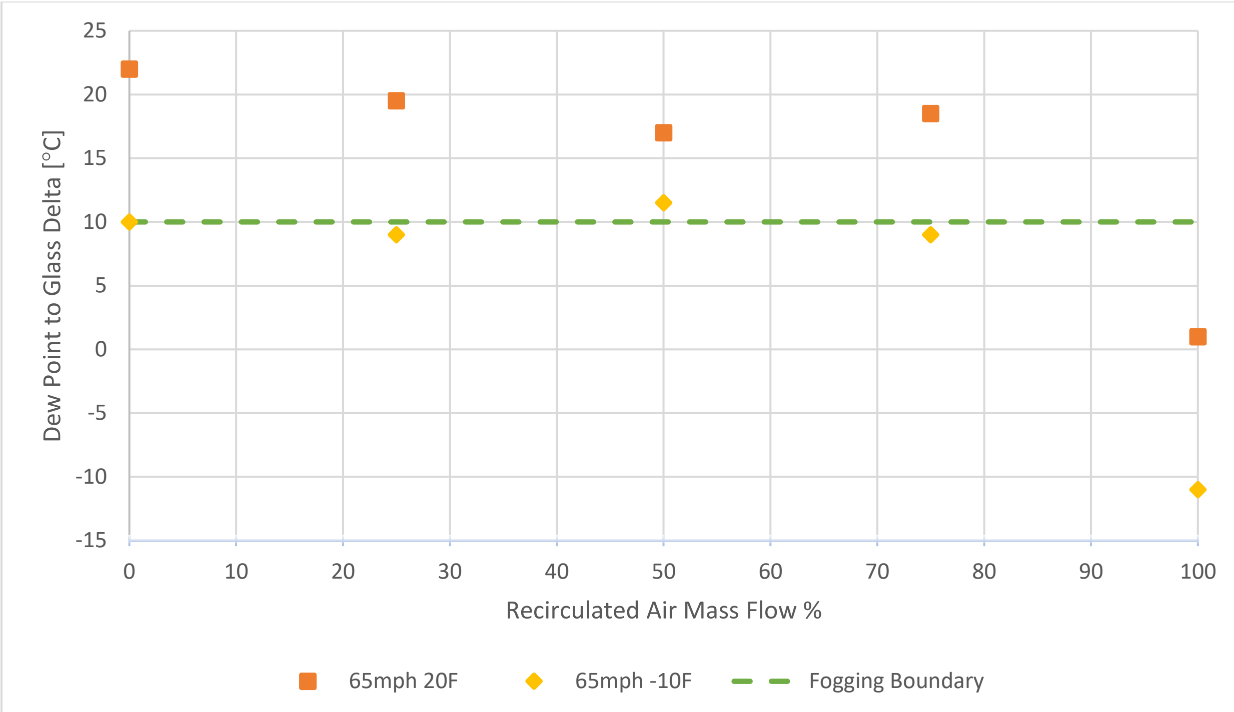

Fig. 27: Dew point to glass temperature delta vs. recirculated air mass flowrate, 65 mph

Based on multiple tests it was observed fogging generally occurred when the delta between the dew point and glass was close to 5 °C for the low-speed, and 10 °C for the high-speed.

During experimentation it is observed that for the specific vehicle platform chosen the blower fan's 'medium speed setting', was the lowest speed that defogged the entire windshield, and is thus the desired blower fan speed for all steady state defogging operations.

6.1.3 Simulation results vs. Experimental results

The trend of the experimental results matches the trend of the simulation, which indicates the mathematics used in this research do align with physics. As expected, higher vehicle speeds and lower temperatures result in higher external convection and a lower glass temperature of the windshield, which reduces the dew point temperature.

Additionally, the power consumption increases with increased external convection to compensate for the energy being lost, which primarily came from the evaporator.

When analyzing the optimal state, it is apparent that the overall heat transfer increases as the internal air speed since it produces more convective heat transfer. The minimum internal air velocity that can reach the full height of the windshield is always the most optimal internal air speed.

An increase from 25 to 65 mph resulted in an average 1.5 ± 1 °C drop in glass temperature, and a drop in 30 °F external temperature resulted in an average 12 ± 1 °C drop in glass temperature. This means a 30 °F drop in external temperature had an approximately 8 times greater effect than the vehicle velocity increasing from 25 to 65 mph.

Due to the tolerance of the thermocouple the change in vehicle velocity has an average deviation of 75% and the change in external temperature has an average deviation of approximately 8%. The large deviation for the vehicle velocity is due to small temperature changes coupled with a large thermocouple tolerance. A more accurate thermocouple is needed to accurately determine the effect of vehicle velocity. The change due to external temperature has an acceptable deviation and has a consistent behaviour on glass temperature for both speeds' changes in external temperature.

The change in vehicle speed has an average difference of 215% between simulated and experimental results, and the change in temperature has an average difference of 29% between simulated and experimental results.

Changing the temperature has a more dramatic effect than changing velocity, however since the evaporator consumes approximately 4 times more power than the blower motor, it is more efficient to increase the blower fan speed than the evaporator to improve defogging performance.

The simulation calculated the windshield temperature to be approximately 4 ± 1 °C colder on average than the experimental results but with a deviation of approximately 69%.

Further tests are necessary to use the deviations in a statistical analysis.

6.2 Accuracy of model

The model does not match the experimental results likely due to the large number of assumptions made in the simulation's mathematics. Some assumptions that may have influenced results are: internal flow may not be laminar due to the exit vents potentially disturbing airflow, Air may leak out of other door vents resulting in less than 100% of the treated air venting onto the windshield, warm cabin air may be partially picked up near the exit vents due to the venturi effect and may mix with the treated air, and the windshield is a curved inclined surface so the flat inclined surface correlation used in Eq.6 and Eq.10 may not be completely accurate, and each step of each experiment was performed for 20 minutes to allow the system to reach steady-state but it is possible the system was not entirely stabilized for some tests.

6.3 Data trends

The model follows a similar pattern to the experimental results, with a similar magnitude of the effect of temperature and vehicle speed.

The windshield temperature decreases 10.5 times faster for a one °C decrease external temperature than a one mile per hour increase in vehicle speed.

Since the data follows the same trend, it is possible to create an empirical correlation; however, further experimental testing is required to develop an accurate correlation of the simulation to experimental results.

Figure 28 compares the windshield glass temperature between simulated and experimental results. Although similar for high temperatures and low speeds, the difference increases as vehicle speed increases and external temperature decreases. Simulated results are consistently colder than experimental results in all experiments performed.

In Fig.28 the tests are as follows: test 1 is 25mph and 20 °F, test 2 is 65mph and 20 °F, test 3 is 25mph and -10 °F, and test 4 is 65mph and -10 °F. The significance of this figure shows that there is a clear similarity in behaviour between simulated and experimental results, but correlations are required to improve the accuracy of results.

Fig. 28: Simulated and experimental windshield temperatures

6.4 Key learnings

The optimal state to minimize power consumption always occurs at the minimum internal air velocity that can reach the full height of the windshield.

If the air outside is a temperature and humidity that poses risk to fogging and has an AH correlating to a RH above 80% at T_2, it is more efficient to use both the condenser and evaporator since little condenser power is required to reduce the air to the point where 100% RH is achieved. The limitation of this strategy is the condenser cannot operate in below-zero temperatures with Stellantis' existing vehicle infrastructure.

If the air outside is a temperature and humidity that poses risk to fogging and has an AH correlating to a RH below 80% at T_2, it is more efficient to use only the evaporator, since the RH target has already been achieved and only heat input is required to prevent fogging.

From analyzing the simulation, Eq.32 is derived.

$$\dot{Q}_{Evap} = \dot{Q}_{Cond \rightarrow 100\%RH} + \dot{Q}_{Cond \rightarrow AH} + \dot{Q}_{Evap \rightarrow T_{Evap}} \qquad \text{Eq. 32}$$

This can be illustrated on the psychrometric chart in Fig.2 as the energy required to progress from the initial state to the heated state. The first two terms of the equation come from the blue arrow which removes energy from the internal air and the third term comes from the orange arrow which adds energy to the internal air. This equation is derived from Fig.2 by observation and is not directly used in the methodology or simulation but may serve some practical use in future development.

To achieve the minimum power consumption the RAR should be as large as possible while still maintaining sufficient oxygen supply to the passengers. Below 0 Celsius since the condenser is not operating the only way to reduce the AH is to reduce the RAR.

Above 0 Celsius the RAR should still be as large as reasonably possible, however the benefit of opening or closing depends on the external RH. If the external RH is above the target 80% then the RAR should remain large; however, if the external RH is below 80% then it is beneficial to allow in fresh, dry air instead of cooling it to the dew point then reheating, as the lower the input RH is the more work must be done to cause it to reach the dew point.

An important piece of information that came out from the methodology is that the lowest internal air velocity that covers the windshield is the most efficient. It may seem counter-intuitive at first since a higher velocity produces greater defogging performance, but in steady-state conditions the higher velocity produces a higher convective loss requiring more heat to be produced.

Glass is a poor thermal insulator and the windshield's thinness leads to conduction having a negligible role in the heat transfer and can be unaccounted for in future analysis or empirically corrected for.

Internal air velocity plays a large effect on defogging performance, but surprisingly the external air velocity does not have a significant effect. The reason for this is due to external air's humidity and temperature having a more dominant effect on fogging than the velocity. The external velocity is still relevant and should not be discounted, but ambient temperature is the largest determining factor on when fogging occurs and how much power must be consumed to defog.

Steady-state defogging is not compatible with cold-start defogging. At a cold-start the HVAC must prioritize defogging performance which needs rapid fog removal necessitating high condenser, evaporator, and blower fan loads. After approximately 10 minutes when the windshield is fully defogged from cold-start it can switch to the steady-state strategy.

Steady-state defogging is compatible with existing reactive defogging strategies. The steady-state defogging model is a predictive model that attempts to minimize power consumption pro-actively during vehicle operation, but due to potential fluctuations in conditions it is still necessary to have a reactive model that adjust the powered components' loads when it detects a higher risk or onset of fogging.

The 1-D simulation may be one of the strongest tools available to predict steady-state defogging during design and development. 3-D simulations are computationally more expensive than 1-D and experimental testing requires a vehicle or subassembly prototype, so although the 1-D may be less accurate that other methods it has strong potential to be cheaper and faster to simulate while providing reasonably accurate predictions if previous correlations have been established for a vehicle model's HVAC and windshield.

Chapter 7: Conclusions

7.1 Conclusion

The 1-D defogging simulation developed can predict defogging with minimal power consumption but still needs further improvement with empirical correlations and integration with other cabin comfort models. This steady-state strategy proposes a method to use less power than the current reactive strategy, though further experimental testing is required to determine how large of an impact it can make.

7.2 Outcomes

The biggest outcome of this simulation shows that there is a mathematical method able to predict windshield defogging with an optimization function for minimum power consumption. The minimum power consumption occurs with the minimum blower fan load that covers the windshield, the minimum condenser load that gives an AH corresponding to a T_{Exit} with 80% RH while the evaporator load provides an equal amount of energy to compensate for the condenser and heat lost through the windshield.

This model was developed solely on mathematical equations and correlations found in literature. Further experimental testing can provide more correlations to correct for inaccuracies. Once the simulation is correlated and validated with development testing it can be used as a predictive tool to program defogging behaviour in advance of prototyping.

This simulation can be combined with an existing cabin heating and passenger comfort model to reduce the energy required in winter conditions, and once empirically correlated it can be applied during the design phase to predict the system's behaviour. The simulation is the final product of the research and can be used by Stellantis' engineers to further improve the efficiency of their defogging strategy and may influence their philosophy on HVAC control.

The research generated key learnings listed in Chapter 6.4 that were of great interest and benefit to Stellantis, and aspects of this research may prove to be invaluable in a highly competitive environment to provide more efficient vehicles.

Only some of the objectives to develop a predictive 1-D simulation were successful. The objectives met are: the simulation can predict defogging, the simulation shares a similar behaviour to experimental results, minimum power consumption can be obtained, and useful information can be provided to Stellantis' engineers. The objectives that still remain for future research are: to make the simulation's algorithm automatic to further reduce simulation times for users, to improve the accuracy through correlations obtained from continued experimentation, some additions such as accounting for rain and wind would allow for a more complete prediction and testing for more conditions, and the simulation is not yet validated to be ready for use by Stellantis' engineers.

7.3 Recommendations for future work

This model can be expanded to the side and rear glass, with relevant changes to each pane of glass' properties. This could allow the model to maintain a defog state on all glass panes to further improve safety.

The model can be improved with the addition of precipitation since water's higher density leads to higher external convection. The level of defogging required could be connected to the windshield wiper's speed or to a rain sensor if equipped.

The model can be integrated into existing HVAC control strategies, such as cabin comfort, to ensure appropriate humidity and oxygen levels are being met while defogging.

Further experimentation should be done to correlate the model for use in the design stage, and correction factors implemented to the simulation for each model. The correlation applies to any model that share the same windshield size, windshield angle, hood geometry, and vent geometry.

Further testing is necessary on solar radiation values used in the simulation. Experiments on the amount of heat radiated by Stellantis' thermal chamber heat lamps, perform experiments with and without the heat lamps to compare results, and better measurements of solar irradiance in northern regions prone to cold temperatures are necessary to validate or improve on the accuracy of the solar radiation value used in the simulation.

An experiment involving a thermocouple placed on the outer windshield surface and a thermocouple placed in the same location on the inner windshield surface may help in validating that the change in temperature through the glass is only 0.3 °C. The experiment should test for multiple conditions to identify if there is any significant change in conduction or if a constant value can be used in the simulation at the expense of a negligible inaccuracy in the simulation results.

Since only four experiments were conducted it is not possible to independently determine whether a change in vehicle velocity or external temperature impacts the glass temperature in a linear or non-linear behaviour.

Exploration of the benefit of using heating on the condenser drain to prevent condensation from freezing could allow for condenser operation in below-zero temperatures. This allows the air to be greatly dehumidified without requiring a significant amount of heat input from the evaporator, improving defogging efficiency.

The simulation can be further improved with iteration. This is not currently in the simulation schematic due to computational and time constraints, but this could allow for the testing of multiple conditions at once.

The simulation can be further improved by reducing the number of assumptions made by performing further experiments. Not all assumptions made may result in a negligible effect on system behaviour and their effect may be accounted for if corrections are made through experimental research.

It is also possible to note that the windshield acts as a heat exchanger in this situation, so it is recommended to explore the mathematical method of solving for defogging assuming the windshield as a heat exchanger with some empirical correlation. This method was not chosen for this research as it requires experimental testing, versus the mathematical method could be solely simulation, though it is a proposed alternative method.

Implementation of this simulation into Stellantis vehicles needs testing to properly correlate the model and validate that it can defog the windshield for multiple conditions as required by customer requirements and regulations. This may lengthen development time in the short-term as more testing is required for validation, but it has the potential to shorten development time in the long-term if it is able to accurately predict the defogging strategy before vehicle testing.

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Appendices

Appendix 1: Derivation of Equation 19

Beginning from Eq.17:

$$\dot{Q}_{Int} = \dot{Q}_{Ext} - \dot{Q}_{Solar}$$

Substitute in Eq.13 and Eq.14:

$$\frac{T_{Cabin} - T_1}{R_{Int}} = \frac{T_1 - T_{Amb}}{R_{Ext\ Conv}} - \dot{Q}_{Solar}$$

$$\frac{T_{Cabin} - T_1}{R_{Int}} - \frac{T_1 - T_{Amb}}{R_{Ext\ Conv}} = -\dot{Q}_{Solar}$$

Use multiplication to achieve a common denominator of $R_{Int}R_{Ext\ Conv}$:

$$\frac{T_{Cabin} - T_1}{R_{Int}}\left(\frac{R_{Ext\ Conv}}{R_{Ext\ Conv}}\right) - \frac{T_1 - T_{Amb}}{R_{Ext\ Conv}}\left(\frac{R_{Int}}{R_{Int}}\right) = -\dot{Q}_{Solar}\left(\frac{R_{Int}}{R_{Int}}\right)\left(\frac{R_{Ext\ Conv}}{R_{Ext\ Conv}}\right)$$

Divide by the common denominator:

$$(T_{Cabin} - T_1)(R_{Ext\ Conv}) - (T_1 - T_{Amb})(R_{Int}) = -\dot{Q}_{Solar}(R_{Int})(R_{Ext\ Conv})$$

Re-arrange for T_1:

$$(T_{Cabin})(R_{Ext\ Conv}) - (T_1)(R_{Ext\ Conv}) - (T_1)(R_{Int}) + (T_{Amb})(R_{Int}) = -\dot{Q}_{Solar}(R_{Int})(R_{Ext\ Conv})$$

$$(T_1)(R_{Ext\ Conv}) + (T_1)(R_{Int}) = \dot{Q}_{Solar}(R_{Int})(R_{Ext\ Conv}) + (T_{Cabin})(R_{Ext\ Conv}) + (T_{Amb})(R_{Int})$$

$$T_1 = \frac{\dot{Q}_{Solar}(R_{Int})(R_{Ext\ Conv}) + (T_{Cabin})(R_{Ext\ Conv}) + (T_{Amb})(R_{Int})}{R_{Ext\ Conv} + R_{Int}}$$

Finally, re-written to conclude with Eq.19:

$$T_1 = \frac{R_{Ext\ Conv}T_{Cabin} + R_{Int}T_{Amb} + R_{Int}R_{Ext\ Conv}\dot{Q}_{Solar}}{R_{Ext\ Conv} + R_{Int}}$$

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