Predicting the effect of powertrain preconditioning on vehicle efficiency

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ABSTRACT
Under extreme climatic conditions, the vehicle fuel consumption can be far from the certified value. Given the growing concern for polluting emissions, it is necessary to investigate a way to improve the overall vehicle efficiency and thus reduce the emissions and fuel consumption gap. One solution is to pre-warm the gearbox and the engine in order to make it work at an optimal temperature to achieve the best efficiency possible. Indeed, low lubricant temperature is a source of reduced vehicle efficiency due to the lubricant viscosity rising exponentially at very low temperature. Using the Powertrain Dynamics library, a vehicle model with a detailed equation-based gearbox model taking into account the temperature-dependent losses and a map-based engine including temperature-dependent emission models (CO, NOx, HC and particulates) is developed.

Introduction
Responding to the ever growing need to reduce vehicle fuel consumption and pollutant emissions, new technologies have been developed and successfully implemented in a large number of vehicles over the last few years. However, if engine efficiency has recently dramatically increased thanks to ongoing design improvements and new technologies, the question is how much further we can push the limits to improve efficiency in use and at what price.

One way to achieve better performance from the powertrain is to improve its efficiency. To do so, we have to keep in mind that our vehicles are rarely operated in their optimal efficiency region due mainly to the road layouts, road traffic, driver behaviour, short range operation and the climatic conditions. We can at least seek to counteract the effects of the latter on vehicle efficiency. Vehicle transmission and engine oil viscosities increase exponentially at low temperatures, affecting the vehicle powertrain efficiency. Until the oil has fully warmed up, which can take a rather long time under extreme cold weather conditions, the transmission losses are very high due to drag on the gears, clutches and bearings caused by the viscous oil and the emissions of pollutants are considerably high. Poor range and fuel economy can result in customer dissatisfaction compounded by the fact that the vehicle is only being used exploiting a small percentage of its certified power. Concerning the emissions of pollutants, a reduction could mean a rebate when buying a new car. The idea is then to put the transmission and the engine in the best conditions whatever the weather is in order to increase their efficiency. Farrant et al. [1] showed the benefits of powertrain preconditioning during a cold start.

In this paper, we build a vehicle model in Dymola* (Figure 1) using components from the Powertrain Dynamics library. We then precondition the transmission and engine lubricants to...
several temperatures and run the vehicle model over the standard NEDC and ARTEMIS drive cycles. The ARTEMIS drive cycle combines an urban and a highway portion. The transmission model involved in this study is a predictive equation-based model in order to show how the efficiency would benefit from higher oil temperatures without the constraints of a map/empirical based model. The engine uses a map-based combustion model and pollutant emission maps that depend on engine speed, engine torque, fuel flow rate and coolant temperature. The benefits of preconditioning are then highlighted as well as the costs of doing so.

The vehicle model

The vehicle includes a predictive thermal model of the transmission to quantify the thermal dynamics of the system including losses such as bearing and gear drag losses. The heat release from friction is evaluated in each area of the model. All gearbox components (moving and non-moving) are interlinked via mechanical and/or thermal ports so that the effects of each component in the system on the others can be evaluated. This physical relationship modelling forms the basis for predicting the oil temperature and viscosity in the whole subsystem.

The vehicle model is built on the Vehicle Interfaces library standard [2] from the Modelica Association. All models use multibody components and are designed for easy assembly and
efficient computation [3]. A driver model is included to exercise the vehicle model over a specific drive cycle in order to evaluate its efficiency and performance. A road model and an atmosphere model are also used in the experiment and interface with the vehicle model.

The model shown in Figure 2 is a view of the vehicle one level lower than that shown in Figure 1. The main subsystems are engine, transmission, driveline and chassis. The engine model uses a performance map giving an output torque depending on pedal position and engine speed. A fuel tank is included to enable the calculation of the instantaneous and average fuel consumptions. The engine used in this experiment is a 2-L four-cylinder petrol engine.

The transmission model will be discussed in detail in the next section. The driveline model can be set to include compliance, stiffness and damping characteristics so that the same vehicle can also be used for detailed driveability studies. The chassis model has a pitch degree of freedom and longitudinal motion as well as models for tyres, aerodynamics and suspension.

**The transmission model**

**Overview**

In this study, focus is on a six-speed automatic transmission which includes a dynamic torque converter model with predictive thermal effects, gear sets with bearing friction and bearing drag models, gear mesh models with temperature-dependent efficiency, a shift actuation system and a heat port (thermal connector) to port the generated heat to other vehicle subsystems.

The green connections in Figure 3 represent translational mechanical flanges to apply the clamp load to the clutches and brakes. The red connector is the heat port which allows us to thermally link all the components together in order to transfer the heat throughout the system and compute the temperatures at various points in the model.

The top left corner of Figure 3 is the thermal network for oil and casing. For each one of these components, a thermal conductor lies between the thermal masses representing the heat transfer to the oil and the casing, respectively, and the remainder of the gearbox’s thermal network. It is a systems modelling thermal model and we must also point out that the distribution of oil in the system is not considered in this paper.

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**Figure 2.** Detailed view of the vehicle model with all the subsystems: (1) ancillaries, (2) engine, (3) gearbox, (4) driveline, (5) chassis, (6) brakes and (7) controllers.
Dynamic torque converter

In automatic transmissions, the torque converter couples the engine to the gearbox. Despite the availability of a steady-state (mapped) torque converter model, here, we choose to use an equation-based dynamic version since the transient response is of prime importance. There are three main components: the impeller connected to the engine, the turbine connected to the gearbox and the stator connected to the gearbox housing via a one-way clutch. The impeller is a centrifugal pump. When the oil inside the torque converter is cold, it affects the efficiency as the high viscosity decreases the impeller performance.

The moment-of-momentum equation is applied to the three control volumes. For example, for the impeller [4,5]:

$$I_i \dot{\omega}_i + \rho S_i \dot{Q} = -\rho \left( \omega_i R_i^2 + R_i \frac{Q}{A} \tan \alpha_i - \omega_s R_s^2 - R_s \frac{Q}{A} \tan \alpha_s \right) Q + \tau_i$$  (1)

The subscripts used in the above equation denote part of the torque converter where i means impeller and s means stator. $\dot{\omega}$ is the rotational speed, $\tau$ is the torque, $Q$ is the volume flow rate, $I$ is the fluid inertia, $\rho$ is the oil density, $S$ is a design constant for the impeller, $R$ is the radius, $A$ is the flow area and $\alpha$ is the blade exit angle. There are similar equations for the turbine and stator.

The conservation of energy equation is [6]

$$\rho(S_i \dot{\omega}_i + S_i \dot{\omega}_i + S_i \dot{\omega}_i) + \frac{\rho L_f}{A} \dot{Q} = \rho \left( R_i^2 \omega_i^2 + R_i^2 \omega_i^2 + R_s^2 \omega_s^2 - R_s^2 \omega_i \omega_s - R_i^2 \omega_i \omega_s - R_i^2 \omega_s \omega_i \right)$$

$$+ \omega_i \frac{Q}{A} \rho (R_t \tan \alpha_t - R_s \tan \alpha_s) + \omega_i \frac{Q}{A} \rho (R_t \tan \alpha_t - R_i \tan \alpha_i)$$

$$+ \omega_s \frac{Q}{A} \rho (R_s \tan \alpha_s - R_t \tan \alpha_t) - p_L$$  (2)

$p_L$ represents the losses, $L_f$ is the fluid inertia length.
\[ p_L = \frac{\rho}{2} \text{sgn}(Q) \left( C_{sh,i} V_{sh,i}^2 + C_{sh,t} V_{sh,t}^2 + C_{sh,s} V_{sh,s}^2 \right) + \frac{\rho f}{2} \text{sgn}(Q) \left( V_{i}^* + V_{t}^* + V_{s}^* \right) \]  

(3)

where \( C_{sh,i} \) is the shock loss coefficient for the impeller and \( V_{sh,i}^2 \) is the shock velocity, \( f \) is a fluid friction factor and \( V_{i}^* \) is the fluid velocity relative to blades. Therefore, we know that the fluid viscosity significantly affects the torque converter efficiency and this is why we need to include the thermal effects in the model. The kinematic viscosity of the fluid (\( \mu \)) can be calculated using the ASTM D341 standard \([7]\) as

\[ \log(\log(V_{vis} + 0.7)) = a - b \log(T) \]  

(4)

where \( a \) and \( b \) are two coefficients calculated from two known viscosity-temperature operating points of the oil. A formula in Ref. \([8]\) gives a performance factor (\( B \)) in order to calculate a correction factor to re-evaluate the impeller efficiency when using a viscous fluid.

\[ B = K \left[ \frac{V_{vis}^{0.5} H^{0.0625}}{Q^{0.375} N^{0.25}} \right] \]  

(5)

where \( V \) is the fluid viscosity, \( H \) is the impeller head, \( Q \) is the fluid flow rate and \( N \) is the impeller shaft speed. We can now use this performance parameter to determine the correction factors \( C_H \) for the head, \( C_q \) for the volume flow rate and \( C_n \) for the efficiency from look-up tables (Figure 4).

These correction factors allow us to modify the losses to account for the thermal effects. First, we introduce the volume flow’s corrected value:

\[ p_{L, \text{thermal effects}} = \frac{\rho}{2} \text{sgn}(Q) \left( C_{sh,i} V_{sh,i}^2 + C_{sh,t} V_{sh,t}^2 + C_{sh,s} V_{sh,s}^2 \right) + \frac{\rho f}{2} \text{sgn}(Q) \left( \frac{V_{i}^* + V_{t}^* + V_{s}^*}{C_q} \right) \]  

(6)

We also need to add a new term to the losses to correct the impeller efficiency:

\[ p_{L2} = \frac{\tau_i \omega_i (C_n - 1)}{Q} \]  

(7)

Figure 4. Correction factors \( C_q \) and \( C_n \) plotted against the performance factor \( B \).
These corrections result in the turbine rotational speed converging more slowly towards the impeller angular velocity at a low temperature (Figure 5).

When the oil is cold and has a high viscosity, the flow’s axial velocity inside the torque converter is reduced and so, the fluid inertia transmitted to the turbine is not as good and it takes longer to reach the coupling point. The overall torque converter efficiency is then affected.

**Roller bearings**

The roller bearings are important components as they can achieve very good efficiencies under appropriate conditions. However, when the oil viscosity is high, their efficiency dramatically drops as the friction torque due to oil drag increases (Figure 6). Since all the components in the transmission model are thermally linked, the heat released by other parts of the model (clutches, torque converter etc.) can affect the bearing behaviour through warming up of the transmission fluid and gradually help to improve the overall system efficiency.

The bearing friction torque is given by

\[
T_{\text{friction}} = fn \times \mu \times R + T_{\text{seals}} + T_{\text{drag}}
\]  

(8)

The friction coefficient \( \mu \) depends on the type of rollers (ball, pin, taper pin etc.) and typically varies between 0.001 and 0.0024. The friction torque due to the seals is not detailed here as it has only a small contribution and does not vary significantly with temperature.

The drag torque formula [9] demonstrates the importance of the oil viscosity:

\[
T_{\text{drag}} = 4 \times V_m \times K_{\text{roll}} \times C_{\omega} \times B \times d_m^4 \times n^2 + 1.093 \times 10^{-7} \times n^2 \times d_m^3
\]

\[
\times \left( \frac{n \times d_m^2 \times f_t}{v} \right)^{-1.379} \times R_s
\]

(9)

\( v \) is the kinematic viscosity at operating temperature. \( C_{\omega} \) and \( K_{\text{roll}} \) depend on the bearing geometry and dimensions. \( V_m \) is the drag loss factor. \( R_s \) is based on the bearing dimensions and oil level (Figure 7). \( n \) is the bearing rotational speed. \( f_t \) depends on the bearing geometry and the oil level.
Clutches

There are several clutches utilized throughout the transmission model. There is a lock-up clutch in the torque converter model and three other clutches are used in the gear set along with two brakes which use the same type of model. Clutches are modelled by multiple rotating plates pressed together via a normal force. In the model, this force is applied via the green connector (Figure 8) which represents a translational mechanical connector in Modelica.

The friction torque between the clutch plates is calculated as follows:

\[
T_{\text{friction}} = \mu \times N_f \times (\varphi_f + \varphi_s) \times \frac{\pi \times w_{\text{rel}} \times (r_o^4 - r_i^4)}{2h}
\]  

(10)

**Figure 6.** Friction torque of a roller bearing depending on the oil temperature.

**Figure 7.** The oil level is the distance above the lowest contact point between the rolling element and the outer ring.
where $\mu$ is the oil viscosity, $\varphi_f$ and $\varphi_s$ are the pressure and shear stress flow factors, respectively, introduced by Patir and Cheng [10]. $h$ is the oil film thickness which reduced under applied pressure [11], $N_f$ is the number of friction plates.

The heat generated by the friction between the clutch plates is expelled through the heat port and stored into a heat capacitor which is linked to further thermal models in the system. The heat capacitor accounts for the thermal mass of driven and driving plates. The temperature of the clutch plates can thus be evaluated as well as the thermal losses to the clutch surroundings.

During slipping, clutches and brakes can generate considerable amounts of heat in the gearbox which contributes to lower the oil viscosity thus affecting the whole transmission efficiency. However, the friction torque is only significant for a small amount of time.

During operation, the clutch temperature can rise very quickly. We can see on in Figure 9 that heat is released when both the relative angular velocity and the torque transmitted are non-zero which essentially happens during the engagement and disengagement phase.

**Gears**

We can divide the gear losses into speed-dependent and load-dependent losses. Speed-dependent losses consist of windage losses and oil churning losses. Load-dependent losses consist of sliding friction losses and rolling friction losses. The windage losses are induced by oil droplets that are in suspension in the gearbox housing and create a thin mist which increases the gear frictional resistance. The churning losses are due to the oil being trapped in the gear mesh and to the gear
rotating in the lubricant. They depend on the gear rotational speed, the oil viscosity, the gear geometry and the proportion of the gear submerged.

The sliding and rolling friction losses are dependent on the gear rotational speed and on the instantaneous coefficient of friction. We are mostly interested in the churning losses since they are closely related to the oil properties. The churning losses for tooth surface are given by

\[ P_{CI} = \frac{7.37f_{gi} \times \mu \times n_i^3 \times D_i^{4.7} \times b_i \times (R_t / \sqrt{\tan \beta})}{A_8 \times 10^{23}} \]  

(11)

\( f_{gi} \) is the gear dip factor that is to say the ratio of the gear dipping into oil. \( D \) is the outside diameter, \( \beta \) is the helix angle and \( R_t \) is the roughness factor.

The oil viscosity \( \mu \) plays a major role in the amount of churning losses and thus, we can easily see the importance of the temperature on these. \( n \) is the gear rotational speed and all the other parameters in the formula are geometrical dimensions. Two other similar formulae exist to calculate the churning losses for smooth outside diameters and smooth sides of discs (i.e. shafts and gear side faces, respectively).

The churning losses increase at low temperatures (Figure 10) and this has an even bigger effect when the dip factor is high as a larger part of the gear is dragged through oil.
The engine model

The generation subsystem

The engine model is map-based and the generation block (circled in red in Figure 11) is a table-based torque source that depends on the throttle angle, engine speed and a fuel cut-off signal to stop injecting fuel during overrun. In this component, we can use either a Brake Mean Effective Pressure or Indicated Mean Effective Pressure map. We use the latter in this study since we want to have a separate temperature-dependent Friction Mean Effective Pressure map.

The emission component controls the creation of pollutants; it is also table based. One 3D table (inputs to this table are engine torque, engine speed and coolant temperature) is used for each type of pollutant: CO, NOx, HC and particulates.

The friction component rejects heat into the system through its heat port. This heat port is connected to the cooling system.

The cooling subsystem

The cooling system has lumped thermal properties to model the coolant and the radiator (Figure 12). Thermal masses are used to account for: the intake and exhaust ports, the cylinder head, the cylinder liners, the outer portions of the cylinder block, the radiator circuit coolant, the solid parts of the radiator and the coolant that is outside the radiator. The losses to the environment through convection are also taken into account.

The heat port at the top allows to connect the components in the engine to their associated thermal representation in the cooling system.

Figure 10. Example of oil churning losses with respect to oil temperature with different dip factors.
Results

Prior to the calculation of vehicle fuel economy, a quick estimation of the cost to per-warm the powertrain can be made. If we apply a heat flow of 100 W, it takes 886 s to heat the gearbox from \(-10^\circ\text{C}\) up to \(+90^\circ\text{C}\) and 1600 s to heat the engine up to the same temperature applying a 200-W heat flow. If we assume that a plug-in electric heater is used to warm the engine and gearbox, it would require 0.1135 kW h of electricity. At an average price at the time of the study (in England) of £0.15/kW h, the electricity needed will cost £0.017. Even if the objective here is not to make money out of this solution, the cost has to be evaluated and the results clearly show that it should not be a financial problem for the customer. This type of preconditioning is more suited in vehicles such as plug-in hybrids or EVs (electric vehicles) or indeed conventional vehicles operating in cold climates where the customer might, by default, plug the vehicle in to recharge the battery and/or precondition the cabin.

NEDC

The NEDC cycle is used to homologate vehicles to various fuel economy and emissions standards including the Euro 6 standard. It is made up from an urban section repeated four times and an extra-urban section. It covers a distance of 11,023 m and lasts for 1180 s. It is often criticized for not representing real-life driving conditions (too light duty). However, it has to be considered as it is not only used for homologation but also because we are interested in slow urban driving conditions to attest the maximum savings possible (slower powertrain warm up).
Figure 13 shows the fuel consumption on the NEDC at different ambient temperatures where the powertrain is also starting at the ambient temperature. We can see that the greatest saving occurs at the beginning of the cycle, in the urban portion at low load. Once the oil temperature reaches its ideal value, the average fuel consumption with a pre-warmed gearbox and engine converges towards the average fuel consumption with a non-pre-warmed powertrain.

If the ambient temperature is +23°C, pre-warming the transmission and engine fluids to +90°C allows a fuel economy improvement of 1.71% (150 mL).

If the ambient temperature is now −10°C, pre-warming the transmission and engine fluids to +90°C yields a fuel economy benefit of 7.66% (660 mL). At the time of writing, the average price per litre of unleaded petrol in England is £1.098 [13]. The improvement in fuel economy means a reduction in operating cost of £0.725 and after taking into consideration the cost of the electricity required to pre-warm the transmission, the customer would still save £0.7075.

Figure 14 shows that the transmission oil temperature reaches its ideal value when starting at 40°C only towards the end of the drive cycle but when we start at a lower temperature, the oil temperature does not even come close; this explains the significant fuel savings.
Figure 13. Average fuel consumption NEDC (l/100 km) under different start temperatures: $-10^\circ$ (solid line), $+23^\circ$ (dashed line), $+40^\circ$ (dotted line), $+90^\circ$ (dash-dot line) versus time (s).

Figure 14. Transmission oil temperature with different start values: $-10$ (solid line), $+23$ (dashed line), $+40$ (dotted line), $+90$ (dash-dot line) versus time (s).
Figure 15 shows the engine oil temperature and the same conclusion can be drawn, even if the oil takes slightly less time to warm-up thanks to higher losses rejected as heat than the transmission.

The emissions of pollutants are also impacted by the engine and transmission oil temperatures. The CO$_2$ is of course coupled to the fuel consumption; so, it benefits from the improvement in fuel economy. The emissions of CO$_2$ decrease when we pre-warm the powertrain as expected. Indeed, since the global powertrain efficiency improves, we can inject less fuel to achieve the same level of performance. Therefore, as the level of CO$_2$ rejected to the environment is a function of the fuel injected, it is reduced at higher engine start temperatures.

Emissions of HC are much higher when the engine is cold (Figure 16) due to unburnt fuel, mainly near the cylinder wall. Indeed, if the wall is cold, then the heat transfer with the flame is high and the combustion temperature in these areas is lower than usual (flame quenching). This results in unburnt quantities of fuel which generates hydrocarbons. The slightly negative values at 90°C are due to the interpolation method used in the tables which gives approximated numbers.

Like the hydrocarbon emissions, the CO emissions are higher during cold engine start (Figure 17). This is also due to poor combustion quality that does not burn enough fuel.

The nitrogen oxides emissions are invariant under all the oil initial temperatures since they are not high enough to have an influence on the combustion temperature to which the formation of NOx is very sensitive.

The time taken to simulate the NEDC drive cycle using this vehicle model, which includes the detailed dynamic torque converter model, is around 30 min whereas the same vehicle with a torque converter model based on the steady state operating curves will simulate the drive cycle in less than 50 s. These simulation times are based on the use of a laptop with an Intel Core i7 4900MQ processor.
Figure 16. Normalized HC emissions under different start temperatures: −10 (solid line), +23 (dashed line), +40 (dotted line), +90 (dash-dot line) versus time (s).

Figure 17. Normalized CO emissions under different start temperatures: −10 (solid line), +23 (dashed line), +40 (dotted line), +90 (dash-dot line) versus time (s).
**ARTEMIS cycle**

The ARTEMIS drive cycle is based on a statistical study and thus fits better to the real usage of the vehicles. It is made of an urban part and a highway portion. It covers a distance of 33,605 m and lasts for 2061 s. In comparison with the NEDC drive cycle, ARTEMIS is much more aggressive.

The potential fuel economy benefit from pre-warming the transmission is less obvious on this cycle (Figure 18). Indeed, as it is more aggressive, the components take less time to heat up and so, the benefit in pre-warming disappears more rapidly.

If the ambient temperature is +23°C, pre-warming the transmission fluid to +90°C yields a fuel economy improvement of 0.24% (37 mL).

If the ambient temperature is now −10°C, pre-warming the transmission fluid to +90°C yields a fuel economy benefit of 1.84% (182 mL). This improvement in fuel economy will reduce the operating costs by £0.20 and after taking into account the cost of electricity to pre-warm the transmission, the customer would still save £0.183.

Figure 19 shows that the oil takes less time to reach its ideal operating temperature with the ARTEMIS drive cycle than with the NDEC drive cycle. This is due to the ARTEMIS cycle being more aggressive.

An interesting occurrence that can be noticed in both drive cycles is that the overall vehicle efficiency does not increase much after the oil reaches +23°C. The fuel economy when submitted to an initial temperature of +90°C instead of +23°C is only 1.71% for the NEDC cycle and 0.24% for the ARTEMIS cycle. This tells us that we could consider pre-conditioning the powertrain to only +23°C as the benefits after this value are much less.

As for fuel consumption, the reduction in the emissions of pollutants is less when driving over this cycle than the NEDC due to the ARTEMIS cycle being more aggressive and causing the powertrain to warm up much faster.

![Figure 18](image-url) Average fuel consumption ARTEMIS (l/100 km) under different start temperatures: −10 (solid line), +23 (dashed line), +40 (dotted line), +90 (dash-dot line) versus time (s).
Applied example

Let’s consider a real-life example to give us a concrete idea of the potential savings that could be achieved at a larger scale. In Göteborg, Sweden (1 million people live in its urban area):

- 100,000 people commute to work every day
- 60,000 people commute by car every day
- 12 km (7.5 mi)/24 min is the average commute ride inside Göteborg
- 51 km (31.7 mi)/50 min is the average commute ride outside Göteborg

Based on the NEDC results, we can estimate that a commuter could save 0.5 L of fuel on a journey that starts at −10°C if we first pre-warmed the powertrain to +23°C. If we make the assumption of 50,000 commuting trips per day, it gives us a saving a 25,000 L of fuel and 62 t of CO₂ per day.

The energy required to pre-warm the powertrains for these 50,000 commuting trips would be 5,675 kW h. The amount and type of fuel used to generate this electricity varies from country to country based on the local energy mix. In Sweden, the energy mix in 2013 was 33% Nuclear, 30% Fossil fuels, 22% from biofuels and the final 15% from hydropower and wind power [14]. If we assumed that 30% of the required energy is from petroleum-based electricity generation sources, then this would require approximately 470 L of fuel. This assumes that 578 kW h can be generated per barrel of petroleum [15].

Conclusions

The benefits of powertrain preconditioning have been studied and show that this strategy could be a real possibility in the future. A device to pre-warm the gearbox and the engine and the way for the customers to choose when to use it has still to be explored though.
As the differences in the results between the NEDC and ARTEMIS drive cycles showed us, the best potential savings could be achieved in a busy urban environment where the lubricant would usually take a long time to reach its ideal working temperature.

Pre-warming of other subsystems like traction batteries would allow even greater fuel economy and performance benefits and will be considered in further work. In the case of EVs, it is even a necessity to heat up the battery in extremely cold climates due to the performance degradation of the battery at very low temperatures. It has been shown in Ref. [16] that the battery discharge can reach 60% when the air temperature drops from 22°C (72°F) to −40°C (−40°F).

Disclosure statement

No potential conflict of interest was reported by the authors.

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