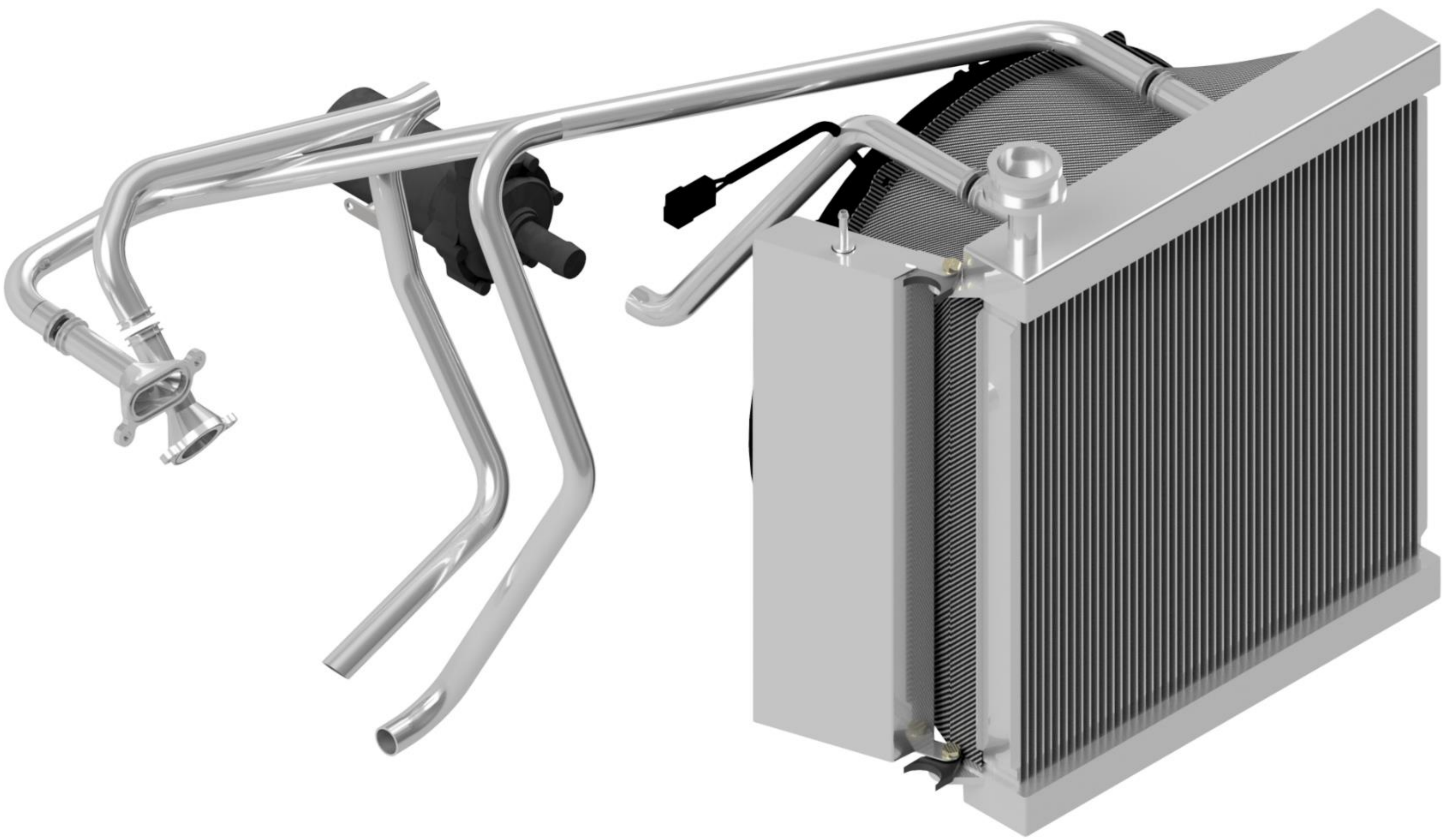
Cooling
ARG18 Spring Technical Report
May 22, 2018
Joseph Benjamin – jkb237



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Team: Engine
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Reviewed by:

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Introduction

Engines obtain their power from latent energy in a combustible fuel. Through combustion of the fuel, an engine is able to obtain power via the rapid expansion of gases produced. However, there are some undesirable byproducts associated with combustion including un-combusted fuel and heat. While some heat is good to keep an engine within operation temperature range, unregulated amounts of heat will degrade materials and reduces efficiency. For this reason, a cooling system is necessary.

The function of a cooling system is to reject the heat generated by the combustion process within the engine. In doing so the heat capacity of the engine is increased, but the primary goal is to increase the rate at which heat is rejected from the system, allowing more fuel to be combusted over a given period of time without exceeding a reasonable maximum steady-state temperature. This is a simple concept made difficult by racing performance requirements. Designing a cooling system that lowers the steady-state temperature without any outside considerations would not be difficult, but for the purpose of racing we must take certain design goals into consideration. A large design driver is that the cooling system must be as light as possible without infringing on its own ability to cool the engine. An ability to sufficiently cool an engine while maintaining reasonable maximum heat rejection per mass of system weight is what makes a successful system.

An engine cooling system itself is comprised of six different elements: heat exchangers, plumbing, pumps, coolant, fans, and the engine. For our purpose, the system is designed currently with the following orientation of coolant flow:

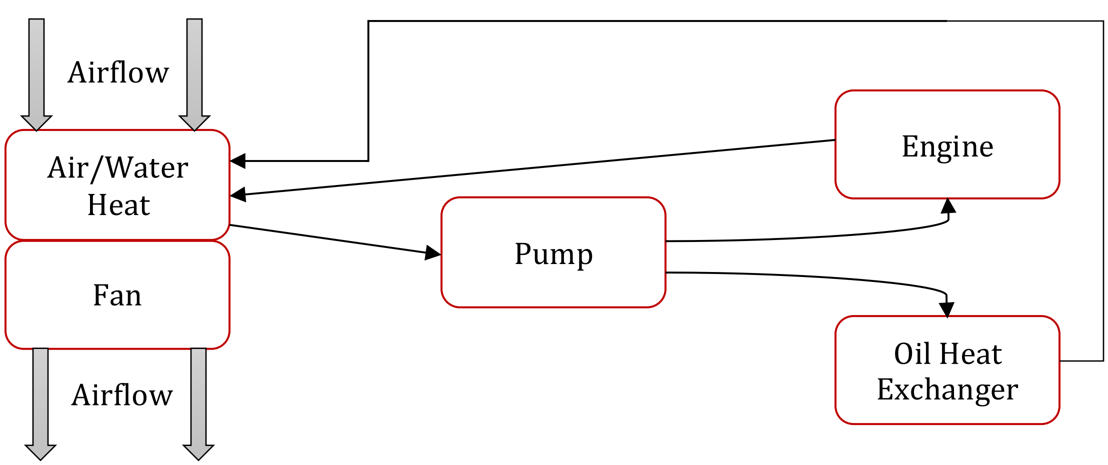


Figure 1: Coolant Flow Diagram

While a large majority of the “design” work done to design a cooling system is actually component selection, it must be done using the same universal design practices of testing, data acquisition, and data analysis used in any other conventional design process. Data-driven design has been a major push for every system this year, and cooling is no exception. Among our highest priorities at the beginning of the year was remedying the oil temperature issue experienced at the MIS 2017 competition (*Figure 2*). Although the maximum temperature seen (128°C) is within the operating range of our oil (10W-40), the Borg-Warner KP35 turbo that we use specifies a maximum operation oil temp of 120°C. This combined with the fact that the oil temp was still rising near the end of endurance was cause for concern, resulting in diagnosing and improving oil cooling being a main goal.

The other high priority issue entering the design semester was reevaluation of the target heat rejection determination and design process as a whole. Past years relied on the same method of calculating the target heat rejection, and entering the year we wanted to investigate the system's design process to uncover any potential areas of improvement.

The general design timeline of this year's system was meant to go along the following structure:

1. Heat Generation Test
2. ARG17 system validation
3. Radiator selection
4. Oil cooler selection
5. Fan selection
6. Pump selection
7. Plumbing design
8. Catch can design
9. Iterate during Spring

The goal of the heat generation test was determining the validity of our calculated heat rejection target. As mentioned above, previously we determined a target heat rejection value by considering all the energy in a given mass flow rate of E85, subtracting reasonable or measured percentages associated with different losses (exhaust heat, incomplete combustion, ambient heat transfer, power produced), and attributing the remaining heat energy to heat rejection by the cooling system. This is a widely-used and recommended practice, but in order to confirm that the values for losses we use are correct we sought to complete the heat generation test. The test was prepared and ready to be run over fall break (10/7-10/10), however due to issues with the engine on the dynamometer the test was continually delayed.

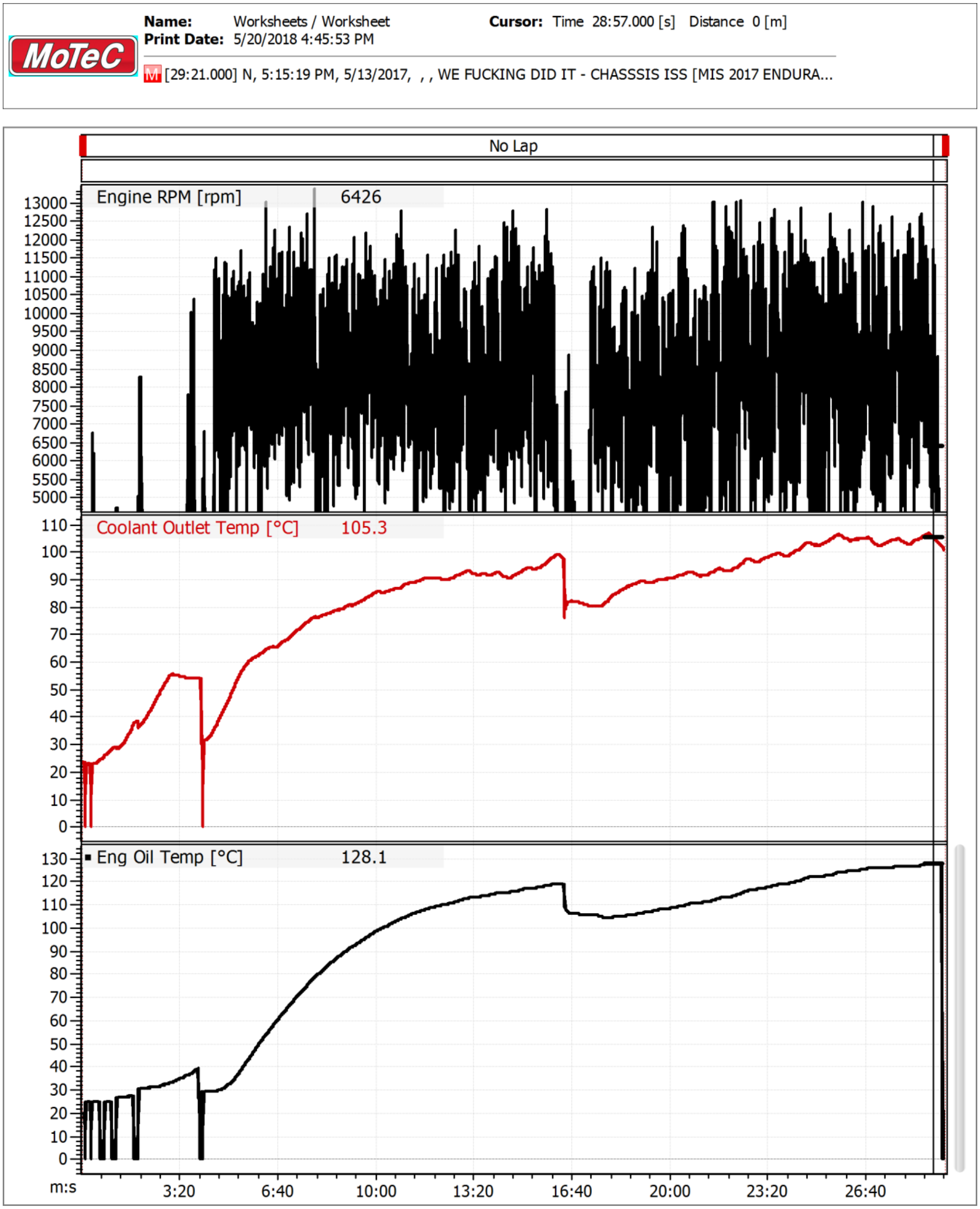


Figure 2: Oil (Top) and Coolant (Bottom) Temp. vs. the length of MIS 2017 Endurance

Obtaining a reasonable heat rejection value to design to plays a key role in designing the cooling system. Without it, the system will be misguided and may end up being an ill fit for the engine even if the rest of the design process was performed correctly. Unfortunately, due to the issues with the engine on our dynamometer we were unable to shift from a calculated to an experimentally determined target value this year. Fortunately, aside from the oil temperature issue, there was nothing noticeably wrong with the design process which Ayinde Crear performed last year [1]. Bearing this in mind, the team leads decided to extend the design timeline of the cooling system to run into the spring semester, and I instead shifted to further validating ARG17's system. The design philosophy during the late fall semester was to ensure that certain design elements from last year would still be able to function had we found that a change to the radiator design was possible in spring. This meant a review of the plumbing design, pump selection, fan selection, catch can design, and mounting points.

The review of the plumbing design was fairly straight forward. A couple of design changes considered were anodization and finned aluminum tubes. Anodized plumbing was taken out of consideration for a couple of reasons: first, that anodization would need to be removed before any welding after the fact, and that the plumbing manufacturing process would be severely lengthened by sending the pieces to be anodized. Finned tubes were more seriously considered, but ultimately the effect they would have on accessibility of the engine bay combined with the anticipated efficacy due to poor airflow over most of the tubes (which are inside of the monocoque) was not worth the weight. More testing may occur in the future to determine whether the added mass of finned tubes is more effective than added mass elsewhere (the radiator), but for the time being the focus remains on the larger issues at hand. Tube sizing remained unchanged, as the pressure drop caused by them was insignificant while remaining a lightweight solution.

Balance of airflow and pressure through the radiator drives fan selection. Jacob Rigos and Carlos de la Torre performed a test in the falls semester to calculate the volumetric airflow through the radiator, which helped in validating ARG17's system. The main design requirement is achieving sufficient airflow while balancing pressure drop through the radiator-fan module, and the fan selection process performed by Ayinde Crear took both of these into account [1].

While not playing a vital role in the performance of the system, it is possible for the catch can to be overdesigned. The rules stipulate that the container must be able to hold .5 liters or 10% of the carried volume of coolant – whichever is greater. Because the current system does not have a drain plug to facilitate draining and refilling, I used a conservative overestimate of the contained volume, 5 liters, to design a smaller catch can.

Design Requirements

The primary design requirement for the cooling system is the ability to reject our target value of heat rejection while at the desired steady state temperature. For our system, the maximum coolant temperature we would like to see is 105°C (due to our 23psi pressure cap). This is an increase from last year, but it should aid in downsizing the radiator while still being in the limit of the pressure cap. This is because a higher steady-state temperature to design to correlates to a higher steady-state heat rejection for a given system, meaning we should be able to decrease the size of the radiator without sacrificing maximum heat rejection capability.

As stated prior, the heat generation test was significantly hindered this year by dynamometer issues, and we were forced to continue design into the spring semester while relying on calculated values. The equation used to calculate the target heat rejection is referred to as the Conservation of Energy equation and goes as follows:

$$\dot{m}_f Q_{LHV} = \dot{Q}_{cool} + P_b + \dot{Q}_{misc} + \dot{H}_{IC} + (\dot{m}_f + \dot{m}_a) h_{ex}$$

$$\dot{m}_f = mass\ flow\ rate\ of\ fuel$$

$$Q_{LHV} = lower\ heating\ value\ of\ E85$$

$$\dot{Q}_{cool} = rate\ of\ heat\ rejection$$

$$P_b = brake\ power$$

$$\dot{Q}_{misc} = miscellaneous\ heat\ losses$$

$$\dot{H}_{IC} = energy\ omitted\ due\ to\ incomplete\ combustion$$

$$\dot{m}_a = mass\ flow\ rate\ of\ air$$

$$h_{ex} = exhaust\ enthalpy$$

Using this equation for several different loading scenarios (RPM-MAP combinations), a table of $\dot{Q}_{cool}$ values can be created, however doing so requires an accurate torque curve. Then, using each load case as the center of a bin, the percent of time spent at each bin while driving (driving an endurance run for example) serves as a weighting method for the given heat rejection values. The value which was used for designing the past two years' systems is 22.5kW, so that is what we used in validating ARG17's system.

The electrical team was not able to give a concrete power budget, so we were forced to use last year's values as an estimate: 9 Amps for the fan and 4.3 Amps for the pump. The selected fan and pump draw 7.9A and 3.8A respectively, so these should be permissible.

Additionally, in order to align the system with this year's car's design goals, we wanted to ensure that even though the system was undergoing potentially drastic changes, the overall mass of the system does not increase dramatically. Last year's system weight, not including fasteners, was roughly 14 pounds so we would like to stay below 17 with the addition of an oil cooler and any new revisions to the radiator design in the spring.

Characteristic	Design Requirement
Steady State Temperature	105°C
Target Heat Rejection @ 105°C/27°C	22.5kW
Fan Current Draw	<9 Amp
Pump Current Draw	<4.3 Amps
Mass	<17lbs (dry mass)

Table A: Design Requirements

Support for Design Decisions

The dynamometer had a plethora of issues during operation during the fall semester. Stemming from this, the design cycle of the cooling system suffered immensely and the team leads chose to, instead of simply delaying further, prepare the main component selections from ARG17 to be manufactured and placed on ARG18. Following this decision, the main job for me became reviewing the designs and selections of components from ARG17's cooling system, ensuring that as many components as possible would be compatible with any pending radiator selection changes in the spring semester. Reviewed designs include the catch can, mounting interfaces, fan selection, and pump selection.

Beginning with the catch can, the design drivers for this specific part are capacity, which must be .5L or 10% of the contained coolant volume; packaging, as the catch can should be close to the radiator; and manufacturability. Beginning with last year's design, I noticed that the enclosed volume was sufficient for a system of over 4.5 gallons (around 17 liters). Although our system cannot be drained easily to obtain an accurate volume by filling with a known volume, a more realistic conservative overestimate of 1.5-gallon (~5L) system capacity was used to design a new catch can. Initially the design I created was a rectangular prism, but this was not possible to package between the fan shroud and monocoque. In the interest of keeping the catch can as close to the radiator as possible the profile was changed to a parallelogram, allowing placement as shown below:

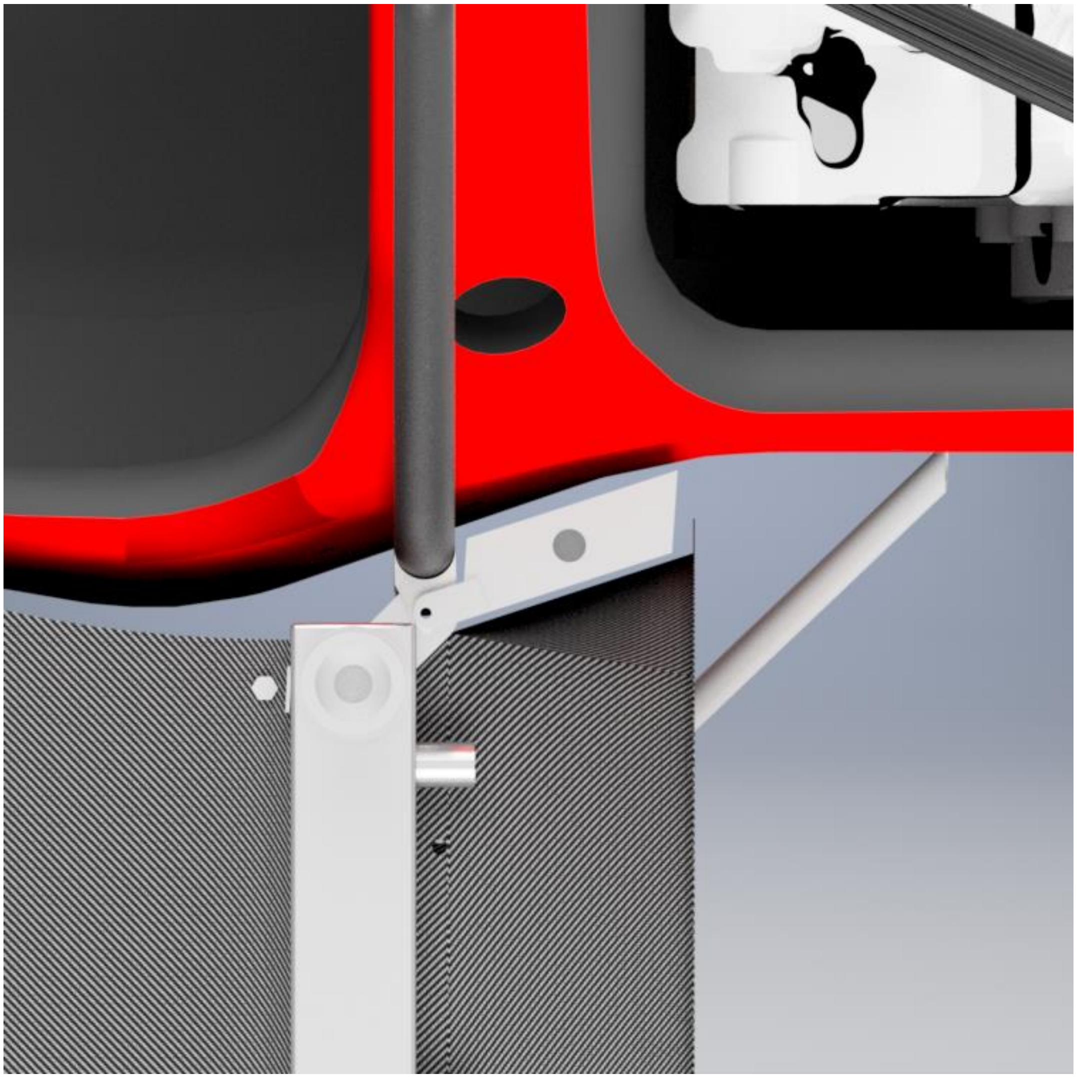


Figure 3: Catch Can Positioning Between Fan Shroud and Monocoque

Mounting tabs were revised for the catch can and radiator so that each may be sent to be cut by waterjet. The mount for the pump remains unchanged, as the current design is very easy to manufacture, as well being a simple solution for constraining the pump.

Pump selection is driven by counteracting the pressure drop through the system. The method employed to design ARG17's system was a very good method of performing this balance. Below is a plot from ARG17's report which shows the characteristic delivery pressure of several BOSCH electric water pumps vs. delivery flow rate as well as curves showing the expected pressure drop vs. flow rate through several proposed radiators from last year [1]. In order to account for other constants in the system, the pump's characteristic curve already has the pressure drop through the engine and oil cooler subtracted from it. The intersection of a radiator and pump curve signifies the steady-state operation flow rate of the pump-radiator combination.

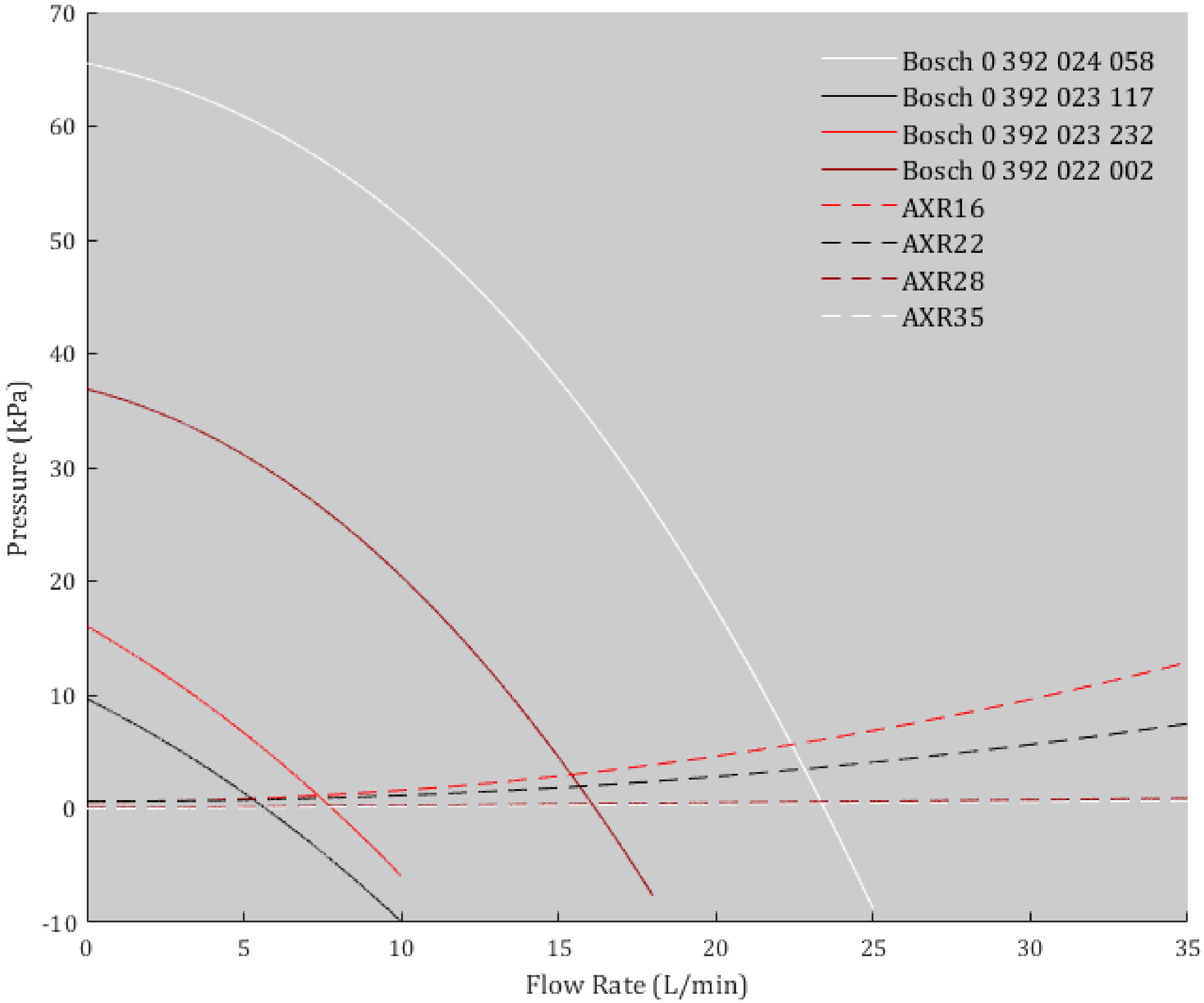


Figure 4: Pressure Delta vs. Flow Rate for BOSCH Pumps and MAHLE Radiators

One design aspect which was not considered last year is the fact that the steady state flow rate may be the flow rate we should design the radiator to operate at. This will require more research, but for the moment this process should be sufficient in ensuring the pump and radiator will be compatible.

The selection of the fan follows a similar process. Below are the curves used to select the SPAL 30100398.

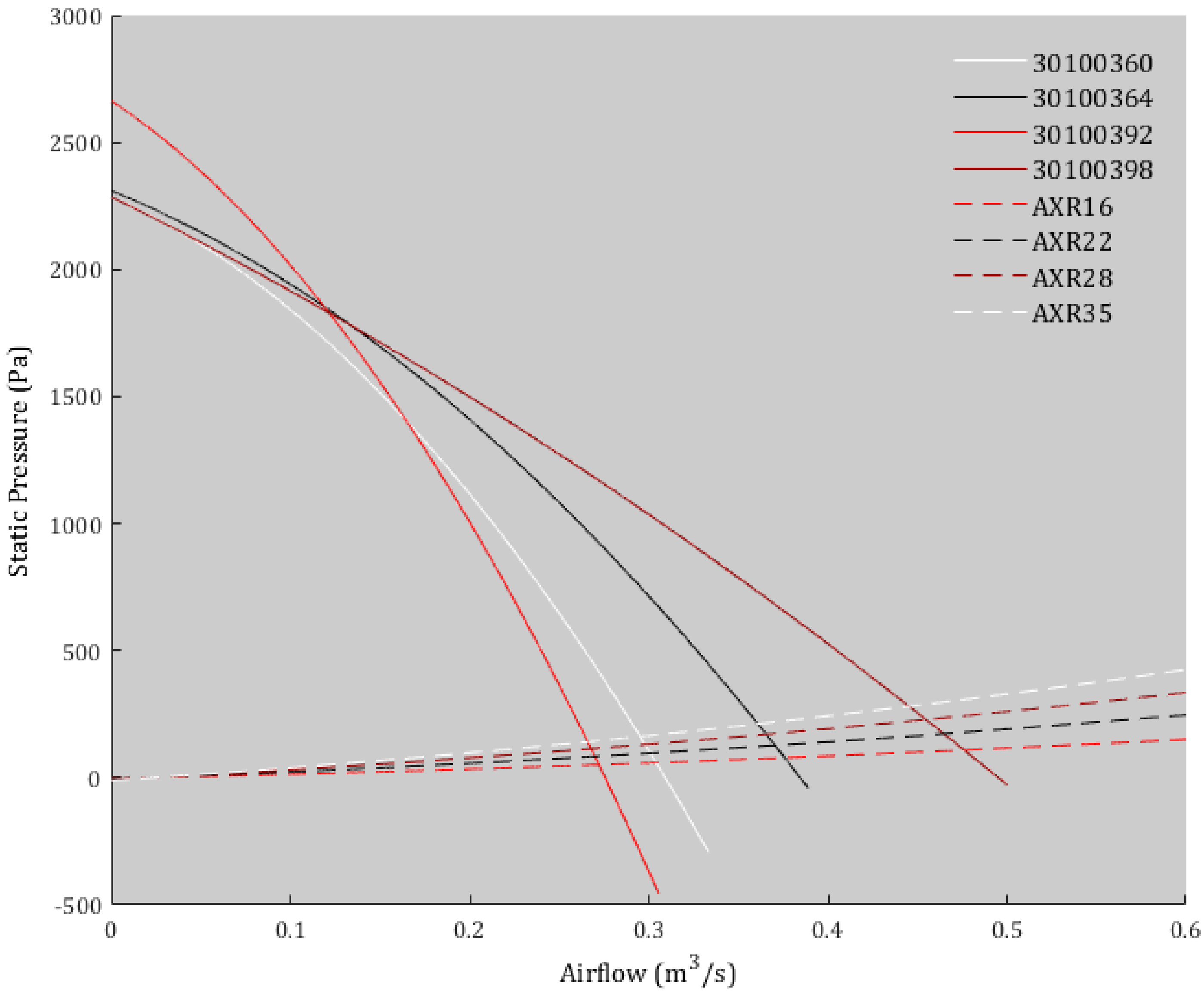


Figure 5: Pressure Delta vs. Flow Rate for SPAL Fans & MAHLE Radiators

Again, the intersection of two curves signifies the steady-state operation airflow point of a radiator and fan module, in our case .477m^3/s. So as to not simply copy the whole of Ayinde Crear's ARG17 justification, any further justification for this system may be found on the S: Drive [1].

Design work continued into the spring semester, with my main goal being to test and fit an air-to-oil cooler to the car. Unfortunately, as the semester picked up and I became engrossed in other obligations (mostly still within the team), my work on the cooling system declined until the very end of the semester when I tried to rush to get the cooler on the car. As usual, such a tactic did not work out, and I was unable to install a new oil cooler.

Systems Integration

The cooling system as a whole has interfaces with the following parts: roll hoop, engine case, oil cooler, rear lower front suspension clevis, and monocoque. Both the radiator and catch can share a mounting tab on the roll hoop, and the water pump's mount is bolted directly into the monocoque. The plumbing is routed such that we will not have to drill holes in the monocoque, as the only line that would somewhat benefit from doing so is the line to the oil cooler, and the accessibility and ease of repair is more desirable than the weight savings. Additionally, the fan shroud and radiator shroud will bolt directly onto the radiator, with the fan bolting to the fan shroud.

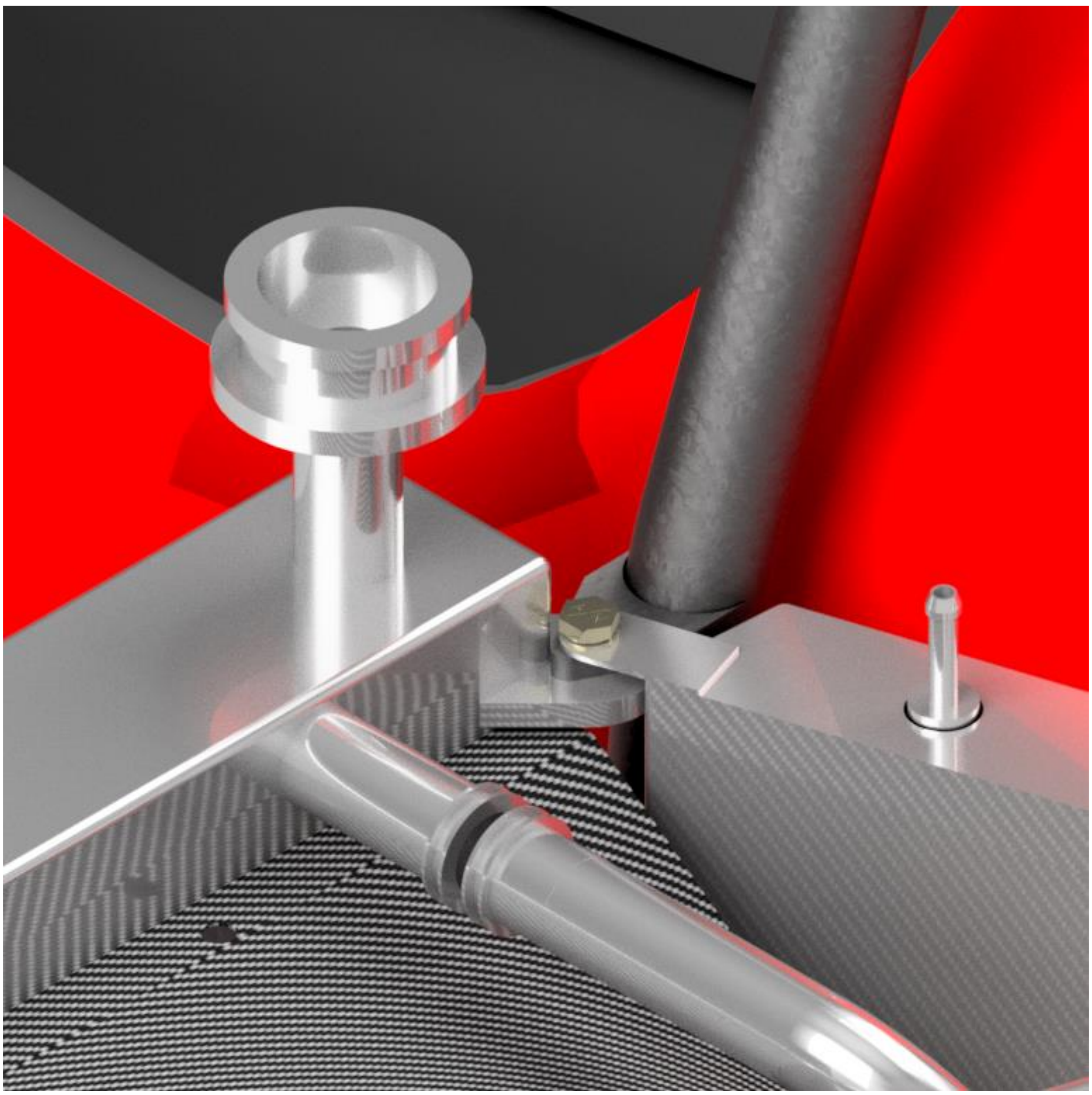


Figure 6: System Interfaces with Roll Hoop

Final Design Specifications

Table B: Final Design Specs

Parameter	Value
Weight	16.5lbs
SPAL 30100398 Current Draw	7.9A
BOSCH 0 392 022 002 Current Draw	3.8A

Table C: CAD Weights & Real Weights (lbs)

Components	Description	CAD Weight	CAD + Fasteners	Real Weight	Real + Fasteners
System	**Overall**	**15.93**	**16.4053**	**16.275**	**16.4878**
Lines	(CO18 01C22) Rad to Pump	0.066	0.066	0.087	0.087
	(CO18 01C18) Pump to LCM	0.068	0.068	0.183	0.183
	(CO18 01C17) Pump to Oil Cooler	0.093	0.093	Incl. Above	Incl. Above
	(CO18 01C23) UCM to Rad	0.202	0.202	0.373	0.373
	(CO18 01C19) Oil Cooler to Rad	0.116	0.116	Incl. Above	Incl. Above
	Coolant	N/A	N/A	5.07	5.07
Other Plumbing	(CO18 01C10) Line Ferrules (6)	0.066	0.066	0.033	0.033
	(CO18 01C16) Manifold Ferrules (2)	0.056	0.056	0.06	0.06
	(CO18 01C13) Bleeder Bungs (3)	0.024	0.024	0.015	0.015
	(CO18 01C14) T3 Bungs (4)	0.036	0.036	0.022	0.022
	(CO18 01C05) LCM	0.056	0.08	0.057	0.081
	(CO18 01C04) UCM	0.063	0.087	0.042	0.066
	(CO18 01C12) Top Oil Cooler Line Reducer	0.014	0.014	0.013	0.013
	(CO18 01C11) Bottom Oil Cooler Line Reducer	0.012	0.012	0.011	0.011
	(CO18 01C20) Catch Can	0.541	0.5802	0.67	0.7092
	Bleeder Valve (3)	N/A	N/A	0.01	0.01

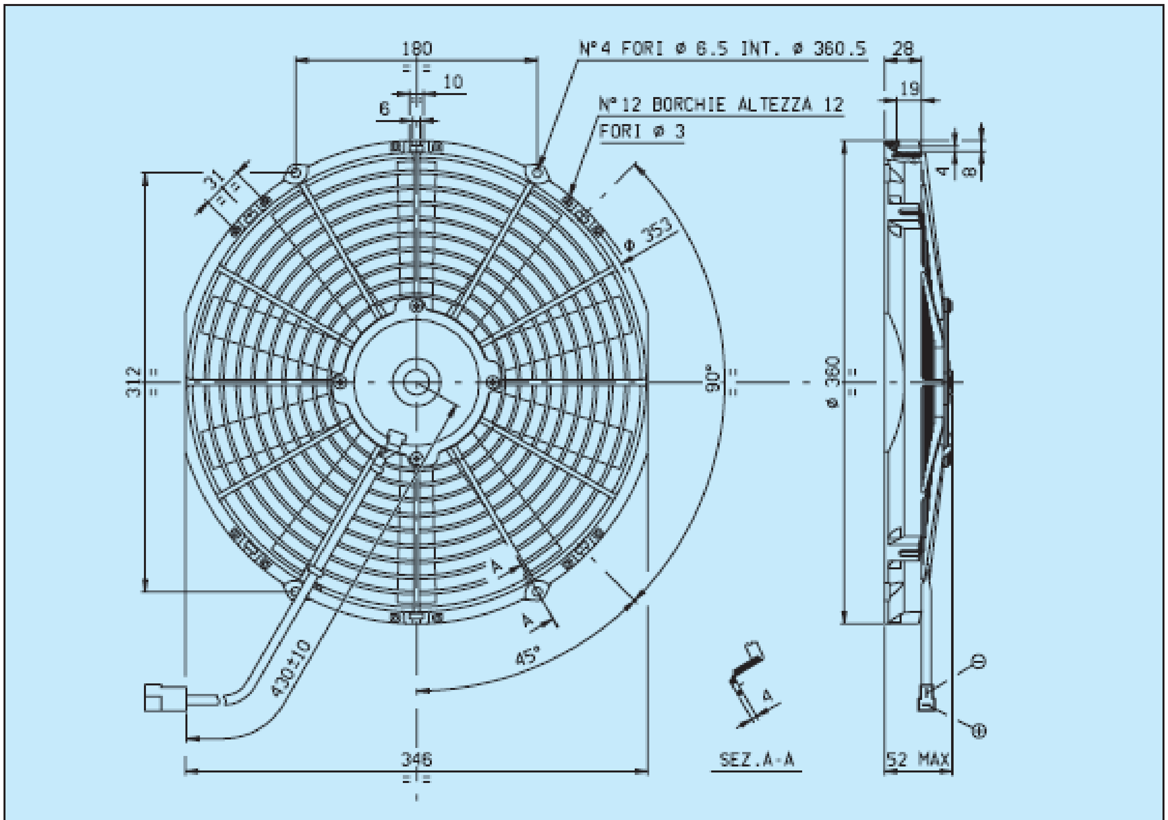
Major Components	Filler Head	N/A	N/A	0.061	0.061
	Filler Cap	N/A	N/A	0.09	0.09
	SPAL 30100398	N/A	N/A	2.47	2.5404
	BOSCH 392 022 002	N/A	N/A	2.4	2.42
	MAHLE AXR28	N/A	N/A	4.21	4.21
Mounting & Fastening	(CO18 01C02) Radiator Mount Tabs	0.036	0.036	0.047	0.047
	(CO18 01C21) Catch Can Mounting Tabs	0.012	0.012	0.006	0.006
	(CO18 01C08) Radiator Support Strut	0.237	0.237	0.139	0.139
	(CO18 01C06) Pump Mount	0.023	0.0582	0.044	0.0792
	Wiggins Clamps (2)	N/A	N/A	0.162	0.162
	Roll Hoop Mounting Tabs (2)	N/A	N/A	0.04	0.04

SPAL AUTOMOTIVE

ELETTROVENTILATORI ASSIALI
AXIAL MOTOR FANS

12V. C.C. - *D.C.*

Cod.	Tipo - *Type*	Diametro ventola - *Fan diameter*
3010.0398 **3010.0399**	**VA13 - AP9/C - 35A** Aspirante - *Suction* **VA13 - AP9/C - 35S** Soffiante - *Blowing*	**330 mm**



Connettore tipo AMP Cod.180908 - Faston maschi tipo AMP Cod.42098 - 2
Plastic connector AMP Cod.180908 - Male flat AMP Code 42098 - 2

* Peso 1,120 kg. circa
* *Weight 1,120 Kg. approx.*
* Motore chiuso, IP 68
* *Waterproof motor, IP 68*
* Lunga durata
* *Long life*
* A richiesta (per applicazioni OEM) disponibile anche in versione LL e VLL
* *Available also in LL and VLL versions for OEM applications only.*
* Accessori disponibili: tutti i kit di fissaggio.
* *Available accessories: all the fixing kits.*

Tensione di prova 13 V cc - *Test voltage 13 V DC*

Pressione statica *Static pressure* mm H_2O	Portata *Airflow* m^3/h	Corrente assorbita *Current input* A
0	1750	7,1
2,5	1550	7,5
5	1400	7,7
7,5	1170	7,8
10	820	7,9
12,5	500	7,8
15	280	7,7
17,5	0	7,7

Static pressure: 1 mm H_2O = 0,04 in. H_2O
Airflow: 1 m^3/h = 0,59 cfm

Water-circulation pumps with D.C. motors

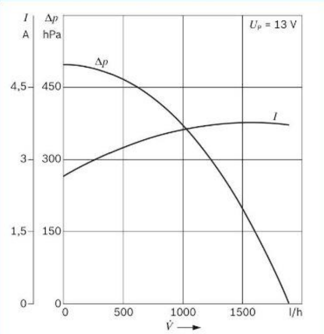
PCA | 12 V

Technical data

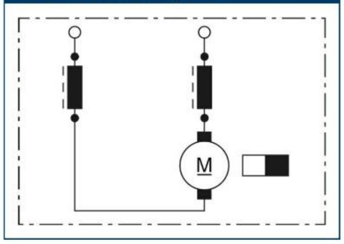
Part number		**0 392 022 002**
Nominal voltage	U_N	12 V
Delivery	$\dot{V}$	1200 $dm^3 \cdot h^{-1}$
Delivery pressure	p	0,3 bar
Direction of rotation		R
Operating mode		S 1
Degree of protection		IP 5 K 4[1])
Weight		approx. 1,0 kg

[1]) Applies only with receptacle housing in place

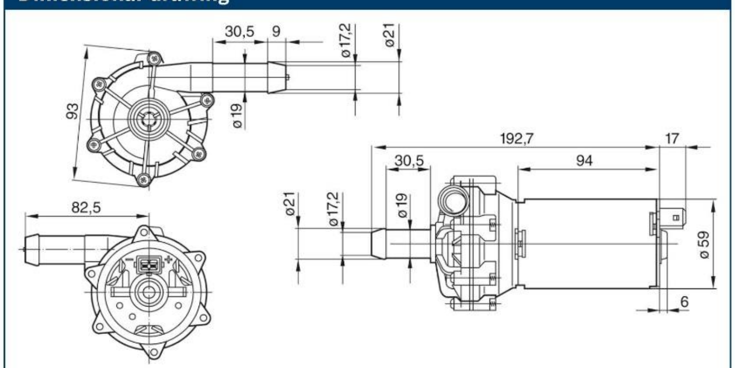
Characteristic curve



Connection diagram



Dimensional drawing



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Automotive Aftermarket
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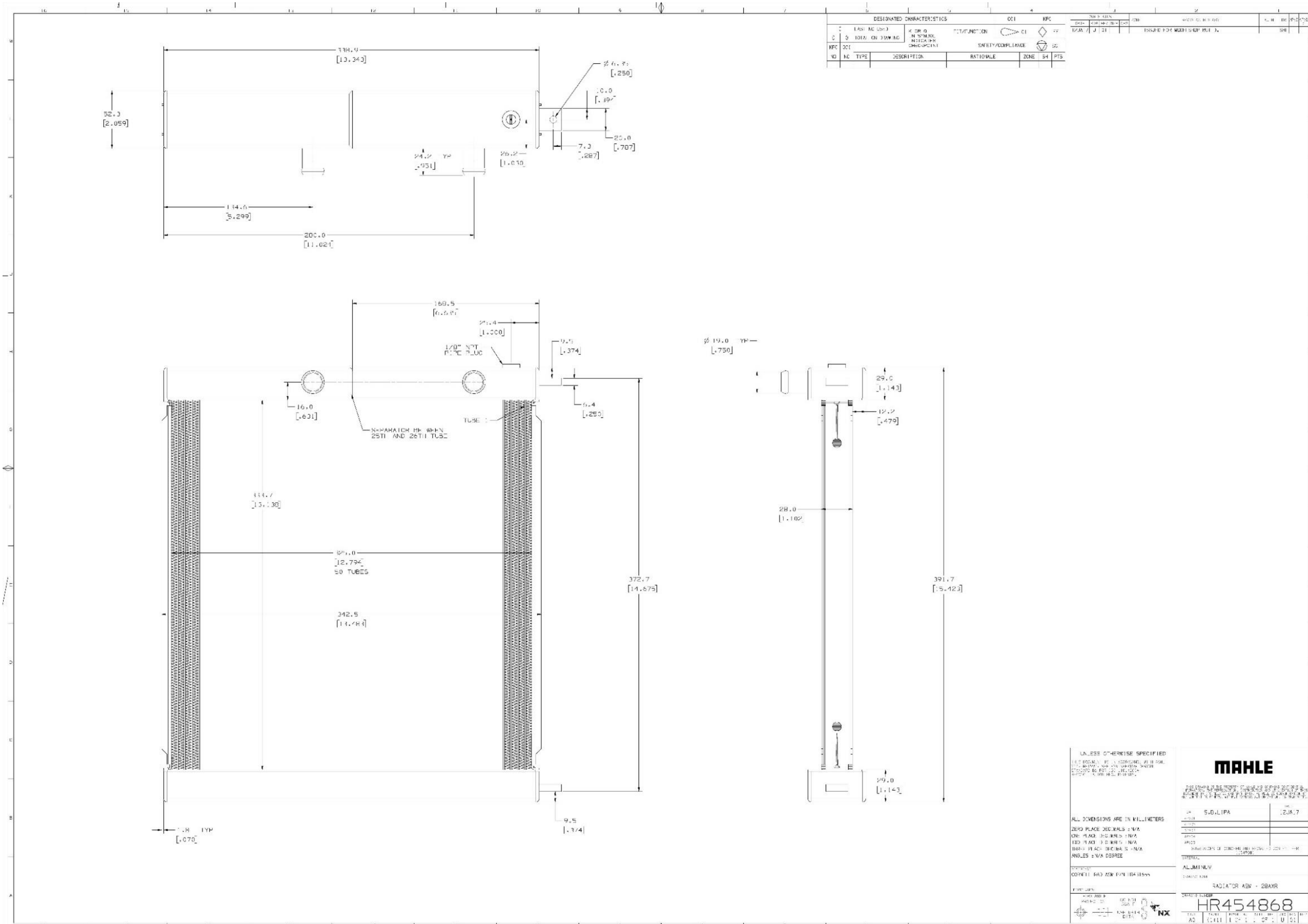
76225 Karlsruhe
Germany

www.bosch-elektromotoren.de

BOSCH
Invented for life



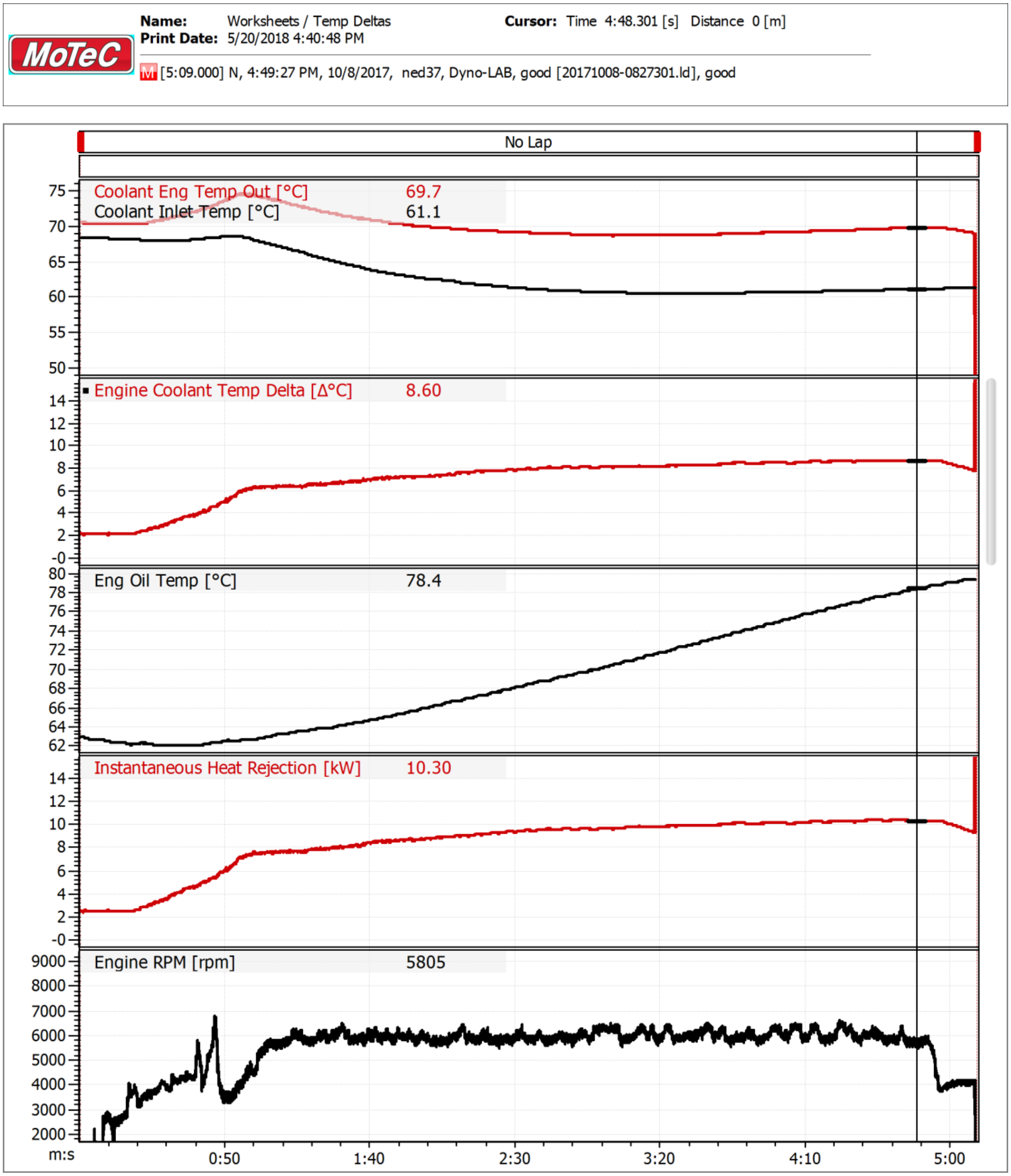
DESIGNATED CHARACTERISTICS
FIT/FUNCTION
SAFETY/COMPLIANCE
DESCRIPTION
RATIONALE
ZONE
338.9
[13.343]
52.3
[2.059]
ø 6.35
[.250]
10.0
[.394]
20.0
[.787]
7.3
[.287]
24.2 TYP
[.951]
26.2
[1.030]
134.6
[5.299]
280.0
[11.024]
168.5
[6.634]
25.4
[1.000]
1/8" NPT
PIPE PLUG
9.5
[.374]
ø 19.0 TYP
[.750]
16.0
[.631]
6.4
[.250]
SEPARATOR BETWEEN
25TH AND 26TH TUBE
TUBE 1
333.7
[13.138]
325.0
[12.794]
50 TUBES
342.5
[13.483]
372.7
[14.675]
29.0
[1.143]
12.2
[.479]
28.0
[1.102]
391.7
[15.423]
29.0
[1.143]
9.5
[.374]
1.8 TYP
[.070]
UNLESS OTHERWISE SPECIFIED
ALL DIMENSIONS ARE IN MILLIMETERS
ZERO PLACE DECIMALS ±N/A
ONE PLACE DECIMALS ±N/A
TWO PLACE DECIMALS ±N/A
THREE PLACE DECIMALS ±N/A
ANGLES ±N/A DEGREE
MAHLE
S.D.LIPA
12JA17
ALUMINUM
RADIATOR ASM - 28AXR
HR454868
NX

5/22/18

Testing

Several tests were performed this year: the heat generation test was attempted, on-car evaluation of ARG17's system, off-car evaluation of the sidepod and fan shroud, pump flow-rate testing, partial oil cooler testing, and on-car evaluation of ARG18's system.

The goal of the heat generation test was to form a table of heat generated by the engine at a list of different RPM-MAP combinations. This is done by running the engine at the desired point until the temperatures entering and leaving the radiator stabilize, indicating that the amount of heat being generated by the engine is equal to that being rejected by the radiator-fan module. This allows us to calculate the heat generated by the engine by adding the heat rejected by the radiator-fan module on the dynamometer to any instantaneous heat added to the engine's oil. This test can also be done by running the engine until the temperature delta through the radiator is stable, then adding the instantaneous heat added to the coolant (based on equally increasing coolant inlet/outlet temp). Unfortunately, the heat generation test was not completed due to issues running the engine on the dynamometer. Below is the data from a sample point from the heat generation test (6000 RPM, 115kPa manifold pressure), with the math used channels outlined below. (*Figure 8*)



$$Engine\ Coolant\ Temp\ Delta\ [dC] = abs('Coolant\ Inlet\ Temp'[C] - 'Coolant\ Eng\ Temp\ Out'[C])$$

$$Instantaneous\ Heat\ Rejection\ [kW] == \frac{'Engine\ Coolant\ Temp\ Delta'[dC] * \left(C_{water}\left[\frac{J}{kg}\right]\right) * \left(\dot{m}_{water}\left[\frac{kg}{s}\right]\right)}{1000}$$

Figure 7: Heat Generation Test Data – 6000RPM, 115kPa

The above math channel does not account for instantaneous rate of heat added to the oil. Account for this by approximating the flow rate of oil through the oil cooler to be the full capacity of the internal oil pump, which has a flow rate of 4.22L/min/1000RPM according to a [2], as well as the density of 10W-40:

$$\dot{Q}_{oil} = (m_{oil})(C_{oil})(\frac{dT}{dt}) = (\rho_{oil})(\dot{V}_{oil})(\Delta t)(C_{oil})(\frac{dT}{dt})$$
$$= \left(\frac{823.9g}{L}\right)\left(\frac{.070L}{s}\right)(53.5s)\left(\frac{2.6J}{g * {^\circ}C}\right)(.069\frac{{^\circ}C}{s})$$
$$= .554kW$$

Adding this to the value for the instantaneous heat rejected by the radiator gives us the total heat generation value for the desired point:

$$.554kW + 10.30kW = 10.854kW$$

The results of the test were inconclusive, as we were unable to complete the table of heat generation values. If complete, we would have been able to calculate a weighted value based on time spent at each load case (the same method of weighting used for the Conservation of Energy equation outlined in the Introduction) and compare this value to past year's design – thus invalidating or confirming our assumptions made in the conservation of energy equation. If completed this test would have been valid for the lifetime of our current engine (CBR600RR with KP35 turbo).

On-car testing was performed to validate last year's design as well as evaluate the accuracy of the simulation spreadsheet GM supplied to us (this is the same spreadsheet used to select the radiator for at least the past five years). In an effort to form a more holistic understanding of the system simulation performed in the past I spoke with our contact at GM Lockport, Steve Showalter. Because Steve will be retiring in the spring of 2018, this effort sought to preserve the method of simulation for future years, had it proven to be accurate (An unmodified copy of the spreadsheet is located in S:\Cars\ARG18\Engine\Cooling\Radiator Sim). To test the accuracy of the spreadsheet we ran the car at idle without the fan plugged in until the coolant reached approximately 100°C. Once up to temperature, the fan was plugged in until the coolant temperature was lowered to around 80°C. We then performed a simulation using Steve's spreadsheet to determine what the predicted target heat rejection is under the same conditions as each trial including zero vehicle velocity, the AXR28 radiator, fluid flow rates, the inlet temperature from the data, and ambient air temperature. Below is the data from the on-car test, note that trial 5 was invalid because the engine stopped briefly (*Figure 4*). Table 1 contains summary data from the test, noting the points of maximum heat rejection in each trial, as well as simulation results from the given conditions.

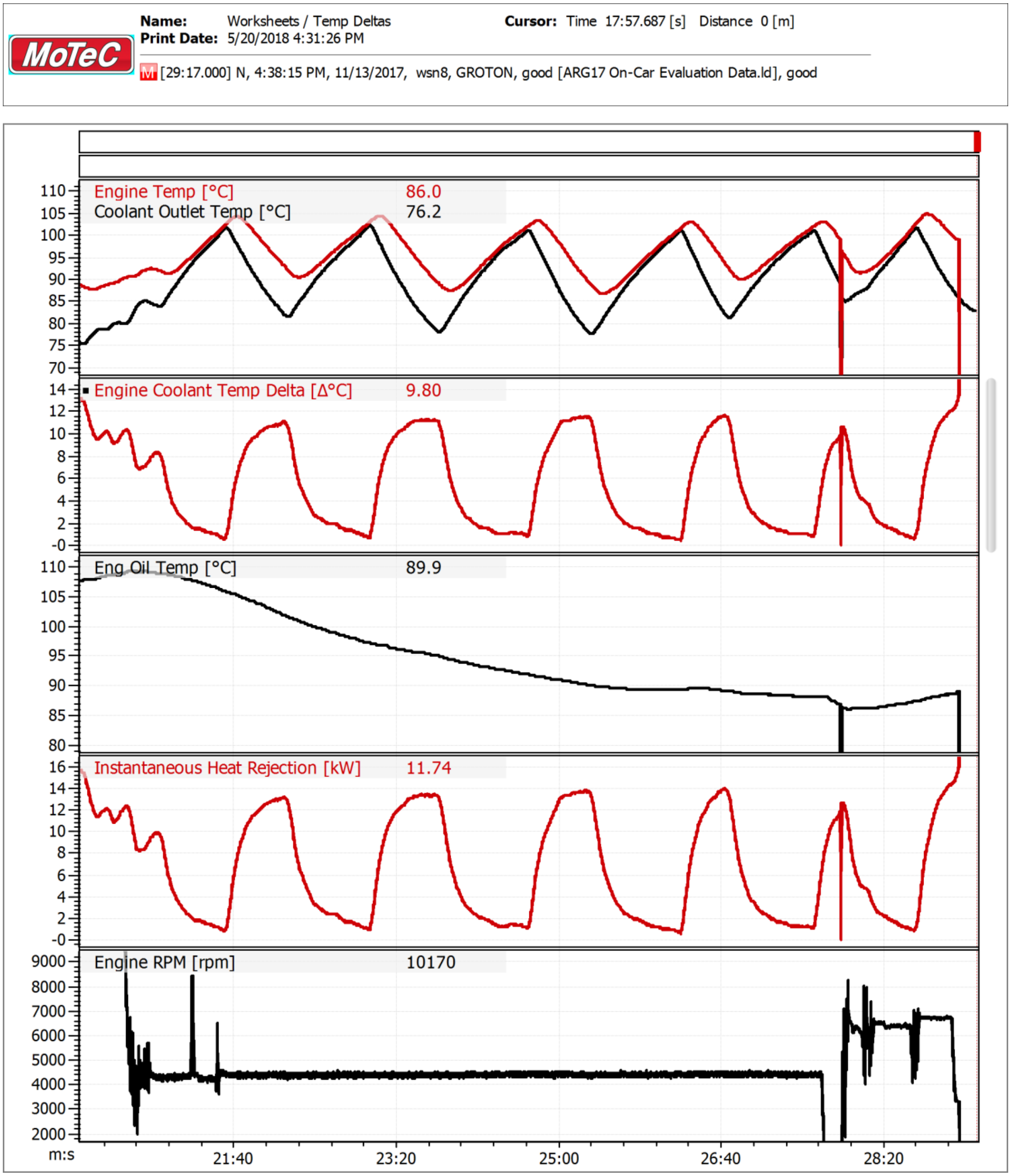


Figure 8: ARG17 On-Car Validation Data

	Trial 1	Trial 2	Trial 3	Trial 4	Trial 6
Inlet Temp (°C)	93.2	90.0	90.3	94.0	99.9
Temp Delta (°C)	11.10	11.3	11.6	11.70	12.10
Heat Rejection (kW)	16.60	16.90	17.35	17.50	18.09
Simulated HR (kW)	22.88	22.05	22.13	23.09	24.63
Coolant Flow (l/m)	20.9				
Air Flow (m^3/s)	.277				
Ambient Temp (°C)	5.3				

Table D: ARG17 On-Car Validation Data Summary

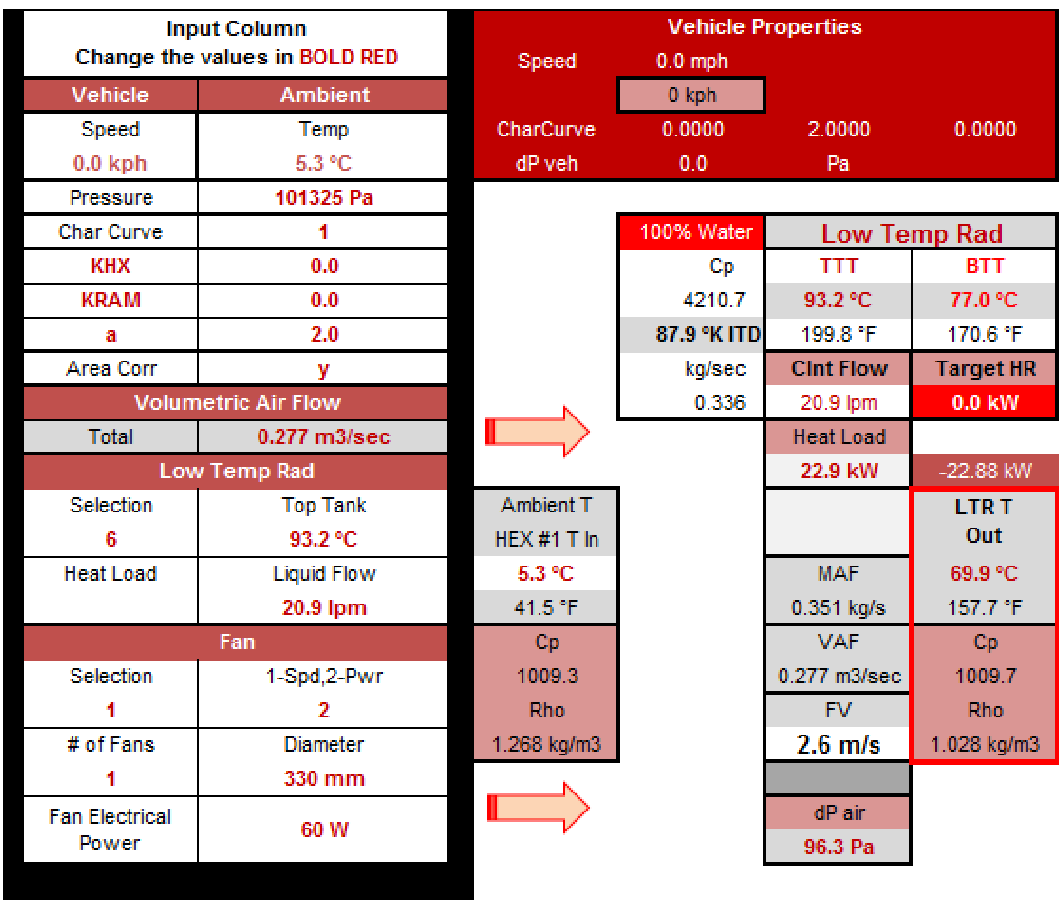


Figure 9: Sample Simulation (Trial 1)

The discrepancy in Table 1 between the tested heat rejection value and the simulated heat rejection value is cause for concern. For this reason, I have since contacted Steve to discuss possible sources of error. These could be anything from poorly organized heat transfer equations (thus an incorrect characterization of the radiator) to improper use of the spreadsheet by myself. At the very least, we can use this as a rough estimate, as there is still a consistent ~75% average ratio between tested and simulated heat rejection.

Off-car tests profited the values used in the above table for Air Flow and Coolant Flow. Coolant flow was tested by myself using the pump on the dyno (the same BOSCH 0 392 022 002 electric water pump as on the car), using a simple bucket test. On a bled and primed system, water flowed for a measured amount of time into a container of known mass. Subtracting the mass of the container and dividing the difference by the measured time nets mass flowrate. Dividing by density offers volumetric flowrate. These values may deviate from those actually achieved using the car's setup, the cause of which is different plumbing geometry and material, resulting in differing values for pressure drop. However, these values should be able to serve as a rough estimate, as the main resistances to flow are still the engine and oil cooler.

Trial	Mass Flowed (kg)	Vol. Flowed (m^3)	Δt (s)	Flow Rate (kg/s)	Flow Rate (m^3/s)	Flow Rate (L/m)
Oil Cooler Outlet						
1	3.365	0.003373	47.43	0.071	0.00007112	4.267
2	2.396	0.002401	33.55	0.071	0.00007158	4.295
3	2.262	0.002268	32.54	0.070	0.00006970	4.182
Average				0.07062	0.00007080	4.248
Engine Outlet						
1	3.926	0.003936	13.70	0.2866	0.0002873	17.24
2	3.929	0.003939	13.90	0.2827	0.0002834	17.00
3	3.932	0.003942	13.79	0.2851	0.0002858	17.15
Average				0.2848	0.0002855	17.13

Table E: BOSCH 0 392 022 002 Flow Rate Testing

Jacob Rigos and Carlos de la Torre obtained the data for Air Flow. The procedure follows as such: sectioning the face of the radiator within the shroud into nine regions of equal area, placing a hot wire film sensor at the center of each region, and measuring the air velocity at that specific point. By averaging the values obtained (because the sections were of equal area) and multiplying by the face area of the radiator (333.7mm x 324mm), total volumetric airflow was obtained.

Air Velocity [m/s]		Position from left edge [in]		
		1.5	6.5	11.5
Depth from top [in]	1	2.6	2.4	3
	6.5	2.6	3.5	2.4
	11	2.2	2.4	2.1

Table F: Air Velocity into the Radiator

Manufacturing

The following tables show a breakdown of which parts were outsourced, machined in-house, or purchased.

In-House Part	Special Processes	Estimated Hours
Upper Cooling Manifold	NC, Welding	8
Lower Cooling Manifold	CNC	8
Radiator Support Strut	Welding	2
Filler Neck		1
Bleeder Bungs (4)		5
Sensor Bungs (4)		5
Ferrules (8)		
Plumbing	Welding	15
Pump Mount	Welding	2
Catch Can	Welding	4
Outsourced Part	**Process (If Applicable)**	**Notes**
Radiator Mounting Tabs (x4)	Waterjet	
Roll Hoop Mounting Tabs (x4)	Waterjet	
Catch Can Mounting Tabs (x4)	Waterjet	

Component	Mass	Quantity	Supplier	Cost	Status
SPAL 30100398	2.47 lbs (1.12 kg)	1	Amazon	$81.35	Received
BOSCH 0 392 022 002	2.4 lbs (1.08 kg)	1	Amazon	$89.00	Received
MAHLE AXR28	4.21 lbs (1.91)	2	MAHLE	$0.00	Received
21J02-12A Clamps	.0034 lb (.016 kg)	0	Ebay	$10.00	Received
Filler Head	.061 lbs(.028 kg)	2	FR.Sport	$15.00	Received
Bleeder Valve	.0034 lb (.016 kg)	0	Ebay	$2.49	Received

Table G: Manufacture/Order Summary

Needed Tool	Location
Line Bender	Lab, behind the grinding wheel
Tube Cutter	Cooling bin, Tool Chest (Scissors)
Tube Beader	Taps & Dies drawer

Table H: Manufacturing Tools

To keep costs low, the only purchased parts were those which were either too complex for us to manufacture ourselves, keeping labor limitations in mind, or very cheap to purchase. The Filler Head, for example, is only $15, but would require CNC lathe and NC time for us to manufacture in-house.

The Cooling Manifolds are the two most time consuming standalone parts to manufacture. Both manifolds require CNC time, however this is unavoidable because the shape of our plumbing does not match the shape of the opening on the engine case (the engine case has an oval-shaped hole). As a result, there will always be a set of lofted profiles unless the outer shape of the manifolds are made into a rectangle, however doing so would result in a heavier, less optimized part.

Several of the remaining machined parts are turned on a lathe including bleeder bungs, sensor bungs, and line ferrules. The Filler neck can be almost any tube stock that fits on the top surface of the radiator's tank and around the bottom of the filler head, and welding this is not a real issue. Bungs and ferrules need to mitered to fit more specific dimensions after the rest of the plumbing is welded, so their machining process can be fairly imprecise (with the exception of the o-ring groove on the ferrules) as they will change later anyway.

The plumbing should take the longest to manufacture. While the CAD model will be roughly correct, there will need to be some degree of trial and error (bending, checking to see if the bend is enough, correcting if necessary). Most of my time spent over Janman this year was working on the cooling lines. Each connection needs careful mitering to fit the radiator's position relative to the manifolds, and welding them takes time, as each mitered area must be marked for jigging somehow (I usually just made the tubes concentric then scribed a couple marks to ensure correct clocking). For each weld joint, there was usually some way to hold each piece in a clamp such that the two pieces meet correctly. This was an easy task until the lines started getting more and more complex as they took shape. I refrained from welding any bungs onto the lines until everything else on the system was completely finished to avoid any clearance issues.

Finally, the catch can design allows it to be waterjet from sheet metal as a box pattern, as this year it will had a parallelogram profile rather than last year's misshapen trapezoid profile. While welding the can together it is also key to drill a relief hole in one of the faces so that the air contained does not interfere with shielding gas. This is fine since the catch can has to have a relief hole anyway. Welding the mounting tabs to the top and bottom of the catch can allow it to fit nicely between the radiator's own mounting tabs for better packaging.

We removed the radiator support strut this year after the first one broke from an insufficient connection with the suspension clevis, because we realized that its existence meant that the failure mode of the radiator mounting was by punching a hole in the radiator's lower tank. It is more desirable to break the cooling lines than to do that, as obtaining a new radiator is more difficult than welding lines back together.

Post Competition Remarks

During the spring semester there were a few complications regarding the cooling system. Primarily the plan to retrofit a second iteration including a newly sized radiator based on a completed heat generation test, which was eventually scrapped in favor of simply continuing to select an air-to-oil cooler based on reasonable estimations of the heat load from the oil (derived from points already swept by the heat generation test). This also failed, as I was unable to complete all the necessary tasks before the hard cutoff for making changes to the car before competition, but not before I designed a blanking plate to bolt onto the engine case. The blanking plate sat where the stock oil cooler sits, in order to remove the weight of the stock oil cooler once an aftermarket one was installed. The full parts are available in ARG18\EN-COOL\Final Assembly on the vault.

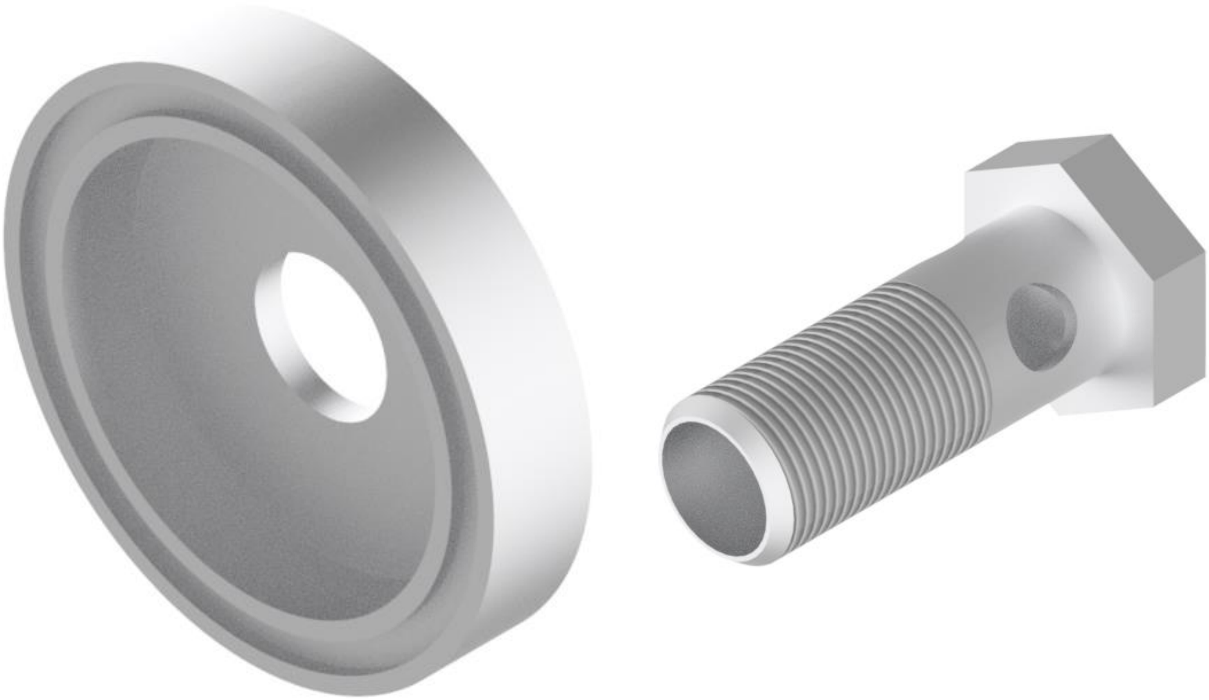


Figure 10: Oil Cooler Blanking Plate (Left) and Oil Cooler Blanking Bolt (Right)

As far as inexplicable failures there were none, however initially it did seem as though that would be the case. After a few rainy driving days spent at the Fairgrounds, we noticed that the system was performing poorly. On a couple of occasions, the system reached its pressure limit during slow driving and actually spewed out of the catch can. At first, we were confused and assumed it was due to mismatched T3 sensor wires causing the fan to regulate off the incorrect temperature (coolant out rather than coolant in). As the issue persisted, we decided to investigate and found that driving through the puddles at the fairgrounds had left the radiator covered in algae – to the point where the radiator suffocated. We tried to remove all of the dirt and algae from the fins however it was simply impossible to remove the blockages without damaging fins. Swapping to a new radiator alleviated the issue.

Competition Research

Rochester Institute of Technology

Figure 11: RIT Sidepod

RIT employs a dual-sidepod setup with radiators about half as big as the AXR28 we run (I was unable to get exact dimensions). According to one of their members who may or may not have truly known what he was talking about, their system runs the radiators in series rather than in parallel. While beneficial from a balance standpoint, cooling efficiency suffers from running heat exchangers in series rather than in parallel due to the reduced temperature of coolant entering the second in series. Their method of affixing the radiator to a structural sidepod is something I like very much, and hope that we can use this is inspiration to move away from the roll hoop-mounted setup we use which is very difficult to remove. The process of removing their radiator is just removing the four bolts pictured above as well as the hose clamp - the radiator then slides out of the sidepod.

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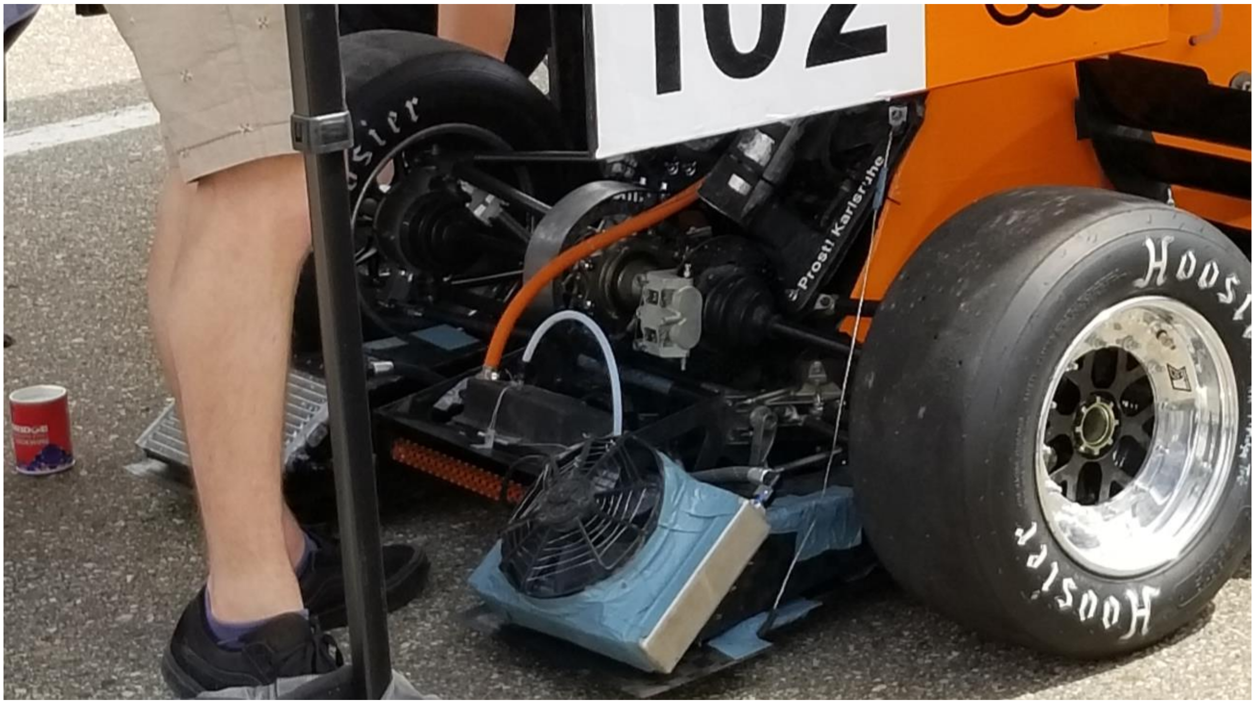

Figure 12: TU München Rear-Mounted Radiators

Of the competing teams discussed here, TU München has the most radically different package from our own – dual rear-mounted radiators. Both times that I visited their paddock in attempt to chat with someone knowledgeable, they were extremely busy. Because of this I was only able to get in a quick conversation about their setup. The left radiator is their massive oil cooler. I do not understand how they are able to use this large of a radiator without overcooling their oil. Perhaps not having a fan truly does detriment the airflow through (something to look into). If so, this seems like an easy way to maintain both symmetry and weight balance L/R while achieving well-calibrated oil cooling. Both of these radiators are mounted facing directly into the diffuser of their undertray, and the water radiator on the right has a pulling fan mounted on the back, with a carbon fiber shroud similar to ours (sealed with the tape pictured here). I would also like to add that TU München won the design event this year, and while that does not necessarily mean that their cooling was perfect it does imply their design process is well organized.

École de Technologie Supérieure

Figure 13: ETS Sidepod

ETS runs a setup very similar to RIT, with the addition of an oil cooler. I like their setup a lot because the oil cooler they selected is very small (around 4 inches square) and uses a computer fan to create airflow, with the oil cooler stacked behind the left radiator. This type of setup is the one I wish we could use because it demonstrates a strong design if implemented correctly, satisfying our cooling needs while being very light, aesthetically appealing, and balanced L/R. I forgot to ask them if the setup is series or parallel, but I would guess that the radiators are in series based on the cooling lines I was able to spot on the engine case not having any Y connections.

Rules

-T2.1a: The top 180 degrees of the wheels/tires must be unobstructed when viewed from vertically above the wheel.

-T2.1c: No part of the vehicle may enter a keep-out-zone defined by two lines extending vertically from positions 75mm in front of and 75mm behind, the outer diameter of the front and rear tires in the side view elevation of the vehicle, with tires steered straight ahead. This keepout zone will extend laterally from the outside plane of the wheel/tire to the inboard plane of the wheel/tire. See the figure “Keep Out Zones” below.

-T8.1: Water-cooled engines must only use plain water. Electric motors, accumulators or HV electronics may use plain water or oil as the coolant. Glycol-based antifreeze, "water wetter", water pump lubricants of any kind, or any other additives are strictly prohibited.

-T8.2.1: Any cooling or lubrication system must be sealed to prevent leakage.

-T8.2.3: Any vent on other systems containing liquid lubricant or coolant, i.e., a differential, gearbox, or electric motor must have a catch-can with a minimum volume of ten (10) percent of the fluid being contained or 0.5 liter (half U.S. quart), whichever is greater.

- T8.2.4: Catch cans must be capable of containing boiling water without deformation, and be located rearwards of the firewall below the driver's shoulder level, and be positively retained, i.e. no tie-wraps or tape.

- T8.2.5; Any catch can on the cooling system must vent through a hose with a minimum internal diameter of 3 mm (1/8 inch) down to the bottom levels of the Frame.

- T8.5.1: During technical inspection, the car must be capable of being tilted to a forty-five-degree (45°) angle without leaking fluid of any type.

- T8.5.2: The tilt test will be conducted with the vehicle containing the maximum amount of fluids it will carry during any test or event.

References

[1] Ayinde Crear, "ARG17 Spring Technical Report," Cornell FSAE, Cornell University. S:\Reports\2017 Car\Spring Technical\ARG17_Sp17_TechnicalReport_Cooling

[2] Ethan Carr and Michael Rogozinski, "FSAE Engine Dry-Sump Oiling System Design," Mechanical Engineering and Mechanics Department, Drexel University. Nov. 24, 2003.

[3] Ayinde Crear, "Line_Diameter_Pressure_Drop," Cornell FSAE, Cornell University. S:\Cars\ARG17\Engine\Cooling\ARG17 Matlab\Line_Diameter_Pressure_Drop.m