UNIVERSITY OF OULU

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DESIGNING A COOLING SYSTEM FOR A FORMULA STUDENT RACE CAR

Janne Kemppainen

DEGREE PROGRAMME IN MECHANICAL ENGINEERING

Bachelor's thesis 2020

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ABSTRACT

Designing a cooling system for a formula student race car

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University of Oulu, Degree Programme of Mechanical Engineering

Bachelor’s thesis 2020, 49 p.

Supervisors at the university: Mauri Haataja, Professor (Emeritus) &
Miro-Tommi Tuutijärvi, Doctoral Student

The aim of this bachelor’s thesis is to document the primary design of the liquid cooled cooling system for a Formula Student race car and to get acquainted with the basics of a modern liquid-cooled cooling system. The physics in the cooling is presented with the working principle of the cooling system itself. Finally, the primary design of a liquid-cooled cooling system for a Formula Student race car is presented with a brief summary of the results.

Another aim of this thesis is also to provide a comprehensive but brief package of information regarding the cooling system and its design especially for the new members of Formula Student Oulu (FSO) to help in the continuity of development and to ease the familiarization of the subject.

The applied theory has been gathered mainly on the subject-related literature. The emphasis in the theory is in the heat exchanger and water pump sizing due to dominant role in the cooling system.

Due to lack of component properties few assumptions had to be made during the design process but as the successful result shows, assumptions have been adequately good. However, this leads in the difficulty of design improvements due to number of possible parameter changes. Though, the design method is very applicable in the automotive use and thoughts about the development and next steps are presented also at the end of the thesis.

Keywords: cooling system, liquid-cooled, heat exchanger, Formula Student

TIIVISTELMÄ

Jäähdytysjärjestelmän suunnittelu Formula Student -kilpa-autoon

Janne Kemppainen

Oulun yliopisto, Konetekniikan tutkinto-ohjelma

Kandidaatintyö 2020, 49 s.

Työn ohjaajat yliopistolla: Professori (Emeritus) Mauri Haataja &
Tohtorikoulutettava Miro-Tommi Tuutijärvi

Tämän kandidaatintyön tarkoituksena on dokumentoida nestejäähdytteisen jäähdytysjärjestelmän suunnitteluprosessi sekä tutustua modernin nestejäähdytteisen jäähdytysjärjestelmän perusteisiin. Jäähtymisen teoria on esitetty jäähdytysjärjestelmän toimintaperiaatteen lisäksi. Viimeisenä esitellään Formula Student -kilpa-autoon suunniteltu jäähdytysjärjestelmä lyhyen tuloskatsauksen kera.

Työn toisena tavoitteena on toimia kattavana mutta tiiviinä informaatiopakettina erityisesti uusille Formula Student Oulun jäsenille helpottaakseen suunnittelutyön kehitystä ja aiheeseen tutustumista.

Teorialähteinä on käytetty pääasiassa alan kirjallisuutta. Painotus teoriassa kohdistuu jäähdyttimen ja vesipumpun mitoitukseen johtuen niiden suuresta roolista jäähdytysjärjestelmän toiminnassa.

Suunnitteluprosessin edetessä muutamia arvioita jouduttiin tekemään puutteellisten komponenttien ominaisuustietojen takia, mutta kuten tulokset osoittavat, arviot olivat riittävän hyviä. Tästä syystä tulevaisuuden kehitystyö on kuitenkin hieman vaikeampaa johtuen muuttujien suuresta määrästä. Suunnittelumetodi on kuitenkin erinomaisesti sovellettavissa autoteollisuuden kohteisiin ja parannusehdotuksia on listattukin työn loppupuolella.

Asiasanat: jäähdytysjärjestelmä, nestejäähdytteinen, jäähdytin, Formula Student

FOREWORD

During the term of 2018-2019 I worked as the Head of Powertrain at the Formula Student Oulu and my main engineering task was to design a new cooling system for our 2019 car, M03. This thesis is written in autumn semester 2019 after busy summer filled with testing and the main event for FSO, Formula Student Austria. The main reason and objective for this thesis is to document the design and also to help new members understand the basics of a liquid-cooled cooling system and the demands in the designing of it.

In the previous car, M02 Daytona, the cooling system design was unsuccessful considering the design and the documentation. Adding the fact that in the beginning of the design process my own knowledge was also very finite, a great amount of research was needed to be done to perform this task decently.

I would like to thank my family and friends for supporting me during all these years in the fascinating field of engineering. A huge thanks to all my team members in FSO, who have become good friends, for interesting discussions during coffee breaks and irreplaceable experience for the future. And finally, the biggest thanks to my beloved wife who still stands beside me after long working hours and always encourages me to be at my best.

Oulu, 5.1.2020

Janne Kemppainen

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NOMENCLATURE

FSO	Formula Student Oulu
ICE	Internal Combustion Engine
ITD	Inlet Temperature Difference
LMTD	Logarithmic Mean Temperature Difference
MTD	Mean Temperature Difference
NTU	Number of Transfer Units
SI	Spark Ignition

A	heat transfer area
A_c	the cross-section area of a flow
b	impeller blade height
C	total heat capacity
C*	heat capacity rate ratio
C_{max}	maximum heat capacity (of C_a and C_c)
C_{min}	minimum heat capacity (of C_a and C_c)
c_p	specific heat capacity at constant pressure
c_v	specific heat capacity at constant volume
D	pipe diameter
F	correction factor
f	Darcy-Weisbach friction factor
g	standard gravity
$H_{e,ic}$	exhaust enthalpy loss due to incomplete combustion
h	heat transfer coefficient
$h_{e,s}$	sensible part of exhaust enthalpy
h_i	ideal or maximum head
h_p	pump head
h_L	head loss
h_{Le}	engine head loss
h_{Lhe}	heat exchanger head loss
h_{Lmajor}	major head losses
h_{Lminor}	minor head losses
i	gear-ratio
K	head loss factor

K_L	loss coefficient
k	thermal conductivity
l	pipe length
l_c	characteristic length
$\dot{m}$	mass flowrate
n	frequency of crankshaft rotation
P_b	brake power of the engine
P_s	shaft power
p	pressure
Δp	pressure loss
Q	heat transfer rate
Q_c	heat rejected to the coolant
Q_{LHV}	lower heating value of fuel
Q_{misc}	heat rejected to oil, and surroundings from engine's external surface
q	heat flux
R	thermal resistance
ΣR_t	total thermal resistance through a wall
R_{cond}	thermal resistance in conduction
R_{conv}	thermal resistance in convection
Re	Reynolds number
r	radius
ΔT	temperature difference
T_e	engine torque
T^m	mean temperature
ΔT_{max}	inlet temperature difference (ITD)
ΔT_{mean}	mean temperature difference
T_s	shaft torque
$\frac{\mathrm{dT}}{\mathrm{dx}}$	temperature gradient
$\dot{\mathrm{V}}$	volumetric flowrate
$\mathbf{V}$	absolute velocity of the fluid particle
V_d	engine displacement
v	average velocity
v_r	radial component of the absolute velocity
v_θ	tangential component of the absolute velocity
$\mathbf{U}$	velocity of the blade

U	overall heat transfer coefficient
z	height of the streamline from a zero-level
α	angle of the absolute velocity
β	angle of the blade
ε	effectiveness
ε_r	equivalent roughness
δ	wall thickness
η_0	extended surface efficiency
ϕ	functional relationship
ρ	density
ω	rotational velocity
μ	dynamic viscosity

Subscripts

a	air
c	coolant
f	fluid
hot	hotter fluid
in	inlet of a control volume
out	outlet of a control volume

1 INTRODUCTION

During the term 2017-2018 I was a junior member of FSO, and alongside own tasks I witnessed major problems in the cooling system of the 2018 car, M02 Daytona. For the next year, the responsibility of Head of Powertrain was granted and the design task of a new, working and *cooling*, cooling system was also chosen. For the new 2019 car, M03, a new engine was also selected for benefits in range of areas. The new engine, from Yamaha MT-07, was a new acquaintance for the whole team and therefore made the design process a bit more complicated since no previous data was available.

At the beginning of the design, my knowledge considering cooling systems was limited to the practical side and therefore a fair amount of research was needed. A wide range of literature was investigated and a chosen selection of it is only referred in this thesis. Briefly, the design process was on its path but the lack of precise properties of certain components resulted in estimations that lead a little inconvenient zone during the design process but was laudably triumphed.

Since the design of the cooling system in the old car was failed also in documentation, and the engine change was done, no orientation or baseline was granted for the new design. Thus, the design is mainly based on literature and considering the past, designed as better safe than sorry, but still for a racing application. Both, thermo- and hydrodynamic, aspects are investigated.

2 THEORY OF COOLING SYSTEM

Cooling system has a very descriptive name due to its working purpose; it cools down the engine. During burning process of internal combustion engine (ICE), the peak temperature of over 2000 °C is reached and overheating of components must be prevented by cooling (van Basshuysen & Schäfer 2004: 555). The demand for cooling high temperatures can be divided in three reasons which are to promote a high volumetric efficiency, ensure proper combustion and secure mechanical operation and reliability (Stone 1993: 425).

A high volumetric efficiency for an ICE means more shaft power from the engine and it is achieved with cooling; the cooler the surfaces of the cylinder, the higher mass of fuel/air mixture is trapped in the combustion chamber. When considering spark ignition (SI) engines, the cooler combustion chamber also prevents spontaneous ignition of fuel/air mixture, known as knocking. This destroys the thermal boundary layer which can overheat components. If overheating is happening in an engine, it can affect to the engine in multiple ways. The higher temperature can cause a loss of strength in some materials, the lubricants will degrade in high temperatures which can lead loss of lubrication and excessive thermal strain can lead to severe damage. (Stone 1993: 426)

2.1 Mechanisms of cooling

To cool the ICE, heat must be transferred from the engine block. This is done by using a cooling media which is air or liquid (Hoag 2006: 153). In other words, there can either be air-cooled or liquid-cooled engines. In this thesis cooling systems based on liquid-cooling is considered since it is nowadays the most common one used in automotive cooling systems, according to van Basshuysen & Schäfer (2004) and Hoag (2006).

2.1.1 The coolant

The cooling media is called as the coolant and is often water/ethylene glycol mixture rather than just plain water (van Basshuysen & Schäfer 2004, Hoag 2006). Hoag (2006: 153) presents that the demands for coolant are serious since it must perform in a wide range of temperature without prominent change of phase. Callister et al. (1997) adds that the coolant should also provide protection against corrosion for metal parts and have good anti-freeze qualities for winter operation. According to Callister et al. (1997) a 50/50 mixture of water and ethylene glycol has the boiling point of 108 °C and freezing point of -38 °C in atmospheric pressure. When comparing to plain water (100 °C/0 °C), the mixture clearly is better in automotive use.

If plain water is considered, it is an excellent coolant due to its high enthalpy of vaporisation, specific heat capacity and thermal conductivity. But like pointed out previously, it has downsides of low boiling point of 100 °C and high freezing point of 0 °C. Water also is corrosive for certain metals. (Stone 1993: 442)

Since the Formula Student rules specifically indicates that as a coolant can be used only plain water, no further consideration of additives is performed (FSG 2019: 49).

It also must be noted, that previously mentioned boiling and freezing points were in atmospheric pressure (101.32 kPa). When the temperature of the coolant rises in the system, it causes expansion in the volume of the coolant which results in a rise of pressure. (Callister et al. 1997) As a property of water, also boiling and freezing points raises due to pressure rise. For example, in a pressure of 120.90 kPa the boiling point of water is 105°C. (Engineering Toolbox 2001)

2.1.2 Heat transfer

The heat is now transported from the engine using the coolant. To find out how the heat transfers to the coolant and eventually away from it, the heat transfer modes of conduction and convection need to be introduced. The radiation term is generally negligible for SI engines according to Heywood (1988: 672).

In conduction thermal energy is transferred by molecular motion through solids and fluids (Heywood 1988: 670). According to Fourier's law, the heat flux q is a proportional of the temperature gradient, stated in one-dimensional form (Lienhard & Lienhard 2019: 11):

$$q = -k\frac{dT}{dx}\,, \tag{1}$$

where k is thermal conductivity [W/(m·K)] and

$\frac{dT}{dx}$ is the temperature gradient in x-direction [K/m].

It also should be noted that heat flux q is a heat rate per unit area, given by relation of the form (Lienhard & Lienhard 2019: 11):

$$q = \frac{Q}{A}, \tag{2}$$

where Q is the heat transfer rate [W] and
A is the heat transfer area [m^2].

In convection thermal energy is transferred through fluid in motion, says Lienhard & Lienhard (2019: 19). Heywood (1988: 670) adds that heat is also transferred between fluid and solid surface in relative motion. Heywood (1988: 670) also clarifies that in engines the fluid motion is turbulent and is called forced convection when motion is produced by other forces than gravity. Lienhard & Lienhard (2019: 19) gives an equation for the heat flux of steady-state heat transfer problems which applies also in forced convection, says Heywood (1988: 670):

$$q = h\Delta T, \tag{3}$$

where h is a heat transfer coefficient [W/(m^2·K)] and
ΔT is the temperature difference [K].

When ICE is considered, a conductive heat transfer is occurring through cylinder head, cylinder walls, piston and engine block, to mention a few. A heat is transferred by forced convection between cylinder gases and the cylinder head, valves, piston and cylinder walls. Also, the heat is transferred by forced convection from the engine block to the coolant. (Heywood 1988: 670) Van Basshuysen & Schäfer (2004: 555) reminds that same heat transferring phenomena is also applicable in heat transfer of heat exchanger.

2.1.3 Fluid mechanics

To make the cooling of the ICE continuous, the coolant flow through the engine must be continuous too. This is done by making the cooling system a closed-circuit system where the coolant flows (Hoag 2006: 159). It is important to notice that a cooling system is an internal flow system (Miller 1978: 3) where basic laws of fluid mechanics applies.

When the fluid flows in a fixed volume that has one inlet and outlet and there is no additional accumulation of fluid within the volume, must mass of the fluid be conserved. This is represented as

$$\dot{m}_{in} = \dot{m}_{out}, \tag{4}$$

where $\dot{m}$ [kg/s] is the mass flowrate and the subscripts *in* and *out* denotes for inlet and outlet of the control volume, respectively. The mass flowrate is given as (Young et al. 2012: 82):

$$\dot{m} = \rho\dot{V}, \tag{5}$$

where $\dot{V}$ is volumetric flowrate [m^3/s] and
ρ is density of the fluid [kg/m^3].

With liquids, the variation of density is small according to Young et al. (2012: 84), therefore it is considered as an incompressible flow. Thus, the flowrates of mass are equal in the inlet and outlet of the control volume stated in equation 4 and the incompressibility applies, the continuity equation is:

$$\dot{V}_{in} = \dot{V}_{out} \tag{6}$$

When the fluid flows across an area with an average speed, Young et al. (2012: 82) gives for the volumetric flowrate an equation

$$\dot{V} = A_c v, \tag{7}$$

where A_c is the cross-section area normal to the flow direction [m^2] and
v is the average velocity of the fluid [m/s].

When equations 6 and 7 are combined, Young et al. (2012: 84) gives the form of the continuity equation as

$$A_{cin} v_{in} = A_{cout} v_{out} \tag{8}$$

Another important parameter related to mass flowrate of the fluid in thermodynamic application is the total heat capacity of the fluid, which implicates how much heat can be included, and thus transported, with a certain amount of fluid. The total heat capacity C is expressed as

$$C = \dot{m} c_p, \tag{9}$$

where c_p is the specific heat capacity at constant pressure [J/(kg·K)].

It should be noted that the specific heat can be defined also in constant volume as c_v. Nevertheless, when the incompressibility applies within the fluid, i.e. the volume is constant for any pressure variation, the two specific heats c_p and c_v are equal. (Lienhard & Lienhard 2019)

2.1.4 Internal flow system characteristics

Thermodynamics first law states that total stored energy of the system equals to energy addition to system done by heat transfer and work transfer. The energy equation is a fluid mechanics application of thermodynamics first law which applies to incompressible flow problems within a control volume, also known as *extended Bernoulli* equation (Young et al. 2012: 160):

$$\frac{p_{out}}{\rho g} + \frac{v_{out}^2}{2g} + z_{out} = \frac{p_{in}}{\rho g} + \frac{v_{in}^2}{2g} + z_{in} + h_p - h_L, \tag{10}$$

where p is pressure [Pa],
g is standard gravity [m/s^2],
z is height of the streamline from a zero-level [m],
h_p is the pump head [m] and
h_L is the head loss [m].

The head is an energy per unit weight notion that is generally used in pump calculations. The amount of head is described with units of length.

In an internal flow system, each component results in change of energy. Due to the conservation law, the kinetic energy remains constant in control volume but the potential energy changes to heat due to the friction with the wall and the viscosity of the fluid (Nordlander 2008: 5). These decreases in potential energy are called as head losses that composes friction losses in the pipes and minor losses in pipe fittings, valves and bends. In heat exchanger the turbulence phenomena also creates losses (Nordlander 2008: 5). For a flow system that uses a pump to raise the fluid from a certain height to another, the energy equation 10 contracts to

$$h_p = z_{out} - z_{in} + \Sigma h_L, \tag{11}$$

where h_p is the actual head gained by the fluid from the pump and Σh_L represents all losses mentioned above. In the case of automotive cooling system being a circuit system, the height difference between inlet and outlet streamlines is negligible. Thus, the elevation head z_{out} - z_{in} contracts. From studies of the pipe flow it is known that typically head losses vary as the volumetric flowrate squared. Therefore, equation 11 gets the form as

$$h_p = K\dot{V}^2, \tag{12}$$

where K depends from loss coefficients, friction factors and pipe dimensions, i.e. $K \approx \Sigma h_L$. With equation 12 a specific curve can be plotted for each circuit of the flow system. The curve is called as a system curve and is presented in Figure 1. (Young et al. 2012: 416)

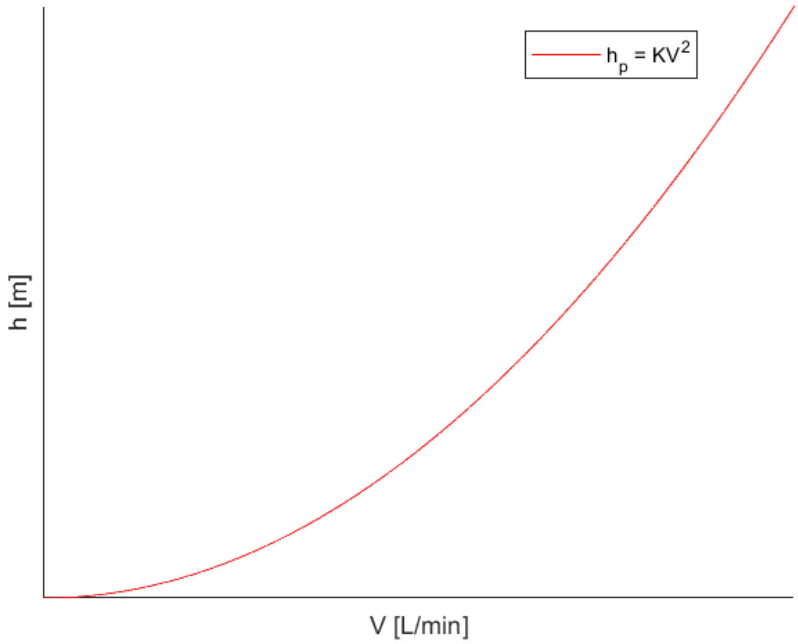

Figure 1. An example of a system curve.

As said, the factor K depends on the head losses, which are geometry of the flow system, pipe dimensions, loss coefficients and friction factors. The head losses includes major and minor losses that roughly said are the pipe losses and other geometrical losses, respectively. The major losses h_{Lmajor} can be computed from the *Darcy-Weisbach equation* as (Young et al. 2012: 287):

$$h_{Lmajor} = f\frac{l}{D}\frac{v^2}{2g}, \tag{13}$$

where f is the Darcy-Weisbach friction factor,
l is the pipe length [m] and
D is the pipe diameter [m].

The Darcy-Weisbach friction factor f depends on the *Reynolds number* Re and relation of the equivalent roughness to diameter ε_r/D. The functional dependency between these is represented as the *Moody chart* and can be found easily on literature. (Young et al. 2012: 287) The Reynolds number is the most famous dimensionless parameter in fluid mechanics that is used in defining whether the flow is laminar or turbulent. The Reynolds number Re expresses the influence between inertia force to viscous force of the fluid and is described as

$$Re = \frac{\rho v l_c}{\mu}, \tag{14}$$

where l_c is a characteristic length and μ is the dynamic viscosity of the fluid. When the flow in a pipe is concerned, the diameter of the pipe D operates as the characteristic length. (Young et al. 2012)

Other losses due to components such as valves, fittings, contractions, expansions and bends are called as minor losses h_{Lminor} and is defined as

$$h_{Lminor} = \Sigma K_L \frac{v^2}{2g}, \tag{15}$$

where K_L is the loss coefficient of a component. The loss coefficients for pipe components are also easily available in literature. (Young et al. 2012)

It should be acknowledged that engine and heat exchanger, being components of internal flow system as well, cause also losses and must be noticed in the design. Shah & Sekulic (2003) presents precise methods how to evaluate losses in a heat exchanger but is neglected due to number of unknown parameters needed. They also states the typical design value of pressure loss in a heat exchanger of 70 kPa (Shah & Sekulic 2003: 380). Since in many flow system calculations heads are used, and with heat exchanger design the pressure loss is more expressed concept, a connection between them is needed. Shah & Sekulic (2003: 380) gives the connection the form of

$$h_L = \frac{\Delta p}{\rho g}, \tag{16}$$

where Δp is the pressure loss [Pa].

3 WORKING PRINCIPLE OF A COOLING SYSTEM

As stated, the cooling system is a closed-circuit system where the coolant is circulated through the engine jacket, where the heat rejects from the engine block to the coolant. The same phenomenon is then repeated in the heat exchanger where the heat transfers from the coolant through the heat exchanger wall to the ambient air. Finally, the coolant returns to the engine jacket. The flow within the cooling system is produced by pressurizing the coolant with a water pump. Adding the fact that pressure rises in the system also when the coolant temperature rises, the management of total pressure is required. This is done by a pressure cap in the system. If pressure increases too much and overflow happens, a reservation of coolant is often used. The flow through heat exchanger is also often controlled if the temperature has a lot of variation and this is done by a thermostat. (Callister et al. 1997, Hoag 2006: 159)

In Figure 2 a simple example of a cooling circuit is presented including the components described above. Hoag (2006: 159) reminds that the cooling installation can include several different heat exchangers for oil cooling, recirculated exhaust gas cooling and charge air cooling. He also makes the note that for charge air cooling the ambient air is used preferably due to cooler fluid. Van Basshuysen & Schäfer (2004) adds that usually different heat exchangers have their own branches in the circuit and that even separate circuits are possible. Also, most installations have a heater for the cabin (Hoag 2006: 159).

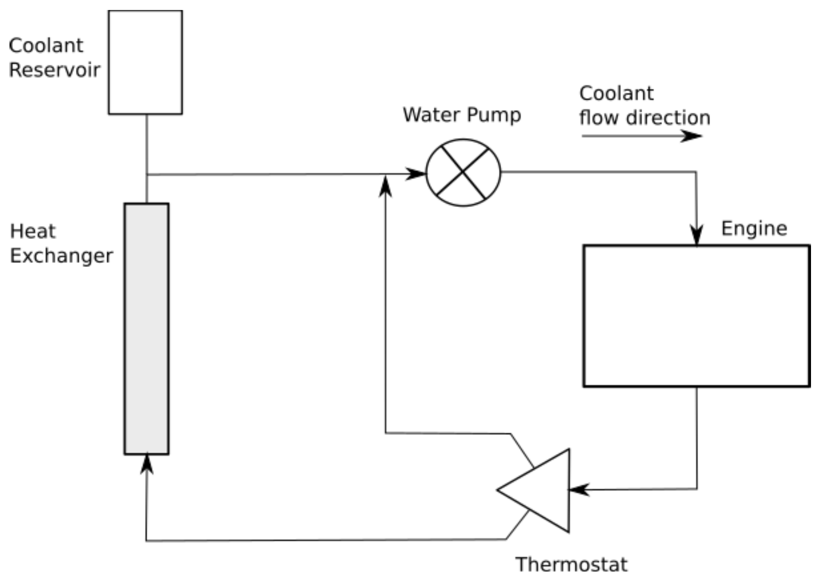

Figure 2. A cooling circuit.

Since the target of this thesis is to get acquainted with the basics of liquid cooling system and its designing, no further consideration is done with advanced cooling concepts. Next a more precise look for the main components of liquid a cooling system is performed.

3.1 The heat exchanger

In the heat exchanger thermal energy is transferred from one flowing fluid to the other, from coolant to ambient air, through the heat exchanger's walls and fins (Shah & Sekulic 2003). In automotive use the everyday term of radiator is replaced with heat exchanger and it is usually positioned in airstream in the vehicle (Hoag 2006: 159, Smith 1978: 98).

The area of heat exchanger where the heat transfer occurs is called core or matrix. It consists of pipes where coolant flows and fins, which are extensions to air side surface area of heat transfer. The pipes can be flat, round or oval shaped for cross-section. A small cross-section is desired for an ability to create a turbulent, high velocity flow which results increased heat transfer rate. Nowadays the most common material for heat exchangers is aluminum due to major weight savings and higher corrosion and pressure resistance. (Callister et al. 1997, Van Basshuysen & Schäfer 2004)

Several different types of heat exchangers exists, and classification could be done by flow arrangement or construction, to mention just two groups. Within flow arrangements, there are counterflow, parallelflow and crossflow, again to mention few. With these types, the differences are quite clear the flow directions are opposite, similar and normal to each other, respectively. (Shah & Sekulic 2003) The crossflow heat exchanger with horizontally assembled pipes are most used in modern cars according to Van Basshuysen & Schäfer (2004: 557). In Figure 3 a crossflow heat exchanger is presented.

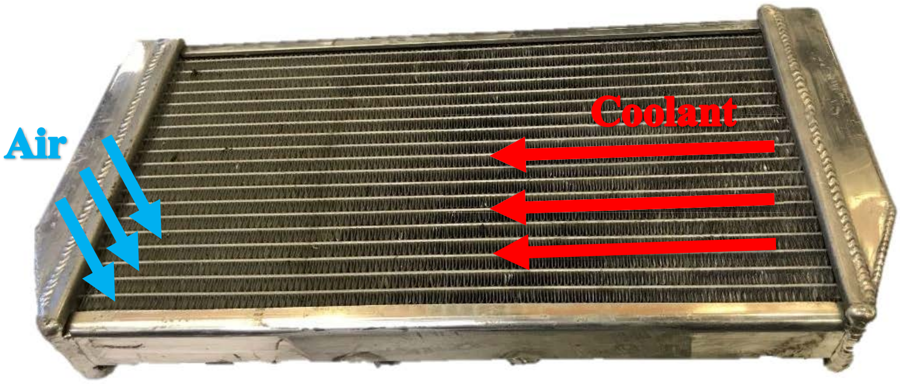

Figure 3. A crossflow heat exchanger with illustrations.

As a notice, crossflow heat exchangers with horizontally assembled piping are often compared to downflow heat exchangers what can be done of course, but according categories Shah & Sekulic (2003) presented, it is not very convenient since the downflow heat exchanger is actually a crossflow heat exchanger with vertically assembled piping. Therefore the correct comparing should be done using construction of both.

3.1.1 Sizing the heat exchanger

In their book Shah & Sekulic (2003) presents numerous different calculating methods for sizing a heat exchanger such as Mean Temperature Difference (MTD) method, effectiveness-Number of Transfer Units (ε-NTU) method and temperature effectiveness-Number of Transfer Units (P-NTU) method, for example.

Torregrosa et al. (2010) clearly refers to MTD method and Nordlander (2008) to ε-NTU method to be used in a heat exchanger designing. According to Shah & Sekulic (2003: 210) the ε-NTU method is used generally in automotive and transport industries where are the need in design of compact heat exchangers and the MTD method is more popular in process and power industries where noncompact heat exchangers are used.

To be clear, both ε-NTU and MTD methods are presented and applicable but ε-NTU is chosen to be used due more practicality. In the MTD method the calculation of heat transfer rate is based on temperature difference between fluids as the name refers and a lot of prediction must be done since outlet temperatures are rarely known before the design is finished (Lienhard & Lienhard 2019: 120). The method requires a lot of incrementation, and calculation could be simplified by using the ε-NTU method, says

Lienhard & Lienhard. (2019: 121) Shah & Sekulic (2003: 208) adds that with the ε-NTU method the determination of the improvement in heat exchanger performance with increasing surface area and sizing problem solution are straightforward.

3.1.2 The ε-NTU method

The physical significance in heat exchanger design theory is carried by explicit use of energy balance and rate equations in the derivation of ε-NTU formulas, clarifies Shah & Sekulic (2003: 208). The heat transfer rate Q is defined by the ε-NTU method as (Shah & Sekulic 2003: 114):

$$Q = \varepsilon C_{min} \Delta T_{max}, \tag{17}$$

where ε is the heat exchanger effectiveness,
C_{min} is smaller of C_a and C_c and
ΔT_{max} is the inlet temperature difference (ITD) [K].

Subscripts a and c denotes for air and coolant, respectively. The heat exchanger effectiveness ε is the ratio between actual heat transferred and maximum heat that could be possibly transferred from one fluid to the other. Mathematically ε is defined as

$$\varepsilon = \frac{C_{hot}(T_{hot_{in}} - T_{hot_{out}})}{C_{min} ITD}, \tag{18}$$

where subscript *hot* stands for hotter fluid. (Lienhard & Lienhard 2019: 121) As pointed out, the outlet temperatures are usually unknown, and equation of effectiveness cannot be used in predesign. Thus, the NTU part of the ε-NTU method must be deployed by introducing two nondimensional groups of Number of Transfer Units (NTU) and the heat capacity rate ratio C* since ε, dimensionless itself too, is dependent by those.

NTU is the nondimensional heat transfer size of the heat exchanger that provides a compound measure of the heat exchanger and therefore is an important design parameter. NTU is expressed as (Shah & Sekulic 2003: 119):

$$NTU = \frac{UA}{C_{min}}, \tag{19}$$

where U is overall heat transfer coefficient [W/(m^2·K)] and
A is overall heat transfer area [m^2].

Shah & Sekulic (2003: 118) explains that the heat capacity rate ratio C* is an operating parameter of the heat exchanger since it has the dependency over mass flowrates and temperatures of the fluids. Mathematically it is the ratio between heat capacity rates of the fluids so that C* is never more than 1. Therefore, the form of C* is defined as

$$C^* = \frac{C_{min}}{C_{max}}, \tag{20}$$

where C_{max} is larger of C_a and C_c.

Next it is needed to pay attention to the dependency of ε, NTU and C*. It is remembered that different types of heat exchangers are existed, hence the flow arrangement for a direct-transfer type heat exchanger affects also to the exchanger effectiveness. Thus, the dependency can be written in general form of

$$\varepsilon = \phi(NTU; C^*; flow\ arrangement), \tag{21}$$

where the functional relationship ϕ is dependent on the flow arrangement. (Shah & Sekulic 2003: 114) Lienhard & Lienhard (2019: 126) presents a simplified relation between ε and NTU is found when C* goes to zero:

$$\varepsilon = 1 - e^{-NTU} \tag{22}$$

This is called as a single stream limit and it occurs when one stream's temperature remains constant. Although, C* can approach zero when flowrate or specific heat for other fluid is greatly larger than the other, as is the case with high mass flowrate of water and low mass flowrate of air. Sekulic et al. (1999) explains that the difficulty of derivation of ε-NTU formulas for heat exchanger flow arrangements rises when flow arrangement becomes more complicated (see Shah & Sekulic 2003: 127). Lienhard & Lienhard (2019) and Shah & Sekulic (2003) provides tables for defining ε with known C* and NTU for different flow arrangements.

3.1.3 The overall heat transfer coefficient

The overall heat transfer coefficient U describes the overall thermal resistance that includes the convection at the walls and the conduction through the wall as a series, shown in Figure 4. (Van Basshuysen & Schäfer 2004: 555, Nordlander 2008: 14)

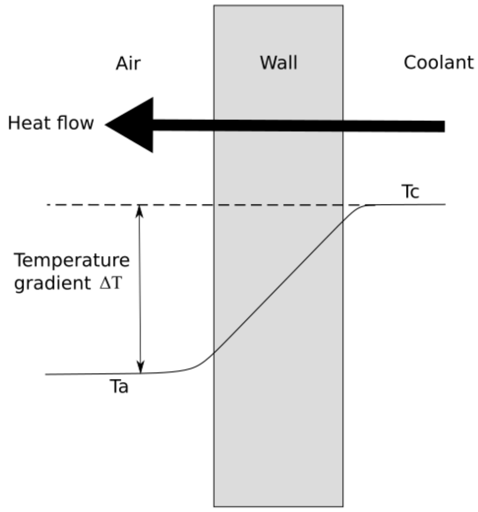

Figure 4. Heat transfer through heat exchanger wall (retell Van Basshuysen & Schäfer 2004).

Lienhard & Lienhard (2019: 79) gives equation for heat transfer for the situation in Figure 4:

$$Q = \frac{\Delta T}{\Sigma R_t}, \tag{23}$$

where ΔT is the temperature difference between fluids [K] and
ΣR_t is the total resistance [K/W].

The total resistance is the sum of two convective heat transfers between the separation wall and a fluid and conductive heat transfer within the separation wall. Since it is known that thermal resistance R is temperature difference per heat transfer rate (Lienhard & Lienhard 2019):

$$R = \frac{\Delta T}{Q}, \tag{24}$$

the thermal resistance for convection can be written by using equations 2, 3 and 24 to the form (Lienhard & Lienhard 2019):

$$R_{conv} = \frac{1}{hA}, \tag{25}$$

With the same principle, with equations 1, 2 and 24, the thermal resistance for conduction gets the form as (Lienhard & Lienhard 2019):

$$R_{cond} = \frac{\delta}{kA}, \tag{26}$$

where δ is wall thickness [m].

Now, the total resistance is

$$\Sigma R_t = R_{conv,c} + R_{cond} + R_{conv,a} = \frac{1}{h_c A} + \frac{\delta}{kA} + \frac{1}{h_a A}. \tag{27}$$

Lienhard & Lienhard (2019: 78) defines that through resistances the heat transfer rate is defined with overall heat transfer coefficient as

$$\mathrm{Q} = \mathrm{UA}\Delta T, \tag{28}$$

therefore, when equation 27 and 23 is combined with 28, the overall heat transfer coefficient is

$$U = \frac{1}{\frac{1}{h_c}+\frac{\delta}{k}+\frac{1}{h_a}}. \tag{29}$$

Equation 29 assumes that the heat transfer is done through a simple wall shown in Figure 4 and the effect of fins are assumed to be included in that (Van Basshuysen & Schäfer 2004: 556). Shah & Sekulic (2003: 109) gives a more precise form for equation 29 included with an extended surface efficiency η_0. The equation is found as

$$U = \frac{1}{\frac{1}{h_c \eta_0}+\frac{\delta}{k}+\frac{1}{h_a \eta_0}}, \tag{30}$$

where η_0 is the extended surface efficiency on one fluid side of the extended surface heat exchanger. In crossflow heat exchanger extended surface refers as fins.

3.1.4 The MTD method

In the MTD method the heat transfer rate Q is computed by using a mean temperature difference as the name of the method implicates, and the equation is given as (Shah & Sekulic 2003: 105):

$$\mathrm{Q} = \mathrm{UA}\Delta T_{mean}, \tag{31}$$

where ΔT_{mean} is the mean temperature difference between air and coolant [°C].

The mean temperature difference between coolant and air is still needed for equation 31 and Torregrosa et al. (2010: 7) defines it as

$$\Delta T_{mean} = (T_c^m - T_a^m), \tag{32}$$

where T_c^m and T_a^m are mean temperatures coolant and air, respectively. The mean temperature for both fluids is calculated by same analogy which Torregrosa et al. (2010: 7) gives the form as

$$T_f^m = T_f^{in} + \frac{\Delta T_f}{2}, \tag{33}$$

where subscript f stands for a fluid and T_f^{in} is the temperature of fluid before the heat exchanger [°C]. Equation 32 is a simplified version of the calculation of mean temperature difference. Lienhard & Lienhard (2019: 111) defines logarithmic mean temperature difference (LMTD) that is computed as

$$\Delta T_{mean} = LMTD = \frac{\Delta T_{out} - \mathrm{ITD}}{\ln(\frac{\Delta T_{out}}{\mathrm{ITD}})}, \tag{34}$$

where ΔT_{out} is the outlet temperature difference [°C].

Unfortunately, the equation of LMTD applies only single-pass parallel- and counterflow heat exchangers and therefore a correction factor F is introduced, which can be used in multi-pass and/or crossflow heat exchangers. The correction factor is a relation between tube and shell flows in the heat exchanger and therefore diagrams should be used to define the accurate value of F, which naturally is between 0 and 1. (Lienhard & Lienhard, 2019: 116)

3.2 The fan

The volumetric air flow through the heat exchanger core is one of the most important parameters in a working and effective cooling system. A fan, an axial-flow turbine, is usually added to ensure adequate air flow at all conditions. Fans can be engine driven or powered electrically or hydraulically. Due to growing need of controllability, electric fans and engine driven viscous coupling fans controlled with coolant and bimetallic element temperature, respectively, have already become a norm, according to Van Basshuysen & Schäfer. (2004) They add that usually the fans are pulling the air but pushing type fans are also an option.

As introduced, often the heat exchanger is installed in the airstream of the vehicle to utilize the ram air effect. Callister et al. (1997) and Van Basshuysen & Schäfer (2004: 564) discovers that in cruising speeds fans are often switched off and SRP (2019) specifies that above a speed of ~60 km/h fans are causing more drag if switched on. For this reason, Smith (1978) gives a precise observation of air duct design and clarifies that in racing applications the velocity of air is often wanted to be lowered (from the high velocity of the vehicle) when entering the heat exchanger core.

Within the air duct, the same analogic of pressure drop or head loss applies and equation 10 is applicable (Smith 1978). In the air duct the heat exchanger core is naturally the most losses causing component and Shah & Sekulic (2003) provides precise tools for analyzing that when all parameters are known. They also state that the typical design value of pressure drop in air side of the heat exchanger is 0.25 kPa to keep the pumping power moderate. As a conclusion, the need of the fan should be estimated at operating conditions of idle and maximum power, or “at rest” and “at speed” like SRP (2019) says, at least.

3.3 The water pump

To overcome the pressure drop within the cooling system, energy must be added to the fluid. This is done by a water pump, where energy is supplied to the rotating shaft and transferred into the coolant. This increases the pressure and the velocity in the fluid and with the shape of the casing, the velocity is decreased at the outlet of the impeller. This decrease is transferred again to rise in the pressure of the fluid. (Nordlander 2008, Young et al. 2012)

In present-day vehicles the rotating shaft of the pump receives the kinetic energy from the engine with an appropriate pulley-ratio, says Van Basshuysen & Schäfer (2004: 564). Hoag (2006: 157) adds that gear-driven pumps are also as typical and that these pumps are usually mounted in the engine block as well. Van Basshuysen & Schäfer (2004: 564) also specifies that due to emerging need of controllability electric pumps might be the solution in the future.

The centrifugal pump is one of the most common pumps and Hoag (2006: 157) adds that it is used almost universally. The centrifugal pump has two main components which are the impeller attached to a shaft and the casing. Hoag (2006: 158) also adds that the impeller can be made of cast iron, stamped steel or plastic resin. (Young et al. 2012)

3.3.1 Properties of a centrifugal pump

When the centrifugal pump is considered, the basic theory of operation can be developed by considering the steady, time averaged flow even though the flow is very complex in reality. Fluid velocities over the inlet and outlet sections are taken to be average velocities. The relationships between velocities are shown in a projection of an impeller in Figure 5. (Young et al. 2012: 410)

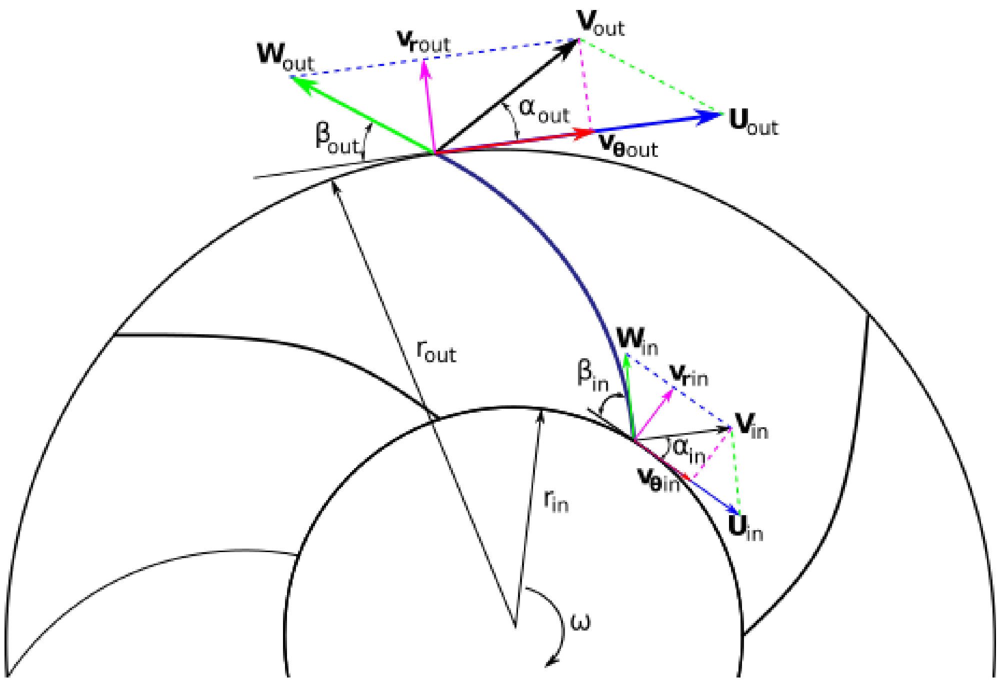

Figure 5. Velocity diagrams of a centrifugal pump (retell Young et al. 2012: 411).

From Figure 5 the relationship between the absolute velocity **V** and velocity of the blade **U** is constituted by using the tangential and radial components of absolute velocity:

$$v_\theta = \omega r - \frac{v_r}{tan\beta}, \tag{35}$$

where v_θ is the tangential component of the absolute velocity [m/s],
ω is the angular velocity [rad/s],
r is the radius [m],
v_r is the radial component of the absolute velocity [m/s] and
β is the angle of the blade [deg].

It should be noted that the forms of geometric equations are idiomatic to inlet an outlet sections of blade passage. (Young et al. 2012: 411)

Since the mass flowrate $\dot{m}$ through pump is constant, Young et al. (2012: 411) gives the required shaft torque T_s as

$$T_s = \rho\dot{V}(r_{out}v_{\theta out} - r_{in}v_{\theta in}), \tag{36}$$

where $\rho\dot{V} = \dot{m}$ and thus, the relation between torque and angular velocity is known, the required power P_s for rotating the shaft is (Mäkelä et al. 2015: 93):

$$P_s = T_s\omega \tag{37}$$

Since the pump is needed in the cooling system for producing flow and more specifically, pressure, the maximum or ideal pressure head is an important parameter and is found as (Young et al. 2012: 411, Lienhard & Lienhard 2019: 127, Nordlander 2008: 17):

$$h_i = \frac{P_s}{\rho g\dot{V}} \tag{38}$$

Young et al. (2012: 412) gives a form for the maximum head pressure rise that takes the impeller geometry and angular velocity straight into account and a curve can be plotted for different velocities. The curve is called as the pump curve and the equation is given as

$$h_i = \frac{U_{out}^2}{g} - \frac{U_{out}cot\beta_{out}}{2\pi r_{out}b_{out}g}\dot{V}, \tag{39}$$

where U_{out} is the velocity of the blade [m/s] and
b_{out} is the blade height at the outlet radius [m].

Since there are simplifying assumptions (i.e. no losses) in equation 39, the ideal pump curve is linear, and it varies over flowrate $\dot{V}$. In reality, there are hydraulic losses due to fluid skin friction, flow separation and impeller blade casing clearance affecting, thus the actual head-rise should vary with flowrate squared $\dot{V}^2$. Both curves are presented in Figure 6. (Young et al. 2012: 412)

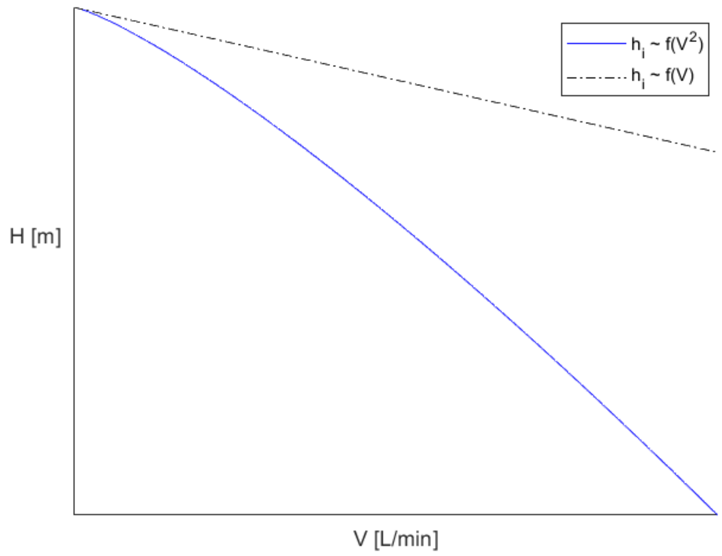

Figure 6. The difference of losses between pump curves.

3.4 The thermostat

When the coolant temperature is below the operating temperature, a thermostat restricts the flow to the heat exchanger. The purpose of thermostat is to give no more heat away from the engine than needed and thus maintaining the temperature as constant as possible. When the coolant is below the wanted temperature, the thermostat is closed, and coolant is circulated directly back to the water pump. As temperature increases, the thermostat begins to open and starts sending a portion of the coolant to the heat exchanger and finally at a desired temperature the thermostat is fully open. This is done by a wax element that expands and shrinks due to temperature change. The set point is generally designed in the range between 85°C – 90°C according to Callister et al. (1997). Van Basshuysen & Schäfer (2004: 563) presents modern thermostat of which wax element is electrically heated that allows independent controllability over engine temperature, thus resulting reduced fuel consumption. (Hoag 2006: 159, Nordlander 2008: 27)

Thermostat from the Yamaha MT-07 engine is shown in Figure 7.

Figure 7. Thermostat construction by Yamaha from MT-07.

3.5 Expansion and de-aeriation of the coolant

As mentioned, the pressure rises in the cooling system due to increase in temperature, and that maximum pressure is controlled with a pressure cap. Callister et al. (1997) explains that a pressure cap works with a simple spring that contracts when the spring rate's maximum allowable pressure is exceeded. Since the boiling point of water increases also with the pressure, the largest pressure a cooling system can handle is desired (Genibrel 2017). For an example, modern-day hybrid Formula 1 cars uses over 250 kPa pressure in a cooling system to run the engine safely over 120°C (Scarborough 2019). The normal maximum pressure is approximately 150 kPa in a cooling system of a motor vehicle, according to Van Basshuysen and Schäfer (2004: 555). Genibrel (2017) adds that pressure in a cooling system is vital for keeping water in contact with the metal surfaces of the cylinder heads and block.

SRP (2019) reminds that keeping the cooling system full reduces aeriation and maintains the pressure. Since pressure cap allows air and coolant escape from the system after specific pressure, a recovery system or an overflow tank is often used to collect the excess water. When the pressure cap is correctly located in low pressure side of the cooling system, the air escapes first. This creates a vacuum when system cools and allows the coolant re-enter to the system resulting in no net loss in coolant. If a recovery system is not utilized, a more common solution is to use an expansion tank that has volume to cope with the expansion of the coolant. Also, it can be used to separate air with correct location and plumping. (Callister et al. 1997, SRP 2019)

Hoag (2006: 159) presents that air in the coolant reduces its effectiveness in controlling engine temperature, and therefore a proper venting and de-aeriation is needed in cooling system. Additionally to recovery system and expansion tank, special de-aeriation devices have been developed. Callister et al. (1997) adds that these are more common in racing cars and not on production cars. A simple solution is to add baffles in heat exchanger inlet tank that forces the portion of coolant with air bubbles to a low velocity area above the baffle when the air rises and separates from the coolant. Other efficient device is a swirl pot with a same basic idea than a baffle; the swirl pot is a collector tank where the coolant flows forming a swirl and from the low velocity area in the middle of the tank the air separates from the coolant. Callister et al. (1997) also makes the note that a swirl pot can have the pressure cap directly and that in racing applications an expansion tank or swirl pot can act as a coolant reserve.

3.6 Engine heat rejection

In order to design an efficient cooling system, one of the most valuable things to be acknowledged is the heat rejection of the engine. The heat dissipated to the coolant is one of the factors of engine's energy balance along with fuel, (associated heat of combustion) intake air, brake power, exhaust loss, and convection and radiation to the surroundings (Abedin et al. 2013). The engine's energy balance is introduced in following equation which is a steady-flow energy-conversation equation for a control volume (Heywood 1988: 673):

$$\dot{m}_{fuel} Q_{LHV} = P_b + Q_c + \mathrm{Q}_{misc} + H_{e,ic} + \dot{m}_e h_{e,s}, \tag{40}$$

where $\dot{m}_{fuel}$ is the mass flowrate of fuel [kg/s],
Q_{LHV} is the lower heating value of fuel [J/kg],
P_b is the brake power [W] of the engine,
Q_c is the heat rejected to the coolant [W],
Q_{misc} is the heat rejected to oil (if separately cooled) plus convection and radiation to surroundings from engine's external surface [W],
$H_{e,ic}$ is the exhaust enthalpy loss due to incomplete combustion,
$\dot{m}_e$ is the exhaust mass flowrate [kg/s] and
$h_{e,s}$ is the sensible part of exhaust enthalpy [J/kg].

Due to large number of variables in energy balance of an engine, the accurate value of heat rejection is often unknown at early stage of design process. Fortunately, there are several studies about the estimation of ICE's heat rejection.

According to van Basshuysen and Schäfer (2004: 556), the amount of heat rejected to coolant from engine is 50–60 % of the brake power. Heywood (1988: 674) presents that 25–28 % and 17–26 % of fuel's heating value transforms into brake power and heat dissipated to coolant, respectively. Besides these estimations, Lahvic (1986) has presented an empirical correlation for thermal load according to his work in Ford Corporate Vehicle Simulation Program, which is shown in equation 41:

$$\mathrm{Q}_c = \frac{8.66V_d n+108.93T_e+1119.74P_b-1010V_d+2890}{3412.2}, \tag{41}$$

where Q_c is the heat rejected to the coolant [kW],
V_d is the engine displacement [l],
n is the frequency of engine rotation [rpm],
T_e is the engine torque [Nm] and
P_b is the brake power of the engine [kW].

Although, the Lahvic's correlation is known to overestimate the amount of heat dissipated to coolant (Parish, 2003), but it is still adequate to use in this kind of application due to this included "safety factor".

4 THE M03 COOLING SYSTEM

The design of the cooling system for 2019 car, M03, was a very dividing situation, due to the amount of problems in the cooling system of the predecessor, M02 Daytona. On the other hand, since there were so many problems in the past, it would be a challenge to be worse to that. Nevertheless, the preferable outcome on thoughts some pressure, mostly due author's lack of knowledge at the beginning of the design process, had to be dealt with.

In the design the emphasis was on the calculation of components and especially the heat exchanger, for it being the major operant in cooling. Due to the extent of the bachelor's thesis, only final designs and results are presented. The design is based on the fundament of worst case -scenario due to the history of the previous car. Since, the design is preliminary and first for the car and the engine, no data is available for this configuration and some assumptions had to be made during the design process. Also, in some aspect, this may have resulted too safe decisions in the design, but for this stage, better safe than sorry.

4.1 The heat

The selected engine for new M03 was two-cylinder Yamaha MT-07 engine, that was also brand-new acquaintance for the team. Therefore, no data for heat transfer rate exists and before anything else could be done, the amount of heat rejected to the coolant for this engine needed to be defined. Yamaha (2019) gives the technical specification of V_d = 0.689 l, P_b = 55 kW and M_t = 68 Nm. Since the engine is meant to be tuned, for the heat rate evaluation the brake power and torque is rounded up to 60 kW and 70 Nm, respectively, and the maximum engine speed is defined as 10 000 rpm. Therefore, from equation 41, the heat transfer rate to coolant is

$$Q_c = \frac{8.66 * 0.689\text{l} * 10000\text{rpm} + 108.93 * 70\text{Nm} + 1119.74 * 60\text{kW} - 1010 * 0.689\text{l} + 2890}{3412.2} = 40.05\ \text{kW}$$

According to Van Basshuysen & Schäfer (2004: 556) the heat transfer rate would be in the range of 0.5…0.6 * P_b = 30…36 kW and after Heywood (1988: 674) in the range of (60 kW / 25) * 17…26 = 40.8…62.4 kW. A lot variation occurs but the big difference in Heywood's estimation seems quite unreasonable. The above calculated value of heat transfer rate utilizing Lahvic's correlation is found as good estimation since it being also in the middle of Van Basshuysen & Schäfer and Heywood and is therefore chosen to be used in the design of cooling system.

4.2 The pressure

The first approach was to use engine's own water pump since it has designed for that engine by Yamaha, but it must be evaluated regardless of that. Unfortunately, no dimensions were given for the pump and it had to be measured. The pump's impeller geometry is shown in Figure 8 and the measured dimensions in the Table 1. The bigger blade was named 1 and smaller 2 as Figure 8 presents.

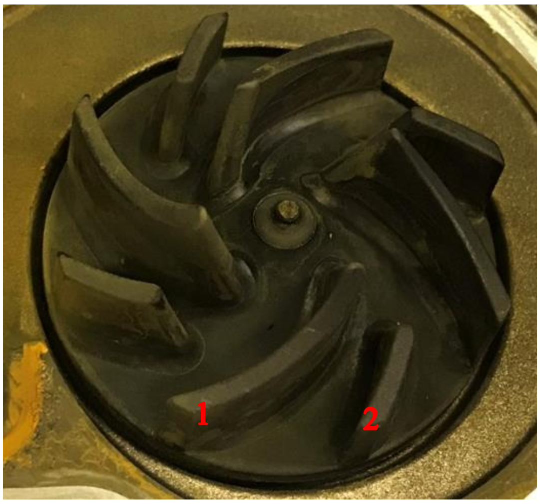

Figure 8. The impeller of the MT-07 water pump.

Table 1. The impeller's dimensions.

Blade	r_{out} [mm]	r_{in} [mm]	b_{out} [mm]	b_{in} [mm]	β_{out} [deg]	β_{in} [deg]
1	22.5	6.3	10.0	12.7	45	25
2	22.5	15.0	10.0	11.3	45	18

Since the dimensions and the rotational velocity of the impeller are the only known parameters, some assumptions must be made that the coolant flowrate could be calculated. Yamaha (2014: 48) states the reduction ratio for water pump (i = 1.309) and with the engine speed known, the rotational velocity of the impeller can be calculated. With the known relation between rotational velocity and rotational speed $\omega = 2\pi n$ (Mäkelä et al. 2015:92), the rotational velocity of the impeller can be pronounced with the equation of gear-ratio (Airila et al. 1995: 491):

$$i = \frac{\omega_1}{\omega_2}, \tag{42}$$

where i is the gear-ratio,
ω_1 is the rotational velocity of the primary gear [1/s] and
ω_2 is the rotational velocity of the secondary gear [1/s].

Therefore, the rotational velocity of the impeller is

$$\omega_2 = \frac{2\pi n_1}{i} = \frac{2\pi * \frac{10000}{60}\frac{1}{s}}{1.309} = 800.0\,\frac{1}{s}$$

To proceed further, an assumption must be made that the impeller is designed reasonably (Young et al. 2012: 412) and the tangential velocity component in pump inlet $v_{\theta in}$ is zero. Also, because the impeller has blades with two different geometries (1 & 2) as Figure 8 shows, must both be evaluated. This is done by separate evaluations of the radial component of the absolute velocity and averaging them to one result. Thus, the dimensions known, equation 35 gives the radial component of the absolute velocity for blade 1 as

$$v_{rin1} = \omega_2 r_{in1} tan\beta_{in} = 800\,\frac{1}{s} * 0.0063\text{m} * \tan\,(25°) = 2.350\,\frac{\text{m}}{\text{s}}$$

and for the blade 2 as v_{rin2} = 3.899 m/s. Thus, the average is then v_{rinA} = 3.125 m/s. For the volumetric flowrate the inlet area A_{in} is still needed. With pump that is defined as the cylindrical area that fluid crosses:

$$A_{in} = 2\pi r_{in} b_{in} \tag{43}$$

Equation 43 gives for the blade 1 A_{in1} = 502.71 mm^2 and blade 2 A_{in2} = 942.48 mm^2, thus the average area is then A_{inA} = 722.60 mm^2. Now, with equation 7 the volumetric flowrate of the water pump is

$$\dot{V} = A_{inA} v_{rinA} = 722.60\mathrm{mm}^2 * 3.125\frac{\mathrm{m}}{\mathrm{s}} = 0.002258\frac{\mathrm{m}^3}{\mathrm{s}}.$$

The torque in the shaft is then calculated with equation 36 that results as

$$T_s = 965.3\frac{\mathrm{kg}}{\mathrm{m}^3} * 0.002258\frac{\mathrm{m}^3}{\mathrm{s}} * \left(0.0225\mathrm{m} * 16.403\frac{\mathrm{m}}{\mathrm{s}} - 0\right) = 0.807\mathrm{Nm}$$

where $v_{\theta out}$ = 16.403 m/s is calculated with equations 7 and 35. Thus, the required power defined by equation 37 is then

$$P_s = 0.807\mathrm{Nm} * 800\frac{1}{\mathrm{s}} = 645.74\mathrm{W}$$

The maximum pressure head for the pump is still needed to verify that it can overcome the head losses in the system. With equation 38 the maximum pressure head is defined as

$$h_i = \frac{645.74\mathrm{W}}{965.3\frac{\mathrm{kg}}{\mathrm{m}^3}*9.81\frac{\mathrm{m}}{\mathrm{s}^2}*0.002258\frac{\mathrm{m}^3}{\mathrm{s}}} = 30.20\ \mathrm{m}$$

At this point of the preliminary design, the approximation of head losses is made. Since no accurate data of head losses in the heat exchanger or engine block are available, an estimation must be done. As pointed out before, Shah & Sekulic (2003: 380) presented that usually the heat exchanger is designed for 70 kPa pressure loss, and when converted to heads with equation 16, the estimated head loss in the heat exchanger h_{Lhe} is

$$h_{Lhe} = \frac{70000\ \mathrm{Pa}}{965.3\frac{\mathrm{kg}}{\mathrm{m}^3}*9.81\frac{\mathrm{m}}{\mathrm{s}^2}} = 7.39\ \mathrm{m}$$

The head loss in the engine block h_{Le} is the most dominant loss alongside the one in the heat exchanger and thus they are estimated equally large as $h_{Le} = h_{Lhe}$. To get the perspective to losses in the piping, the major losses h_{Lmajor} are also evaluated. First, the Reynolds number is calculated with equation 14 as

$$Re = \frac{965.3\frac{\mathrm{kg}}{\mathrm{m}^3}*\left(\frac{0.002258\frac{\mathrm{m}^3}{\mathrm{s}}}{\pi*(0.0125\mathrm{m})^2}\right)*0.025\mathrm{m}}{0.000315\ \mathrm{Pa}*\mathrm{s}} \approx 352\ 000,$$

where 0.000315 Pa*s is the dynamic viscosity of water at 90 °C (Engineering ToolBox 2001). When new aluminium pipe is considered as smooth, the equivalent roughness ε_r is thus zero and the Darcy-Weisbach friction factor f is then 0.015 from the Moody chart (Young et al. 2012: 288). Therefore, equation 13 gives the major losses as

$$h_{Lmajor} = 0.015 * \frac{1.5\mathrm{m}}{0.025\mathrm{m}} \frac{\left(\frac{0.002258\frac{\mathrm{m}^3}{\mathrm{s}}}{\pi*(0.0125\mathrm{m})^2}\right)^2}{2*9.81\frac{\mathrm{m}}{\mathrm{s}^2}} = 0.97\ \mathrm{m},$$

where 1.5 m is an estimation of maximum pipe lengths. Since the major losses are severely smaller than losses in the heat exchanger and engine block, the minor losses are neglected. Overall, with these losses summed (15.75 m), the ideal pressure head of the pump has still the safety factor twice as big as the losses and therefore the pump is trusted to be used in the cooling system.

4.3 The heat transfer

The ε-NTU method is chosen to be used in the heat exchanger design and equation 17 defines the heat transfer rate that should naturally match the heat transfer rate from the engine. Several parameters of heat exchanger effectiveness, minimum total heat capacity and inlet temperature difference are needed for evaluating the heat transfer rate. The Figure 3 presents the selected crossflow heat exchanger and the dimensions of it and its related properties is shown in the Table 2. The dimensions are measured from the heat exchanger due no given data was available. Therefore, the value of overall heat transfer coefficient U must be assumed. The Engineering ToolBox (2001) gives the typical value of 600…750 W/(m^{2}K) and Lienhard & Lienhard (2019: 82) up to 600 W/(m^{2}K) for U in air-to-water heat exchangers, thus 600 W/(m^{2}K) is chosen. The airduct design was a responsibility of team's aero-department and the specification of 12 m/s of minimum air velocity through the heat exchanger at racing speed was given to them, that was outstandingly fulfilled.

Table 2. The heat exchanger properties.

The heat transfer area A [m^2]	Frontal cross-section area A_c [m^2]	Air velocity v_a [m/s]	Overall heat transfer coefficient U [W/(m^{2}K)]
2.6	0.078	12	600

As said, the cooling system is designed for the worst case and that would be extremely hot weather. That was decided to be 35 °C ambient temperature and from MT-07 service manual (Yamaha 2014) the optimum operating temperature of 90 °C for the engine was received, thus the inlet temperature difference is then ITD = 90 °C – 35 °C = 55 °C. The needed properties for air and water in these selected temperatures are presented in the Table 3 (Engineering ToolBox 2001).

Table 3. Air and water properties.

	(Inlet) temperature T^{in} [°C]	Density ρ [kg/m^3]	Specific heat capacity c_p [kJ/kgK]
Air	35	1.146	1.006
Water	90	965.3	4.21

The minimum total heat capacity C_{min} and heat capacity rate ratio C* are defined with total heat capacities of air and coolant with equations 9, 5 and 7. Since the volumetric flowrate of coolant is known, equations 9 and 5 gives:

$$C_c = 965.3\,\frac{\mathrm{kg}}{\mathrm{m}^3} * 0.002258\,\frac{\mathrm{m}^3}{\mathrm{s}} * 4.21\,\frac{\mathrm{kJ}}{\mathrm{kgK}} = 9.176\,\frac{\mathrm{kW}}{\mathrm{K}}.$$

The equation 7 defines the volumetric flowrate of air with the frontal surface area of the core and the velocity of air through it. Now, A_c = 0.078 m2 and v_a = 12 m/s, the volumetric flowrate is

$$V_a = 0.078\mathrm{m}^2 * 12\frac{\mathrm{m}}{\mathrm{s}} = 0.936\frac{\mathrm{m}^3}{\mathrm{s}},$$

and then with equations 9 and 5 the total heat capacity for air is C_a = 1,079 kW/K. Therefore, the minimum total heat capacity C_{min} is then C_a and equation 20 defines the heat capacity rate ratio C* as

$$C^* = \frac{1.079\mathrm{kW/K}}{9.176\mathrm{kW/K}} = 0.118$$

For the heat exchanger effectiveness ε, the C* is defined but NTU is still needed and equation 19 specifies that as

$$NTU = \frac{600\frac{\mathrm{W}}{\mathrm{m}^2\mathrm{K}}*2.6\mathrm{m}^2}{1.079*10^3\frac{\mathrm{W}}{\mathrm{K}}} = 1.446$$

Now, the heat exchanger effectiveness can be checked from the known heat exchanger effectiveness figure (Lienhard & Lienhard 2019: 129) and with the above values of C* and NTU for crossflow heat exchanger ε ≈ 0.72. Since, the heat capacity rate ratio C* is quite close zero, the evaluation of ε can be performed also using equation 22 that gives

$$\varepsilon = 1 - \mathrm{e}^{-1.446} = 0.76$$

Finally, with all the parameters, the heat transfer ratio can be calculated. The value for ε = 0.72 is chosen due to larger safety margin. Thus, equation 17 gives the heat transfer ratio of

$$Q = 0.72 * 1.079\frac{\mathrm{kW}}{\mathrm{K}} * 55\mathrm{K} = 42{,}73\mathrm{kW},$$

that is just exceeds the heat transfer rate from the engine Q_c = 40.05 kW, thus the heat exchanger is chosen.

4.4 The circuit

In Figure 9 the original cooling circuit from the Yamaha MT-07 (Yamaha 2014) is presented. An interesting thing to be noted is, that it seems that the MT-07 uses the oil cooler more as an oil warmer than a cooler due to circuit layout.

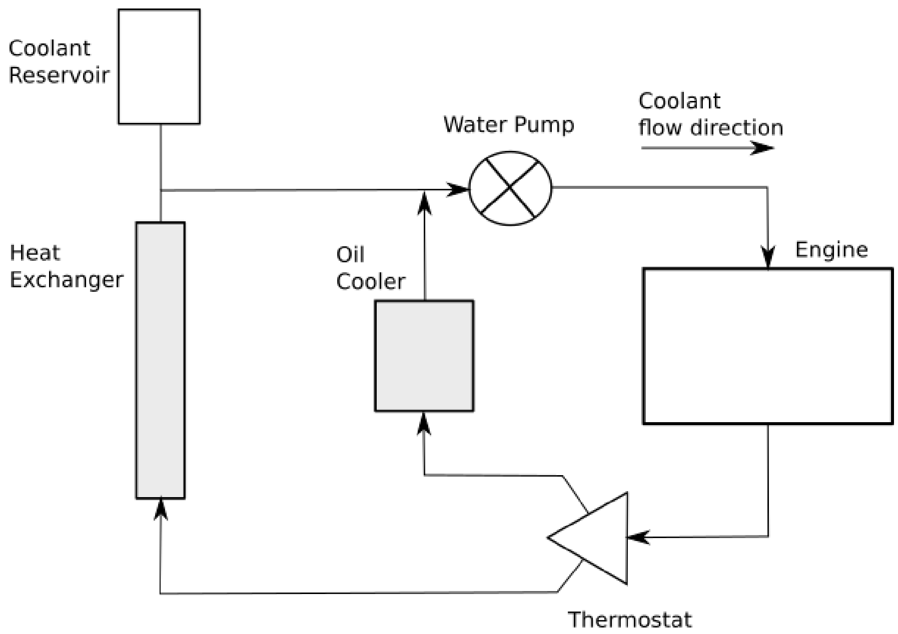

Figure 9. Yamaha MT-07 cooling circuit.

The Figure 10 shows the layout of the final cooling circuit for the M03 and the difference is clear for the original one. The oil cooler is now used as a cooler and the design has the option for it to be used if necessary, depending on the conditions. A swirl pot is designed to be used as an expansion tank to save weight and to remove possible trapped air in the system. At this point of the design, these both are neglected from the calculations due to complexity of the analysis. This is trusted to be safe, since these are more of a safety device and the margin in the pump pressure head exists. The swirl pot/expansion tank was equipped with a 1.4 bar pressure cap to with the boiling point of the water could be raised in 109.3 °C (Engineering ToolBox 2001). This cautious design relates to the problems with the old M02 Daytona car.

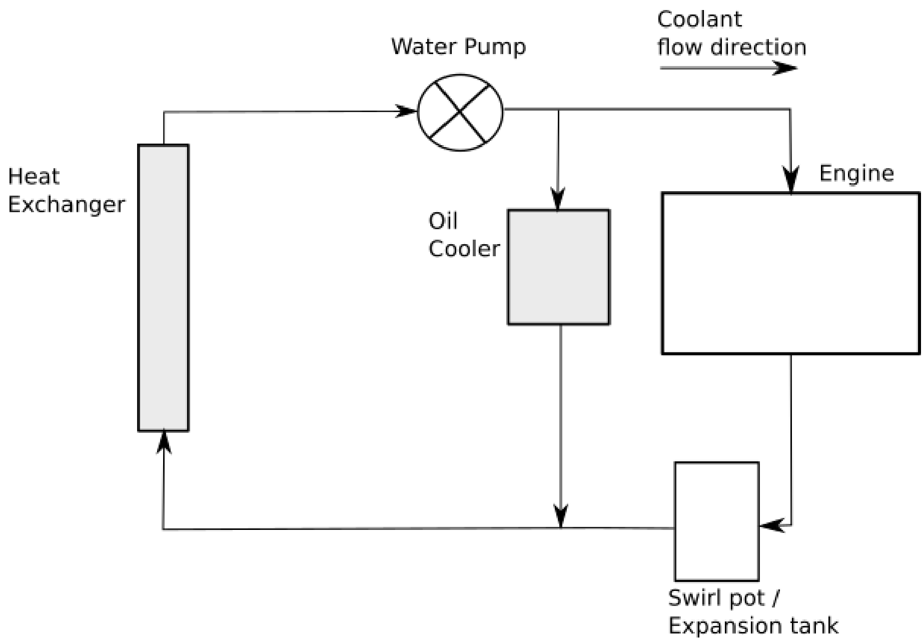

Figure 10. The M03 cooling circuit.

Due to changes in the cooling circuit, the thermostat is thus removed due to alternated flow direction within the oil cooler, and for the unnecessity of thermostat in racing application.

4.5 The idle

The design is finalized with a small cooling fan to cool the idle when the car is stationary. Due to difficulty of knowing the correct idle conditions, the heat rejected from the engine in idle was also evaluated with the Lahvic's correlation (Equation 41) with parameters of 2000 rpm, 5 kW and 5 Nm. This resulted in 5.94 kW that was trusted to be well enough for idle.

Since specific fans were already available within the team, these were evaluated. The selected fan was SPAL VA31-A101-46A with a frontal area of 0.0133 m^2 and the volumetric flowrate of 0.161 m^3/s (SPAL Automotive 2019). Since the fan is placed directly behind the heat exchanger, these parameters can be used in the calculation of the heat transfer ratio for the fan. The same procedure in the calculation is used than with the heat exchanger and thus is only briefly explained.

With this new volumetric flowrate, the total heat capacity for air is C_a = 0.186 kW/K. For the coolant, the volumetric flowrate in the idle is 0.00034 m^3/s and therefore the total heat capacity for coolant is C_c = 1.380 kW/K. Thus, the NTU is 1.433 and the heat exchanger effectiveness is still approximately 0.7. Finally, the heat transfer ratio for idle is calculated with equation 17 and thus Q = 7.16 exceeds the rejected heat, the fan is chosen.

5 CONCLUSIONS & RECOMMENDATIONS

By the time that this thesis is written, the cooling system has already been used for the 2019 season and for author's greatest relief, no problems were detected within the cooling system during the whole season. As a conclusion, the design and more specific, the accuracy of the design could be said to be successful.

The cooling system was augmented with two temperature sensors, placed in the engine block and after the heat exchanger, that were used during the season for monitoring and gathering data. Unfortunately, the logging was not used in every run and therefore the amount of reliable data is fairly small. Nevertheless, two major notifications were made during the season: the engine temperature was never above 95 °C and the average temperature difference between sensors was 5 °C.

These are good signals from the cooling system but solely unfortunately pretty useless. There are so many different parameters and estimations that all affects another and that is why a proper validation of the system and design methods should be done next. After that the profound improvement could be achieved when a change in design is done.

For example, additionally to temperature sensors, pressure sensors in the system would help the design and validation of the pump characteristics and calculation. The same benefit would lie in the use of velocity sensors, both in the coolant and air side of the system. Also, a simple flowrate test for the water pump would help the validation of accuracy of the flowrate calculation and the impeller measurements.

From the design point-of-view, a thorough analysis with a wider scale of racing conditions should be integrated in the design. Of course, this kind of worst-case scenario analysis should always be proceeded but the system could be designed to be more adaptable in different racing conditions, so that the system would benefit with space and weight alongside a steadier non-condition-pendent operating temperature for the engine. In the coolant side an integration of 1D-(CFD)simulation could be beneficial in this considering the simulation time and work load. Of course, it should be noted that the 1D-simulation is not a stairway to heaven, but parameters and equations must still be in the knowledge of the designer. Within the heat exchanger and the airduct design 3D CFD has already been used and the development should continue in that side also, this as a side notice.

6 SUMMARY

Since the main aim of this bachelor's thesis was to document the primary design of the liquid cooled cooling system for a Formula Student race car and to get acquainted with the basics of a modern liquid-cooled cooling system, this can be considered as successful. Also, even though the design of the cooling system was cautious, it can be considered successful too, since no problems occurred during 2019.

As a downside, the continuation of development for this cooling system is a bit complicated due few estimations that needed to be done during the design process. Fortunately, with a profound testing and validation of this cooling system, and the design procedure, severe benefits can be found in the future. Some ideas and following actions were presented in conclusions and recommendations chapter.

Another aim of this thesis was to provide a comprehensive but brief package of information regarding the cooling system and its design, for which it has succeeded. With the help of the information in this thesis the topic of a cooling system is more approachable for individuals with no previous experience.

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