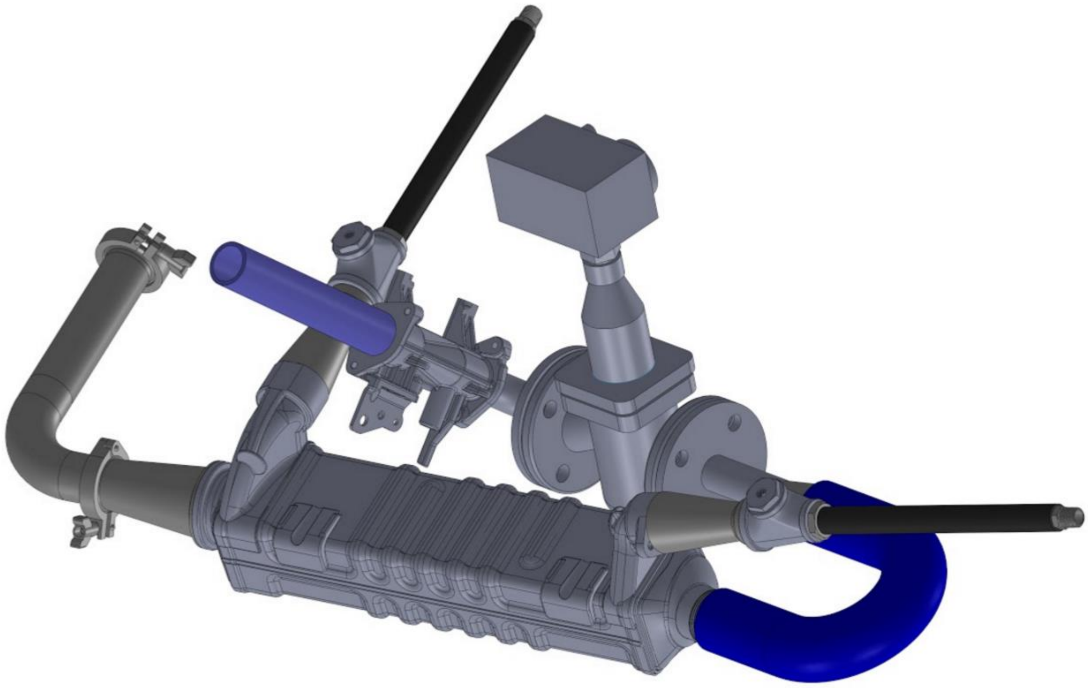


Design and Assembly of an EGR-circuit for an MD11 Research Engine

Bachelors's thesis in Mechanical Engineering

EMIL BJÄREHÄLL[1]
TOVE BURMAN[1]
ELAMIN HAMID ELAMIN[1]

JOHN LACY[2]
NICHOLAS PISCIOTTA[2]
EVAN VRABEL[2]

[1]Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

[2]Department of Mechanical and Nuclear Engineering
PENNSYLVANIA STATE COLLEGE
University Park, PA, USA 2017

BACHELORS THESIS 2017:02

Design and Assembly of an EGR-circuit for an MD11 Research Engine

EMIL BJÄREHÄLL
TOVE BURMAN
ELAMIN HAMID ELAMIN
JOHN LACY
NICHOLAS PISCIOTTA
EVAN VRABEL

Department of Applied Mechanics

CHALMERS UNIVERSITY OF TECHNOLOGY
Gothenburg, Sweden 2017

Department of Mechanical and Nuclear Engineering

PENNSYLVANIA STATE COLLEGE
University Park, PA, USA 2017

Design and Assembly of an EGR-circuit for an MD11 Research Engine
EMIL BJÄREHÄLL
TOVE BURMAN
ELAMIN HAMID ELAMIN
JOHN LACY
NICHOLAS PISCIOTTA
EVAN VRABEL

Supervisor at Chalmers University of Technology: Dr. Alf H. Magnusson
Examiner at Chalmers University of Technology: Prof. Sven B. Andersson
Supervisor and Examiner at Pennsylvania State College: Prof. Jason Moore

Bachelor thesis 2017:02
ISSN 1654-4676
Institution of Applied Mechanics
Chalmers University of Technology
SE-412 96 Gothenburg
Sweden
Telephone: + 46 (0)32-772 1000

Cover: A CAD-drawing of the finalized EGR-circuit featuring the cooler, the control valve, the venturi and all the intermediate pipes and flanges.

Compiled in LaTeX
Printed by: Chalmers Reproservice
Gothenburg, Sweden 2017

Abstract

Volvo Group, North America will be performing combustion research in conjunction with the University of Michigan to increase the efficiency of a new engine being developed. This research will be conducted with an 11 L static engine (model MD11) in a test cell that will undergo a variety of different tests to advance industry knowledge of combustion, engine design, and engine efficiency. One of the major components of diesel engines is the exhaust system. Volvo assigned the construction and design of an exhaust gas recirculation (EGR) system, for use in a Michigan test cell, to engineering students from Penn State and Chalmers Universities. The exhaust gas recirculation system on commercial diesel engines reduce the NO_x-emissions. This is achieved by recirculating a portion of exhaust gas into the intake manifold.

The objective of this project was for the student teams at Penn State and Chalmers to design and physically assemble portions of the EGR system with the support of numerical calculations to prove that the system would operate as intended. The team was responsible for sourcing components and proper system connections and outfitting specific sensors to the system to ensure effective monitoring of the engine during testing.

The Chalmers team was given thermodynamic data from a similar system in Gothenburg, Sweden, to calculate key parameters needed to understand the flow of exhaust from the engine in Michigan. Calculations that partly aided the system design. The Chalmers students also consulted suppliers about potential appropriate electrically actuated control valves needed to control the flow in the system. The Penn State students were responsible for the physical design, connections and the system components.

The students fully designed the system using computer automated design software, and ran thermodynamic calculations to verify that the system would theoretically operate as intended. The team effectively sourced, purchased, and assembled the main components of the EGR system.

Keywords: **EGR · EGR cooler · Control Valve · Venturi**

Sammandrag

Volvo Group North America genomför förbränningsforskning tillsammans med University of Michigan för att förbättra effektiviteten hos en ny motor som är under utveckling. Forskningen utnyttjar en uppställning med en motor på 11 L (modell MD11) som väntas genomgå flera olika tester för att bidra till industrins samlade kunskap gällande förbränning, motordesign och verkningsgrad. En av de mest omfattande delarna i en dieselmotor är avgassystemet. Volvo tilldelade arbetet att konstruera ett system för avgasrecirkulation (eller *Exhaust Gas Recirculation*, EGR), som ska användas i forskningsmotorn i Michigan, till ingenjörsstudenter från Pennsylvania State College och Chalmers tekniska högskola. EGR-systemet i kommersiella dieselmotorer används för att reducera skadliga NO_x-emissioner. Detta åstadkommes genom att EGR-systemet cirkulerar tillbaka en del av avgaserna till den inkommande friskluften.

Målsättningen för projektteamet, bestående av studenter från Chalmers och Penn State, var i detta projekt att konstruera, och montera, delar av EGR-systemet. Som hjälp användes numeriska beräkningar för att se till att systemet kommer fungera som önskat. Teamet var ansvarigt för att identifiera och beställa komponenter, röranslutningar och lämpliga sensorer som möjliggör noggrann övervakning av systemets prestanda under operation.

Chalmersstudenterna erhöll termodynamiska data från en liknande forskningsmotor i Göteborg som underlag för att beräkna de viktigaste egenskaperna hos det förväntade flödet genom systemet i Michigan. Beräkningarna underlättade samtidigt delar av konstruktionsfasen. Chalmersstudenterna kontaktade även flera tillverkare i sökandet efter en lämplig elektriskt styrd ventil för att reglera flödet genom systemet. Studenterna på Penn State ansvarade huvudsakligen för systemdesignen, alla röranslutningar och komponenter.

Projektteamet levererade en komplett design av ett EGR-system i form av en sammanställd CAD-ritning och genomförde termodynamiska beräkningar som verifierade att systemet, i teorin, upplever de ställda kraven. Vidare identifierades, beställdes och monterades de huvudsakliga systemkomponenterna.

Nyckelord: **EGR · EGR-kylare · Reglerventil · Venturi**

Acknowledgements

We would like to give thanks to all who have helped us during our work on this project. We are especially grateful for our examiner Prof. Sven B. Andersson and our supervisor Dr. Alf H. Magnusson for all their help and input on our progress, Magnus Christensen at Volvo Trucks Technology for providing us with operational data from his research engine and helping us when we got stuck in our calculations and also our contacts at Volvo Group North America and the University of Michigan, mainly Sam McLaughlin and Prof. Andre Boehman, for providing us with all the information needed to complete this project. Furthermore, we would like to extend thanks to Prof. Mikael Enelund at Chalmers and Prof. Jason Moore at Pennsylvania State College who both helped guide us students through this international collaboration.

Finally we want to give a special thanks to Volvo Group and Monica Ringvik, Director of Research & Innovation Policy at the Volvo Group, Gothenburg, for financially supporting the Chalmers students' visit to Penn State and vice versa. The interesting and fun visits across the Atlantic helped the student cooperation immensely and contributed to a better project overall.

Nomenclature

AFR, AFR_{sto}	The air-to-fuel ratio and stoichiometric air-to-fuel ratio. AFR represents the ratio between the air and fuel masses in the combustion while the stoichiometric equivalent, AFR_{sto}, is the mass ratio required between air and fuel for complete combustion.
C_c, C_h, C_r	C is the heat capacity rate, i.e. the product of the specific heat capacity and the mass flow. The index 'c' stands for cold fluid, the index 'h' stands for hot fluid and C_r is the fraction of min(C_c, C_h) and max(C_c, C_h).
EGR	EGR stands for Exhaust Gas Recirculation, which is a system for bringing exhaust gas back into the engine.
gal	Unit of volume, in this report gal denotes US gallon and 1 gal = 3.7854 L.
h	The heat convection coefficient [W/(m^{2}K)].
hp	Abbreviation for horse power, this report uses metric horsepower i.e. 1 hp = 735.499 W.
k	The thermal conductivity [W/(m K)].
Nusselt, Nu_D	The Nusselt number is the fraction of the convective and conductive heat transfer across the border. The index 'D' stands for diameter and means that the boundary is circular.
NTU	NTU stands for Number of Transfer Units and is defined as the fraction between UA and C_{min}.
Prandtl, Pr	The Prandtl number is a dimensionless number used in fluid calculations. It is defined as the ratio of momentum diffusivity to thermal diffusivity.
q	q is the heat transfer rate [W].
Reynolds, Re_D	The Reynolds number is a dimensionless number used in fluid calculations. It helps predict flow patterns in fluid. The index 'D' means that the number is used for internal flows.
UA	UA is the product of the overall heat transfer coefficient, U, and the heat transfer area, A [W/K].
ε	ε is the heat transfer effectiveness, the ratio of the actual heat transfer rate to the maximum possible heat transfer rate.
λ	The ratio between the AFR and AFR_{sto}, i.e. the amount available air relative the amount required for complete combustion.

Table of Contents

1 Introduction

This report covers a project that is a collaboration between Chalmers University of Technology and Pennsylvania State University. Volvo Group North America sponsored this project with the University of Michigan as the intended customer.

Volvo Group North America, a major manufacturer of trucks, will be conducting combustion tests with a single-cylinder diesel engine in a test cell at the University of Michigan. The engine is a modified six cylinder, 11 L Volvo engine called MD11 with a maximum power of 425 hp. Five of the cylinders will be made inactive, leaving one operational. This means that the modified engine had one sixth of the normal 11 L volume and one sixth of the maximum power. Due to the experimental engine's unique configuration, it requires a special exhaust gas recirculation (EGR) system to reduce its NO_x emissions, since the previously installed EGR system was not dimensioned for small exhaust flow.

Included in this report is a brief introduction to EGR as a concept, the problem statement and limitations found under Section 1. Under Section 2 the details of the cooperation between the two students groups at Chalmers and Penn State, the customers at the University of Michigan and the sponsor Volvo is described. Section 3 through 5 features a more in-depth problem description acquired through communication with the customer, a brief overview of existing solutions to similar problems and a summary of the target system specifications that the final product must satisfy. The concept generation, concept selection and design work is treated in Sections 6 and 7. A more detailed system analysis, which serves to substantiate the chosen concept and predict the final performance of the system, is found under Section 8. Section 9 describes the construction of the system in detail and lastly Section 10 and 11 features discussion, conclusions and recommendations for the future development of the test cell.

1.1 Background

As of 2017, the global transportation industry continues to rely, essentially exclusively (International Energy Agency, 2009), on the combustion of petroleum. Since the invention of modern, petroleum driven, internal combustion cycles in the late 19^{th} century[1], combustion technology has advanced rapidly in the areas of performance, fuel efficiency and reliability. Consequently, the different types of internal combustion engines were one of the most important and commercially successful technologies of the 20^{th} century.

However, the burning of fossil fuels on a global scale has given rise to environmental and health concerns, which gained serious public, political, and scientific attention starting in the 1950s (Moseley, 2014). For the environment, the net increase of carbon dioxide (CO_2) in the atmosphere from combustion gives rise to the greenhouse effect alongside other exhaust products such as water vapor and nitrous oxide (N_2O) (IPCC, 2008).

[1] The first two-stroke spark ignited (SI) "Lenoir engine" (after Jean Joseph Étienne Lenoir) was patented in 1860, Nikolaus Otto built the first four-stroke SI engine, which utilized a compressed charge in 1876 (Ratiu, 2003) and Rudolph Diesel patented his compression-ignition (CI) engine in 1892. (Massachusetts Institute of Technology, 2008)

Public health is at risk with the indirect creation of ground level ozone (O_3) from nitric oxide (NO) (World Bank Group, 1998) and the emission of soot particles (Smith, 2012).

To address the serious environmental and health related consequences of every type of combustion, legislative guidelines and laws have been instituted on both global and nationwide scales to regulate the composition of exhaust gas (EG) from passenger cars, trucks, boats, airplanes, industry, power generation, motor driven machinery, et cetera (Jääskeläinen and Khair, 2016). These stricter requirements have been an important factor in the latest chapter of engine development and forced rapid advancements in exhaust catalysis and exhaust gas recirculation (EGR) to reduce nitrogen oxide (NO_x) emissions.

EGR is based on the idea that exhaust gas from the combustion can, in part, be recirculated and added to the intake air on its way into the engine cylinders. The composition of the exhaust gas differs from the ambient air since it contains much higher concentrations of CO_2 and water vapor while the oxygen (O_2) concentration is lower (Jääskeläinen and Khair, 2016). Subsequently, the chemical and thermal properties of the intake air can be somewhat controlled by steering the amount of EG that is recirculated; the EGR rate. If the EGR rate is controlled properly the formation of NO_x can be decreased because (a) the diluted oxygen concentration contributes to a lowered flame temperature during combustion, (b) the flow rate is increased by introducing EGR, (c) the heat capacity of exhaust gas is greater than that of regular air (Jääskeläinen and Khair, 2016). All of these factors contribute to lower the peak temperature in the combustion chambers during engine operation which inhibits, the highly temperature dependent (Heywood, 1988), formation of NO_x from nitrogen and oxygen (Jääskeläinen and Khair, 2016).

1.2 EGR and Engine Description

A conventional EGR circuit in a combustion engine usually consists of three main components. A controllable EGR valve which can steer the amount of exhaust gas that is recirculated, a dedicated EGR cooler (not the same intercooler that is meant for the intake air) that cools the exhaust to keep the mixed intake gas below a certain temperature threshold and intermediate pipes to transport the gas. (Jääskeläinen and Khair, 2012) The EGR cooler lowers the temperature of the recirculated exhaust gas so that the gas mixture does not reach a too high temperature and mixes well with the intake air. The valve controls the EGR rate by controlling how much of the exhaust gas enters the EGR circuit. The EGR rate is often found by calculating the ratio of the CO_2 concentration in the exhaust gas (*eg*) and the difference in concentration between the ambient air (about 0.04%) the intake gas mixture (*gm*) in the engine (Jääskeläinen and Khair, 2016),

$$\mathrm{RATE}_{EGR} = \frac{[\mathrm{CO_2}]_{gm} - [\mathrm{CO_2}]_{ambient}}{[\mathrm{CO_2}]_{eg}}. \tag{1}$$

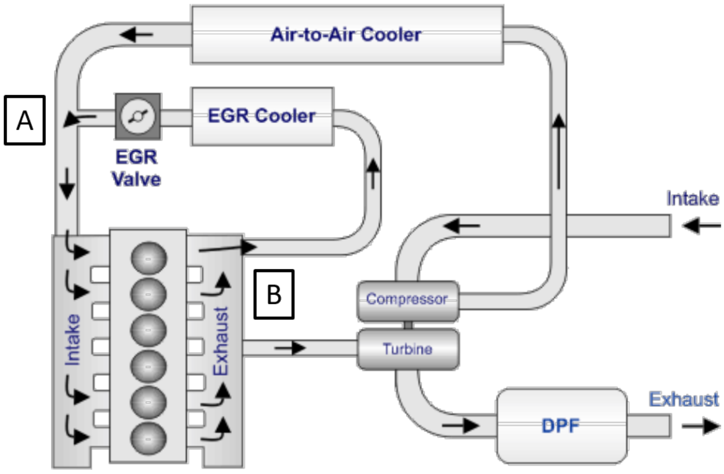


Figure 1: A schematic showing a six cylinder engine with a high pressure EGR-loop. The air enters the system at `Intake` on the right and travels through the `Compressor` and the `Air-to-Air Cooler` before being mixed with recirculating exhaust at `A`. At `B` some of the exhaust is led through the EGR-circuit while the rest exits the system, through the turbine and Diesel Particulate Filter (`DPF`). Picture reference: (Jääskeläinen and Khair, 2016).

A schematic picture of a six cylinder engine with a high-pressure EGR circuit is presented in Figure 1.

The intake air enters the system at `Intake`, on the right in Figure 1 and is first compressed to increase the pressure and then cooled in the `Air-to-Air Cooler` before being mixed with recirculated exhaust gas. The mixing of air and exhaust gas happens at `A` just before the engine. The gas mixture then enters the engine, at `Intake`, and exits as exhaust gas at `Exhaust`. (Jääskeläinen and Khair, 2016) Most of the exhaust gas exits the engine block through the turbine and Diesel Particulate Filter (`DPF`), but some of it enters the EGR circuit at `B`. The EGR-rate is changed for different engine speeds and load levels to ensure best possible NO_x reductions while at the same time maintaining the target air-to-fuel ratio, AFR. (Jääskeläinen, 2016)

1.3 Initial Project Statement

The objective of this project is to design an EGR system for the research engine in the test cell at University of Michigan. The system should be partly assembled at Penn State and delivered in sections that are easily assembled and mounted in the under construction research engine in Michigan. A block map describing the test cell can be found in Appendix A.

One of the key technical components of the EGR system is already available, the system will use an EGR cooler provided by the sponsors at Volvo Group, North America. The final system must feature a method to measure the gas flow and an electrically actuated EGR-valve for control. The EGR system must also connect to the pipe inlets and outlets of the test cell, fit entirely within specified dimensions and be designed for the flow specifications and EG temperatures of the research engine.

1.4 Objectives

The main objective of this project, as seen in Section 1.3, is to create a design for a functional EGR system, to be used with a six-cylinder 11 L diesel engine operating with a single cylinder. The main objective can be divided into three smaller objectives, sub-objectives.

The first sub-objective is to find operational values for the engine, desired operational performance of the EGR system working in tandem with the engine and other specifications required to design and perform calculations on the EGR system. Some of the important data to define about the EGR system can be seen in Table 1.

In addition to the necessary engineering specifications provided in Table 1, the team will also need to define which materials to use to safely contain gas with possibly high temperatures and acquire details of the space availability in the test cell.

The second sub-objective originates from the fact that the EGR system will be used in the single cylinder MD11 research engine. The exact dimensions, specifications and placements of the points of contact (i.e. the pipe connections) with the rest of the test cell must be taken into account when the system is designed. Its crucial that the available space, its geometrical shape and possible weight limits, is respected and the system is assembled in a way the customer finds sufficient. This sub-objective can be divided into three parts that ensure an appropriate design:

1. A choice of materials and components that handles the specified flows and allows the design to stay within the space constraints
2. Decide the order of components and methods for assembly with customer requests in mind
3. 3D design with respect to the space availability that ensures and proves the design

Table 1: List of operational values and test cell specifications to define for the EGR system.

	Data
1	Coolant inlet temperature
2	Coolant type
3	Coolant flow rate
4	Heat exchange rate of EGR cooler
5	Exhaust gas inlet temperature
6	Exhaust gas inlet flow rate
7	Desired exhaust outlet temperature, i.e. gas exiting the EGR-circuit
8	Actual exhaust temperature from engine

Table 2: List of key Sub-objectives that describes the project.

	Sub-objective
1	Find operational values and specifications
2	Design EGR system
2.1	Find material and components suitable for the EGR system
2.2	Decide the orders of components and methods for assembly
2.3	Create a CAD model of the designed EGR system
3	Do calculations on the EGR cooler
4	Make a cost estimate of the design
5	Acquire and assemble parts

The third sub-objective is to perform calculations on the EGR cooler, regarding its cooling capacity. To guarantee or debunk its performance working in the premise of the test cell.

The final two sub-objectives of this project is the financial aspect, to stay within a set budget and create a cost analysis of the design. As well as the actual acquiring and assembling of the EGR system parts. A list of all the key sub-objectives can be found in Table 2.

1.5 Scope of Work and Limitations

The project stretches over 20 weeks (January 9th to May 23rd) for the Chalmers students and 17 weeks (January 9th to May 1st) for PSU students, with reservations for Easter holidays and breaks for exams. The project is done on part-time, i.e. 20 hours per week, by the Chalmers students while the PSU students has 15 hours per week. The limited amount of allocated project-time for all group members is a major limitation which restricts the level of sophistication of the final product. Furthermore, administrative documents as well as written project milestones and deadlines compete for the limited time spent on the actual deliverables. The Final Report, Design Specification and Proposal all serve as very useful guidelines and repeated affirmation of what has been accomplished and what remains to be done but thorough document writing could require an undesired amount of effort.

Another project limitation is the fact that the engine system is located at University of Michigan which means that PSU and Chalmers students almost exclusively (except for a visit to Hagerstown by PSU students) have to work without hands-on contact with the actual system. This means that component fitting and testing will be difficult to arrange on short notice and the acquiring of any system properties will be restricted to email and Skype conferences. The 6 hour time difference between the two student teams also means that communication will be suffering some delays. However, a visit to State College and Penn State by the Chalmers students (April 18th to 26th) and a visit to Gothenburg and Chalmers by the PSU students (May 19th to 26th) is made possible by a contribution from Volvo, which aids the cooperation immensely and provides an opportunity to bring everyone involved up to speed.

2 Team and Project Management

The team and project management includes an economic analysis, a project schedule and management, a risk analysis, ethics and environmental statements and a plan of communication with the sponsors of the project. The details surrounding team and project management were important to define for the project to run smoothly and provide good conditions for achieving the objectives of the project.

The economic analysis helped the team prioritize and stay within budget, the project schedule and management made sure all the team members knew what to do and made sure the project was completed on time. The risk analysis helped the team avoid major setbacks and provided a plan for dealing with problems that appeared during the course of the project. The plan for communicating with the sponsors and customers was important to make sure that all necessary questions for the sponsors were answered and that the input from the sponsors was received and taken into account. It was especially the exact dimensions and strict requirements imposed by the test cell at University of Michigan that demanded a close contact with sponsors and customers.

2.1 Preliminary Economic Analyses - Budget and Vendor Purchase Information

The students from Chalmers had a preliminary budget of $226 and the students from Penn State had a budget of $1000. The money was used to purchase the EGR system parts, pay for assembly fees and to cover travel expenses connected to the project. The vast majority of the funds were spent on the EGR system components. However, once it was made evident that a control valve with the level of sophistication that the sponsor required, on its own, were to cost more than $1000 this item was lifted out of the student budget once University of Michigan made the decision to purchase the component.

2.2 Project Schedule and Management

The Gantt Chart[2] shown in Table 3 outlines how much time was earmarked for each deliverable, along with corresponding deadlines and possible dependencies. This schedule, and more importantly the students' dedication to follow-through accordingly, helped the team project by reducing the amount of needed administrative actions and, consequently, by allowing more time to be spent on deliverables, tasks, research and writing.

[2]In Table 3, only the lists of activities in the Gantt Chart and their start and finish dates are shown, not the graphics showing how the work is spread out over the total time of the project which is an intuituve part of any Gantt Chart. The complete Gantt Chart takes up too much space to be attached into this report.

Table 3: Gantt Chart over the project schedule with all the activities with corresponding responsible student group, together with possible dependencies between activities and time frame.

Activity	Responsible	Key	Dep.	Start	Finish
Project Start		**1**			
Write group contract	PS & Chalmers	1.1		17-01-24	17-01-27
Write project plan	PS & Chalmers	1.2	2.1	17-01-26	17-02-12
Create Gantt chart	PS & Chalmers	1.3		17-01-24	17-01-26
Scope of Work: 1		**2**			
Research EGR-systems	PS & Chalmers	2.1		17-01-27	17-02-17
Write background of project	PS & Chalmers	2.2	2.1	17-02-06	17-02-20
Write project scope, limits	PS & Chalmers	2.3		17-02-06	17-02-20
Scope of Work: 2		**3**			
Calculations based on spec.	Chalmers	3.1		17-02-21	17-03-10
Design system based on calc.	PS & Chalmers	3.2	3.1	17-02-28	17-03-15
Make system model in CAD	Penn State	3.3	3.2	17-03-15	17-03-24
Scope of Work: 3		**4**			
Implement system in engine (CAD)	Penn State	4.1	3	17-02-20	17-03-17
Scope of Work: 4		**5**			
Research possible instrumentation	Chalmers	5.1		17-03-20	17-03-24
Choose most suitable and available components	PS & Chalmers	5.2	5.1	17-03-24	17-03-28
Scope of Work: 5		**6**			
Provide engineering layout (finished CAD-model)	PS & Chalmers	6.1	5.2	17-04-05	17-04-10
Provide information on how to procure components	PS & Chalmers	6.2	5.2	17-03-28	17-03-31
Testing		**7**			
Create test plan	Penn State	7.1	5.1	17-04-20	17-04-30
Chalmers finishing work		**8**			
Find information on valves	Chalmers	8.1		17-05-02	17-05-07
Summarize and compare valve information	Chalmers	8.2	8.1	17-05-03	17-05-11
Trips					
PS students to Hagerstown	Penn State			17-02-14	17-02-14
Chalmers students to PS	Chalmers			17-04-18	17-04-26
PS students to Chalmers	Penn State			17-05-19	17-05-25
Deadlines					
Statement of Work	PS & Chalmers			17-02-12	
Midterm Report	PS & Chalmers			17-03-27	
PS Design Showcase	Penn State			17-04-27	
PS Final Report	Penn State			17-05-01	
Chalmers Final Report	Chalmers			17-05-12	
Chalmers Presentation	Chalmers			17-05-23	

2.3 Risk Plan and Safety

During the course of the project, the team experienced setbacks and adverse conditions. Particular risks we dealt with included scheduling risks, communication risks, technical risks, and manufacturing risks. The most significant risk, due to the separation by distance of the teams was customer satisfaction i.e. how well the designed system lives up to the specifications put forth by University of Michigan. The team ensured that the customer would be satisfied by remaining in contact with the sponsor to fully understand the sponsor's expectations. Table 4 shows a summary of risks along with methods to minimize them as well as fallback strategies. The first entry, "Change in customer specification", was from the start rated "Moderate" however during the course of the project this proved to be the most serious of the concern presented in Table 4. As the test cell was still under construction new information continuously reached the sponsor and eventually the student team, which made for a highly dynamic project.

The safety of project team members and those who would use final the product was taken very seriously. All proper guidelines and precautions for manual part production and assembly were taken into account. The parts produced and acquired were in accordance with the operating procedures of the test cell at the University of Michigan.

Table 4: A risk plan portraying categories of project risks that ultimately could influence the final product.

Risk	Level	Actions to minimize	Fallback strategy
Change in customer specifications.	High	-Attend weekly meeting with Volvo and University of Michigan.	-Include extra meeting time until problem is resolved. -Large percentage of money as emergency funds.
Product does not function as expected.	Moderate	-Use available resources at PSU and Chalmers. -Test product as early as possible.	-Use other designs from concept generation.
Customer is not satisfied.	High	-Fully understand the deliverables that are expected of us.	-Work with sponsor and Uni. of Michigan to resolve any problems that arise.
Lack of communication overseas	Low	-Use WhatsApp to communicate any questions or thoughts on the project.	-Take advantage of class Skype meetings to communicate face-to-face.
Schedule delays	Moderate	-Constantly check Gantt Chart to track progress. -Try to work ahead of schedule.	-Build in a week of extra time to account for future delays.

2.4 Ethics Statement

The team adhered to the ASME Code of Ethics of Engineers (ASME, 2006). The code of ethics provides means to "uphold and advance the integrity, honor and dignity of the engineering profession". All members of the team familiarized themselves with the code, to make sure it was adhered to.

2.5 Environmental Statement

The Volvo Group's goal is to be ranked as one of the world's leading companies in environmental care (Volvo Group, 2007). Penn State and Chalmers helped Volvo inch towards this goal by also striving to minimize the environmental impact of carrying out the project. The purpose of an EGR system is to reduce NO_x emissions. Therefore, by assisting Volvo and University of Michigan with the completion of this project, added a necessary building block towards helping the environment. However, it should be noted that every aspect of the resulting EGR system's operation is to be taken into account when evaluating its environmental performance. The environmental profile of the final system cannot be deduced solely from the concentration of NO_x in the exhaust gas since this must be weighed against possibly higher fuel consumption and emission of particulates (Jääskeläinen and Khair, 2016).

When selecting components and materials, the project abided by Volvo Group's environmental requirements for suppliers (Volvo Group, 2007), which laid out requirements for suppliers and contractors to reduce environmental impact.

2.6 Communication and Coordination with Sponsor

The team of students communicated with the sponsor via e-mail, with Evan Vrabel being the team's main point of contact. Sam McLaughlin was the primary contact for Volvo and Prof. Andre Boehman was the primary contact for the University of Michigan. Weekly meetings with Penn State, Chalmers, Volvo, and the University of Michigan were conducted via Skype on Tuesdays at 9 am Eastern Time (3 pm Central European Time). Additional meetings were arranged as needed. The Penn State team members visited Volvo's facilities in Hagerstown, MD on Tuesday, February 14th to tour the plant, meet regarding the project, and gather information on the engine.

Magnus Christensen at Volvo Trucks Technology also provided the students with very valuable information and design inspiration from his test cell via Tove Burman as main point of contact. The test cell featured the exact same EGR-cooler and is in general a setup similar to the one at University of Michigan. The main difference being that the test cell at Volvo Trucks Technology uses a specially made single cylinder engine, originating from a six cylinder 13 L engine.

3 Customer Needs Assessment

The customer of this project was University of Michigan, but the project was sponsored by Volvo who also assisted in mediating and developing customer needs. The EGR system was only designed for use in the specific test cell at University of Michigan, which meant that there were no other end buyers or possibility for mass production.

The team identified the needs of University of Michigan and Volvo to be a functional EGR system at the conclusion of the term. Michigan provided a list of potential deliverables that they needed for the test cell to function properly. However, the requirements for the team were narrowed to focus on the EGR system itself. If the EGR system was capable of being more easily developed, there were tiers of other deliverables that would be available to the team to further build on the system. Originally the deliverables were specified as:

1. CAD Model for the EGR system.
2. A functional EGR flow control using available and designed components.
3. Validation test specifications and test plan.
4. Cost estimate for the design.
5. Provide design documentation for procuring components and possible failure modes analysis.

However, over the course of the project, the deliverables changed as a response to the changes in budgeting (moving the purchase of the costly control valve to University of Michigan) and the demands of customer. Towards the end of the project only deliverable one and four remained without change and the list had developed into:

1. CAD model for the EGR system.
2. A partly assembled EGR system ready for direct use in the test cell.
3. Validation calculations.
4. Cost estimate for the design, with price offer for out of budget control valve.
5. A researched collection of control valves and actuators compatible with the system.

3.1 Gather Customer Input

The team shared the customer needs gathered during Skype meetings with the sponsors as well as through e-mails from Sam McLaughlin, and Prof. Andre Boehman. The customer input was continuously gathered and the project goals were revised when necessary. Measurements, pictures and CAD-drawings of the test cell were especially important formats for customer input.

3.2 Weight of Customer Needs

The customer needs were divided into four groups: functionality, safety, efficiency and cost. The need functionality was defined by the need for the designed system to meet all requirements for a functional EGR system. For the functionality group there were two essential aspects. First, the system needed to function similarly to the commercial EGR, meaning controllable and NO_x reduction; Secondly, the designed system needed to be compatible with the test cell.

The need for safety ensured a quality product that will operate without damage to the nearby parts or anyone working with it. Efficiency related to optimizing the EGR system so it is not oversized or over specified. Cost is defined by designing the system and choosing components without exceeding the budget.

In order to make decisions and compromises easier during the design project, a weighted listing of customer needs was used. Besides the benefits for decision making, a hierarchical register of needs also served as reassurance factor for customer satisfaction. inter was the most important customer need. Table 5 shows the Analytical Hierarchy Process table, which shows how the customer needs rank. As seen the Functionality and Safety of the system are of highest priority.

Table 5: AHP diagram comparing the customer needs.

	Functionality	**Safety**	**Efficiency**	**Cost**	**Total**
Functionality	1.00	1.33	2.00	4.00	8.33
Safety	0.75	1.00	1.50	3.00	6.25
Efficiency	0.50	0.67	1.00	2.00	4.17
Cost	0.25	0.33	0.50	1.00	2.04

4 External Search

An external search was performed to understand if there were any publicly available patents that would have had to be taken into consideration when developing the concepts in the project. This also helped the project members find what existing products and concepts are already on the market. This search was necessary to make sure that the developed concepts did not use technology which is patented. It was also a way to gather different ideas on how to develop a system which satisfies the objectives and problem statement.

EGR systems are an old technique and this project only focused on using existing, commercially available, components and parts. This means that there were no patents that either helped or hindered the progress of the project. The studied reference engine and easily accessible literature regarding internal combustion engines was more than enough to inspire a system design and selection of components. The project is in this sense not focused on innovation, as much as construction of a highly constrained and specific EGR-system.

4.1 Existing Tests

The EGR cooler used in the test cell at the University of Michigan will undergo an emissions test to identify concentrations of chemical compounds like CO_2 and NO_x. The team's external research focused on other tests that are applied to the EGR system. We found two other tests that removed the EGR cooler from the engine in order to be tested.

The first test found was a leak test. A leak test pressurizes the EGR cooler with air to check for any leaks at the exhaust or coolant connections. The cooler is pressurized to 20 psi and soapy water is sprayed over the connections. If any bubbles appear at the connections from the soapy water, the connection has a leak and has to be fixed before being remounted to the engine. Other leak tests pressurize the exhaust flow section of the EGR cooler to check for tube cracks within the cooler. (Detroit Diesel, 2008)

Another off engine EGR cooler test is a thermal cycling test that can calculate the EGR cooler's effectiveness and the thermal fatigue life. In a thermal cycling test, as many as four EGR coolers can be tested. The EGR cooler is set up in a test rig with exhaust and coolant connections to inlet and outlet ports. While the thermal cycling test is running, exhaust gas temperatures fluctuate from maximal engine conditions to idle conditions. The test is run until the number of cycles is met or there is a significant change in the delta pressure. A change in the delta pressure can signify a crack or soot buildup (fouling) in the cooler (Southwest Research Institute).

5 Engineering Specifications

The engineering specifications are determined by the desired effectiveness of the EGR cooler and functionality of the EGR system, the space availability and location of connections in the test cell, different operating conditions and specifications and conditions of the connected systems. The connected systems are the coolant system and the engine. The various resulting values of the EGR system are dependent on the coolant flow and temperatures and the engine's load and speed.

5.1 Establishing Target Specifications

The target values relate to the customer needs: functionality, efficiency, cost and safety of the designed EGR system, as made apparent in Section 3.2.

Safety relates to the safety of the people handling the EGR system, there should not be any risks of injury if the EGR system is handled correctly. Therefore, it is critical that the system is assembled and handled correctly.

Product demands from the customer regarding functionality and efficiency that the final EGR needs to satisfy can be broken down into operational aspects as "Achieves proper flow rate", "Controls EGR rate", "Completely sealed system" and "Works between minimum and maximum temperatures". A completely sealed system is achieved by proper component fitting and pipe design and the same controllability of the EGR flow is achieved by selecting a suitable control valve.

The system should also produce satisfying results by "Reducing NO_x emissions" and by "Measuring relevant parameters". The current limit for NO_x emissions from heavy-duty engines such as the MD11 is roughly $0.4\,\mathrm{g\,kWh^{-1}}$ since 2011 (Jääskeläinen and Khair, 2016). As for the relevant measurement parameters there are the ones needed for a successful control system (EGR mass flow and EGR valve status). Lastly the system will adhere to financial limits and "Cost less than $1000", as is stated in Section 2.1.

The provided target values for the operational aspects are presented in Table 6. These values were provided by the researchers at the University of Michigan.

Table 6: Target values for the operational aspects of the EGR system.

Operational aspect	Target value
Operating temperature of exhaust gas	600 °C
Maximum engine power	70.8 hp (one sixth of 425 hp)
Coolant flow rate	4 gal/min (at 60 psig)
Coolant inlet temperature	5 to 90 °C
Desired EGR rate	0 to 40% of total gas exhaust flow
Desired outlet temperature from EGR cooler	≈ 80 °C
Desired EGR cooler effectiveness	90%

5.2 Relating Specifications to Customer Needs

In the Needs/Metrics matrix in Table 7 the correlation between the target specifications of the final EGR system and the corresponding customer needs group is shown. As seen, the cost aspect is fairly isolated from other parameters. This is due to the fact that this project only sees to the construction and mounting of the system and, besides assuring functional, efficient and safe operation, does not account for the cost of its continued operation.

Table 7: Needs-Metrics matrix which correlates target specifications with corresponding customer needs.

Needs/Metrics	**Achieves proper flow rate**	**Reduces NO_x emissions**	**Controls EGR rate**	**Operates within temp. limits**	**Sealed system (no leaks)**	**Measures parameters**	**Costs less than $1000**
Functionality	X	X	X	X	X	X	
Efficiency	X	X	X				
Cost							X
Safety				X	X		

6 Concept Generation and Selection

To generate concepts, the team sought clarification on the details of the problem, which was then divided into parts. Each section had unique spatial and temperature requirements that had to be taken into consideration.

The concepts were generated through analyzing the function of a given part and then researching and asking the sponsors how that function has been solved previously. The generated concepts were then compared with each other to identify the most suitable concept, based on technical specifications, sponsor demands and the budget of the project.

6.1 Problem Clarification

There were five main physical sub-components involved in this EGR system:

1. Piping from engine exhaust plenum to the new EGR cooler.
2. The EGR cooler itself with corresponding original valve (if its performance is satisfactory) and a mount.
3. Connections between the EGR cooler and the already present chiller/circulator.
4. Piping from the EGR outlet to the Volvo venturi intake of the engine.
5. A mass flow sensor (preferably located downstream of the EGR cooler).

The physical properties of the current engine test cell imposed strict constraints for the EGR assembly. Volume and dimension restrictions for the final system as a whole were specified by University of Michigan and must be followed. The dimensions of the connections from the exhaust plenum (2 in), from/to the chiller/circulator ($1/2$ in), to the intake manifold (approximately 85.35 mm according to CAD-file provided by sponsors) and from/to the EGR cooler itself imposed further restrictions.

6.2 Concept Generation

The team generated two concepts respectively for the placement of the EGR valve and for different methods of measuring the airflow. Because of the highly specific nature of the final product, the EGR-circuit, the choice to move one component up-stream or down-stream directly influenced the design of the intermediate piping and placement of other components. As a consequence the generation of perfectly defined system concepts which could be compared with regard to one or even multiple parameters were not applicable in this project. However, many design choices were made in collaboration with University of Michigan as well as Volvo representatives and the development and motivation behind each decision is treated in this section.

6.2.1 Placement of EGR Valve

The EGR valve could be placed on either the hot side of the EGR cooler or the cool side of the EGR cooler.

If it were placed on the hot side it would need to manage operational temperatures of 600 °C, specified in Table 6. Materials that can manage such high temperatures are often very expensive which imposes budgeting issues.

If it were placed on the cool side it would not have to manage as high temperatures, allowing for a less expensive material. Including a safety margin a maximum temperature of 200 °C is to be expected on downstream of the cooler. The downside of placing the valve on the cool side of the EGR cooler was that the cooler temperatures may lead to condensation inside the valve, which could result in fouling due to the particles in the exhaust air.

6.2.2 Method of Measuring Air Flow

The sponsor preferred to conduct flow measurements using a commercially available venturi. There were two venturi options to choose between, a large venturi by Volvo and a smaller Renault venturi.

The large venturi, already installed on the test cell engine, is designed for use with the full mass air flow of the 11 L engine. Since the engine will only be used as a single cylinder engine, the venturi will only see one sixth of the EGR flows associated with the MD11 engine. There was possibility that the venturi would not be sensitive enough to measure the flow accurately.

The smaller venturi is designed for a 5 L Renault engine, and was thought to be better suited for measuring smaller airflows. The downside of using this venturi was that the venturi needed to be installed separately into the EGR system while the larger venturi was already included.

Each venturi uses an associated sensor to measure the pressure difference it creates, by outputting a voltage signal. The pressure measurement and venturi properties is the used to calculate volume flow.

6.3 Concept Selection

The concepts were selected by analyzing the ability to fulfill the technical specifications and demands from the sponsor and customer. Over the course of the project the different concepts were changed multiple times and in this section the development of the most important design choices are described.

6.3.1 Selecting Placement of EGR Valve

The placement of the EGR valve was determined by how it could manage the high temperatures of the exhaust gas and if the cost of the EGR valve was within budget. Other criterion for the EGR valve in general is that it must be controllable at University of Michigan and have such dimensions that it can be installed in the design of the EGR system. Those latter criteria was not to be taken into account when selecting the concept since these are not influenced by the placement of the EGR valve.

The EGR valves and flow control valves found through research, contacting suppliers and through communication with the sponsors varied in cost and quality. The main similarity between the different valves was that none of them fulfilled both important requirements. The valves made of material heat-resistant enough to withstand the high temperatures on the hot side of the EGR cooler were all too expensive.

After consultation with the sponsors working with the research engine at University of Michigan, it was decided to place the valve on the cool side of the EGR cooler. This meant that the valve would only be exposed to exhaust gas with a temperature of approximately 80 °C. However, at the request of the sponsor, the team looked for a valve that could manage approximately 200 °C to ensure it would not get damaged if the exhaust gas for some reason was not cooled enough.

6.3.2 Selecting Method of Measuring Air Flow

The selection between the larger or smaller venturi was based on calculations which can be seen under Section 8.2.3. The calculations helped determine what the relative pressure drop would be in the component. These calculations were made for both the smaller and larger venturi and then overlaid with the sensitivity of the Δp-sensor, which measures the pressure difference in the venturi, provided by the sponsors. The team wanted to determine whether the larger venturi generated a large enough pressure difference to be detected by the sensitivity of the sensor or if the smaller sensor would be more accurate. Based on the calculations and comparison in Section 8.2.3, it was determined that the smaller, 5 L venturi would provide the most accurate measurements.

6.4 Final Concept

The chosen concept with all the interconnections and main components is presented as a block chart in Figure 2, with a component key in Table 8. Exhaust gas drawn from the exhaust plenum goes through the Volvo EGR cooler to lower the gas temperature. The system uses an external wall mounted chiller/circulator, with circulating cool water, to provide constant coolant to the EGR cooler. The cooled exhaust gas then passes through the control valve into the venturi for flow and temperature measurement, before being fed back to the engine intake manifold.

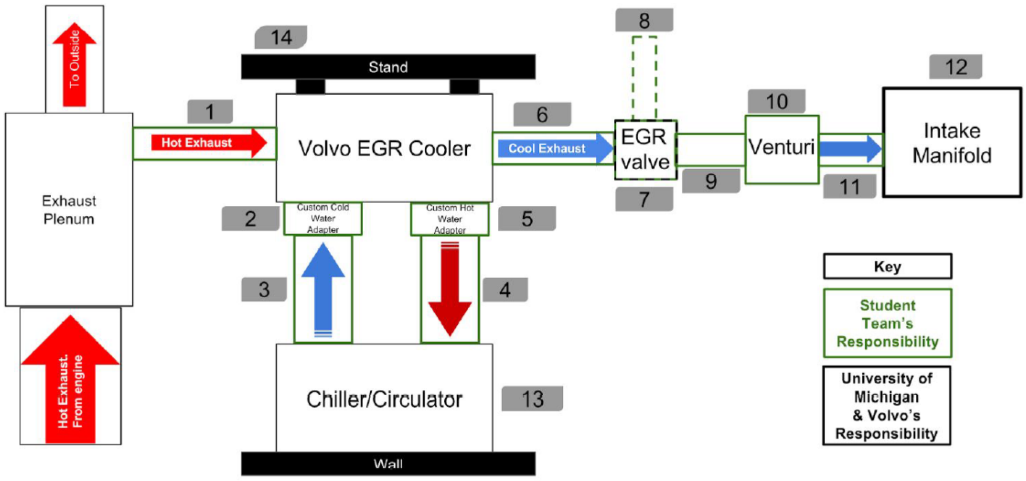


Figure 2: Block diagram describing the interconnection between the physical deliverables with labels from Table 8.

Table 8: Key of components for the block diagram in Figure 2.

Key	Physical Deliverable
1	Hot exhaust pipe from back pressure valve to EGR cooler
2	Cold water chiller to EGR cooler adapter
3	Cold water chiller to EGR cooler pipe
4	Hot water EGR cooler to chiller pipe
5	Hot water EGR cooler to chiller adapter
6	Cold exhaust pipe to EGR valve
7	Electrically actuated EGR valve
8	Possible connection for lubrication to EGR valve
9	Pipe between EGR valve and venturi
10	Renault 5 L venturi with corresponding Δp-sensor for flow measurement
11	High temperature rated hose connecting the EGR circuit to the intake manifold
12	The intake connection to the intake manifold of the MD11 engine block
13	The external chiller/circulator of the test cell
14	A stand for mounting the system in test cell

7 System Level Design

The team has designed an EGR system to be placed into a test cell at the University of Michigan. Our design was drafted in SolidWorks in order to verify component and system fit. Volvo provided the centerpiece of the system, the EGR Cooler. There are three sections that connect to the EGR cooler; The pre-cooler, post-cooler, and coolant systems. Figure 3 shows the total EGR system deliverable design rendered in SolidWorks. The pre-cooler, or exhaust inlet, is located at the far left of Figure 3. The coolant lines look identical and are located at the two ports on top of the EGR cooler, pointing towards the upper center and far right of the Figure. After the 180° silicone hose, which connects directly to the outlet of the EGR cooler, the gas travels through the control valve which is rendered with the actuator system attached. Lastly, the gas travels through a 90° turn and goes through the venturi while going over the EGR cooler. The exhaust outlet is seen right after the venturi, in the top left of the Figure, the short and straight silicone hose represents a much longer curved hose which leads the gas back to the intake manifold of the engine.

The pre-cooler section shown in Figure 4 contains high-polish stainless steel tubing which uses triclover style clamps to connect the system to the exhaust plenum. A welded 90° elbow aligns the tubing with the EGR cooler mount before a reducer clamps to the tube to match the diameter with that of the EGR cooler inlet. The reducer is welded to the EGR cooler.

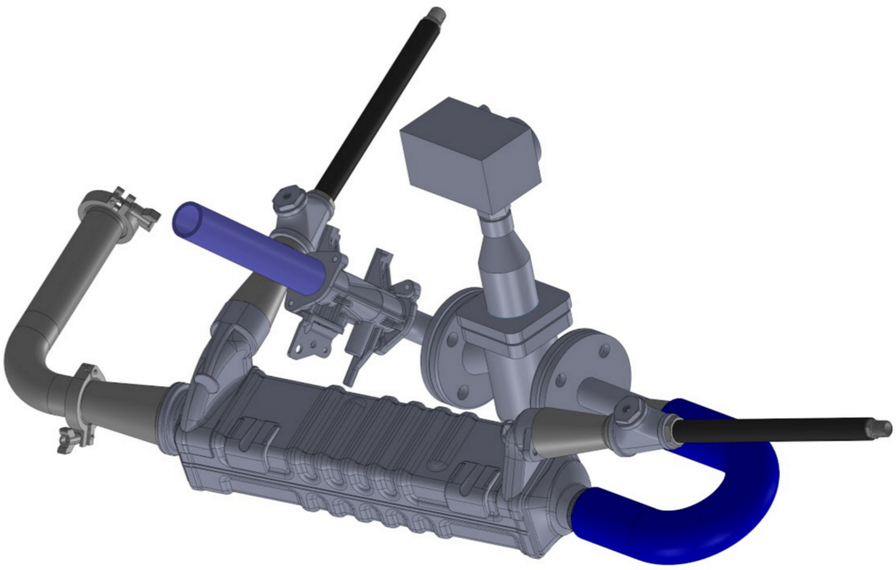


Figure 3: SolidWorks model of the entire EGR system. The exhaust enters from the exhaust plenum at the top left right before a 90° turn leads the gas through the cooler. Attached to the cooler are the coolant lines witch supplies the coolant from an external chiller. After the cooler the gas travels through a 180° silicone hose and the control valve before making another 90° turn that takes the gas through venturi and over the cooler. The outlet of the EGR is the blue hose after the venturi.

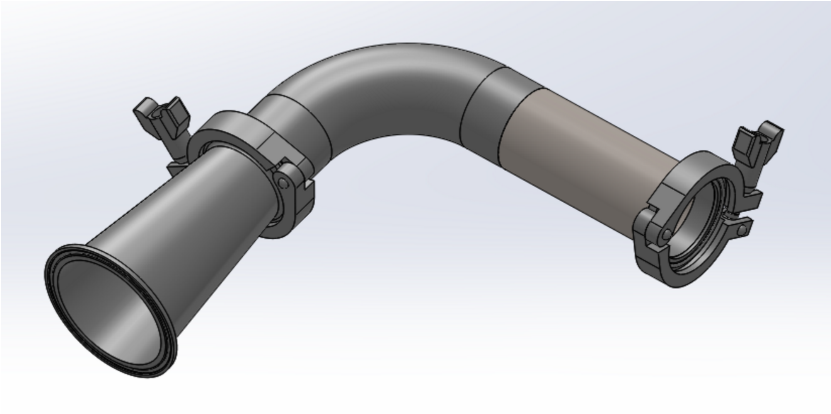


Figure 4: SolidWorks model of parts connecting exhaust plenum to EGR cooler. The opening on the right attaches to the exhaust plenum with a clamp while the opening on the left is to welded to the EGR cooler.

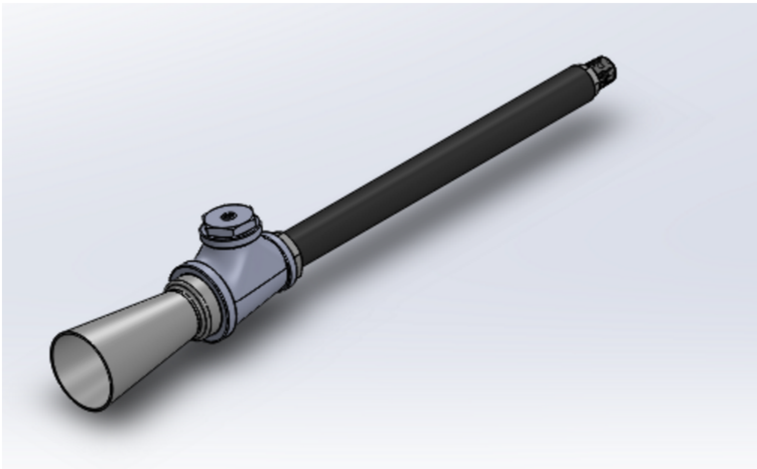


Figure 5: SolidWorks model of the coolant lines, connecting to the EGR cooler (bottom left) and the external chiller of the test cell (top right). Its to be noted that the black, thin, pipe represents a flexible, and much longer, coolant hose.

The two coolant lines, which are almost identical, use flexible hosing to connect the wall mounted chiller to the EGR cooler. Each coolant line has a T-shaped pipe fitting which will accommodate thermocouples in order to monitor the temperature of the coolant. The fitting is connected to a reducer that will be welded onto the EGR cooler. A silicone coolant hose makes up the majority of the coolant lines, as they are 12 ft long in order to reach the chiller on the opposite side of the test cell. Figure 5 shows a SolidWorks model of the coolant lines.

The post-cooler section was the most complicated section of the system to design. This is because it was required to reorient flow, maintain pressure, measure pressure, and measure temperature. Due to limited space in the test cell the team used a 180° silicone elbow to reorient the system to allow the heavy EGR valve to rest on the same stand as the EGR cooler. Welded components including a stainless steel reducer, reduces the diameter to 1.5 in, making it compatible with the inlet diameter of the EGR valve. ANSI Class 150 flanges connect to the EGR valve.

A stainless steel elbow welded to straight tube sections and a water-jet cut flat plate was utilized to connect the EGR valve to the Renault venturi. Another precisely cut flat plate is welded to a 2.25 in stainless steel tube to allow flexible tubing to take exhaust flow from the outlet of the venturi to the intake manifold of the engine.

8 Detailed Design

The detailed design of the EGR Cooler system includes a manufacturing process plan, a numerical analysis, material and material selection process, component and component selection process, a plan for the test procedure and economic analyses.

The manufacturing process plan describes how the system was assembled. The analysis portion consists of calculations used for analyzing the effectiveness and function of different parts of the system. The material and material selection process describes the materials used for the different parts of the system and how those materials were selected. The component and component selection process describes which components are included in the system and how they were selected. The test procedure lays out a plan for how the system will be tested before being delivered to University of Michigan. Lastly, the economic analysis includes a Bill of Materials and information on identified problems regarding the budget.

8.1 Manufacturing Process Plan

The manufacturing process plan, shown in Table 9, was developed to connect the entire system to the EGR cooler. As previously, three major sections connect to the EGR cooler. The pre-cooler (exhaust plenum to cooler), coolant lines (chiller to/from EGR cooler), and post-cooler (cooler to intake manifold).

8.2 Analysis

In order to guarantee safe and functional operation of the EGR system, the preliminary calculations focused on the properties of the exhaust gas going through the EGR cooler and the venturi. The calculations centered around the EGR cooler are necessary to ensure that the specified flow of coolant (4 gal/min) is enough to cool the gas sufficiently for different engine loads. The analysis of the gas flow through the venturi determined the expected pressure difference, Δp. These requirements helped to provide reliable data to determine if the reading of the pressure difference for flows of different magnitudes (depending on the EGR-rate) and temperatures (depending on cooler performance and engine exhaust temperature) would be adequate for the pressure sensor to accurately read.

Table 9: Manufacturing Process Plan describing the operations necessary to assemble the EGR system.

Assembly type	**Material Type**	**Raw Stock Size**	**Operations**
Exhaust Plenum to Cooler			
2 in Straight Tube	304 Stainless	1 ft	Cut from 1 ft to 7 in
Ferrule and 2 in Straight Tube	304 Stainless		Butt weld to Straight Tube
High-Polish Tube Elbow	304 Stainless		Butt weld Straight Tube to non-clamp side of elbow
Chiller to and from EGR cooler			
2.5 in - 1.5 in Reducer and 1.5 in Fitting	304 Stainless		Butt weld l.5 in ends together
EGR Cooler to Intake Manifold			
2.25 in Straight Tube	304 Stainless	1 ft	Cut two 3 in sections
1.5 in Straight Tube	304 Stainless	1 ft	Cut three 4 in sections
2.5 in - 1.5 in Reducer	304 Stainless	2.5/1.5 x 4 in	Cut larger diameter down to 2.25 in
2.25 in Straight to Reducer to 1.5 in Straight	304 Stainless		Weld straights to their respective side of the reducer
1.5 in Straight to 90° Elbow to 1.5 in Straight	304 Stainless		Weld straights to each end of the elbow
Venturi Connection Plates	304 Stainless	12 in x 12 in	Cut outline of inlet and outlet sides of venturi from steel plate using water jet
1.5 in Straight to 90° Elbow to 1.5 in Straight to Venturi Connection Plate (inlet)	304 Stainless		Weld straight to connection plate
2.25 in Straight to Venturi Connection Plate (outlet)	304 Stainless		Weld straight to connection plate

8.2.1 Exhaust Flow Calculation

In order to approximate the exhaust gas flow from the engine a simple calculation originating from the engine power output was utilized.

For example, the engine operates at max capacity, i.e. generating one sixth of 425 hp (approximately 70.8 hp or 53 kW) in power, P. The average mass flow of diesel is determined through the chemical energy density of light diesel[3] $E_{\text{diesel}}^{\text{chemical}} = 43.2\,\text{kJ}\,\text{kg}^{-1}$ (Heywood, 1988) and the assumed engine efficiency[4] of $\eta = 0.4$ as

$$\dot{m}_{\text{diesel}} = \frac{P}{\eta \cdot E_{\text{diesel}}^{\text{chemical}}} = \frac{53 \cdot 10^3}{0.4 \cdot 43.2 \cdot 10^6} = 0.0031 \text{ kg/s}. \tag{2}$$

The corresponding air flow is found as a function of the λ-value defined as

$$\lambda = \frac{\text{AFR}}{\text{AFR}_{\text{sto}}} \tag{3}$$

i.e. the ratio of the air-to-fuel-ratio, AFR, and the stoichiometric air-to-fuel-ratio, AFR_{sto}. The stoichiometric AFR is the ratio between the required air mass and diesel mass required for precisely complete combustion. Since regular diesel fuel is primarily composed of a mixture of hydrocarbon chains with lengths varying between 9 and 23 carbon atoms the value of AFR_{sto} can only be provided as an average. For the blend of hydrocarbons that is diesel $\text{AFR}_{\text{sto}} \approx 14.5$ (Jääskeläinen and Khair, 2016). Consequently, since $\lambda \in [1, 5]$ during normal engine operation, considering $\lambda = 2$ implies

$$\text{AFR} = \lambda \cdot \text{AFR}_{\text{sto}} = 2 \cdot 14.5 = 29.0 \tag{4}$$

and a resulting mass flow of air at $\dot{m}_{\text{air}} = \text{AFR} \cdot \dot{m}_{\text{diesel}} = 29 \cdot 0.0031 = 0.0899\,\text{kg}\,\text{s}^{-1}$. This in turn makes the total mass flow through the cylinder as

$$\dot{m}_{\text{air + diesel}} = \dot{m}_{\text{air}} + \dot{m}_{\text{diesel}} = 0.0031 + 0.0899 = 0.0930 \text{ kg/s}. \tag{5}$$

A portion of which (depending on the EGR-rate, usually between 0 to 40%) will be recirculated through the EGR cooler and venturi back into the intake manifold.

By changing the assumed output power of the engine, P, and the λ-value in calculations the corresponding mass flow for different operation states of the engine can be simulated. The output power depends on the rotational speed of the piston and the fuel injection while the λ-value is a function of the intake charge pressure. A higher charge pressure will correspond to a higher AFR and more lean combustion.

[3] That is the chemical energy density for diesel without the energy that can potentially be acquired through flue-gas condensation i.e. the so called "lower heating value".

[4] The assumed efficiency is representative for diesel engines of this size i.e. approximately 45% (EPA, 2004).

To determine the corresponding volumetric flow somewhere in the EGR system, this value is only a division with the density ρ away from the corresponding mass flow. However, the density depends on both the temperature T (in $^\circ$K) and the pressure P as

$$\rho(T, p) = \frac{p}{R_{\text{spec}} \cdot T} \tag{6}$$

were R_{spec} is the specific gas constant for air at $285\,\mathrm{J\,kg^{-1}\,K}$. This approximation, that the properties of diesel exhaust is the same as those of air, is sufficient in this case, since the errors involved in such an assumption is less than 2%. (Jääskeläinen, 2011)

8.2.2 EGR Cooler Analysis

The function of the analysis of the EGR cooler was to investigate if the cooler will be able to cool the exhaust gas enough. It is supposed to be able to cool the exhaust gas from 600 °C to approximately 80 °C. The cooling capacity of the cooler depends on the design of the cooler, which can not be altered, and the mass flow of the coolant, which could be altered if necessary by building a new loop for coolant in the test cell instead of using the existing one with a flow of 4 gal/min. Another factor that can affect if the exhaust gas will be cooled enough is the cooling capacity of the chiller. The coolant flows through the EGR cooler collecting heat and then it flows through the chiller, where the heat should be released. If the coolant holds to much heat it will not be able to release it all in the chiller, meaning that the temperature of the coolant before the EGR cooler will rise, and the exhaust gas will not be cooled enough.

The first part of the analysis was to determine if the coolant flow is large enough to cool the exhaust gas.

The following calculations are based on operational data from an EGR cooler connected to another engine. The other engine is a single cylinder 13 L engine which means that the values must be scaled down to match the values of the single cylinder 11 L engine used in this project. The data was provided by Magnus Christensen.

The data was gathered under four different speed and load stages with two EGR levels for each stage. The test data is based on VGT-EGR engine, i.e. with high pressure (short route) EGR. The speed/loads provided[5] were:

- 1200 rpm/25% load (cruise speed, low load)
- 1200 rpm/50% load (cruise speed, medium load)
- 1200 rpm/100% load (cruise speed, high load)
- 1800 rpm/100% load (full power, corresponds to 500hp D13 six-cylinder)

[5] See Appendix C for a more detailed overview of the operational data.

The data was used to calculate the product of the heat transfer area, A, and the overall heat transfer coefficient, U. The product UA was scaled down to work for the MD11 engine. This was used to calculate the efficiency of the EGR cooler used in this project and to calculate the temperature of the exhaust gas on the cool side of the cooler, with different engine speeds and loads.

The following equations and data that were used for the calculations were found in *Principles of heat and mass transfer* by Incropera, Dewitt, Bergman and Lavine, published 2013. The method of calculating the product UA and the temperatures was the effectiveness-NTU method. (Incropera *et al.*, 2013)

The first step in calculating the product UA of the MD13 is to calculate the heat capacity rate of the exhaust gas and of the coolant.

$$C = c_p \cdot \dot{m} \tag{7}$$

c_p is the specific heat capacity for air or water, respectively. (Incropera *et al.*, 2013, Table A.4) The properties of the exhaust gas were approximated to be the same as for air. The specific heat capacity of the coolant, which in the MD13 engine is water, was assumed to be $4200\,\mathrm{J\,kg^{-1}\,K^{-1}}$.

The mass flow of the coolant in the MD13 was calculated using the measured volume flow and the density of the coolant, using the mean temperature through the EGR cooler.

To calculate the mass flow of the exhaust gas through the EGR loop the following equation was used.

$$\mathrm{RATE}_{EGR}[\%] = 100 \cdot \frac{\dot{m}_{EGR}}{\dot{m}_{air} + \dot{m}_{EGR}} \tag{8}$$

$$\Leftrightarrow \dot{m}_{EGR} = \frac{\dot{m}_{air} \cdot \mathrm{RATE}_{EGR}/100}{1 - \mathrm{RATE}_{EGR}/100} \tag{9}$$

When the mass flow and the specific heat capacity had been calculated for both the coolant flow and the exhaust gas flow the heat capacity rates could be calculated. The variables and values for the exhaust gas have the index 'h' because it is the hot fluid and the variables and values for the coolant have the index c because it is the cold fluid.

$$C_h = \dot{m}_{EGR} \cdot c_{p,h} \tag{10}$$

$$C_c = \dot{m}_{coolant} \cdot c_{p,c} \tag{11}$$

The heat capacity rates are used to calculate the transferred heat between the coolant and the exhaust gas.

$$q_h = C_h \cdot (T_{h,i} - T_{h,o}) \tag{12}$$

$$q_c = C_c \cdot (T_{c,o} - T_{c,i}) \tag{13}$$

The heat transfer rate of the cold side was in all speed/load stages much higher than the heat transfer rate of the hot side. This may be because the coolant also exchanges heat with the air outside the cooler and because it is much more sensitive to the measurement of temperature, since the temperature difference between the coolant in and out is small. Because of this the heat transfer rate was assumed to be equal to the calculated heat transfer rate of the hot side, the exhaust gas.

$$q = q_h \tag{14}$$

The next step in the effectiveness-NTU method was to calculate the maximum possible heat transfer rate, the fraction of the maximum and minimum heat capacity rate, and the effectiveness of the cooler, which is the ratio of the actual heat transfer rate and the maximum possible heat transfer rate.

$$q_{max} = C_{min} \cdot (T_{h,i} - T_{c,i}) \tag{15}$$

$$C_r = \frac{C_{min}}{C_{max}} \tag{16}$$

$$\varepsilon = \frac{q}{q_{max}} \tag{17}$$

The effectiveness, ε, and the ratio of the heat capacity rates can be used to calculate the NTU (the Number of Transfer Units) which is defined as

$$\mathrm{NTU} \equiv \frac{UA}{C_{min}}. \tag{18}$$

The NTU for a counter flow cooler, which is what is installed in the MD13 engine, could be calculated through the following equation. (Incropera *et al.*, 2013, Eq. (11.29b))

$$\mathrm{NTU} = \frac{1}{C_r - 1} \cdot \ln\left(\frac{\varepsilon - 1}{\varepsilon C_r - 1}\right) \qquad (C_r < 1). \tag{19}$$

When both NTU and C_{min} were known the product UA could be calculated as

$$UA = \mathrm{NTU} \cdot C_{min}. \tag{20}$$

The UA acquired for the different speed/load stages are presented in Table 10. These values were scaled down and used for the calculations on the EGR cooler used in this project, for the MD11 engine, as can be seen in Table 11.

The first step in doing the calculations for the smaller MD11 engine was to scale down the mass air flow. It was done by using the ratio of stroke volumes for the MD11 and the MD13 engines. The stroke volume was calculated using the dimensions of the stroke and bore of the engine.

$$V_{stroke} = \frac{\pi}{4} \cdot \mathrm{bore}^2 \cdot \mathrm{stroke} \cdot \mathrm{number\ of\ cylinders} \tag{21}$$

Table 10: The product UA.

Speed/Load stage	EGR rate	UA [$\mathrm{W\,m^{-1}\,K^{-1}}$]
A25	24.4%	251.04
A25	32.5%	324.84
A50	23.1%	70.28
A50	29.4%	62.72
A100	18.8%	69.87
A100	23.6%	91.19
C100	19.3%	87.89
C100	25.9%	105.78

Both engines are single cylinder. The MD11 has a bore of 123 mm and a stroke of 152 mm while the MD13 has a bore of 131 mm and a stroke of 158 mm.

$$V_{stroke,MD11} = \frac{\pi}{4} \cdot 0.123^2 \cdot 0.152 \cdot 1 = 1.81l \tag{22}$$

$$V_{stroke,MD13} = \frac{\pi}{4} \cdot 0.131^2 \cdot 0.158 \cdot 1 = 2.13l \tag{23}$$

The ratio of these volumes multiplied with the mass air flow through the MD13 provided the mass air flow through the MD11. This is because the ratio of the stroke volumes is equal to the ratio of the mass air flows.

$$\dot{m}_{air,MD11} = \dot{m}_{air,MD13} \cdot \frac{V_{stroke,MD11}}{V_{stroke,MD13}} \tag{24}$$

Using Eq. (9) the mass flow of the exhaust gas through the EGR loop was calculated. The EGR rates used for the calculations on the MD11 are the same as those provided for the MD13.

The UA was also scaled down to apply to the MD11 engine. This was done through by finding proportionalities for UA.

$$\frac{1}{UA} = \frac{1}{h_i A_i} + \frac{\ln(D_o/D_i)}{2\pi k L} + \frac{1}{h_o A_o} \tag{25}$$

The conduction coefficient is probably high, since this is a heat exchanger and therefore designed to maximize the heat transfer rate. The convection coefficient on the outside of the pipes, h_o, is also high since the fluid on that side is water. Considering that those coefficients are much larger than the convection coefficient on the gas side means that the equation (25) could be approximated to be

$$\frac{1}{UA} \approx \frac{1}{h_i A_i} \tag{26}$$

Assuming that the inner area is equal to the heat transfer area, A, means that the equation could be simplified further

$$U \approx h_i \tag{27}$$

The convection coefficient is proportional to the Nusselt number (Nu_D), which is the ratio of convective to conductive heat transfer across the boundary, assuming that the diameter of the boundary and the conduction coefficient are constants

$$h = \frac{Nu_D \cdot D}{k}. \tag{28}$$

For turbulent flow in circular tubes the correlation for the Nusselt number is obtained from the *Dittus-Boelter equation*. (Incropera *et al.*, 2013, Eq. (8.60))

$$Nu_D = 0.023 \cdot Re_D^{4/5} \cdot Pr^n \tag{29}$$

Assuming that the Prandtl number (Pr) was constant, the change of the convection coefficient is proportional to the change of $Re_D^{4/5}$. If the properties of the fluid are unchanged the Reynolds number (Re_D) is proportional to the velocity of the fluid. The area is constant and assuming that the density also is means that the Reynolds number is proportional to the mass flow,

$$Re_D^{4/5} \propto v^{4/5} \propto \dot{v}^{4/5} \propto \dot{m}^{4/5}. \tag{30}$$

Connecting this back to UA means that

$$UA \propto h \propto Nu_D \propto Re_D^{4/5} \propto \left(\frac{v_{EGR,MD11}}{v_{EGR,MD13}}\right)^{4/5} \propto \left(\frac{\dot{m}_{EGR,MD11}}{\dot{m}_{EGR,MD13}}\right)^{4/5}. \tag{31}$$

This means that the product UA of the MD11 could be estimated to be

$$UA_{MD11} = UA_{MD13} \cdot \left(\frac{\dot{m}_{EGR,MD11}}{\dot{m}_{EGR,MD13}}\right)^{4/5}. \tag{32}$$

Table 11: The product UA, scaled down for 11 L engine.

Speed/Load stage	EGR rate	UA [$\mathrm{W\,m^{-1}\,K^{-1}}$]
A25	24.4%	220.06
A25	32.5%	284.76
A50	23.1%	61.60
A50	29.4%	54.98
A100	18.8%	61.25
A100	23.6%	79.93
C100	19.3%	77.04
C100	25.9%	92.72

The scaled down UA, presented in Table 11 was used to calculate the NTU using Eq. (18).

Using the calculated mass flow and the given coolant flow, which is 4 gal/min, and the intake temperatures for the exhaust gas flow and coolant flow provided by University of Michigan, which are 600 °C and 5-90 °C respectively, the heat capacity rates could be calculated for the hot and cold flows as well as the density of the coolant. The heat capacity rates were calculated using equation (9) and (10), with the specific heat capacity assumed to be $4200\,\mathrm{J\,kg^{-1}\,K^{-1}}$ for the coolant flow and for the hot flow it was estimated, using a mean temperature of 340 °C since the desired outlet temperature of the exhaust gas is 80 °C, to be $1054\,\mathrm{J\,kg^{-1}\,K^{-1}}$. The outlet and inlet temperatures of the exhaust gas through the EGR cooler were assumed to be the same for all load and speed stages. The density of the coolant was used to calculate the coolant mass flow.

The ratio of the maximum and minimum heat capacity rates was calculated using equation (15) and the maximum possible heat transfer rate was calculated using equation (14). When the NTU, C_r and q_{max} were known these three values were used to calculate the effectiveness. (Incropera *et al.*, 2013, Eq. (11.28a))

$$\varepsilon = \frac{1-\exp[-\mathrm{NTU}(1+C_r)]}{1+C_r} \tag{33}$$

The resulting effectiveness is presented in Table 12.

The values for the effectiveness in Table 12 are much higher than what is usual for parallel flow coolers. This is possibly because sources of errors in the operational data from the MD13 engine. The errors occur in the measurement of inlet and outlet temperatures of the EGR cooler. The temperature is measured a few centimeters upstream of the inlet and downstream of the outlet which means that the temperatures used for calculations on the EGR cooler are incorrect, resulting in a higher effectiveness. The following calculations will be made with the values in Table 12 and thereafter the calculations will be made with an assumed lower effectiveness for comparison.

Table 12: The effectiveness of the EGR cooler for different speed/load stages.

Speed/Load stage	EGR rate	Effectiveness (ε)
A25	24.4%	99.4%
A25	32.5%	99.2%
A50	23.1%	99.1%
A50	29.4%	98.4%
A100	18.8%	98.5%
A100	23.6%	98.4%
C100	19.3%	97.2%
C100	25.9%	96.3%

The effectiveness multiplied with the maximum possible heat transfer rate gives the actual heat transfer rate, which was used to calculate the outlet temperature of the exhaust gas and the coolant.

$$q = \varepsilon \cdot q_{max} \tag{34}$$

$$q = C_h \cdot (T_{h,i} - T_{h,o}) \tag{35}$$

$$\Leftrightarrow T_{h,o} = T_{h,i} - \frac{q}{C_h} \tag{36}$$

$$q = C_c \cdot (T_{c,o} - T_{c,i}) \tag{37}$$

$$\Leftrightarrow T_{c,o} = T_{c,i} + \frac{q}{C_c} \tag{38}$$

The outlet temperature of the exhaust gas and the coolant depends on the inlet temperature of the coolant. The temperature can be controlled and lies within the range of 5-90 °C with an absolute maximum at 92 °C, which is the highest temperature the chiller can handle. For the outlet temperature of the exhaust gas to reach approximately 80 °C the inlet temperature of the coolant should be approximately 60-70 °C according to the results of the calculations. The resulting heat transfer rates and outlet temperatures are presented in Table 13.

The results in Table 13 show that the coolant flow of 4 gal/min is enough to cool the exhaust gas down to approximately 80 °C, assuming that the sources of error does not affect the effectiveness of the EGR cooler too much.

The next step in analyzing whether the exhaust gas can be cooled enough is to compare the heat collected by the coolant with the chiller's cooling capacity. The temperature of the coolant will be in the range of approximately 71-73 °C. The cooling capacity of the `ThermoFlex 5000` will then be just under 6000 W (see Appendix D). (Thermo Fisher Scientific, 2016) Comparing this to the heat transfer rates in Table 13 shows that the chiller can manage the seven first speed/load stages but that the three last stages have too high heat transfer rates. The sponsors at University of Michigan has stated that the motor will mostly be used in the lower stages, so if they only stay on the higher stages for short periods of time the exhaust gas should still reach the desired low temperatures.

Table 13: The resulting temperatures and heat transfer rates for the different speed/load stages.

Stage	EGR-rate	Inlet EG-temp. [°C]	Inlet coolant temp. [°C]	Heat transfer rate [W]	Outlet EG-temp. [°C]	Outlet coolant temp. [°C]
A25	24.4%	600	70	3356	73.2	73.2
A25	32.5%	600	70	4324	74.2	74.2
A50	23.1%	600	70	4537	74.8	74.4
A50	29.4%	600	70	5590	78.4	75.4
A100	18.8%	600	70	5909	77.9	75.7
A100	23.6%	600	70	7526	78.6	77.0
C100	19.3%	600	65	9174	80.2	73.9
C100	25.9%	600	60	11940	80.3	71.5

The conclusions drawn from the results above are that the exhaust gas can be cooled down to approximately 80 °C with the 4 gal/min coolant flow at a temperature between 60 and 70 °C and that the chiller's cooling capacity will be enough if the engine does not stay on the three highest speed/load stages for long.

These results are achieved through assuming that the values for the effectiveness presented in Table 13 are correct. To verify that the existing chiller and coolant flow would be enough even if the effectiveness was lower additional calculations were made. Assuming that the effectiveness of the cooler in the MD13 engine at the lowest speed/load stage is 90% and that the effectiveness of the other stages have the same relations to each other means that a new effectiveness can be found for all stages of the MD13 engine and then scaled down, as was done before, to provide new values for the effectiveness of the cooler in the MD11 engine.

Table 14: The effectiveness of the EGR cooler, when the effectiveness of the cooler in the MD13 engine is assumed to be 90% in the lowest speed/load stage.

Speed/Load stage	EGR rate	New effectiveness (ε), MD13 system [%]	New effectiveness (ε), MD11 system [%]
A25	24.4%	90.0	89.6
A25	32.5%	90.0	89.5
A50	23.1%	89.9	89.5
A50	29.4%	89.4	88.9
A100	18.8%	89.6	89.2
A100	23.6%	89.7	89.3
C100	19.3%	88.7	88.2
C100	25.9%	88.2	87.4

Table 15: The resulting temperatures and heat transfer rates for the different speed/load stages.

Stage	**EGR-rate**	**Inlet EG-temp.** [°C]	**Inlet coolant temp.** [°C]	**Heat transfer rate** [W]	**Outlet EG-temp.** [°C]	**Outlet coolant temp.** [°C]
A25	24.4%	600	20	3310	80.5	23.2
A25	32.5%	600	20	4269	80.9	24.1
A50	23.1%	600	20	4484	80.9	24.3
A50	29.4%	600	20	5528	84.2	25.3
A100	18.8%	600	20	5856	82.7	25.7
A100	23.6%	600	20	7206	82.2	27.0
C100	19.3%	600	10	9179	79.9	18.9
C100	25.9%	600	10	11852	84.1	21.5

This results in new outlet temperatures and in new heat transfer rates. With an effectiveness just below 90%, a coolant temperature of 60-70 °C can not sufficiently cool the exhaust gas. By lowering the coolant temperature to 10-20 °C the desired gas temperature of approximately 80 °C can be reached. This proves that the coolant flow of 4 gal/min is sufficient for cooling the exhaust gas even at a lower effectiveness. These results can be seen in Table 15.

8.2.3 Venturi Options Detailed Analysis

The EGR system will use one of two venturis to measure the EGR flow, either the original Volvo MD11 venturi which was a part of the stock engine and designed for full, six-cylinder exhaust gas flows or a smaller Renault venturi designed for 5 L engines. By temporarily obstructing the flow with a decrease in diameter a pressure difference is created by this component, which can be utilized to calculate the gas volume flow.

Using the larger Volvo venturi would be beneficial because the original Δp-sensor is already in place and the component is already present at the University of Michigan test cell. However, since the Volvo venturi is designed for six times the flow expected from a single cylinder it is not clear whether this component would generate pressure differences large enough to be detectable with the Δp-sensor. The smaller Renault venturi would perform better in the sense that it will generate larger pressure differences but the component might not be necessary if a sufficient measurement signal could be obtained with the Volvo venturi and associated Δp-sensor. An overview of the small and large diameter, D_1 and D_2, at the points where the Δp-measurement is conducted is presented in Table 16 for the two venturi systems.

Table 16: Throat diameters for the two different ventury systems.

Dimension/Make	**Volvo**	**Renault**
D_1	27.0 mm	15.0 mm
D_2	65.3 mm	36.3 mm

Assuming the situation when the engine operates at 75% of its peak power output, $P = 0.75 \cdot 53 \approx 40\,\mathrm{kW}$, no intake charge and a pressure gradient of 0.3 bar between the exhaust plenum and the intake manifold the procedure from Section 8.2.1 can be followed to determine the mass flow through the cylinder. In this case the fuel flow, using $\eta = 40\%$, is easily found as $0.0023\,\mathrm{kg\,s^{-1}}$ and without intake charge the cylinder volume fills with 1.83 L air at $1.24\,\mathrm{kg\,m^{-3}}$ or 0.0027 kg per every second cylinder retraction. If the engine is operating at 1800 RPM this translates to a mass air flow of $0.0405\,\mathrm{kg\,s^{-1}}$ and an AFR-value of $0.0405/0.0023 = 17.61$. Consequently it is found that

$$\lambda = \frac{\mathrm{AFR}}{\mathrm{AFR_{sto}}} = \frac{17.61}{15.5} = 1.21 \tag{39}$$

which means lean combustion (Heywood, 1988) (since $\lambda > 1.00$), and the total mass flow is $0.0023 + 0.0405 = 0.0428\,\mathrm{kg\,s^{-1}}$. Now the volumetric flow through the venturi in the EGR circuit depends on temperature, EGR-rate and pressure. If the pressure in the exhaust plenum is stationary at 0.3 bar above atmospheric the resulting Δp can be plotted as a function of the two remaining variables, temperature and EGR-rate (as a portion of the total exhaust mass flow). The Δp calculation in a venturi of given dimension is a simple application of Bernoulli's principle[6] for incompressible flow (White, 2011)

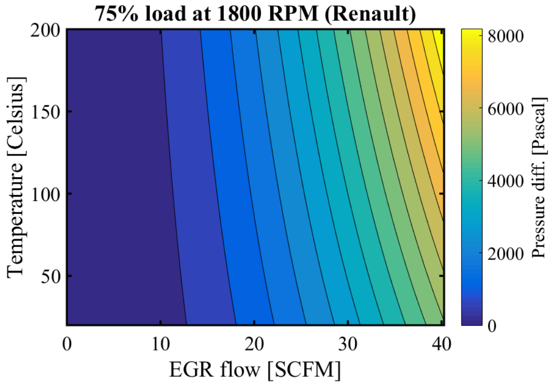
$$\Delta p = \frac{\rho(T, p)}{2}\left(v_1^2 - v_2^2\right) \tag{40}$$

were v_1 and v_2 are the gas flow speeds through the small and large throat of the venturi (found through the flow rate and throat dimensions). The resulting Δp for the smaller Renault venturi for flows equivalent to between 0 and 50% EGR (0 to roughly 40 SCFM[7]) and temperatures between 20 °C and 200 °C is presented in Figure 6(a). A comparison between the expected Δp values, at 100 °C, for the smaller Renault and larger Volvo venturi is found in Figure 6(b).

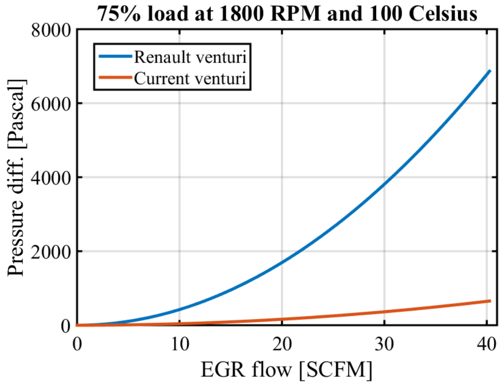
From Figure 6(b) it is clear that the smaller Renault venturi, as expected, offers much higher pressure differences across all volume flows. These results must in turn be analysed in contrast with the sensitivity characteristics of the Δp-sensor. In order to (a) possibly dismiss the usage of the Volvo venturi and (b) determine within which engine operational conditions the Δp-reading, from the chosen venturi, will be reliable.

[6] In the flow situations where Bernoulli's principle apply (for instance in the analyzed venturi) it states that the value of $gz + \frac{v^2}{2} + \frac{p}{\rho}$, where g is the standard acceleration due to gravity, is constant for the gas.

[7] The Standard CFM unit (SCFM) is defined from the standard temperature 16 °C (40 °F) and standard pressure of 1 atmosphere. Consequently 1 SCFM corresponds to a mass flow of $34.7\,\mathrm{g\,min^{-1}}$ by this definition.



(a) Venturi Δp over T and flow.



(b) Δp for the two venturi systems.

Figure 6: The expected pressure differences in the venturi systems which will be measured by the Δp-sensor. Figure (a) features a mapping of the expected Δp values across temperatures in the range from 20-200 °C and different flows for the Renault venturi while Figure (b) is a comparison of the expected Δp from the Renault and Volvo venturi at 100 °C for different flows.

Since the pressure sensor has a linear sensitivity curve, emitting voltages of between 0.5 to 4.5 V for pressure differences in the range of 0 to 35 kPa it is clear that the smaller venturi is much more suitable for use in the circuit. For instance, the larger Volvo venturi would result in sensor voltage change of only 26 mV when the EGR-rate changes from 0% to 50% with the results presented in Figure 6(b). The smaller Renault venturi would on the other hand provide much more easily detected voltage changes of 256 mV for the same change in flow. This, combined with the fact that the gas flows from operating a single cylinder will not come close to the maximum 35 kPa rating of the sensor, means that smaller Renault venturi is the best option out of the two.

However, as indicated by the large blue region for small and cool flows in Figure 6(a) the pressure changes slowly, even for the better performing Renault venturi. In this region the corresponding change in output voltage from the Δp-sensor also changes slowly, which in turn means a lower accuracy for the flow measurement and a lower signal-to-noise ratio.

8.3 Material and Material Selection Process

Different sections of the EGR system required different materials based on their operating constraints. The section of tubing going from the exhaust plenum to the EGR cooler needed to withstand temperatures of up to 600 °C, be resistant to corrosion, and match the material of the plenum and the EGR cooler. This led to the selection of 304 stainless steel aluminum for all components between the exhaust plenum and the inlet of the EGR cooler.

The section of tubing after the EGR cooler did not have to withstand temperatures as high as the pre-cooler tubing. It needed to be able to tolerate temperatures up to 200 °C while remaining flexible enough to be bent around to connect the EGR cooler to the engine intake manifold. The tubing also had be resistant to corrosion. Based on these constraints, flexible tubing made of silicone was selected. The metal parts of the post-cooler tubing network, such as the EGR valve, are made of steel.

The tube fittings used to connect tubes to the chiller needed to be the same material or close to each other on the Anodic Index. The chiller outlet connection is stainless steel and the chiller inlet connection is bronze. The chiller outlet fitting selected is stainless steel and the chiller inlet fitting selected is brass. The voltage difference between brass and bronze is minimal which allows the use of the brass on bronze connection. The only constraint for the coolant lines is that everything fits together and can reach the chiller. The coolant hoses used in the system are 3/4 in inner diameter and should be 12 ft long to reach from the EGR cooler to the chiller.

8.4 Component and Component Selection Process

The major components that make up the EGR system had to go through a selection process to be adequately selected. The intermediate piping and arrangement of the cooler, valve, and venturi are all key components and subject to design. The component arrangement (for example the placement of the EGR valve before versus after the cooler) vastly influenced the properties of the intermediate piping. The selection process of one component was never independent from the system as a whole.

8.4.1 EGR Valve Selection Process

Initially, the choice of the EGR valve stood between the current, stock, EGR valve mounted on the exhaust manifold of the engine block or an alternative valve which could be purchased. The benefit of using the stock valve for the new EGR system was that a valve control system was already in place. However, the control system was designed for regulating flow from operating all six cylinders and the valve is specifically made to be attached directly to the engine. In the new EGR circuit, which draws exhaust from the exhaust plenum, the complicated geometry of the stock valve made it difficult to implement. Consequently, the decision to find an alternative valve was made at an early stage.

Since it is preferable to place the valve upstream of the cooler to minimize the risk of condensation the sought after alternative valve had to be able to handle gas temperatures of 600 °C. An additional requirement from the University of Michigan was that the valve should be electrically actuated.

Electrically actuated control valves (designed for industrial applications) with a high enough temperature tolerance for use in the hot side of the EGR circuit were found to be too expensive. Consequently, production EGR valves meant for conventional 2 L diesel cars were identified as the next best option.

EGR valves meant for conventional 2 L diesel cars are dimensioned for an exhaust flow equivalent to that of an engine with similar cylinder displacement, are much cheaper and built to withstand high temperatures. Unfortunately, the problem with complicated geometry resurfaces and the electrical control system could possibly introduce further issues since these valves are meant to communicate with the ECU (Engine Control Unit) of a production car.

In accordance with recommendations from University of Michigan and sponsors the decision was made to move the EGR valve to the cool side of the circuit. The lower temperatures allow usage of a possibly cheaper, simple geometry and highly controllable, control valve.

After reviewing EGR valve options, it was determined that there was no way to purchase an adequate EGR valve while staying within the $1000 budget. The team spoke with Volvo and the University of Michigan and agreed that the University of Michigan would purchase the EGR valve based on recommendations from the team. The team contacted suppliers and identified a potential valve for the system. The system was designed based on a 40 mm outer diameter globe valve. A request from the University of Michigan was that the valve should be a globe valve while an electric actuator was still a requirement. A globe valve combined with an electric actuator is also what is used in the test cell at Volvo Trucks Technology. It was decided that the research should be focused on electrically actuated globe valves specifically.

University of Michigan is responsible for approving the valve that has been selected. If they approve of the valve, they will be responsible for ordering the valve and installing it within the system.

The most important features of the identified valve options that can be used in the finalized EGR circuit is presented in Table 17, with corresponding actuators specified in Table 18.

The first option, which initially was meant for usage on the hot side of the EGR cooler, is the `GL40` globe valve and `PSL202` electric actuator from Tianjin Bell. The system is according to its specifications able to withstand gas temperatures up to 500 °C and the actuator and valve is purchased for $570, and therefore comparatively cheap, bundle. However, since the system is the heaviest option, weighing 24 kg, and the supplier is located in China, the total cost including shipping to University of Michigan is almost twice the cost of the hardware, at $1147.

The second option from Samson, a supplier with a global market presence, was identified after the budget constraints were loosened and the decision to move the control valve downstream of the cooler was final. This system consists of the `Type 3321 DN40` globe valve (with a valve curve presented in Appendix E) and `Type 5824-30` electric actuator which differs from the Bell options since this system is almost half the weight (12.75 kg versus 24 kg). The Samson alternative also includes a more sophisticated, but less powerful, actuator (0.7 kN versus 2 kN). The actuator is more sophisticated in the way that it features feedback voltage signal of 0 to 10 V proportional to the current position of the valve and has the option to control using both AC and DC signals.

Lastly, the high-end valve and actuator system that Magnus Christensen is using in his research setup at Volvo Trucks Technology has been included in Table 17 and 18 for reference and comparison. The valve is also a 40 mm `Series 035000` globe valve manufactured by Kämmer and the high performance electric actuator is a `P30L` from Oden Control which is more precise, powerful and faster than the other options. However, the `P30L` has recently gone out of production and has been replaced by the upgraded successor `V30QL`, which is similar but more precise and with higher temperature tolerances.

Table 17: An overview of the identified 40 mm globe valve options which could be used together with corresponding actuators from Table 18 to control the EGR flow.

Property/Valve	Series 035000	Type 3321 DN40	GL40
Manufacturer	Kämmer	Samson	Bell
Diameter [mm]	40	40	40
Stroke [mm]	20	15	N/K
Width [mm]	200	200	222
Temp. range [°C]	≤ 200	-20 to 220	< 500
Weight [kg]	13.5	12	24
Price	$2819.41*	$1966.48†	$1147‡

* The valve could be purchased including an unspecified Kämmer actuator for $4497.7.
† The price offer from Samson is for a package with the `Type 5824-30` electrical actuator and `Type 3321 DN40` valve.
‡ This price is for the entire valve and actuator system, including shipping costs of ≈ $500.

Table 18: The electrical actuators which controls the opening and closing of the globe valves from Table 17.

Property/Actuator	P30L*	V30QL	Type 5824-30	PSL202
Manufacturer	Oden Cont.	Oden Cont.	Samson	Bell
Force [kN]	3 - 12	3 - 12	0.7	2
Max stroke [mm]	40	40	15	25
Max speed [mm/s]	6.67	6.67	0.36	0.5
Precision [mm]	± 0.10	± 0.02	N/K	± 0.25
Weight [kg]	7	5.5†	0.75	24‡
Temp. range [°C]	-20 to 60	-40 to 80	0 to 50	N/K
Power supply	24 V DC	24 V DC	24 V DC 230 V 50 Hz AC	220 V 50 Hz AC
Max power [W]	200	200	N/K	N/K
Avg. power [W]	100	100	8 (AC), 5 (DC)	N/K
Control signal	4 - 20 mA	4 - 20 mA	0 - 20 mA 0 - 10 V	4 - 20 mA 0 - 10 V
Price	—	N/K	$1966.48**	$1147††

* The `P30L` which is used at Volvo Trucks Technology is no longer in production and has been replaced by `V30QL`.
† The weight of the `V30QL` actuator is excluding the linear unit connecting to the valve, including this component will result in a weight of ≈ 7 kg.
‡ This value is for the entire valve and actuator system, including valve, linear unit and actuator.
** The price offer from Samson is for a package with the `Type 5824-30` electrical actuator and `Type 3321 DN40` valve.
†† This price is for the entire valve and actuator system, including shipping costs of ≈ $500.

8.4.2 Venturi Selection Process

Since the start of the project many alternative ways of measuring the exhaust gas flow has been proposed. The EGR-rate can be determined through CO_2 sensors located in the EGR circuit and intake manifold, but the addition of a more precise flow measurement in the EGR system would provide University of Michigan with a much more capable setup.

Alternatives such as a heated element and complete aftermarket MAF sensors were researched, but the high temperatures and possible accumulation of soot shifted focus towards the already available venturi or an equivalent component. The stock venturi with its 27 mm throat diameter and pre-installed Δp-sensor was however designed for exhaust flow from six cylinders. That is, if this component were to be used in the new circuit, a recalibration of the sensor was necessary, provided a large enough signal-to-noise ratio could be acquired at all.

A more flow appropriate alternative was identified as a Renault made venturi designed for 5 L engines and, according to the calculations from Section 8.2.3, this system generates much higher, and therefore more easily detected, pressure differences. These calculations, alongside recommendations from University of Michigan, became the basis for the final decision to use the smaller Renault venturi.

8.4.3 The Selection Process for the Parts of the Intermediate Piping

The pre-cooler section was complicated due to the high temperatures of exhaust before the EGR cooler and the positioning of the connection to the exhaust plenum. The gaskets that will be used in the pre-cooler system will be provided by the University of Michigan. The University of Michigan has an excess supply of gaskets that were used for a previous test that are the same size that is needed for the design. 304 Stainless steel tubing was selected due to the temperature range reaching 600 °C. An elbow is needed in the pre-cooler section due to the odd angle that is mounted to the exhaust plenum.

The selection process for the inlet and outlet coolant lines were almost identical. The inlet and outlet for coolant on the EGR cooler have the same inner and outer diameter, which made it easier to design. The main component of the coolant lines is the tee connection, which connects the hose from the chiller, the reducer from the cooler, and the fitting for a thermocouple. The hose from the chiller will be $^3/_4$ in tubing that the University of Michigan will provide. The hose is connected to the chiller and tee by barbed hose fittings at each end. The reducer had to be of the same material as the EGR cooler in order to weld right onto the cooler. In order to connect the reducer to the tee connection, a butt-weld tube fitting that had an end to weld to and an end to screw into the tee connection was selected. The fitting for the thermocouple needed to have a $^1/_2$ NPT thread for the Swage-lok fitting. A simple adaptor was the only part needed to include thermocouple into the system. The only difference between the inlet and outlet coolant lines is the material of the hose fitting to avoid galvanic corrosion at the chiller.

The post-cooler section was the most complicated selection process due to the number of vital parts that needed to be integrated into the section. The venturi and EGR valve were two major parts of the entire system that determined the details of the rest of the parts for the post-cooler system. Once the flow calculations showed that the smaller Renault venturi was necessary and it was determined that the electrically actuated 1.5 in control valve from Samson could be used, despite being out of the $1000 budget, the rest of the decisions could be made. Silicone tubing can be used to allow flexible and easy connection back to the intake manifold and a 180° silicone tube was chosen as the outgoing tubing from the EGR cooler, to allow for the relatively heavy valve and venturi to rest on the same triangular platform as the EGR cooler does.

8.5 Test Procedure

This procedure included three different types of testing prior to the delivery of the system to the University of Michigan. In order to identify the proper components for the system, the design specifications relied on validating numbers and calculations. Once the calculations validated the use of specific components, the team generated a computer model of the system to test if the system would fit properly. The final test that the system faced was a manual fitting test. The system was tacked together and bolted together. The team manually checked the all angles, dimensions, and connections prior to permanently welding and fitting the system. These three tests ensured that the system will operate both theoretically, and physically as intended.

Once the system is fully assembled at the University of Michigan, it will have to undergo pressure testing to ensure a tight fit, however this testing will not affect the predicted performance of the system and will only result in manufacturing corrections. In addition, all welds will have to be checked to ensure there are no leaks. Additional tests that could be relevant are tests to the load vs. EGR mass air flow to determine whether the wall mounted chiller would be adequate for the estimated loads. This testing however would probably confirm the results of the calculations, showing that the chiller cannot handle extreme operating environments for extended periods of time.

8.6 Economic Analysis - Budget and Vendor Purchase Information

The majority of parts that will be used in the EGR system have been identified and are purchased. The parts that have not been purchased are the EGR valve, flanges, and flexible silicone hose leading to the intake manifold. Due to the price of the EGR valve and flexible silicone hose, the project would exceed its budget of $1000. The flanges are dependent on the valve and will not be purchased because the valve has not been purchased. The Bill of Materials can be found in Appendix B.

9 Construction Process

The construction process was split up into the three governing sections of the project design. The pre-cooler, coolant lines, and post-cooler. All sections start at the EGR cooler. Table 19 lists all of the parts in the construction process. A more detailed list with pictures of each separate part can be found in Appendix F.

Table 19: Overview of EGR system parts for construction.

Part Key	Part Description
1	2 in Elbow & Straight
2	Reducer to Cooler
3	Quick Clamp
4	1 NPT Stainless Steel Hose Fitting
5	$^1/_2$ NPT Stainless Steel Hose Fitting
6	$^1/_2$ NPT Brass Hose Fitting
7	$^1/_8$ NPT Fitting for thermocouple
8	1.5-1-1 NPT Tee Connection
9	1.5 NPT Threaded Reducer
10	$^{13}/_{16}$ in - 1 $^1/_2$ in Hose Clamps
11	180° Silicone Hose
12	2.25 in Straight - Reducer - 1.5 in Straight
13	2 $^1/_{16}$ in - 3 in Hose Clamps
14	EGR Valve
15	1.5 in Straight - Elbow - Straight - Venturi Conn. Plate
16	Renault venturi
17	Venturi Connection Plate - 2.25 in Straight
18	Flexible Silicone Hose

9.1 Pre-Cooler Construction Process

The construction of the pre-cooler part of the system was made simple by the Quick Clamp connections. The procedure consisted of three steps:

1. Weld Reducer (2) to EGR Cooler
2. Use Quick Clamp (3) to clamp Reducer (2) to 2 in Elbow & Straight (1)
3. Use Quick Clamp (3) to clamp 2 in Elbow & Straight (1) to exhaust plenum

9.2 Coolant Lines Construction Process

For the coolant inlet the construction process is summarized in nine steps:

1. Weld 1.5 NPT Threaded Reducer (9) to EGR Cooler
   (a) Make sure that the Inlet Reducer is used
   (b) Line up lines on EGR Cooler and Reducer
2. Screw on Tee Connection (8) to 1.5 NPT Threaded Reducer (9)
3. Screw in $^1/_8$ NPT Fitting (7) to top of Tee Connection (8)
4. Screw in 1 NPT Stainless Steel Hose Fitting (4) to Tee Connection (8)
5. Connect Coolant Hose to Hose Fitting (4)
6. Use Hose Clamp (10) to clamp Coolant Hose to Hose Fitting (4)
7. Screw in $^1/_2$ NPT Stainless Steel Hose Fitting (5) to Chiller
8. Connect Coolant Hose to Hose Fitting (5)
9. Use Hose Clamp (10) to clamp Coolant Hose to Hose Fitting (5).

Meanwhile, the construction process of the coolant outlet adapter differed only slightly from the list above. The process is identical up to the last three steps, where the details are changed:

7. Screw in $^1/_2$ NPT Brass Hose Fitting (6) to Chiller
8. Connect Coolant Hose to Hose Fitting (6)
9. Use Hose Clamp (10) to clamp Coolant Hose to Hose Fitting (6)

9.3 Post-Cooler Construction Process

Since the EGR control valve will be purchased, and thereafter mounted in the system, by University of Michigan the construction of the post-cooler part of the system was not finished before the student involvement in the project came to an end. However, all component except the valve itself and corresponding flanges have been acquired and the proposed construction process can be described in a fashion very similar to already completed parts of the system.

1. Connect 180° Silicone Hose (11) to Exhaust Outlet of EGR Cooler
2. Use Hose Clamp (13) to clamp Silicone Hose (11) to EGR Cooler
3. Connect 2.25 in Straight - Reducer - 1.5 in Straight (12) to Hose (11)
4. Use Hose Clamp (13) to clamp 2.25 in Straight (12) to Hose (11)
5. Weld 1.5 in Straight (12) to Flange
6. Weld 1.5 in Straight of the 1.5 in Straight - Elbow - Straight - Venturi Connection Plate (15) to Flange
7. Connect Flanges to EGR Valve (14)
8. Bolt Venturi Connection Plate (15) to inlet of Venturi (16)
9. Bolt Venturi Connection Plate (17) to outlet of Venturi (16)
10. Connect Flexible Silicone Hose (18) to 2.25 in Straight (17)
11. Use Hose Clamp (13) to clamp Hose (18) to 2.25 in Straight (17)

10 Results and Conclusions

The Exhaust Gas Recirculation system satisfies the customer needs that Volvo and University of Michigan outlined at the beginning of the design process. The minimum requirements of the project was to complete a design that fits the spatial requirements with the addition of calculations that proved that the designed system will work. The CAD-design of the EGR system contains all the necessary parts and is well integrated into the test cell, with consideration for the spatial availability.

Comparing the outcome of the project with the objectives listed in Table 2 it is clear that the first sub-objective was resolved by communication with Volvo and representatives from University of Michigan. The team completed the second sub-objective by conducting research of publicly available and suitable components, including those that were used for similar purposes elsewhere in the industry, and through communication with University of Michigan. To complete the predictive calculations on the EGR cooler performance, sub-objective three, reference data from the single cylinder D13 research engine at Volvo Truck Technology in Sweden was used, and served as the basis of the analysis. The team has compiled a Bill of Materials (see Appendix B) to stay within budget and fulfill sub-objective four. Lastly, the fifth sub-objective of Table 2 was resolved primarily at Penn State where a sufficiently equipped workshop was used during assembly and as temporary storage during the delivery phase.

The calculations on the EGR cooler resulted in an effectiveness of approximately 96.3-99.4 % and based on this the exhaust gas could be cooled down to 80 °C using the 4 gal/min coolant flow at 60-70 °C. The limiting factor was the chiller, which cools the coolant. The calculations show that the chiller should be able to handle the majority of loads produced by the engine, but not the highest load, A100 with an EGR rate above approximately 18% and C100. If the higher speed/load stages are run for an extended amount of time, the coolant will retain more and more heat and eventually exceed the maximum coolant temperature of 92 °C. As long as the engine only stays on the higher stages for limited periods of time the cooling capacity of the chiller should not be a problem.

To make sure that the exhaust gas could still be cooled enough even if the effectiveness of the cooler was lower additional calculations were done. Using an effectiveness of approximately 87.4-89.6 °C the coolant flow of 4 gal/min had to be between 10 °C and 20 °C to cool the exhaust gas to the desired outlet temperature. The heat transfer rate is a bit lower than with the better effectiveness meaning that the chiller would work at a lower effectiveness. Altering the coolant flow does not affect the cooling capacity much which leads to the conclusion that another coolant loop, with a higher flow rate, would not be necessary. The best way for reaching a lower temperature on the exhaust gas would be to lower the temperature of the coolant.

The calculations that were aimed to predict and compare the pressure differences that will emerge within the Volvo and Renault Venturi respectively was enough to show that the expected signal from the Δp-sensor will differ with approximately one order of magnitude between minimum and maximum flows through the EGR-system. A high resolution Δp-measurement for small flows is however difficult to achieve, as seen in the small Δp in Figure 6(b) and the small change in Δp from Figure 6(a) for flows below 10 SCFM, but the smaller Renault Venturi remains as the better option for all flows. On the contrary to using an over dimensioned cooler, which did not feature any serious downsides, an over dimensioned venturi directly affects the accuracy of the flow measurements.

A Bill of Materials was written and updated as parts and components were acquired. The final Bill of Materials can be found in Appendix B and contains all the purchased parts and components. The components that are yet to be bought are a silicone hose, the EGR control valve, actuator and flanges for connecting the valve to the system. These parts are not included in the Bill of Materials as these will be purchased by the customers at University of Michigan after the end of this project. The reason for not acquiring these parts was that the silicone hose, EGR valve and actuator did not fit within the $1000 budget. In addition, the flanges for the valve were not purchased due to not having the physical valve present during the assembly and, as it was requested, a summary of the researched valves and actuators that have proposed during the project is included in Section 8.4.1. Extrapolating from Christensen's experience with similar valves it was also determined that the lubrication of the valves was not a concern.

As stated in the paragraph above most parts and components have been acquired. These have also been assembled to fit into the test cell. The assembly of parts were either welded, screwed, or clamped together. The detailed construction process can be found under Section 9 and in Appendix F with pictures. The parts that are not yet purchased will be assembled into the EGR system by the customer University of Michigan but the system is design to easily integrate with any of the proposed, or similar, valves.

11 Recommendations

The first recommendation for the customers at the University of Michigan is to purchase one of the valves and actuators presented in this report, or others with similar characteristics. When this is done the system should be assembled into the test cell and the test procedures mentioned in Section 8.5 should be performed, along with additional tests that are deemed necessary by the customers working with the research engine in the test cell.

One thing which was not included in the design or assembly of the EGR system was components for measuring the inlet and outlet temperature of the exhaust gas going through the cooler. Therefore, the second recommendation is to install thermocouples before and after the EGR cooler. The thermocouples should preferably be installed very close to the inlet and outlet respectively, so that the temperature does not drop between the measuring point and the actual inlet or between the actual outlet and the measuring point.

Installing thermocouples in the pipes leading the exhaust gas is easily achieved by drilling, inserting the thermocouple and then sealing. It would be much more difficult to install thermocouples in the coolant hoses, which is why dedicated thermocouple T-connections were included in the design of the coolant adapters, placed close to the coolant inlet and outlet of the cooler.

A third recommendation is to verify the effectiveness of the cooler, to make sure the conclusions drawn are valid. The verification could be done through measuring the inlet and outlet temperature of the exhaust gas going through the EGR cooler and checking if they are in line with the results presented in Section 8.2.2.

References

ASME. *Code of Ethics of Engineers*, 2006. URL https://www.asme.org/getmedia/3cd4a989-d024-47cd-93b7-74cf6b257977/Student-Design-Competition-Engineering-Code-of-Ethics.aspx. Viewed on: March 2017.

Detroit Diesel. *Inspection and Testing of the EGR Cooler*, 2008. URL https://ddcsn-ddc.freightliner.com/cps/rde/xbcr/ddcsn/08604.EPA07.pdf. Viewed on: February 2017.

EPA. *Medium and Heavy Duty Diesel Vehicle Modeling Using a Fuel Consumption Methodology*, 2004. URL https://www3.epa.gov/otaq/models/ngm/may04/crc0304c.pdf. Viewed on: March 2017.

J. B. Heywood. *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988.

F. P. Incropera, D. P. DeWitt, T. L. Bergman, and A. S. Lavine. *Principles of Heat and Mass Transfer*. John Wiley & Sons, 2013.

International Energy Agency. *Transport, Energy and CO2*, 2009. URL https://www.iea.org/publications/freepublications/publication/transport2009.pdf. Viewed on: February 2017.

IPCC. *Appendix - Annex II - Glossary*, 2008. URL http://www.ipcc.ch/pdf/assessment-report/ar4/syr/ar4_syr_appendix.pdf. Viewed on: February 2017.

H. Jääskeläinen. *Diesel Exhaust Gas*, 2011. URL https://www.dieselnet.com/tech/diesel_exh.php. Viewed on: May 2017.

H. Jääskeläinen. *EGR Control Strategy*, 2016. URL https://www.dieselnet.com/tech/engine_egr_control.php. Viewed on: January 2017.

H. Jääskeläinen and M. Khair. *EGR Systems & Components*, 2012. URL https://www.dieselnet.com/tech/engine_egr_sys.php. Viewed on: January 2017.

H. Jääskeläinen and M. Khair. *Exhaust Gas Recirculation*, 2016. URL https://www.dieselnet.com/tech/engine_egr.php. Viewed on: January 2017.

Massachusetts Institute of Technology. *Internal Combustion Engines*, 2008. URL https://ocw.mit.edu/courses/mechanical-engineering/2-61-internal-combustion-engines-spring-2008/lecture-notes/lecture1.pdf. Viewed on: February 2017.

S. Moseley. *Environmental History of Air Pollution and Protection*, 2014. URL http://www.eolss.net/sample-chapters/c09/e6-156-15.pdf. Viewed on: February 2017.

S. Ratiu. *The History of the Internal Combustion Engine*, 2003. URL http://annals.fih.upt.ro/pdf-full/2003/ANNALS-2003-3-21.pdf. Viewed on: February 2017.

Samson. *Information Sheet T 8000-3*, 2016. URL https://www.samson.de/pdf_en/t80003en.pdf. Viewed on: April 2017.

M. Smith. *WHO: Diesel Exhaust Causes Lung Cancer*, 2012. URL http://www.medpagetoday.com/HematologyOncology/OtherCancers/33226. Viewed on: February 2017.

Southwest Research Institute. *Exhaust Gas Recirculation (EGR) Cooler Testing*. URL http://www.swri.org/3pubs/brochure/d03/EGR/EGRFlyer.pdf. Viewed on: February 2017.

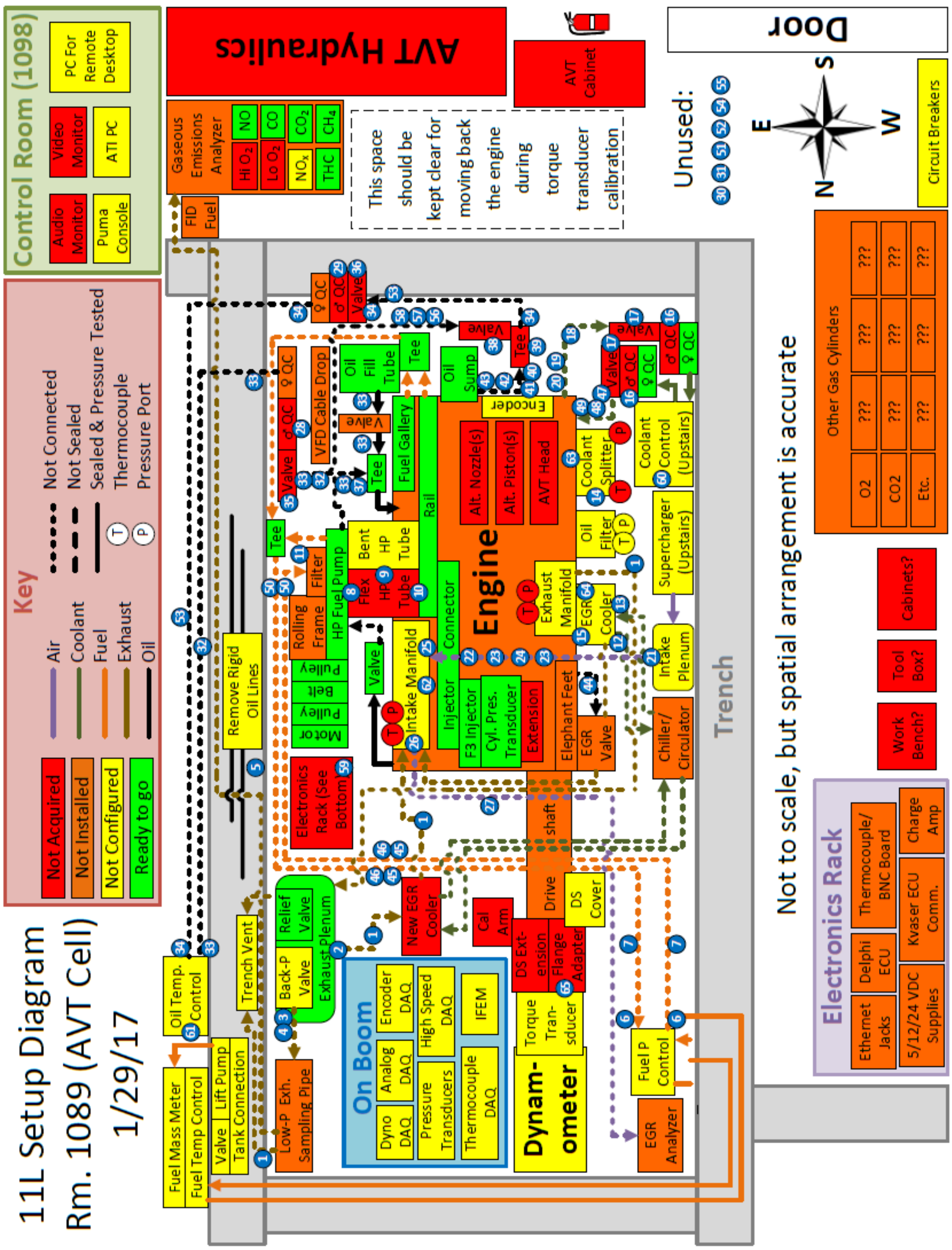
Thermo Fisher Scientific. *Thermo Scientific ThermoFlex Recirculating Chillers*, 2016. URL https://tools.thermofisher.com/content/sfs/brochures/Brochure-ThermoFlex-1113-Final.pdf. Viewed on: April 2017.

Volvo Group. *NAP Environmental Assessment*, 2007. URL http://www.volvogroup.com/SiteCollectionDocuments/suppliers/attachments/nap/NAP%20Environmental%20Self%20Assessment.pdf. Viewed on: February 2017.

F. M. White. *Fluid Mechanics*. McGraw-Hill, 2011.

World Bank Group. *Ground-Level Ozone*, 1998. URL https://www.ifc.org/wps/wcm/connect/dd7c9800488553e0b0b4f26a6515bb18/HandbookGroundLevelOzone.pdf?MOD=AJPERES. Viewed on: February 2017.

A Engine Test Cell Setup



Appendix A: A block schematic of the test cell at University of Michigan. For this project its the EGR cycle that includes the `Exhaust Manifold` (15), `Exhaust Plenum` (2), `New EGR Cooler` (45), `Intake Manifold` and `Chiller/Circulator` that is of interest.

B Bill of Materials

Appendix B: A table summarizing the expense of components for the EGR system.

Quantity	Part Description	Material	Price per Part
Exhaust Plenum to Cooler			
2	2 in Gasket	Graphite	-
1	2 in Butt-Weld Adaptor	304 Stainless	$4.89
2	Quick-Clamp	304 Stainless	$11.62
1	2 in OD High-Polish Tube	304 Stainless	$24.24
1	High-Polish Tube Elbow	304 Stainless	$31.43
1	3 in-2 in Reducer	304 Stainless	$4.83
Coolant Lines (Chiller to Cooler)			
1	Hose Fitting	304 Stainless	$12.18
1	Hose Fitting	Brass	$3.80
2	12 ft Coolant Hose	Silicone	-
2	Hose Fitting	304 Stainless Steel	$26.40
2	Tee Connection	Iron	$17.37
2	1/8 NPT Fitting	Steel	$4.82
2	Butt-Weld Tube Fitting	304 Stainless Steel	$19.75
2	2.5 in to 1.5 in Reducer	304 Stainless Steel	$36.52
1	Pack of Hose Clamps	301 Stainless Steel	$6.74
EGR Cooler to Intake Manifold			
1	2.25 in 180° Elbow	Silicone	$64.00
1	2.25 in OD Straight Tube	304 Stainless Steel	$53.77
1	2.25 in-1.5 in Reducer	304 Stainless Steel	$37.02
1	1.5 in OD Straight Tube	304 Stainless Steel	$64.54
2	Tube Flange	304 Stainless Steel	-
2	Gaskets	Graphite	-
1	Control valve and actuator	Iron	-
1	1.5 in OD Elbow	304 Stainless Steel	$18.18
1	12 x 12 in Sheet Metal Piece	304 Stainless Steel	$40.64
1	Renault Venturi	-	-
1	Flexible Hose	Silicone	-
1	Pack of Hose Clamps	301 Stainless Steel	$9.00
		Shipping costs	$55.54
		Total	**$733.76**

C Operational Data - EGR Cooler

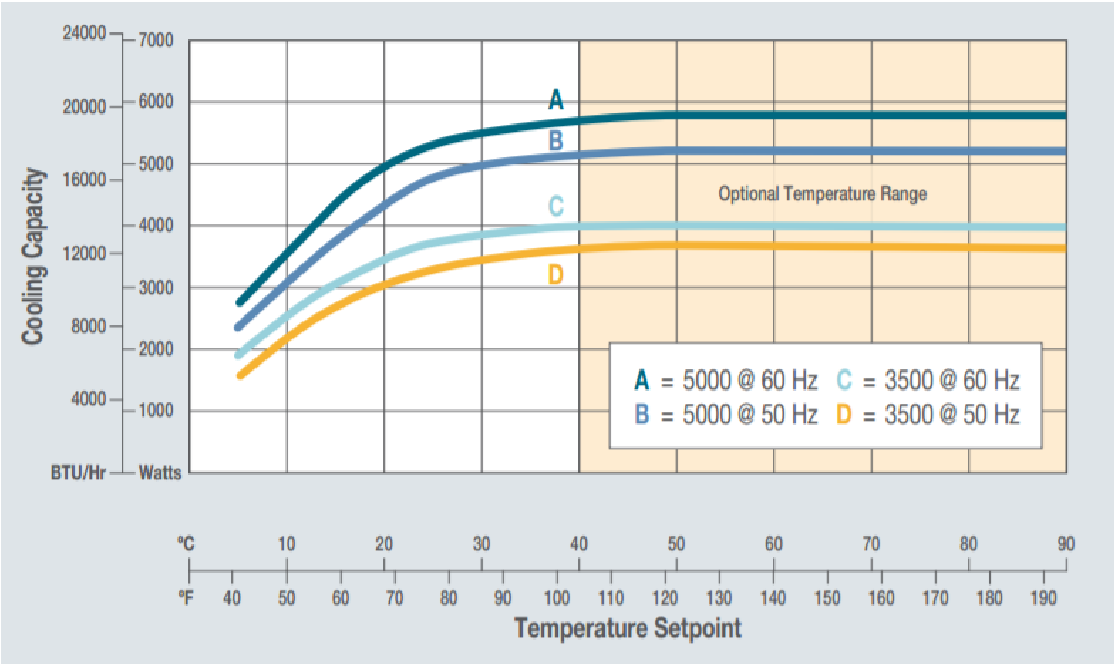
Build Nr	Comment	Comment2	RunState	Speed Load	Engine Speed	Torque measured (1cyl)	Injected Fuel	Charge Pressure
					rpm	Nm	mg/stroke	kPa
459	A25	VGT-EGR S	r5660s01	1200-25	1200	98,9	76,8	135
459	A25	VGT-EGR S	r5660s04	1200-25	1200	97,9	76,1	135
459	A50	VGT-EGR S	r5662s01	1200-50	1200	192,4	135,2	200
459	A50	VGT-EGR S	r5662s04	1200-50	1200	192,2	136,0	200
459	A100	VGT-EGR S	r5659s01	1200-100	1200	390,2	263,9	330
459	A100	VGT-EGR S	r5659s04	1200-100	1200	390,8	270,1	330
459	C100	VGT-EGR S	r5671s01	1800-100	1800	323,1	224,1	333
459	C100	VGT-EGR S	r5671s06	1800-100	1800	324,4	229,2	333

Exhaust Pressure	Charge Temp	BSNOx Corrected	Air Mass Flow	Exhaust Temp	Lambda	O2 exhaust dry measured	CO2 exhaust dry measured	CO2 intake dry measured
kPa	°C	g/kwh	g/s	°C		%	%	%
153	40,0	4,94	22,06	342,43	1,940	10,46	7,67	1,87
153	46,1	1,97	19,12	353,69	1,690	8,83	8,85	2,87
225	45,0	4,90	32,21	427,56	1,609	8,18	9,31	2,15
225	51,0	1,93	28,83	444,18	1,431	6,53	10,52	3,09
360	50,0	4,83	54,68	553,24	1,405	6,21	10,73	2,02
360	56,0	1,98	50,45	585,73	1,270	4,52	11,92	2,81
380	58,0	4,89	82,59	520,43	1,677	8,71	8,92	1,72
380	67,0	1,95	73,33	554,41	1,456	6,73	10,33	2,68

Corrected Engine Power	**EGR Gas Temp in**	**EGR Gas Temp out**	**EGR Coolant flow**	**EGR Coolant Temp Out**	**EGR Coolant Temp In**	EGR Cooler Power	**EGR rate**	**EGR Gas Pressure (abs)**
kW	°C	°C	l/s	°C	°C	kW	%	kPa
13,1	276,6	68,0	0,799	70,2	69,4	2,62	24,4	153
12,9	265,8	69,7	0,800	71,1	70,5	1,76	32,5	153
25,1	318,9	70,0	0,800	71,0	69,8	3,76	23,1	225
25,0	329,7	71,1	0,800	71,0	69,5	4,39	29,4	225
50,4	414,5	71,1	0,799	71,2	69,4	5,27	18,8	360
50,5	473,1	71,2	0,799	71,6	69,7	5,64	23,6	360
63,5	455,0	74,7	0,800	72,0	69,3	7,69	19,3	380
63,8	482,4	77,5	0,800	73,0	69,4	10,69	25,9	380

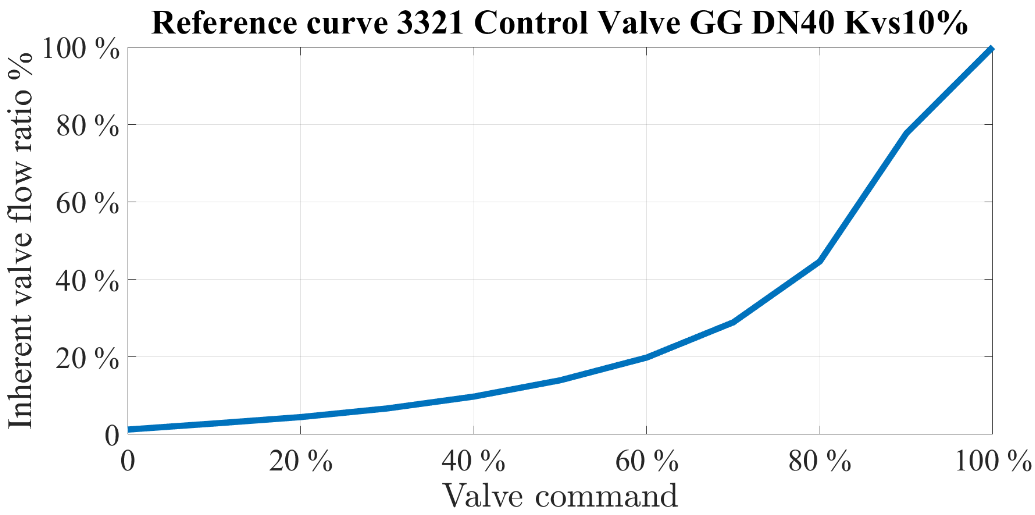
Appendix C: Operational data from Magnus Christensen's research engine. The table consists of 10 rows and 27 columns, and should be read from left to right and top to bottom as indicated by the contouring black lines.

D External Chiller Cooler Capacity



Appendix D: The curve "A" shows how the test cell's chiller's cooling capacity varies with the temperature of the coolant. Picture reference: (Thermo Fisher Scientific, 2016).

E EGR Control Valve - Data

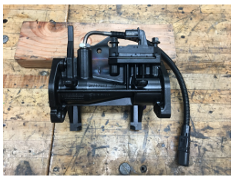


Appendix E: The reference curve shows how large the effective flow area is for different opening stages for the Type 3321 DN40 globe valve. The data is collected from the product manual (Samson, 2016).

F Construction Process Parts

Part Key	Part Description	Part Picture
1	2 in Elbow & Straight	
2	Reducer to Cooler	
3	Quick Clamp	
4	1 NPT Stainless Steel Hose Fitting	

Part Key	Part Description	Part Picture
5	1/2 NPT Stainless Steel Hose Fitting	
6	1/2 NPT Brass Hose Fitting	
7	1/8 NPT Fitting for thermocouple	
8	1.5-1-1 NPT Tee Connection	
9	1.5 NPT Threaded Reducer	

Part Key	Part Description	Part Picture
11	180° Silicone Hose	
12	2.25 in Straight to Reducer to 1.5 in Straight	
15	1.5 in Straight to Elbow to Straight to Venturi Connection Plate	
16	Renault 5 L Venturi	
17	Venturi Connection Plate to 2.25 in Straight	

Appendix References

Samson. *Information Sheet T 8000-3*, 2016. URL https://www.samson.de/pdf_en/t80003en.pdf. Viewed on: April 2017.

Thermo Fisher Scientific. *Thermo Scientific ThermoFlex Recirculating Chillers*, 2016. URL https://tools.thermofisher.com/content/sfs/brochures/Brochure-ThermoFlex-1113-Final.pdf. Viewed on: April 2017.