Waste Heat Recovery in a Heavy-Duty Engine using Rankine Cycles and Humid Air Cycles

PRAKASH NARAYANAN ARUNACHALAM | DIVISION OF COMBUSTION ENGINES
DEPARTMENT OF ENERGY SCIENCES | LUND UNIVERSITY | 2016

Waste Heat Recovery in a Heavy-Duty Engine using Rankine Cycles and Humid Air Cycles

Waste Heat Recovery in a Heavy-Duty Engine using Rankine Cycles and Humid Air Cycles

Prakash Narayanan Arunachalam

LUND UNIVERSITY

DOCTORAL DISSERTATION
by due permission of the Faculty of Engineering, Lund University, Sweden.
To be defended at M-building, LTH. Date 17/6 2016 and time 10.15.

Faculty opponent
Vicente Dolz Ruiz, CMT-Motores Térmicos, UPV Universitat Politècnica de Valéncia

Organization LUND UNIVERSITY	Document name DOCTORAL DISSERTATION
	Date of issue 2016-06-17
Prakash Narayanan Arunachalam	Swedish Energy Agency

Waste Heat Recovery in a Heavy-Duty Engine using Rankine Cycles and Humid Air Cycles

The majority of the fuel energy in any internal combustion engine is lost as heat loss, namely, exhaust gas heat, exhaust gas recirculation cooling, charge air cooling, and cylinder cooling. Utilizing this heat improves the overall efficiency. This thesis analyzes two such waste heat recovery cycles; the more mature Rankine bottoming cycle and the less investigated humid air cycle (HAM), from a heavy duty (HD) diesel engine perspective. The Rankine cycle potential was studied using 0-D and 1-D simulation tools. More emphasis was laid on understanding engine operation with a humid air cycle, which was investigated both virtually using simulation tools and also experimentally using a modified multi-cylinder heavy duty engine.

The Rankine cycle potential was studied using a thermodynamic analysis of the different heat sources in an effort to maximize the waste heat recovery (WHR) potential. The Rankine cycle circuit was designed in IPSE-Pro (Thermodynamic modeling software) with various heat sources in both single and dual loops. The possibilities and challenges involved in coupling these multiple sources in a single Rankine cycle, and the selection of suitable working fluid were analyzed. The study shows that it is possible to recover 5-10% of the otherwise wasted heat energy, which results in ~5% increase in power or reduced fuel consumption. Also, a comparative investigation of the WHR potential between the existing conventional diesel combustion and the novel concept of partial premixed combustion (PPC) was carried out. The results implies that PPC offers improved WHR potential compared to conventional combustion.

Humid air cycle was studied as a means of waste heat recovery technology intended to improve efficiency and decreasing emissions. The potential benefits of the HAM cycle were investigated in a virtual environment, with a heavy duty Volvo engine model modified with a humidifier (modeled as vapor injection). The commercial software GT-SUITE was used to build the system model and to perform the simulations. The results suggested a decrease in bulk in-cylinder gas temperature and a potential increase in power. The results showed that the efficiency improvement was on par with Rankine cycle along with the in-cylinder emission reduction potential.To investigate HAM experimentaly a HAM system was built around a 13-litre six cylinder Volvo diesel engine. The performance and emissions were investigated in three different modes, namely conventional operation with or without EGR and with the humid air cycle. The experiments carried out at part load showed that HAM reduced the emissions but without any influence on fuel economy. The experiments were further extended to two other load points in an effort to study the effect of engine load and speed. The results show that HAM reduces NOx emissions significantly together with the soot emissions without affecting the efficiency. CO and HC emissions increase marginally. At higher speeds the brake efficiency was improved by 3% for HAM along with the emissions reduction. With the fact that the HAM engine requires no charge air cooler (HAM tower acts as a cooler) and no EGR, this could further reduce losses in the form of aerodynamic losses in a vehicle application.

Waste heat recovery, Rankine cycle, Humid air cycle

Classification system and/or index terms (if any)

Supplementary bibliographical information		Language English
ISSN and key title 0282-1990		ISBN 978-91-7623-872-1 (print) ISBN 978-91-7623-873-8 (pdf)
Recipient's notes	Number of pages 138	Price
	Security classification	

I, the undersigned, being the copyright owner of the abstract of the above-mentioned dissertation, hereby grant to all reference sourcespermission to publish and disseminate the abstract of the above-mentioned dissertation.

Signature Date 2016-05-11

Waste Heat Recovery in a Heavy-Duty Engine using Rankine Cycles and Humid Air Cycles

Prakash Narayanan Arunachalam

LUND UNIVERSITY

Faculty of Engineering | Department of Energy Sciences
Division of Combustion Engines

ISRN LUTMDN/TMHP-16/1119-SE
ISSN 0282-1990
ISBN 978-91-7623-872-1

Printed in Sweden by Media-Tryck, Lund University
Lund 2016

Abstract

The majority of the fuel energy in any internal combustion engine is lost as heat loss, namely, exhaust gas heat, exhaust gas recirculation cooling, charge air cooling, and cylinder cooling. Utilizing this heat improves the overall efficiency. This thesis analyzes two such waste heat recovery cycles; the more mature Rankine bottoming cycle and the less investigated humid air cycle (HAM), from a heavy duty (HD) diesel engine perspective. The Rankine cycle potential was studied using 0-D and 1-D simulation tools. More emphasis was laid on understanding engine operation with a humid air cycle, which was investigated both virtually using simulation tools and also experimentally using a modified multi-cylinder heavy duty engine. The thesis can be broadly divided into two parts. The first part deals with the Rankine cycle simulation and results and the second part examines the humid air cycle potential with simulation and experiments.

In the first part, the Rankine cycle potential was studied using a thermodynamic analysis of the different heat sources in an effort to maximize the waste heat recovery (WHR) potential. The Rankine cycle circuit was designed in IPSE-Pro (Thermodynamic modeling software) with various heat sources in both single and dual loops. The possibilities and challenges involved in coupling these multiple sources in a single Rankine cycle, and the selection of suitable working fluid were analyzed. The study shows that it is possible to recover 5-10% of the otherwise wasted heat energy, which results in ~5% increase in power or reduced fuel consumption. Also, a comparative investigation of the WHR potential between the existing conventional diesel combustion and the novel concept of partial premixed combustion (PPC) was carried out. The results imply that PPC offers improved WHR potential compared to conventional combustion.

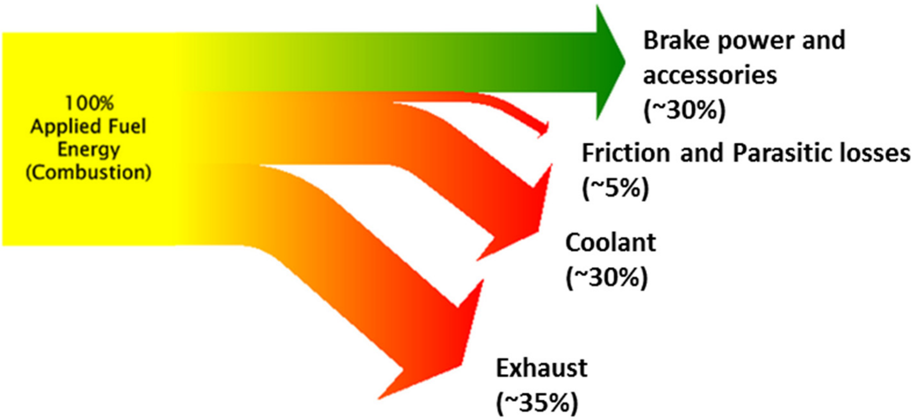
In the second part the humid air cycle was studied as a means of waste heat recovery technology intended to improve efficiency and decreasing emissions. The potential benefits of the HAM cycle were investigated in a virtual environment, with a heavy duty Volvo engine model modified with a humidifier (modeled as vapor injection). The commercial software GT-SUITE was used to build the system model and to perform the simulations. The results suggested a decrease in bulk in-cylinder gas temperature and a potential increase in power. The results showed that the efficiency improvement was on par with the Rankine cycle along with the benefit of emission reduction potential.

To investigate HAM experimentaly a HAM system was built around a 13-litre six cylinder Volvo diesel engine. The performance and emissions were investigated in three different modes, namely conventional operation with or without EGR and with the humid air cycle. The experiments carried out at part load showed that

HAM reduced the emissions but without any influence on fuel economy. The experiments were further extended to two other operating points in an effort to study the effect of engine load and speed. The results show that HAM reduces NOx emissions significantly together with the soot emissions without affecting the efficiency. CO and HC emissions increase marginally. At higher speeds the brake efficiency was improved by 3% for HAM along with the emissions reduction. With the fact that the HAM engine requires no charge air cooler (HAM tower acts as a cooler) and no EGR, this could further reduce losses in the form of aerodynamic losses in a vehicles application.

Populärvetenskaplig sammanfattning

Förbräningsmotorn effektivitet har förbättrats avsevärt sedan dess introduktion för mer än hundra år sedan. Det som har möjliggjort denna förbättring är en rad ändringar som: förbättringar i förbränningsrummet, mixningen av bränsle, gasväxlingen, minskade kylförluster, bränslekvaliteten, förbränningskoncepten (tex PPC, HCCI, RCCI), förbättrade mekaniska komponenter, material etc. Men fortfarande är den totala verkningsgraden inte bättre än ca 45% och den största boven är värmeförluster.



En typisk energikaskad för en förbränningsmotor.

Nyttan av att ta tillvara på den annars förlorade energin ökar i betydelse. Värmeåtervinning är en metod som använder annars förlorad värmeenergi till att utvinna nyttigt arbete. Det finns många sätt att göra detta på. Några av de äldre teknologierna är turboladdning, turbocompound och Rankine cykel (ångcykel) men det finns även nya tekniker som thermoelektriska generatorer och evaporativa cykler.

I detta projekt har två värmeåtervinningsprinciper undersökts med lyckat resultat. En av teknikerna innefattar Rankine cykeln, som i generationer har används för kraftgenerering i ångkraftverk, till att i senare tid användas som en bottencykel för en förbränningsmotor. En annan mindre känd teknik som undersökts i projektet är den evaporativa cykeln (HAM). Denna teknik innebär att återvunnen värme används för att förånga vatten som i sin tur öka luftfuktigheten i insugsluften till motorn. Detta ökar inte bara verkningsgraden utan även minskar utsläppen av kväveoxider och partiklar, som båda är väldigt hälsovådliga.

Denna avhandling kan delas upp i två delar. Den ena delen innefattar Rankine cykel simuleringar och resultat från dessa medan den andra delen innefattar den evaporativa cykelns potential med både simuleringar och experiment.

I den första delen går det att läsa om Rankine cykelns potential för att maximera värmeåtervinningen med en termodynamisk analys av olika värmekällor. Denna studie visar att den är möjligt att återvinna 5-10% av energi som i annat fall hade förlorats, vilket resulterar i cirka 5% effektökning i motorn eller minskning i bränsleförbrukning. Det har också gjorts en jämförelse av värmeåtervinningspotentialen för en vanlig dieselmotor och ett nytt motorkoncept vid namn PPC. Resultatet visar att potentialen för värmeåtervinningen kan öka för det nya PPC-konceptet jämfört med en vanlig dieselmotor.

Den andra delen av denna avhandling, den evaporativa cykeln, har studerats med avseende på att förbättra motorns effektivitet samt för att minska utsläppen. Potentialen för HAM-cykeln utvärderades först teoretiskt och resultaten visar att bulktemperaturen i cylindern minskade och effekten potentiellt ökade. Studien visade även att effektiviteten ökade i paritet med Rankine cykeln men med fördelen av samtidigt minskade utsläpp.

För att verifiera de teoretiska resultaten har ett HAM-system konstruerats och installerats på en 13-liters sex-cylindrig Volvo diesel motor. De experimentella resultaten bekräftar de teoretiska och visar att kväveoxider kan minskas signifikant samtidigt med partikelemissionerna medan CO och HC emissioner bara ökar marginellt. Vid högre motorvarvtal så förbrättrades den bromsade verkningsgraden med 3% samtidigt som NOx och partikel emissioner minskade. Faktum är att en HAM-motor inte behöver någon kylning av den turboladdade insugsluften i och med att HAM tornet fungerar som en luftkylare och att HAM inte heller behöver någon EGR så kan även aerodynamiska förluster minska vid användning av HAM i en fordonsapplication.

List of Publications

I. Arunachalam, P., Shen, M., Tuner, M., Tunestal, P. et al., "Waste Heat Recovery from Multiple Heat Sources in a HD Truck Diesel Engine Using a Rankine Cycle - A Theoretical Evaluation", SAE Technical Paper 2012-01-1602, 2012, doi: 10.4271/2012-01-1602.

II. Arunachalam, P., Nyberg, B., Tuner, M., and Tunestal, P., "A Comparative Analysis of WHR System in HD Engines Using Conventional Diesel Combustion and Partially-Premixed Combustion," SAE Technical Paper 2012-01-1930, 2012, doi: 10.4271/2012-01-1930.

III. Arunachalam, P., Tuner, M., Tunestal, P., Johansson, B, Thern, M., et al., "System Simulations to Evaluate the Potential Efficiency of Humid Air Motors," SAE Technical Paper 2013-01-2646, 2013, doi: 10.4271/2013-01-2646.

IV. Arunachalam, P., Tuner, M., Tunestal, P., B, Thern, M., "Experimental Analysis of Humid Air Motor". Submitted to SAE PFL conference for 2016

V. Arunachalam, P., Svensson, E., Tuner, M., Tunestal, P., B, Thern, M., "Emission, and Efficiency Potential of Humidified Air Motor (HAM)". To be submitted.

Acknowledgements

First and foremost I would like to thanks the seniors at our division, Bengt Johansson, Martin Tuner, Öivind Andersson, and Per Tunestål for giving me such an excellent opportunity to learn all these years. Equal thanks go to all the technicians, without them this thesis wouldn't have been possible. I would also like to thank the administrators Catarina Linden, Maj-Lis Roos and Elna Andersson for helping me with numerous administrative things.

Special thanks to my supervisors Martin Tuner for supporting and motivating me throughout my study, Per Tunestål for helping me with the ever confusing control program and Marcus Thern for teaching thermodynamics.

I would also take this opportunity to thank the Swedish Energy agency for funding the project. I also extend my thanks to all of the reference group members. I would also like to thank my counterparts Gunnar Latz (Chalmers) and Habib Aghaali (KTH) in the waste heat recovery project for sharing their findings. Special thanks to Petter Strandh (Volvo GTT) for his control system, advice, and timely contributions.

I thank all my colleagues from the division of engines for the memorable time. Thanks to Slavey, Guillaume and Yann for making me more social. Thanks to Nhut, Ashish, Eric, Sam and Lian hao for all the help in the lab and discussions.

I would also like to thank my inspirations, Sven Andersson (CTH), Ingemar Denbratt (CTH), Jakob Fredriksson (Volvo GTT), Fredrik Ekström (VCC), Johan Lindeyrd (Scania), Olof Erlandsson (TitanX), Arne Andersson (Volvo GTT) and Bengt Johansson (KAUST).

My hearty thanks to my family and friends from my hometown for all their support and encouragement. I would like to thank Arjun, Lal, Lokesh, Ram, Jothi, and Wolfgang for making my time more enjoyable in Sweden.

Finally, I would like to thank Sweden for teaching many things in these past seven years.

Content

Abstract 7
Populärvetenskaplig sammanfattning 9
List of Publications 11
Acknowledgements 12

1. Introduction 15
    1.1 Background 15
        1.1.1 History of combustion engine development 15
        1.1.2 Current scenario 16
    1.2 Motivation 17
    1.3 Approach 17
    1.4 Limitations 18
    1.5 Contribution 18

2 Waste heat recovery in I.C. Engines 19
    2.1 Rankine bottoming cycle 19
    2.2 Humid Air Motor (Humid air cycle) 20
    2.3 Turbocharger 22
    2.4 Thermoelectric generators 22
    2.5 Heat pipes 23
    2.6 Turbocompound 24

3. Methodology 27
    3.1 Rankine cycle 27
        3.1.1 Selection of heat sources 27
        3.1.2 Selection of working fluid 28
        3.1.3 Simulation setup 29
        3.1.4 Availability of energy 29
        3.1.5 Configuration of Rankine cycle 30
    3.2 Humid Air Motor 32
        3.2.1 Simulation 32

4. Internal combustion engine analysis 37

4.1 Heat release analysis 37
4.2 Energy utilization 40
4.3 Emissions 42
4.3.1 Nitrogen Oxides 43
4.3.2 Carbon monoxide 44
4.3.3 Unburned hydrocarbon emissions 45
4.3.4 Particulate emissions 45
5. Experimental Setup 47
5.1 Volvo D13 Engine 47
5.2 Instrumentation 47
5.2.1 Gas flow management 47
5.2.2 Sensing 48
5.3 Data Acquisition 49
5.3.1 Control system 49
5.3.2 Hardware 50
5.3.3 Software 50
6. Results 51
6.1 Rankine simulation (Paper I and II) 51
6.1.1 Waste heat utilization at different load points 53
6.1.2 Rankine cycle heat recovery with PPC combustion and conventional diesel combustion 53
6.2 Humid Air Simulation (Paper III) 55
6.3 HAM Experiments (Paper IV and V) 58
6.3.1 Emission 58
6.3.2 Efficiency 63
7. Future Work 69
8. Conclusion 71
9. Bibliography 73

1. Introduction

1.1 Background

1.1.1 History of combustion engine development

Heat was used to perform work from the times of Leonardo da Vinci as early as 1509. Atmospheric engines were developed first to perform work. Development in the atmospheric engine was conducted by Otto Von Guericke followed by a gun power fed engine by Christiaan Huygens, after which Denis Papin introduced steam as working media which was further developed by Thomas Savery and then improved by Thomas Newcomen, who invented the steam engine which ran with 0.4% efficiency. James Watt further developed this to become a proper steam engine with a separate boiler and condenser. After this period, numerous inventions were made, although some notable ones are the Stirling engine made by Robert Stirling and the Ericsson engine by John Ericsson (C. Lyle Cummins, 1976).

After better understanding of thermodynamic properties like heat, specific heat and latent heat, experiments performed by people like Count Rumford and Humpfrey Davy established that heat is work. The notable scientist Sadi Carnot then improved the understanding of the heat through closed cycle analysis, and he was the first one who proposed using compressed air for combustion to reach bigger temperature difference which would then give better efficiency. After Julius Robert Mayer, James Prescott Joule, Rudolf Clausius and James Clerk Maxwell’s contributions to the thermodynamic laws, studies of heat devices have grown exponentially.

After several developments in reciprocating engines had started, it was De Rivaz who developed the first internal combustion engine with spark ignition. Jean Joseph Lenoir further developed De Rivaz engine and then mass produced the engine. Inspired by Lenoir gas engine, Nicolaus August Otto developed the famous Otto engine (the SI engine). Another important invention made by Rudolf Christian Karl Diesel is the world renowned Diesel engine. These two are the most common reciprocating engines used until today.

1.1.2 Current scenario

Today, the transport sector uses about 19% of the global energy supply. 96% of this energy, i.e., 18%, of global energy supply, comes from oil of which road transport accounts for 76% (Council, 2011). Increased awareness of global warming and its relation to carbon dioxide (CO_2) has forced the reduction of CO_2 in the transport sector (Robin Hickman, 2007) (Can Wang, 2007). Combustion of carbon-based fossil fuels results not only in increased carbon dioxide emissions but also the more harmful NOx, particulate emissions, carbon monoxide, unburnt hydrocarbons, etc. Hence, it is crucial to make the transport sector efficient and clean. Every small step towards improved efficiency affects the global CO_2 production.

Since the introduction of combustion engines (about a century ago), engine efficiency has improved substantially. This was contributed by numerous changes to the engine, notably by optimizing combustion chamber geometry, fuel mixing, gas exchange, cooling, fuels, combustion itself for instance (PPC, HCCI, RCCI), improved mechanical parts, materials …etc. Still most of the modern engines haven't been able to exceed 45% brake efficiency; this is because of the heat losses. Figure1 shows the energy split from an I.C engine.

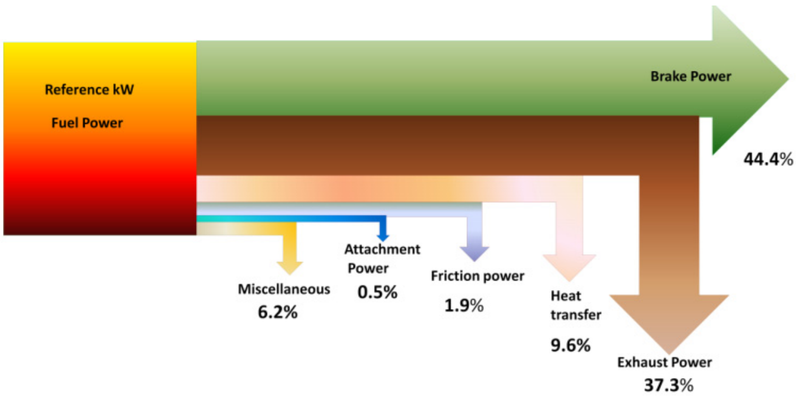


Figure 1.
Power split from an HD diesel Engine

More than 45% of the fuel energy is wasted in the form of heat losses in engines. Thus, the need for waste heat recovery arises. Waste heat recovery is a method to recuperate the otherwise wasted heat to produce useful work. There are many ways to recover heat. Some of the mature technologies include turbocharging, turbo-compounding, and Rankine cycle (Leising, 1978). A couple of novel technologies include thermal electric generators and humid air cycles.

1.2 Motivation

Global warming is currently the biggest threat the world is facing. The transport sector is one of the greatest polluters, contributing about 26% of the global CO_2 emissions (Chapman, 2007). The transportation sector, apart from CO_2 emissions, emits a lot of black carbon and ozone-producing gasses (Nadine Unger, 2009). A primary challenge for the fossil dependent transport sector is the fuel depletion with a peak of oil production likely to appear before 2030 (Steve Sorrell, 2010). The problem further escalates with the increased vehicle population in developing countries (Yunshi Wang, 2011). Electrification of vehicles could be a solution, but then, the source of electricity production becomes a problem. Electrification of the vehicles for meeting the entire transportation demands is not feasible in the near future (Christian Berggren, 2009). Biofuels offer an alternative for the current fossil fuels, but, it can only replace a few percent of the energy demand (Jason Hill, 2006). Hence, the optimization of the current internal combustion engines becomes a necessity to reduce the carbon dioxide emissions and fuel consumption. Irrespective of the fuels used, the majority of the loss is heat loss. As explained in the previous chapter, despite various advancements in the engine, heat loss is still the significant loss from an internal combustion engine. This loss can be recuperated in numerous ways. Familiar and well-studied methods include turbocharging, turbocompounding, and Rankine cycle, and can be used to recover this heat to produce some useful work. In the previously mentioned technologies, the combustion is hardly affected and hence the emissions are little affected. Another heat recovery technique, humid air cycle, uses a novel method of using waste heat to increase the efficiency and to decrease the emissions of the engine (PrimeServ). Humid air cycles were primarily developed for gas turbines to increase the efficiency (Rosen, Evaporative Cycles - in Theory and in Practise, 2000). The humid air cycle was later modified to suit internal combustion engines to reduce the NOx emissions. Humid air cycles for internal combustion engines, to improve efficiency and reduce emission simultaneously have been reserached to a very limited extent and this motivated us to perform theoretical and experimental research of humid air cycles in a heavy-duty diesel engine.

1.3 Approach

A typical Rankine cycle in automotive applications uses the exhaust gas or exhaust gas combined with one other heat source (Tianyou Wang, 2011) (Charles Sprouse III, 2013). To increase the power output from the waste heat and to increase the Rankine cycle efficiency, different heat sources were combined and analyzed for various pressure levels. The analysis was performed with a commercial software IPSE-Pro. Rankine cycle with both, single and dual loops for better heat utilization were studied (Arunachalam Prakash Narayanan M. C., 2012). The Rankine cycle studies were extended to an engine operated with futuristic partially premixed combustion (PPC) concept. PPC offers higher efficiency, but one challenge with the PPC engine is the low-temperature exhaust. A comparison between conventional diesel combustion and the PPC, both with a Rankine cycle, was analyzed. Another novel heat recovery concept, humid air cycle, was studied in a truck engine. Humid air cycle simulations were performed with the commercial software GT-Suite and MATLAB. The humid air cycle simulation study was then extended to experimental work in a multicylinder Volvo heavy duty (HD) truck engine.

1.4 Limitations

Only HD diesel engines were considered in the heat recovery studies in this thesis. For waste heat recovery comparison between the conventional diesel combustion and PPC, only the operating point with the best efficiency was analyzed. Rankine cycle simulations did not take turbocharger efficiency into account. For Humid air cycles, a calibrated engine model from Volvo was used to study exhaust exergy and the turbocharger efficiency was not optimized for the humid air cycle. HAM experiments were performed at low-mid speeds and low-mid loads; this is because of the design limitations, like the inability of the heat exchangers to utilize all the exhaust heat, feed pumps to supply high mass flows and to control the water level inside the humidifier.

1.5 Contribution

This thesis presents two waste heat recovery techniques, namely Rankine cycle and humid air cycle. The Rankine cycle was studied for an HD engine with water as working fluid. The study gives insight into how different heat sources can be utilized (Arunachalam Prakash Narayanan M. C., 2012). Comparative waste heat recovery potential of conventional combustion and PPC engine is presented (Arunachalam Prakash Narayanan B. N., 2012). Potential efficiency of the humid air cycle was evaluated for a HD diesel engine in simulation (Arunachalam Prakash Narayanan M. T., 2013). Emission and efficiency potential of humid air cycle was finally evaluated and compared to conventional diesel operation experimentally.

2 Waste heat recovery in I.C. Engines

2.1 Rankine bottoming cycle

A Rankine cycle can be used as a bottoming cycle to recuperate the power from the exhaust. It is fairly a mature technology investigated by many researchers (Yiping Dai, 2009) (V. Dolz, 2012) (Donghong Wei, 2007) (T.C. Hung, 1997) (Teng H. K., 2011) (Teng H. R., 2007) (Ringler, 2009). The working principle of a Rankine cycle includes heating (isobaric), boiling (isobaric), superheating (isobaric), expanding (isentropic), condensation (isobaric), and compression (isentropic). The exhaust heat is transferred to usable power by means of a working fluid. Several studies have been carried out in finding a suitable working fluid for the Rankine cycle (Latz, 2012) (Bo-Tau Liu, 2004) (Hung, 2001) (E.H. Wang, 2011). The working fluid is heated and vaporized by absorbing heat from any external source which for automotive applications can be exhaust gas, EGR, coolant or hot charge air. The vaporized fluid is made to expand over an expander like a turbine, piston expander or scroll expander to produce power. The vapor is then condensed to liquid form in a condenser. The condensed liquid is then compressed to increase the pressure and then the process continues. The process is explained with a T-S diagram for a wet working fluid (water).

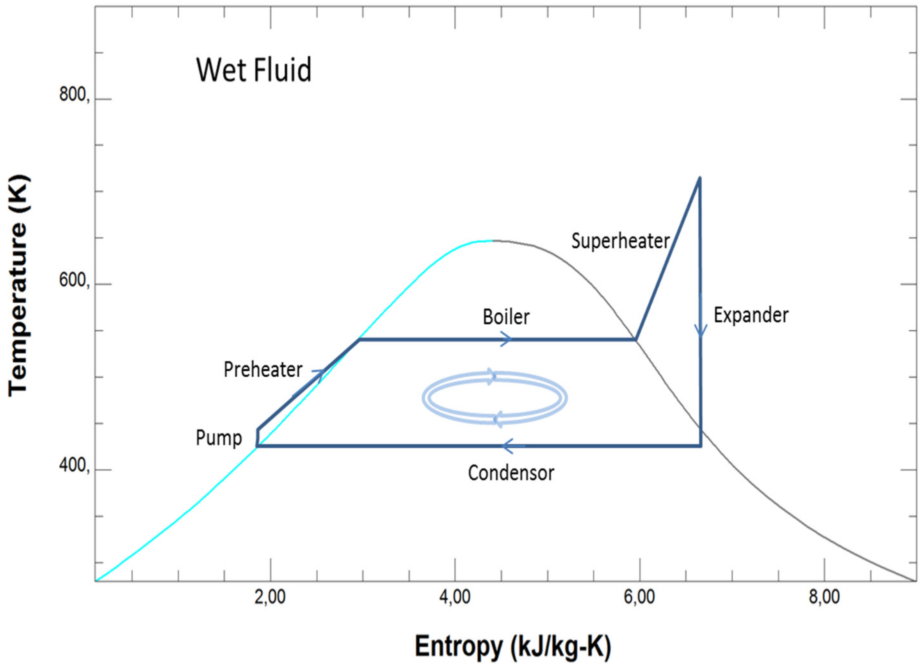


Figure 2.
T-S diagram of water with typical Rankine cycle

2.2 Humid Air Motor (Humid air cycle)

A humid air motor (HAM) uses the waste heat from the exhaust of the engine to humidify the inlet air; this humid air, with higher specific heat capacity, reduces combustion temperature and thus reduces NOx emissions to a greater extent (Entec, 2005) (Rahai, 2011) (Nord, 2001) (Park, 2011). The vapor has higher specific heat capacity than air, as shown in Figure 3, and so its presence reduces the temperature in the combustion process, resulting in lower peak temperatures and thus decreasing the emissions associated with high-temperature combustion process. The specific heat capacity of vapor is almost twice that of air, and so this method is twice as effective as exhaust gas recirculation (EGR); this brings in the possibility of the removal of EGR for NOx suppression.

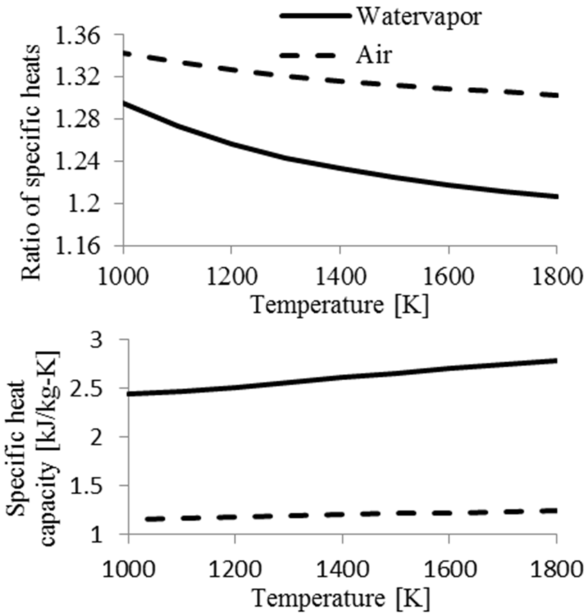


Figure 3.
Specific heat capacities and ratios of specific heat capacities of air and water vapor

The vapor is produced using the heat from the exhaust. This vapor is fed to the engine and hence, this vapor mass flow from the humidification tower (HT), which has not been compressed in the compressor, can be expanded over the turbine, producing free additional work. This could be perceived as a Rankine cycle operated within the engine cycle. Along with the benefit of emission reduction potential, the total system efficiency can be increased by using the vapor mass flow (vapor + air) to lower the pumping losses.

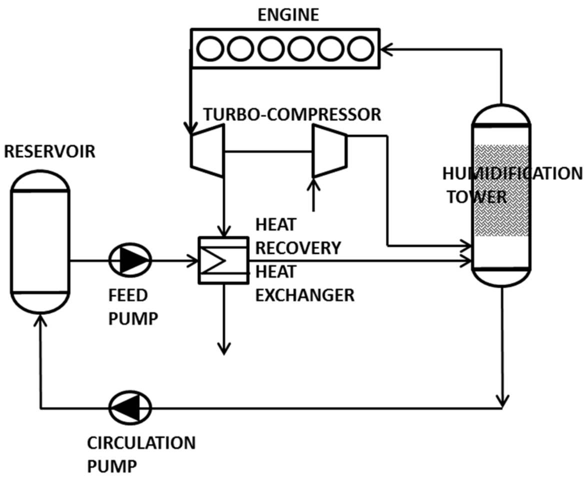


Figure 4.
HAM engine layout

2.3 Turbocharger

A turbocharger is typically a waste heat recovery device which extracts heat from the exhaust stream and uses it to compress the inlet air to increase the air charge density. A turbocharger utilizes the waste exhaust heat, but it puts a penalty in the form of backpressure in the engine cycle.

2.4 Thermoelectric generators

The TEG principle applies the Seebeck effect, where temperature differences in metals/alloys/semiconductors produce electricity. The TEG have been investigated in automotive exhaust waste heat recovery (Crane.D., 2013) (Brito, 2013).

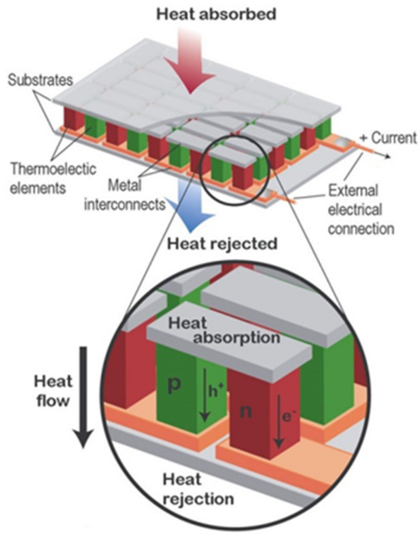


Figure 5
Representation of TEG structure

2.5 Heat pipes

Heat pipes are heat transferring devices which can transfer the heat from point A to point B by utilizing the latent heat of evaporation. The basic working principle is shown in Figure 6. ‘Heat in’ refers to the hot source, the working fluid takes the heat from this hot source and gets vaporized. The hot vapor then travels to the cold side, ‘heat in’, of the heat pipe where the vapor gets condensed to the fluid. This liquid is absorbed by the wick structure like a sponge and feeds it back to the hot end. There are few other types of heat pipes which doesn’t have the wick structure, but uses gravity to feed the water back (heat pipe thermosiphons) or alternatively has an additional reservoir to regulate the mass flow of the working fluid.

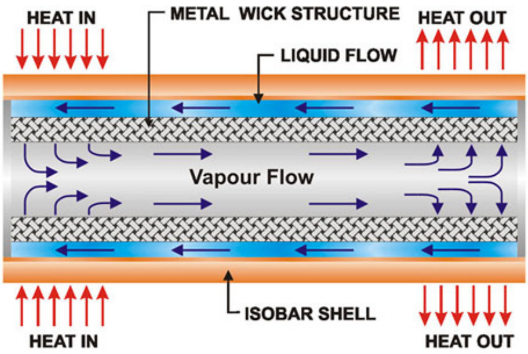


Figure 6.
Heat pipe working principle

Heat pipes with different working fluids can either be used as a hot source or cold source for TEG. Numerical simulations suggests that a fuel saving potential of 7% can be achieved using BiTe elements (max working temp 250 °C) for long-distance traveling commercial vehicles (Steyr Engineering Center, 2016). Experimental results indicate that heat-pipes combined with TEG only give a small improvement. Heat pipes are used for various automotive purposes. Mainly for aiding the cooling process of the coolant or piston (MAESC, 2009) (Hendriks, 2002) (Mignano, 1998). To better the process the heat pipes can be removed, and the TEGs can be directly equipped with exhaust, radiator, etc. The challenge in fitting TEG directly in the exhaust is the maximum working temperature of the alloys/semiconductors used in TEGs (Brito, 2013).

2.6 Turbocompound

In a typical HD engine operation, turbo-compound can be considered as an extension of the existing turbocharger concept, where usually a separate turbine is fitted in the downstream of the turbocharger to produce mechanical work. Similar to a conventional turbocharger this also increases the back pressure of the engine, thus affecting the efficiency. A careful design with this trade-off helps in improving the efficiency (Aghaali, 2014). Turbocompound is beneficial especially at higher loads (Hountalas D, 2007) (C.O. Katsanos, December 2013).

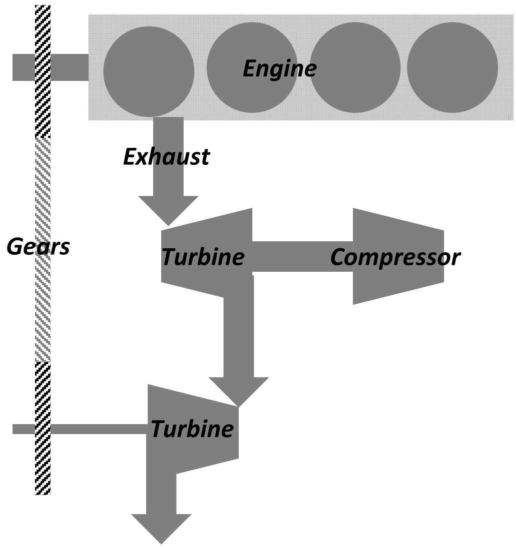


Figure 7.
Typical configuration of a turbocompounded engine

3. Methodology

3.1 Rankine cycle

3.1.1 Selection of heat sources

There are a couple of heat sources in an I.C engine, namely engine coolant, exhaust gas, and hot charge air; these sources often have more heat than the EGR source and yet due to the low differential temperatures these sources have seldom been exploited. A theoretical evaluation of waste heat recovery from these different sources was made using a Rankine cycle. Because of low grade heat from the engine coolant (low temperature difference across the heat exchangers ~20°C which in real life would result in massive size heat exchangers), this source was omitted in the simulations. Several sources were tested with both single and dual loop Rankine cycle. A suitable working fluid was selected for the Rankine cycle. The Rankine cycle was simulated for different configurations for a particular operating point, C75 in ESC 13 point cycle.

The heat balance of an engine can be described with Equation 1 for an engine with system boundary as shown in Figure 8.

$$\dot{m}_{fuel} = \dot{Q}_{Exhaust} + \dot{\boldsymbol{Q}}_{\boldsymbol{EGR}} + \dot{\boldsymbol{Q}}_{\boldsymbol{CAC}} + \dot{\boldsymbol{Q}}_{\boldsymbol{coolant}} + \dot{\boldsymbol{Q}}_{\boldsymbol{catalyst}} + \dot{\boldsymbol{Q}}_{\boldsymbol{radiation}} + P_{brake} + P_{misc-losses} \quad (1)$$

A calibrated HD Volvo diesel engine model was created with GT-Suite. The results from this engine model were used to evaluate the selection of heat source. The cycle was simulated in a virtual environment using the commercial software IPSEpro, and the cycle was optimized for different configurations for maximum power.

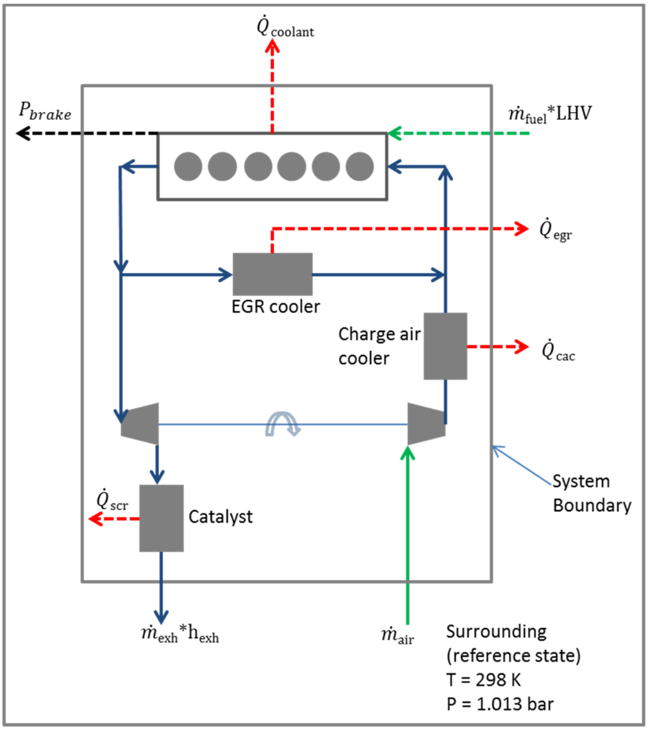


Figure 8.
Energy flows in an engine with system boundary

3.1.2 Selection of working fluid

The latent heat of evaporation (evaporation enthalpy) of a liquid is a good indicator when choosing a working fluid for a given temperature level (Ringler, 2009); Because of the higher latent heat of liquid, the flow rates required will be lesser in volume and thus smaller heat exchangers can be used, which is an important factor in automotive applications. Water has the highest latent heat of evaporation (~2250 kJ/kg) followed by organic working fluids and alcohols (methanol ~1100 kJ/kg, ethanol ~820 kJ/kg). Ethanol and methanol have low boiling point temperatures when compared to the typical exhaust gas recirculation temperature (EGR ~750K).

Further, alcohols have flammability issues; ethanol and methanol have high flammability potential (Latz, 2012). Water, on the other hand, has very high operating temperature limits making it ideal for EGR waste heat recovery and has no flammability issues. Since the engine model has the source temperatures ranging from 500K to 750K, water was chosen as a suitable working fluid.

REFPROP by the National Institute of Standards and Technology (NIST) was used to determine the fluid properties. Transport properties were estimated by using REFPROP (v9.0) (Lemmon, 2011).

3.1.3 Simulation setup

IPSEpro was used to build the Rankine cycle. A built-in optimizer to the software tool enabled us to optimize the cycle for maximum power output for different cycle pressures. The heat exchangers and boilers were not dimensioned; instead, the principle of pinch point was used and set to 20ºC. The turbine isentropic efficiency set to 78%, and the pump efficiency set to 60%. The condensers were modeled to condense the steam after expansion in the turbine and further sub-cool the vapor by 2ºC to ensure that the water saturation. By heat and mass balance equations, the cycle power output was calculated at the turbine. The vapor after the expansion was set to have more than 90% vapor quality at ambient pressure in order to make the condenser work with air at ambient pressure and temperature. Due to practical application issues, the maximum pressure level was set to 60 bar.

3.1.4 Availability of energy

Heat flow across the heat exchanger devices gives the available heat to recuperate. This is calculated using Equation 2, with average values of specific heat entering and leaving the device, temperature difference between the gas entering and leaving the device, and mass flow of the gas flowing through the device. Ultimately, the heat that was earlier let into the ambient is considered available for recuperation. The available heat energy for different sources found in this study is presented in Figure 9.

$$\dot{Q} = \dot{m} * \frac{Cp_{in} + Cp_{out}}{2} * (T_{in} - T_{out}) \tag{2}$$

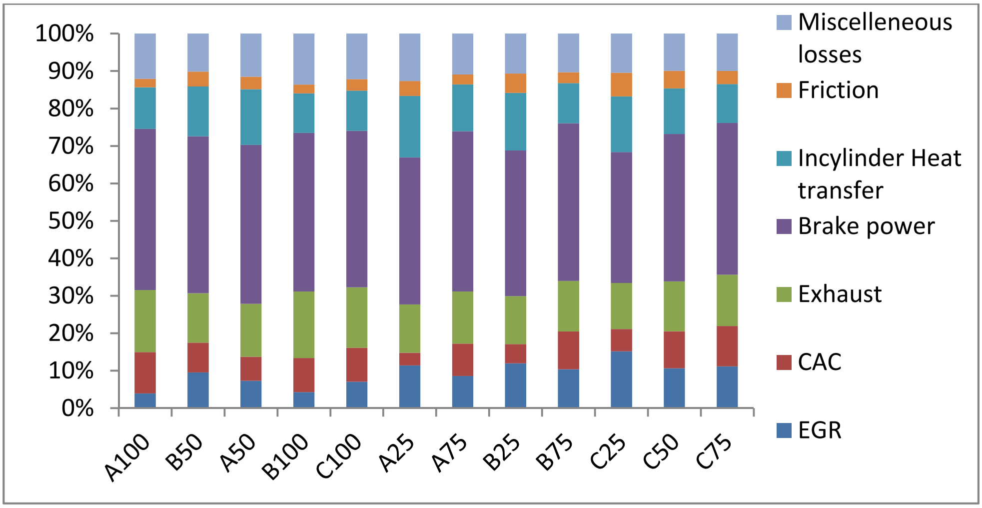


Figure 9.
Energy availability of an HD truck diesel engine

3.1.5 Configuration of Rankine cycle

Different configuration of Rankine cycle setup is shown in Table 1

Table 1.
Rankine cycle - all configurations

Loop	EGR	Exhaust	CAC
Single loop	✓	-	-
Single loop	-	✓	-
Single loop	✓	✓	-
Single loop	✓	✓	✓
Dual loop	✓	✓	✓

When using all the three sources in a single loop, there is a limiting factor of cycle pressure. The EGR source has a higher temperature, allowing us to work with higher pressures. On the other hand, the CAC and exhaust have lower temperatures when compared to EGR. In the dual loop Rankine cycle, the lower temperature exhaust, and CAC, works with a lower pressure cycle and the high-temperature EGR work with higher pressure cycle. The cycle have two turbines: a high-pressure turbine and a low-pressure turbine. A single loop Rankine cycle with all the sources combined and a dual-loop Rankine cycle with all the sources combined are shown in the Figures 10 and 11.

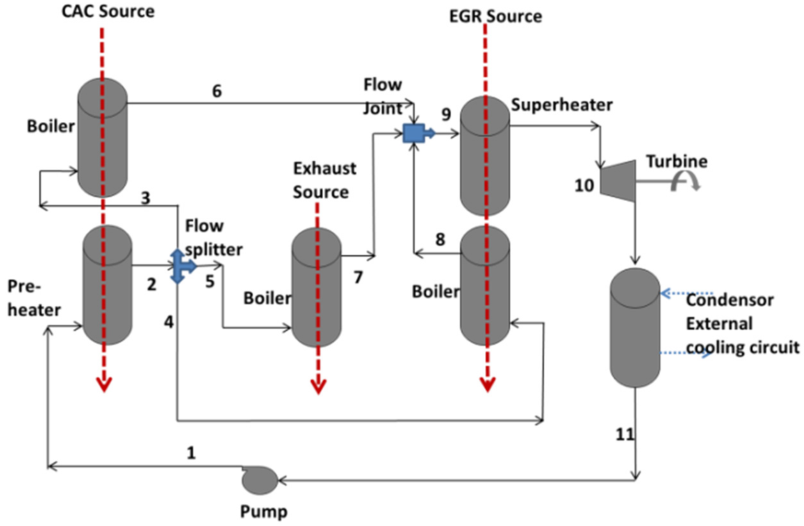


Figure 10.
Single loop Rankine cycle with all three sources

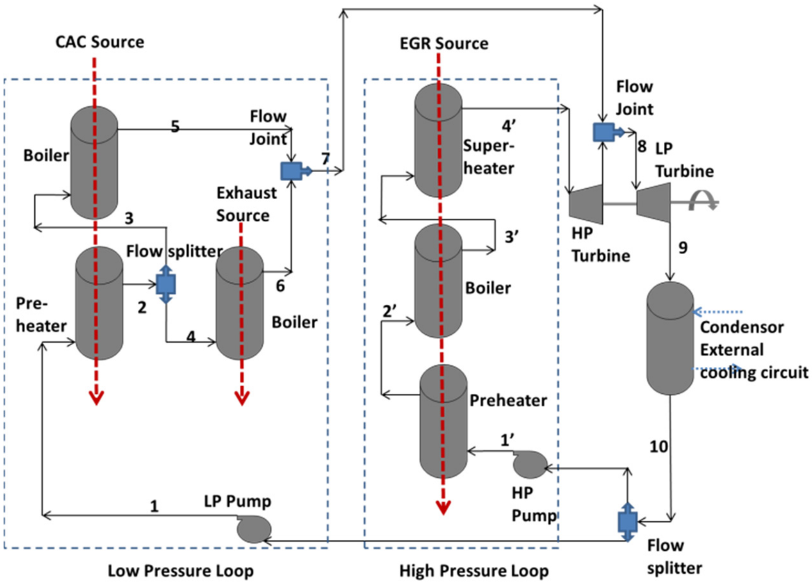


Figure 11.
Dual loop Rankine cycle

3.2 Humid Air Motor

3.2.1 Simulation

Water/steam injection is a promising technology which could be used for both emission reduction and efficiency improvement. While researchers have already studied emissions when operating with in-cylinder water injection (U. Asad, 2012) (Bedford, 2000) (Udayakumar, 2003) (Rahai, 2011) and water-fuel emulsification (Nazha, 2001) (Hountalas, 2007) (Iwai, 2011) technologies, there have been very few investigations of the efficiency improvement in water/steam injection (Kaneko, 2002) for diesel engines. When the water is directly applied to the engine, the mixture becomes undesirably heterogeneous; by using evaporative cycles, the steam, and air mixture become homogeneous (Rosen, Evaporative gas turbine cycles – A thermodynamic evaluation of their potential, 1993). One such steam injection evaporative cycle is the humid air motor (HAM). HAM is not a new concept; it was developed and patented by Per Rosen of Lund University over 20 years ago, with the idea of increasing the electrical efficiency of gas turbines and reducing emissions in reciprocating combustion engines (Lindquist, 2001). Humid air cycles have been successfully demonstrated in gas turbines (Traverso, 2010) (Thern. M., 2003) (Nyberg. B., 2012), and the HAM technology has been successfully tested in a marine application for emission reduction. HAM is also a waste heat recovery technology as it uses the waste heat in the engine cycle for steam production. The objective of this work is to analyze the potential efficiency that a humid air cycle can create in reciprocating heavy duty diesel engines.

3.2.1.1 Simulation setup

The primary advantage of HAM is derived from the reduction of pumping losses, although thermodynamic efficiency might also be improved. This study mainly focuses on the pumping aspect, and for the sake of simplicity the combustion parameters are kept constant. The turbocharger map in the baseline Volvo model was originally developed for the requirements of the production engine, and so applying the same turbocharger model for HAM will affect the turbocharger performance. The turbine and compressor work have been calculated with constant efficiency to neglect the effect of turbocharger efficiency.

3.2.1.2 Layout of HAM cycle

The HAM cycle can be made as a closed loop cycle meaning no additional water would be required for the functioning of HAM; rather the vapor is condensed from the exhaust and can be made to recycle. The layout in the Figure12 shows such cycle.

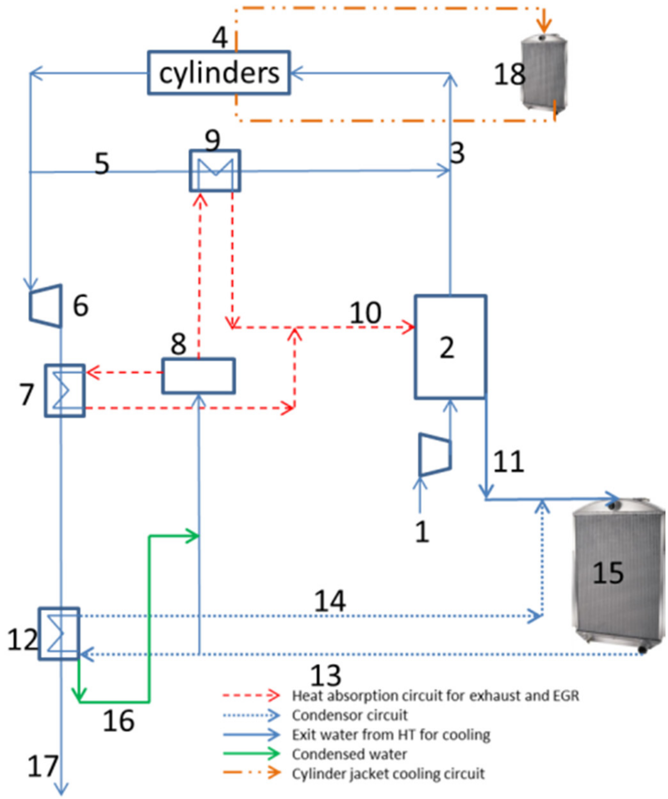


Figure 12.
Theoretical HAM cycle layout

3.2.1.3 Principle of HAM cycle

The working principle of the HAM is shown in Figure 12. The inlet air (1) enters the humidification tower (HT) (2) where it makes contact with the hot water (10), and the resulting humid air (3) is fed through an inlet to the engine (4). The engine exhaust is divided into two parts; a short-route EGR (5) and exhaust which expands at the turbine (6). Downstream of the turbine is the first heat exchanger (7), through which water is sent from the tank (8) to take the exhaust heat. Heat is also taken off from the EGR cooler (9). Both water streams are heated to 90°C and sent to the HT. The exhaust stream, after flowing through the first heat exchanger (7), continues to flow through the second heat exchanger (12), which is a condenser; here, the exhaust is further cooled with a radiator/dedicated heat exchanger (15), and the condensed water (16) is again fed to the water circulation tank (8).

3.2.1.4 Thermodynamic model of the humidification tower

The three driving mechanisms behind the water-to-vapor transformation (Rosen, Evaporative gas turbine cycles – A thermodynamic evaluation of their potential, 1993) are: 1) flashing caused by the higher temperature and pressure of the injected water in comparison to the conditions prevailing in the tower, 2) vaporization through cooling of the air, and 3) enthalpy exchange between water and air caused by the temperature drop between the water inlet and outlet of the tower. The enthalpy exchange is the main source of evaporation.

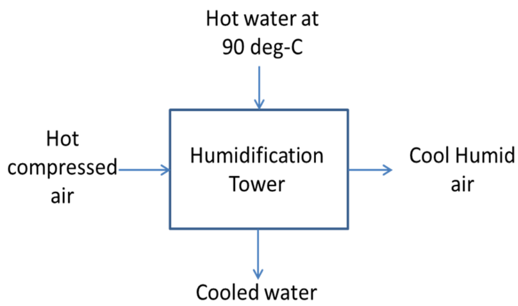


Figure 13.
Simplified humidification tower illustration

In the HT, heat and mass transfer take place between water and air as sensible heat and latent heat. The mass flows, and temperatures of water vapor are predicted with a thermodynamic model of the HT, based on a model by Rosen (Rosen, Evaporative gas turbine cycles – A thermodynamic evaluation of their potential, 1993) and built in MATLAB. Figure 13 shows the simplified humidification diagram for counter-current operation. The temperature and mass flows of the inlet air and inlet water are known. Firstly, an assumption is made regarding the relative humidity of the vapor exiting the HT; on large towers, this is typically close to 100% (Traverso, 2010). Experiments conducted at Lund University have shown that the dynamic performance of the humidification tower model is fast (Thern. M., 2003). Humidification tower models have also been verified by static experiments. Secondly, a relation connecting the departing water temperature with the inlet air temperature is formed with pinch point. This pinch point is the temperature between the lowest possible temperature of the exiting water (saturation temperature of the incoming compressed air) and the incoming compressed air (between points 2 and 3 in Figure 14). A pinch point of 5°C is set to account for the inefficiencies connected with the limitation of physical properties (Rosen, Evaporative gas turbine cycles – A thermodynamic evaluation of their potential, 1993) (Nyberg. B., 2012).

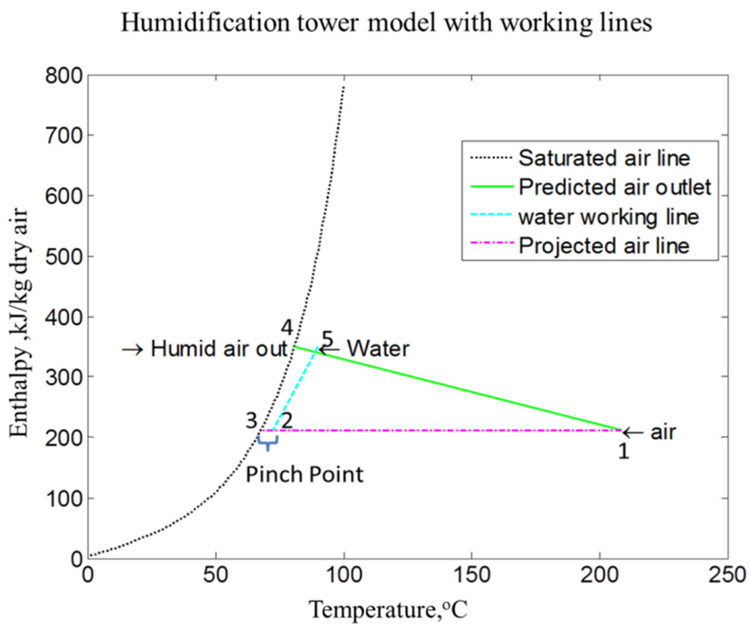


Figure 14.
Humidification tower model with working lines

4. Internal combustion engine analysis

4.1 Heat release analysis

The rate of combustion or rate of heat release (RoHR) inside the cylinders can be calculated with the pressure data from the cylinders. The combustion rate can be calculated by the pressure change with volume along with the heat transfer, crevice losses and blow-by losses. The combustion cycle is considered to be a thermodynamically closed cycle to calculate the rate of heat release.

From the first law of thermodynamics, the energy added into a system can alter the internal energy *(U)* of the system and may produce work *(W)*.

$$\frac{dQ}{dt} = \frac{dU}{dt} + \frac{dW}{dt} \tag{3}$$

Where dQ/dt is the added heat to the system (in the engine this is the fuel energy), dW/dt is the rate of work done, and dU/dt is the change in the internal energy. Internal energy U can be expressed as:

$$U = mC_vT \tag{4}$$

where m is the mass in the system, Cv is the specific heat capacity at constant volume and T is the temperature of the system. Since it is a closed system, the mass is constant and hence the change in the internal energy is expressed as:

$$\frac{dU}{dt} = mC_v\frac{dT}{dt} \tag{5}$$

The gas is considered to be ideal gas, using the ideal gas law

$$pV = mRT \tag{6}$$

p is pressure, V is volume, m is the moles of gas, R is the gas constant, and T is the temperature. Derivative of ideal gas law gives:

$$p\frac{dv}{dt} + V\frac{dp}{dt} = mR\frac{dT}{dt} \tag{7}$$

Substituting equation (6) in equation (7) for mR.

$$p\frac{dv}{dt} + V\frac{dp}{dt} = \frac{pV}{T}\frac{dT}{dt} \tag{8}$$

Combining the ideal gas law equation (5) and internal energy equation (5) for m gives:

$$\frac{dU}{dt} = \frac{p\,V}{RT}C_v\frac{dT}{dt} \tag{9}$$

Substituting equation (8) in (9).

$$\frac{dQ}{dt} = \frac{C_v}{R}\left(p\frac{dv}{dt} + v\frac{dp}{dt}\right) \tag{10}$$

The specific gas constant R can be expressed as where Cp is the specific heat capacity at constant pressure.

$$C_p - C_v = R \tag{11}$$

The ratio of specific heat capacity, γ is given by:

$$\gamma = \frac{C_p}{C_v} \tag{12}$$

Combining equations.10, 11, and 12

$$\frac{dQ}{dt} = \frac{\gamma}{\gamma - 1}p\frac{dv}{dt} + \frac{1}{\gamma - 1}V\frac{dp}{dt} \tag{13}$$

In engines, it is practical to use crank resolved data (CAD, θ) and hence the rate of heat released can be expressed as:

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dv}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \tag{14}$$

Equation (14) is called apparent heat release, but the chemical energy from the fuel is also lost in the cylinder in the form of heat losses, crevice losses and blow-by losses. Crevice and blow-by effects are very less and hence neglected in experiments. Convective heat transfer of the engine is calculated by:

$$Q_{ht} = h \,.\, A_{surface} .\, (T_{gas} - T_{Surface}) \tag{15}$$

Qht is the convective heat transfer, h is the heat transfer coefficient, A is the surface of the wall and piston head, T is the temperature. The heat transfer coefficient is calculated using the Woschini model (Woschni, 1967).

$$h\ (W/m^2 \,.\, K) = \ 3.26 B(m)^{-0.2}\ p(kPa)^{0.8}\ T(K)^{-0.55}\ w\ (m/s)^{0.8} \tag{16}$$

h is the convective heat transfer coefficient, *B* is the diameter of the bore of the cylinder, *p* is pressure, *T* is temperature, *w* is the average cylinder gas velocity, which can be expressed as

$$w = \left[C_1 \bar{S}_p + C_2 \frac{V_d T_r}{p_r V_r} (p - p_m) \right] \tag{17}$$

C1 and *C2* are constants and can be adjusted to suit a particular engine (nominal values for combustion and expansion is *C1=2.28, C2 =3.24x10^3* (Woschni, 1967), *Sp* is the mean piston speed, *Vd* is the displacement volume, *p* is the instantaneous pressure, *T* is the temperature, and *Pr*, *Vr*, *Tr* are the pressure, volume and temperature of working fluid taken at reference state (inlet valve closing or start of injection). The instantaneous gas temperature can be calculated from the ideal gas law. The specific heat ratio was calculated with the Gatowski model (J.A. Gatowski, 1984) which calculates the ratio of specific heats as a linear function. It can be expressed as:

$$\gamma = \ \gamma_{300} + b\ (T - 300) \tag{18}$$

where γ is the specific heat ratio, b is a constant (0.08 is used in experiments) and T is the gas temperature. Including the heat transfer the chemical energy released by fuel can be written as:

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dv}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \tag{19}$$

4.2 Energy utilization

The entire fuel energy is not utilized in the engine. There are numerous losses between the input fuel and output shaft power. The energy split of the fuel can also be expressed in terms of mean effective pressure. Mean effective pressure is a useful measure as it is a function of work produced by the displacement volume which enables to compare different engines irrespective of the sizes. Figure 15 shows the Sankey diagram of energy split. The input is the total fuel energy (*FuelMEP*), and the output is the useful power to the wheels.

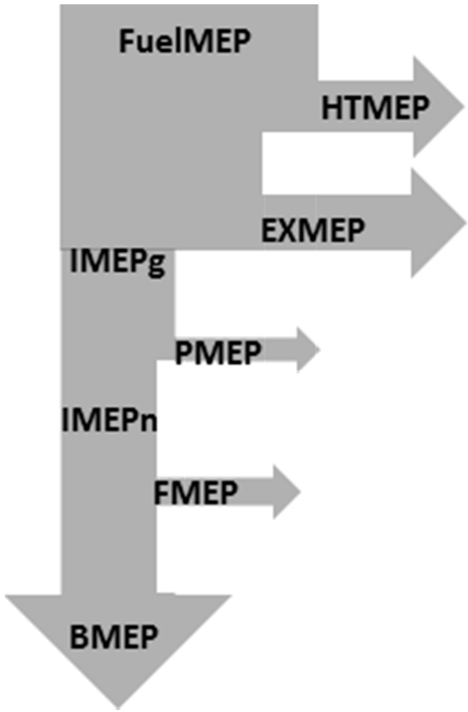


Figure 15.
Energy transformations in mean effective pressures [bar]

Fuel mean effective pressure (*FuelMEP*) is expressed as:

$$FuelMEP = \frac{m_f \, Q_{LHV}}{V_d} \tag{20}$$

m_f is the mass flow of fuel, Q_{LHV} is the fuel's lower heating value, and V_d *is the* displacement volume. The heating value of diesel fuel used in the experiments is 43 MJ/kg. Heat released mean effective pressure (*QMEP*) is the combustion process generating heat. Heat transfer mean effective pressure (HTMEP) and exhaust mean effective pressure (EXMEP) are the heat losses.

From the pressure measurements (as a function of CAD) the work done in a cylinder can be calculated. *IMEP* is indicated mean effective pressure can be expressed as the work per displaced volume.

$$IMEP = \frac{W_i}{V_d} \tag{21}$$

IMEP can be differentiated into gross and net. *IMEPg* is the gross indicated mean effective pressure, which measures the work done in the cylinder over the compression and expansion stroke.

$$IMEPg = \frac{1}{V_d}\int_0^{360} pdV \tag{22}$$

IMEPn is net indicated mean effective pressure which includes all four strokes in the cycle.

$$IMEPn = \frac{1}{V_d}\int_0^{720} pdV \tag{23}$$

PMEP is the pumping losses which occur over inducting and expulsion phase. *PMEP* can be simplified as a difference between inlet and exhaust pressure. *FMEP* is the friction mean effective pressure; friction can be evaluated with friction models (Chen S. K.). Brake Mean Effective Pressure (*BMEP*) is the engine output. *BMEP* is the useful energy produced by the engine per operating cycle divided by displacement volume. Since the brake is used to evaluate the power, the term brake is attached to the MEP.

$$BMEP = \frac{W_b}{V_d} \tag{24}$$

The thermodynamic efficiency shows the conversion of heat to work on the piston, which can be expressed as:

$$\eta_T = \frac{IMEPg}{QMEP} \tag{25}$$

The gas exchange efficiency measures the effectiveness with which the exhaust gasses are replaced with fresh charge/air, which can be expressed as:

$$\eta_{GE} = \frac{IMEPn}{IMEPg} \tag{26}$$

An alternative way of measuring the effectiveness of the engine's induction process is of volumetric efficiency. Volumetric efficiency can be above 100 % and can be expressed as:

$$\eta_V = \frac{m_a}{\rho_{in}\dot{V}d} \tag{27}$$

ρ is the density of the air.

4.3 Emissions

Vehicular emissions lead to global warming by emitting large amounts of carbon dioxide and also carry serious health hazards to human beings. Emissions from internal combustion engines lead to a variety of health ailments. Nitrogen oxides emissions aids in ozone production which are harmful to both the environment and human respiratory system. Carbon monoxide, Sulfur dioxide, and particulate matter lead to various cardiovascular problems. Emissions generated from internal combustion engines can be deadly as it can lead to cancer (Marie-Élise, 2006) (Kennedy, 2007). Hence, it becomes vital to control the emissions from the internal combustion engines. For the emission control in the engine, lambda serves an essential parameter, which can be measured using a lambda sensor or calculated from the exhaust emissions (Heywood, 1988).

Table 2.
EU Emission norms for HD engine

Stage	Date	Test	CO	HC	NOx	PM	PN	Smoke
			g/kWh				1/kWh	1/m
Euro I	1992, ≤ 85 kW	ECE R-49	4.5	1.1	8.0	0.612		
	1992, > 85 kW		4.5	1.1	8.0	0.36		
Euro II	1996.10		4.0	1.1	7.0	0.25		
	1998.10		4.0	1.1	7.0	0.15		
Euro III	*1999.10 EEV only*	ESC & ELR	*1.5*	*0.25*	*2.0*	*0.02*		*0.15*
	2000.10		2.1	0.66	5.0	0.10[a]		0.8
Euro IV	2005.10		1.5	0.46	3.5	0.02		0.5
Euro V	2008.10		1.5	0.46	2.0	0.02		0.5
Euro VI	2013.01	WHSC	1.5	0.13	0.40	0.01	8.0×10^{11}	

a - PM = 0.13 g/kWh for engines < 0.75 dm^3 swept volume per cylinder and a rated power speed > 3000 min^{-1}

4.3.1 Nitrogen Oxides

Nitric oxide (NO) and nitrogen dioxide (NO_2) are grouped together as NOx emissions. NOx emissions are formed at high temperatures when atmospheric nitrogen combines with oxygen. Thus, NO formation is strongly dependent on the temperature and oxygen availability. Current technologies use exhaust gas recirculation (EGR) to reduce NOx emissions. EGR is burnt gas which mainly consists of carbon dioxide which is inert in the combustion reaction and thus reducing the peak flame temperatures. Along with the oxygen availability and temperature, the initial fuel mixing is also an important factor in NOx formation.

The mechanism of nitrogen oxide formation is given by Zeldovich, which is expressed as:

$$O + N_2 = NO + N \tag{28}$$

$$N + O_2 = NO + O \tag{29}$$

The OH molecule formed during combustion also significantly contributes to the NO formation; the equation is given by Lavoie.

$$N + OH = NO + H \tag{30}$$

NO_2 formation in diesel engines in relation to NO formation is usually higher than for SI engines (Heywood, 1988). A possible mechanism for NO_2 formation is given by

$$NO + HO_2 = NO_2 + OH \tag{31}$$

Equation 31 could be converted back into NO, unless the NO_2 formed is quenched by cooler fluid or cold regions. This effect is predominant at low loads in diesel operated engines where the cooler regions are widespread (Heywood, 1988).

$$NO_2 + O = NO + O_2 \tag{32}$$

4.3.2 Carbon monoxide

Carbon monoxide (CO) emissions are primarily formed at low air-fuel ratios. For fuel-rich mixtures, the CO emissions increase steadily with increasing equivalence ratio (Heywood, 1988). This becomes important for spark-ignition engines, which is operated at near stoichiometric conditions. Diesel engines usually are operated on the lean side and hence the CO emission is less important for diesel operation (Heywood, 1988). CO formation mechanism is given by hydrocarbon combustion mechanism which is given as:

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO \tag{33}$$

R is the hydrocarbon radical. The CO can be oxidized to CO_2 at lower rate.

4.3.3 Unburned hydrocarbon emissions

Unburned hydrocarbons (UHC or HC) mostly consist of unburnt fuel due to the combustion inefficiency. Low load operation and also fuel rich operation can produce higher UHC. Cylinder misfiring, improper combustion and cyclic variations also lead to unburned hydrocarbon emissions.

4.3.4 Particulate emissions

Particulate matter (PM) emissions from diesel engines are mainly combustion generated carbonaceous material, also known as soot. Most PM emissions are caused due to incomplete combustion of a hydrocarbon fuel.

5. Experimental Setup

5.1 Volvo D13 Engine

All experiments were performed in the combustion engine research lab facility at Lund University. A Volvo heavy-duty engine was used during the course of the project. The engine was a standard production engine adapted with some modifications both in hardware and instrumentation for research (Henningsson, 2012).

The engine was earlier used for dual fuel study. Delphi E3 unit injectors were used for direct injection and only one injection event was possible per cycle. A modified ECU interface allowed the start of injection, injection duration, and fuel injection pressure to be set from the computer using CAN communication.

5.2 Instrumentation

5.2.1 Gas flow management

The engine was equipped with a long-route (low pressure) EGR system and a variable geometry turbocharger (VGT) from Holset. The EGR was controlled with two servo operated valves, namely, one Exhaust back pressure valve (EBP) and an exhaust recirculation valve (EGR). Servo motors from JVL Industri Elektronik A/S of type MAC140-A1 were used to actuate the valves and controlled by the computer using serial communication. The turbocharger actuator was controlled using a CAN interface to the central computer.

The engine speed was governed by an ABB M2BA 355 electrical motor with a rated power of 355 kW.

5.2.2 Sensing

The engine layout with sensor locations is shown in Figure 16.

Synchronization: A Leine and Linde crank angle encoder was attached to the crankshaft to measure crank angle based sampling. The encoder gives out 5 pulses per crank angle degree. This enabled the cylinder pressure to have a resolution of 0.2 CAD. An optical camshaft sensor was used to synchronize the engine events.

Water cooled Piezo-electrical pressure sensors with a range of 0-250 bar (Kistler 7061B) were used for In-cylinder pressure measurements. The pressure sensors were calibrated with an in-house calibration tool.

Pressures and temperatures were measured at different locations on the engine and on the subsystems. The sensors were of type Keller PAA-21S and had measurement ranges of either 0–5 bar or 0–10 bar.

Pressure transducers for measuring pressure outside the cylinder were used for

- Pressure after compressor
- Inlet manifold pressure
- Exhaust manifold pressure
- Pressure after turbine
- Engine oil pressure
- Fuel pressure to the engine

K-type thermocouple from Pentronic was used as temperature sensors. Temperature sensors were used for

- Temperature measurement after compressor
- Temperature in/out heat exchangers
- Temperature in/out humidification tower
- Temperature of gas at inlet/exhaust
- Temperature after turbine
- Temperature from individual cylinders
- Oil temperature

A torque measurement sensor (HBM) was used to measure the torque from the dynamometer while the engine speed was measured from the engine crankshaft encoder. An optical sensor was used to gauge the speed of the turbocharger shaft.

Fuel flow was computed from the weight measurements from a fuel scale. An AVL 7131-09E fuel balance was used for fuel weighing.

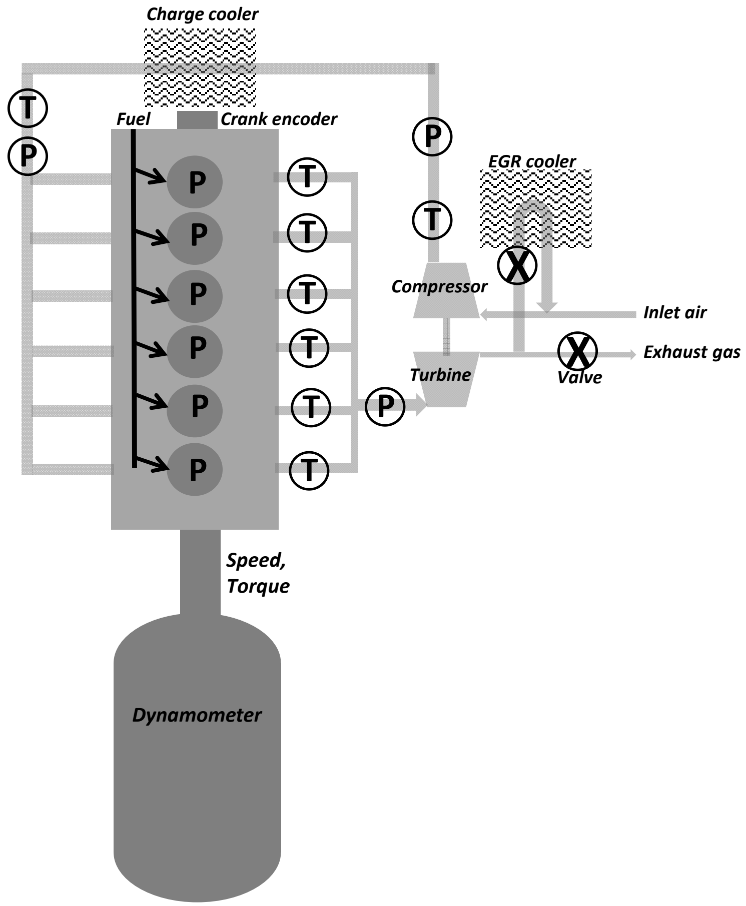


Figure 16.
Engine Layout with valves, pressure, and temperature sensors.

5.3 Data Acquisition

5.3.1 Control system

The control system based on C/C++ was developed by (Strandh, 2006) and (Henningsson, 2012). The control system enables to control the injectors and Variable geometry turbine position via modified CAN communication.

5.3.2 Hardware

The program (DAPMEAS) runs on a Linux PC. A Microstar DAP 5400a/627 PCI data acquisition board was used for the high-frequency sampling. The board had 16 analog input channels with eight 1.25 MHz AD-converters with 14 bits resolution and was used for in-cylinder pressures. Crankshaft encoder was used to trigger every 0.2 CAD.

A PCIcanX II card from Kvaser was used for communication with engine ECU and VGT.

Communication with electrical servos used for EGR, EBP, and VGT actuation were handled through the serial port of the central computer. An HP 2852A data logger was used to log temperatures, pressures, fuel scale weights, engine torque, and turbo and engine speed with a frequency of 0.4 Hz. The logger was connected to a second computer and communicated with the main computer over Ethernet. Emission measurement systems were connected to the main computer through Ethernet.

5.3.3 Software

The control system responsible for handling communication with sensors and actuators, logging data and interaction with GUI. The GUI was developed with Python (Henningsson, 2012).

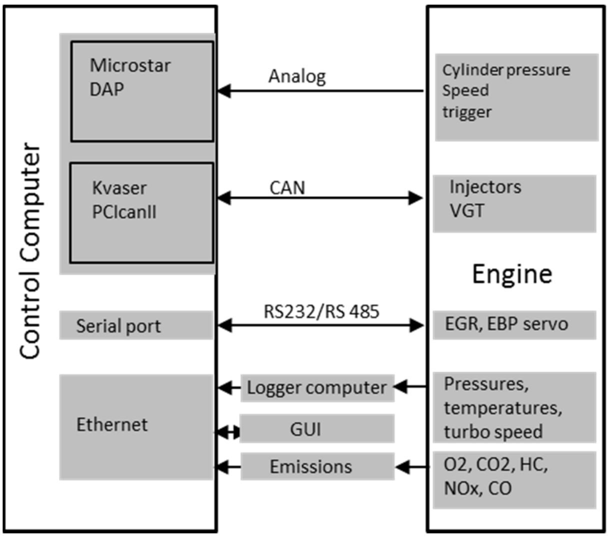


Figure 17.
Engine communication

6. Results

6.1 Rankine simulation (Paper I and II)

Table 3 shows the potential of all the different configurations explained in the methodology chapter. For the engine considered in this study, it was most economical to recuperate only the EGR source, which gave a Rankine cycle efficiency of 22% and used only four heat exchangers (including condensers). On the other hand, the dual loop Rankine cycle could recover more heat, but had only 15% cycle efficiency with additional heat exchangers and turbine. Although the power output was lower when only the EGR source was used, the cycle efficiency (Power turbine/Q̇used) was higher. To recuperate higher quantity of heat energy from the CAC and exhaust sources, a suitable way can be to use a working fluid that has a condensation temperature close to the ambient temperature at ambient pressure and the critical point temperature close to the temperature of the exhaust and CAC source.

Table 3.
Heat Utilized in different configurations

Sources	$\dot{Q}_{in}$	$\dot{Q}_{out}$	$\dot{Q}_{used}$	$P_{turbine}$	$P_{turbine}/\dot{Q}_{used}$
	kW	kW	kW	kW	%
EGR	96	44	52	11	22
Exhaust	108	78	29	2	9
EGR, Exhaust	96 108	40 85	55 22 77	10	13
EGR, Exhaust, CAC	96.2 108 90.6	36 87 69	59 20 20 100	11	11
EGR, Exhaust, CAC, dual loop	96.2 108 90.6	42 79 68	53 28 22 104	16	15

Table 4 shows the percentage power improvement over the engine power at operating point C75. The EGR only case and all sources combined produce similar improvement. It was noted that quality of the heat was more important than the quantity of heat energy.

Table 4.
Power produced with Rankine cycle for different source combinations

Sources	Turbine Power, kW	Total power	% Power Increase
EGR	11	286	4,1
Exhaust	2.7	277	0,9
EGR, Exhaust	10	285	3,7
EGR, Exhaust, CAC	11.8	286	4,1
EGR, Exhaust, CAC-Dual loop	16,3	291	5,6

It's known that the energy carrying capacity of a fluid increases with temperature. From Figure 18, it can be seen that the effective energy used from the energy carrying fluid also depends on these temperatures. This effective energy takes part in the Rankine cycle energy conversion. Thus, the higher the temperatures are, the higher the power output will be for the energy utilized.

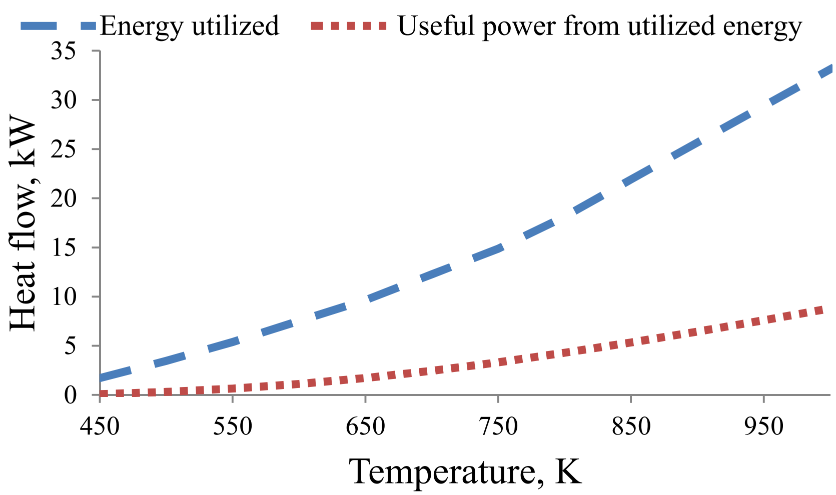


Figure 18.
Useful power vs. temperature

6.1.1 Waste heat utilization at different load points

The waste heat recovery using a Rankine cycle proves to be significant. The evaluation was extended for other load points to analyze the benefits. An analysis similar to what was carried out for C75 was made for different load points– 25 %, 50 %, 75 %, and 100 % in speed range B (1500 RPM). For the operating point B25 the inlet charge pressure was low and thus the charge temperature was too low to allow any recuperation and hence neglected. The results of the analysis, presented in Figure 19, shows that for operating points B75, B50, and B25 the power generated with only EGR as the source was almost equivalent to all the sources combined, confirming the results of the operating point C75 presented in the previous section.

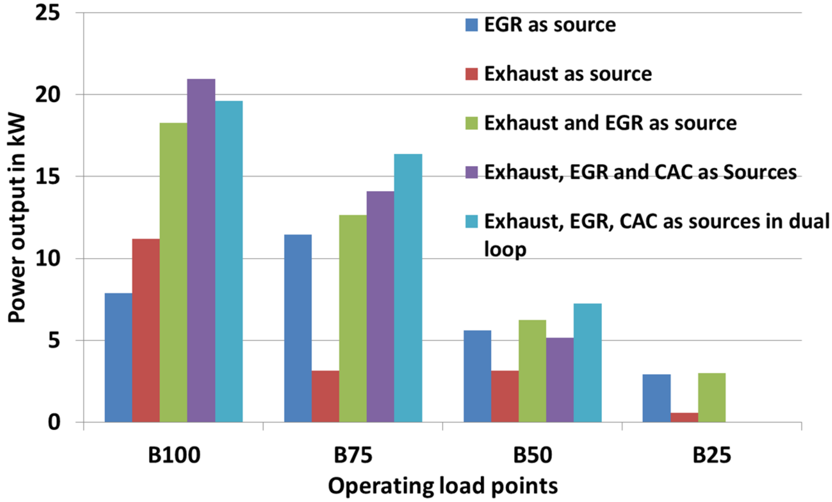


Figure 19.
Rankine cycle output for different loads at B speed

6.1.2 Rankine cycle heat recovery with PPC and conventional diesel combustion

The EGR from conventional diesel combustion operation had higher temperatures, but reduced mass flows when compared to operation with PPC. For both combustion methods, Rankine cycle was employed to recuperate the heat energy from their EGR source, and analysis was carried out to determine which of these two combustion strategies that have a higher potential for waste heat recovery. EGR temperatures from both combustion types were given as input to the Rankine cycle. For the conventional diesel operation, the operating points A75, B75 and C75 were chosen as the IMEP was similar to the PPC as shown in Table 5.

Table 5.
Operating characteristics of conventional and PPC

Combustion type	IMEP (Gross)	Speed	Power	Indicated efficiency (Gross)	Mass flow (EGR)	Temperature (EGR)
	Bar	RPM	kW		g/s	K
Conventional, C75	17	1798	275	45	182	744
Conventional, B75	20	1504	280	47.8	152	774
Conventional, A75	19.5	1211	228	47	96	795
PPC	20.5	1250	225	53	276	677

From Table 6 it can be seen that the PPC combustion with combined cycle has power output equal to that of a conventional engine. The reduced used energy percentage can be attributed to the lower output temperature of the PPC.

Table 6.
Results of IPSEpro simulation for conventional and PPC

Combustion	EGR inlet temperature	Cycle pressure	Heat energy used [1]	Mass flow of working fluid	Useful power out	Power to Used energy
	K	bar	kW	g/s	kW	%
Conventional C75	744	60	52.5	20	11.7	22
Conventional B75	774	60	49.8	17	11.5	23
Conventional A75	795	60	35.9	14	8.0	22
PPC	677	35	59	24	11.6	20

This additional power produced by WHR system corresponds to a BSFC reduction of 3.4 % - 4.0 % for the conventional combustion engine (75 % load) and 4.9 % reduction for the PPC engine. Even with comparable power output from the WHR system, the net waste heat recovery percentage was greater for PPC than for a conventional combustion.

[1] Reference temperature in enthalpy calculation in IPSEpro was 273 K

6.2 Humid Air Simulation (Paper III)

Figure 20 shows the inlet mass fraction of air and vapor; it can be seen that the vapor mass flow was around 10% of the air mass into the engine. This mass flow which was not compressed by the compressor could be expanded over the turbine producing additional work which improves the overall engine efficiency.

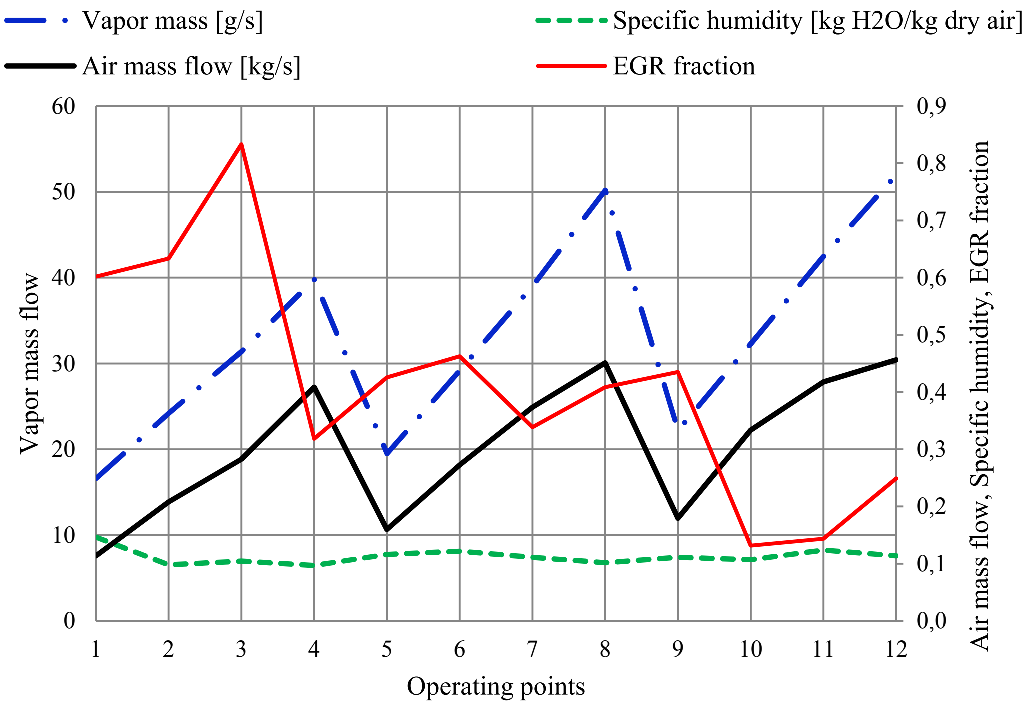


Figure 20.
Inlet charges mass fractions

Figure 21 shows the comparison of in-cylinder pressures for different models at operating point A100 as explained in the methodology section. The HAM and HAM+EGR models had additional water vapor mass flow to pump in along with the air mass. HAM+EGR has higher inlet temperature compared to the baseline condition (CDC40) which required additional boost pressure at the intake and resulted in higher pressures than the rest of the models. It can be seen that the pressure of HAM closely followed the baseline condition. CDC80 has a higher temperature at the inlet compared to the baseline, and thus required a higher boost pressure at the inlet for the same mass flow. Hence, the pressure level of CDC80 was higher compared to the baseline.

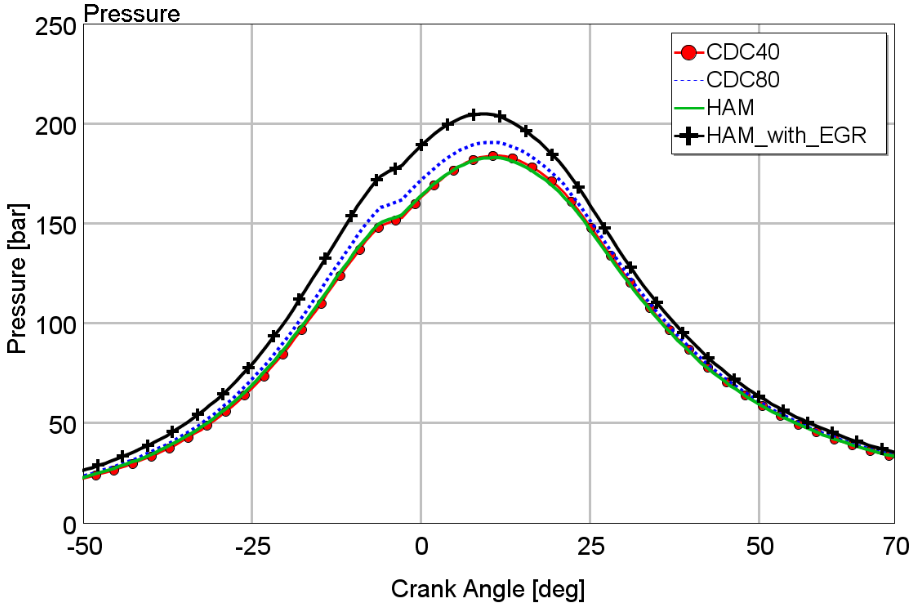


Figure 21.
In-cylinder pressure

Figure 22 shows the global in-cylinder temperature of different models at the operating point A100. CDC80 has higher inlet temperature and thus higher peak temperature. The HAM model with water vapor has almost the same peak temperature as the baseline case (which has EGR); showing the possibility of EGR removal. HAM+EGR have had both the EGR and water vapor, and hence lower temperatures than the other models.

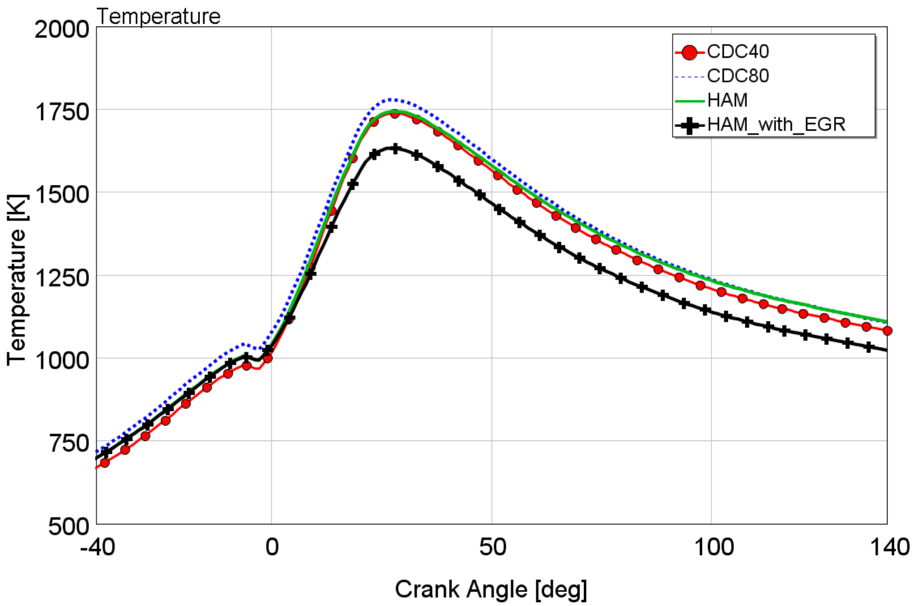


Figure 22.
In-cylinder temperature

A humid air system (HAM) that was studied using system model simulations show clear potential for improving efficiency. The net benefit was calculated by summation of the in-cylinder power, compression work and turbine expansion; this was then used to compare the three models explained in the methodology chapter. It was only reasonable to compare the engine models at similar inlet temperatures. Hence, a conventional engine with inlet temperature similar to HAM (CDC80) was considered for comparison. CDC80, HAM, and HAM+EGR are compared to see the potential benefits; Figure 23 shows the possible benefit of HAM and HAM+EGR over CDC80. At lower speed and loads (A25 and A50), CDC80 was operated with higher EGR amounts that gives a heat transfer advantage over HAM; hence, HAM has lower benefits at these operating points. At higher speeds, HAM delivered greater benefits compared to CDC80.

The HAM system reduced the in-cylinder temperatures compared to EGR. At moderate and high-speed operation, where EGR mass was not too high, HAM was beneficial over CDC80. A combination of factors brought the increment in efficiency: 1) lower heat losses, 2) reduction of in-cylinder pumping losses, and 3) free vapor expansion over the turbine.

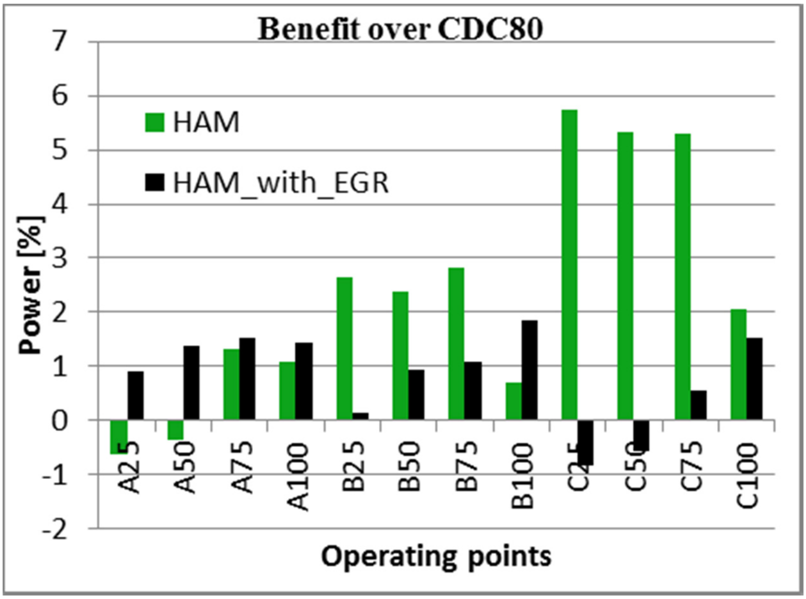


Figure 23.
Net power benefit

6.3 HAM Experiments (Paper IV and V)

HAM experiments are conducted as explained in the methodology chapter. The following sections describe the emission and efficiency potential.

6.3.1 Emission

6.3.1.1 NOx emission

Figure 24 shows a temperature-equivalence (T-ϕ) ratio diagram to comprehend how the peak flame temperature changes for diesel like combustion. A 2ms time interval was considered for the quasi-static study. The pressure was kept at 60 bar, which approximates the cylinder pressure during the onset of combustion. The three different colored lines shows the flame propagation path through the vertical soot cloud (ϕ>2) and the horizontal NOx cloud (ϕ<2) for conventional diesel combustion (base), with 10% EGR, and with 10% water vapor. From the plots, it can be seen that when the water vapor was mixed with air, the flame temperature limits entrance to the NOx and soot formation zone better than EGR case.

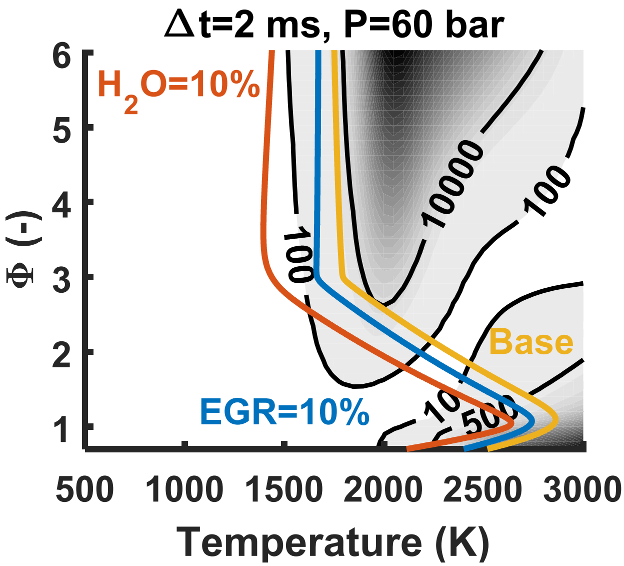


Figure 24.
Temperature-Equivalence ratio plot

From the experiments, we see in Figure 25 that the NOx reductions were significant, both with EGR and with HAM. At the lowest speed (1200 RPM) and load, the NOx produced from HAM was higher than for the EGR case. At higher speed (B25, 1500 RPM) the NOx from HAM increased and became almost equal to the EGR case and with increased load (A50), the NOx reduction potential of HAM was even higher than with EGR; this was brought by both vapor introduction and reduced ignition delay. Lower NOx production can be attributed to the lower instantaneous flame temperature brought by, the higher specific heat capacity of the vapor mixture.

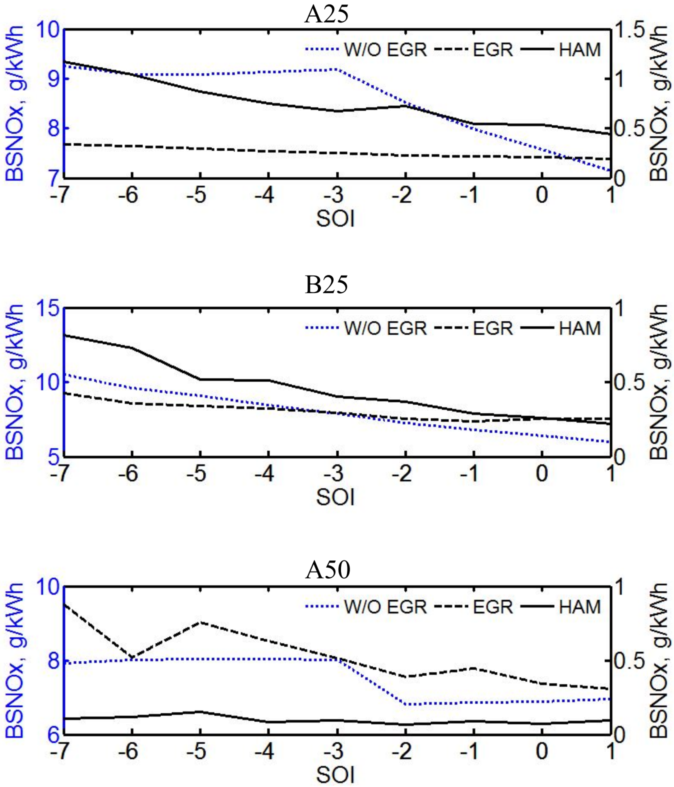


Figure 25.
BSNOx vs. SOI. Top A25, Middle B25, Bottom A50

6.3.1.2 CO Emission

From Figure 26 it can be seen that CO was much lower at low load (A25) compared to the engine operation with EGR. As the speed increased (B25), the CO increased marginally. But increased load (A50) increases CO for HAM significantly to levels comparable to the EGR case. The lower flame temperature caused by the humidification reduces the oxidation rate for CO and hence the increase in CO.

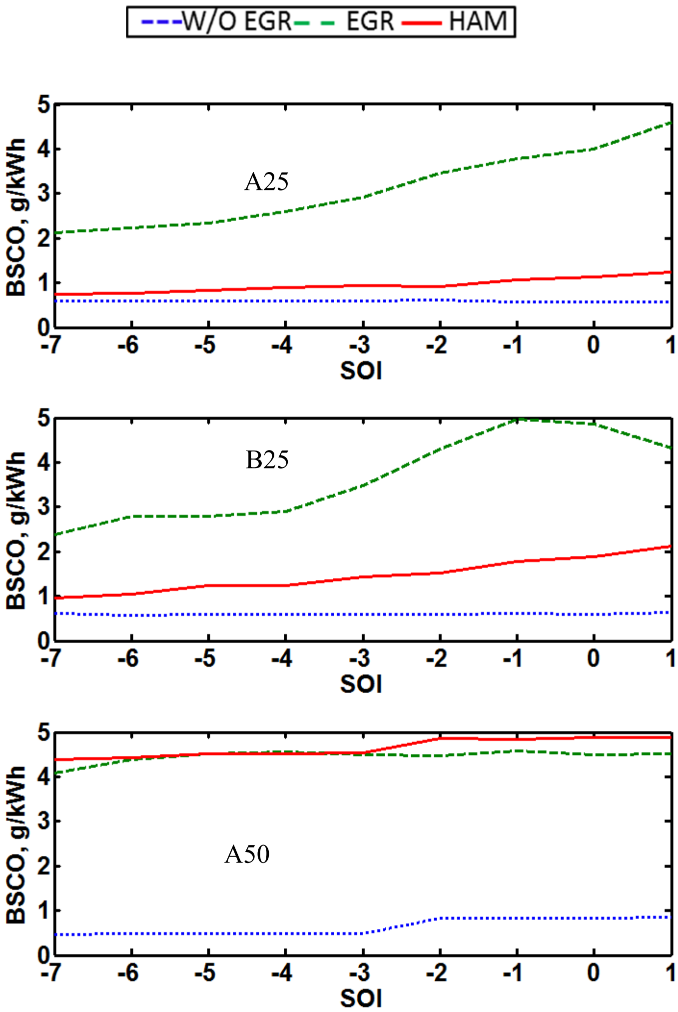


Figure 26.
BSCO vs. SOI. Top A25, Middle B25, Bottom A50

6.3.1.3 HC emission

HC emissions were noted to increase marginally compared to the EGR case at all speeds and loads (Figure 27).

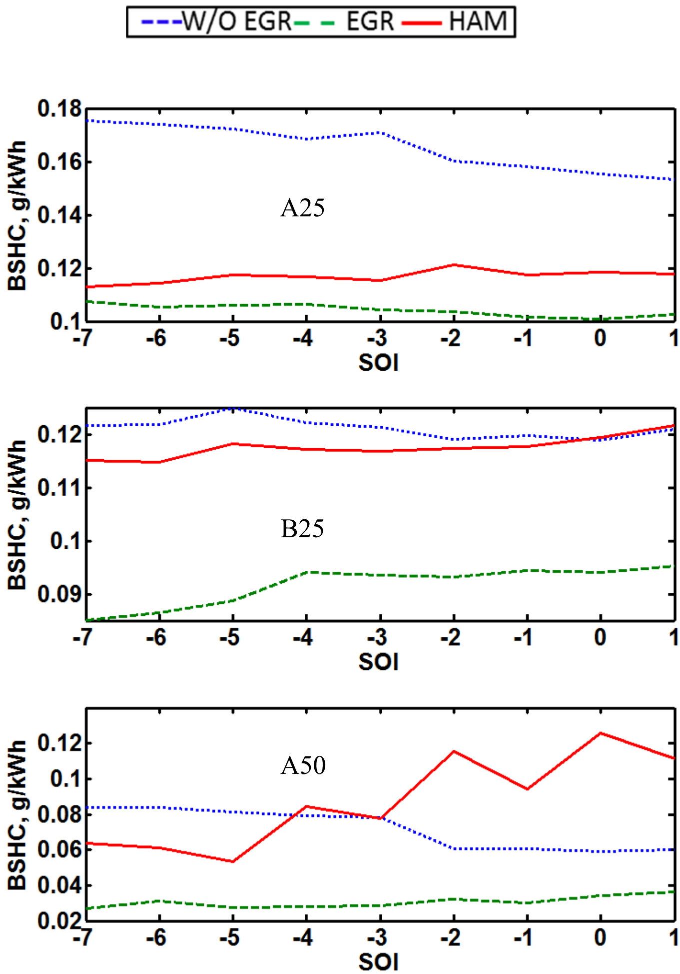


Figure 27.
BSHC vs. SOI. Top A25, Middle B25, Bottom A50

6.3.1.4 Soot Emissions

Figure 28 shows the soot emissions from the experiments. Soot emissions were reduced significantly at low load (A25). At higher load (A50) soot increases for HAM, although still on levels much lower than for the case with EGR. Soot emissions are dependent on the oxidation rate and thus, when the peak flame temperature goes down, the rate of oxidation reduces leading to increased Soot emissions.

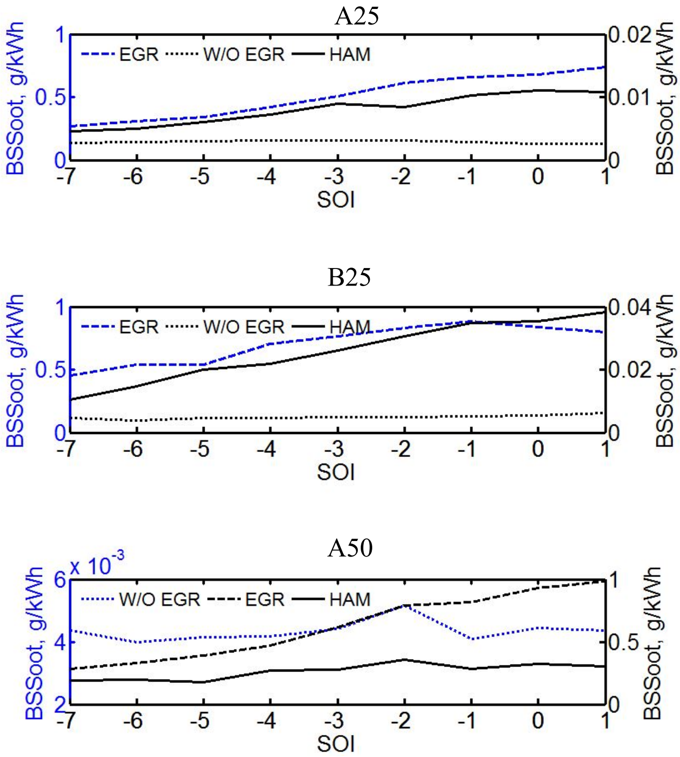


Figure 28.
BSSoot vs. SOI. Top A25, Middle B25, Bottom A50

6.3.2 Efficiency

Efficiency can be affected by numerous factors. Although some of the main parameters which influence the brake efficiency are combustion (timing), in-cylinder heat transfer, friction, pumping losses, and cooling losses in the engine. Figure 29 shows the efficiency plots of all three operating points in three modes. Point A25 indicates that the HAM mode of operation had better efficiency for most of the SOI except for the latest injection timings. For operating point B25 the efficiency of the HAM engine was clearly superior to the other modes for any SOI. At point A50, the HAM engine efficiency was slightly lower than either with or without EGR mode of operation.

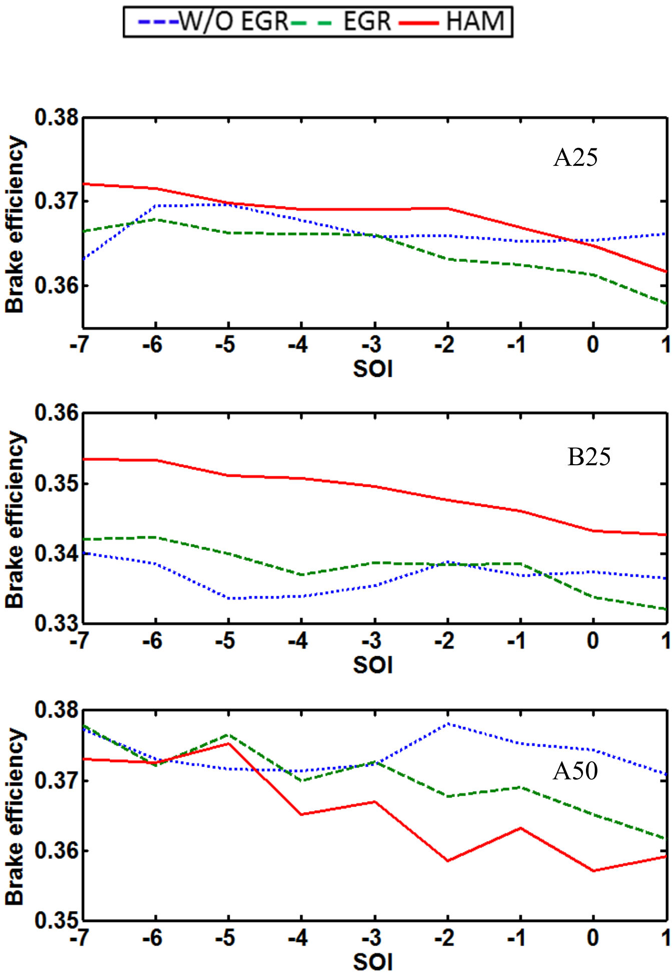


Figure 29.
Brake efficiency vs. SOI. Top A25, Middle B25, Bottom A50

The energy distributions for all the three modes of operation at three different operating points are shown in the pie charts (Figures 30, 31, 32). The comparison was made for different modes and operating points at their best brake efficiency point as a function of SOI.

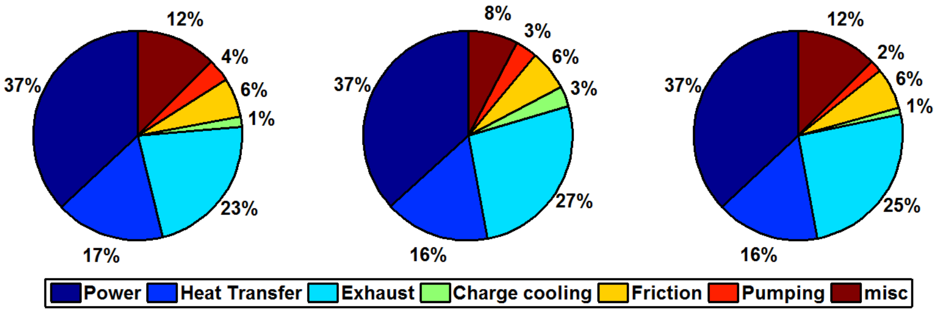


Figure 30.
Energy Split for operating point A25 (Left: Without EGR, middle: With EGR, Right: HAM)

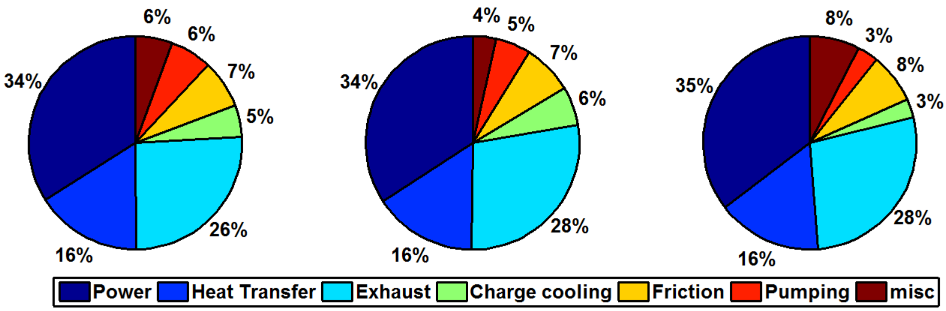


Figure 31.
Energy Split for operating point B25 (Left: Without EGR, middle: With EGR, Right: HAM)

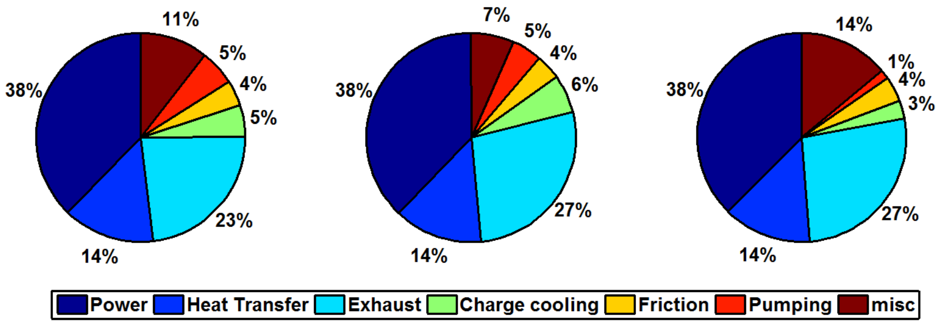


Figure 32.
Energy Split for operating point A50 (Left: Without EGR, middle: With EGR, Right: HAM)

6.3.2.1 Turbine and compressor efficiency

Figure 33 shows the isentropic turbine and compressor efficiencies which were calculated from the inlet and exit temperatures from the experiments. The turbocharger was a 16-liter Volvo HD engine turbocharger fitted on a 13-liter diesel engine. At low load engine operation (25-50% max load), the turbocharger was operated at low speeds and thus operated out of range of the design point.

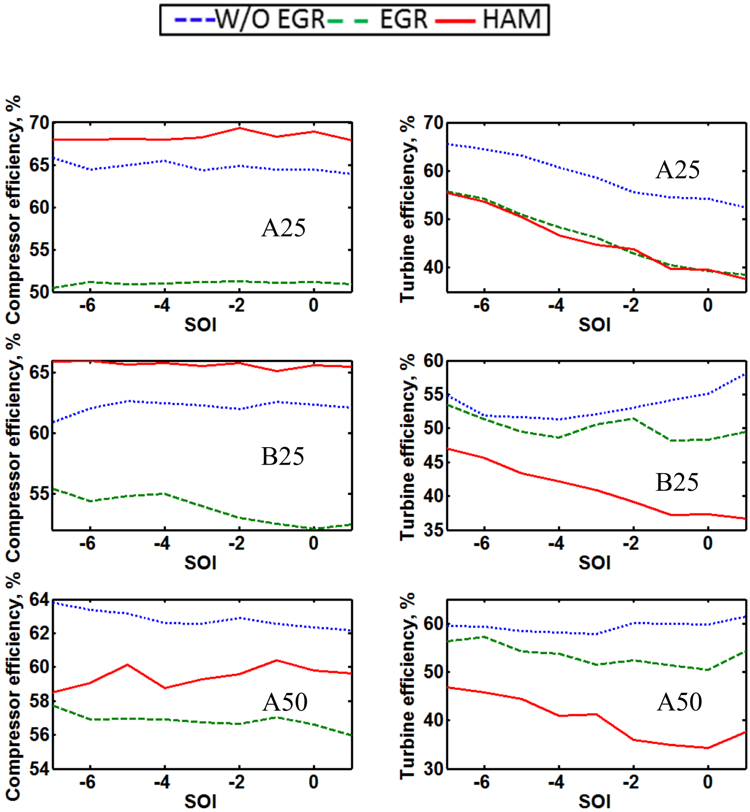


Figure 33.
Turbine and compressor efficiency. Top A25, Middle B25, Bottom A50

6.3.2.2 Variable geometry study

During the experiments, the inlet pressure was kept constant to compare at the same loads. By this, the variable geometry turbine (VGT) position was opened to let the exhaust pass through quickly and thus not extracting the potential work from the additional vapor. This could have limited the potential of the variable geometry turbine (VGT). A VGT sweep was performed at the operating point A25 with SOI -6 (best brake efficiency for the EGR case) to improve the efficiency further. The brake efficiency results presented in Figure 34 shows that varying the VGT position doesn't yield significant improvements.

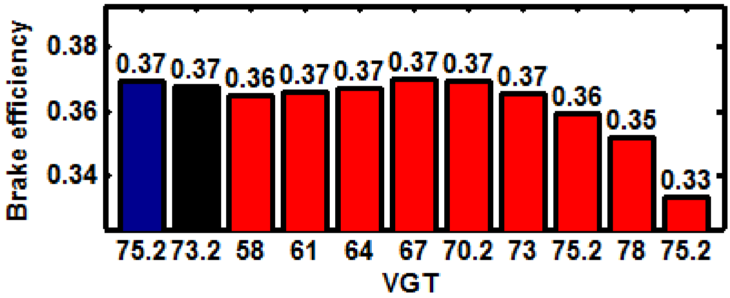


Figure 34.
Brake efficiency vs. VGT at SOI: -6 (*Blue: Without EGR, Black: With EGR, Red: HAM* VGT sweep)

7. Future Work

The work done over the limited number of load points could be further extended to other operating points. The current turbocharger design is not intended for different mass flow between turbine and compressor, which is required for the humid air cycle. Hence, turbomatching should be performed to utilize the additional vapor energy. This would further enhance the efficiency potential of the humid air cycle. This could be done with 1-D tools based on the results obtained from the experiments.

8. Conclusion

From the simulations and experiments performed, it was clear that waste heat recovery can enhance the overall efficiency of the engine. By using the waste heat recovery technology within the engine cycle through the humid air cycle, both emissions and efficiency could be improved. The conclusions for two methods of waste heat recovery are presented.

- For the Rankine cycle, it was evident that even with a higher quantity of heat energy it was the quality of the heat that determines the useful power output (or BSFC reduction).
- When all of these sources were coupled in a single loop, the effectiveness of the high-temperature EGR source was affected. This was overcome by employing a dual loop Rankine cycle, which has better power output among the investigated configurations.
- Although the power output was higher than when only the EGR source was used, the Rankine cycle efficiency was higher when only the EGR source was used because of the elevated cycle pressure.
- For the engine considered in this study, it was more economical to recuperate only the EGR source, which provided better Rankine cycle efficiency compared to the dual loop Rankine cycle which uses more heat exchangers and turbine. Ultimately, the dual loop Rankine would increase cost, weight, and complexity of the system.

Also, the waste heat recovery potential using a Rankine cycle has been analyzed for two different types of combustion, conventional diesel combustion, and partially premixed combustion (PPC).

- From IPSEpro simulations, it was evident that with both combustion types, the useful power output from Rankine cycle was similar.
- The additional power produced by the WHR system corresponds to BSFC reductions of 3.0% - 4.0 % in the conventional combustion engine (75 % load) and ~5.0 % in the PPC engine. This shows that even with the comparable power output from a WHR system, the net waste heat recovery percentage was greater for PPC operation than for conventional diesel combustion operation.
- Furthermore, the thermal efficiency of PPC is higher than the conventional diesel combustion. Thus, the overall engine efficiency would be higher for

PPC combined with a Rankine cycle than for a conventional diesel combustion engine combined with a Rankine cycle.

A humid air system (HAM) was studied using system model simulations.

- The results show clear potential for improving efficiency at higher speeds.
- When the EGR mass used in the conventional engine was not too high, HAM was beneficial over the conventional engine (at similar inlet temperature).
- Also, the HAM system reduced the in-cylinder temperatures compared to conventional operation with EGR.
- The increment in efficiency was brought by a combination of factors: 1) lower heat losses, 2) reduction of in-cylinder pumping losses, and 3) free vapor expansion over the turbine.

From the experiments,

- The reduced oxygen concentration for HAM reduces NO formation and lowers the instantaneous flame temperature which brings down the NOx emissions significantly.
- The combustion in HAM closely follows conventional combustion with EGR.
- Furthermore, the fuel consumption was not affected significantly by the three operating modes in the operating point A25 as suggested by the system simulations.
- At greater speed (1500 rpm from 1200 rpm) it was also possible to improve brake efficiency by about 3% by reducing the pumping losses and cooling loss due to the HAM cycle. This shows the possibility to recuperate more power at higher speeds than the simulation study predicted.
- Furthermore, no EGR cooler or charge air cooler are required and hence the cooling requirement for a vehicle could be reduced.

9. Bibliography

Aghaali, H. (2014). *Exhaust Heat Utilisation and Losses in Internal Combustion Engines with Focus on the Gas Exchange System.* Stockholm: KTH Royal Institute of Technology.

Arunachalam Prakash Narayanan, B. N. (2012). A Comparative Analysis of WHR System in HD Engines Using Conventional Diesel Combustion and Partially-Premixed Combustion. *SAE.* Chicago.

Arunachalam Prakash Narayanan, M. C. (2012). Waste Heat Recovery from Multiple Heat Sources in a HD Truck Diesel Engine Using a Rankine Cycle - A Theoretical Evaluation. *SAE Technical Paper.*

Arunachalam Prakash Narayanan, M. T. (2013). System Simulations to Evaluate the Potential Efficiency of Humid Air Motors. *SAE/KSAE International Powertrains, Fuels & Lubricants Meeting.* Seoul.

Bedford, F. R. (2000). Effects of Direct Water Injection on DI Diesel Engine Combustion. *SAE Technical Paper.*

Bo-Tau Liu, K.-H. C.-C. (2004). Effect of working fluids on organic Rankine cycle for waste heat recovery. *Energy, 29*(8), 1207-1217.

Brito, F. G. (2013). Influence of Heat Pipe Operating Temperature on Exhaust Heat Thermoelectric Generation. *SAE Int. J. Passeng. Cars - Mech. Syst., 6*(2).

C. Lyle Cummins, J. (1976). *Internal Fire.* Oregon: Carnot Press.

C.O. Katsanos, D. H. (December 2013). Simulation of a heavy-duty diesel engine with electrical turbocompounding system using operating charts for turbocharger components and power turbine. *Energy Conversion and Management, 76*, 712-724.

Can Wang, W. C. (2007, July). CO2 mitigation scenarios in China's road transport sector. *Energy Conversion and Management, Volume 48, Issue 7*, pp. 2110-2118.

Chapman, L. (2007, September). Transport and climate change: a review. *Journal of Transport Geography*, pp. 354-367.

Charles Sprouse III, C. D. (2013). Review of organic Rankine cycles for internal combustion engine exhaust waste heat recovery. *Applied Thermal Engineering, 51*(1-2), 711-722.

Chen S. K., a. F. (n.d.). Development of Single Cylinder Compression Ignition Research Engine. *SAE Technical Paper .*

Christian Berggren, T. M. (2009). Hybrids, diesel or both? The forgotten technological competition for sustainable solutions in the global automotive industry. *International Journal of Automotive Technology and Management, 9*(2), 148-173.

Council, W. E. (2011). *Global Transport Scenarios 2050.* London: World Energy Council.

Crane.D., L. J. (2013). TEG On-Vehicle Performance and Model Validation and What It Means for Further TEG Development. *Journal of Electronic Materials, 42*(7).

Donghong Wei, X. L. (2007). Performance analysis and optimization of organic Rankine cycle (ORC) for waste heat recovery. *Energy Conversion and Management, 48*(4), 1113-1119.

E.H. Wang, H. Z. (2011). Study of working fluid selection of organic Rankine cycle (ORC) for engine waste heat recovery. *Energy, 36*(5), 3406-3418.

Entec, U. L. (2005). *Service Contract on Ship Emissions: Assignment, Abatement and Market-based Instruments, Task 2b NOx Abatement.* European Commission, DG Environment.

Hendriks, T. (2002). Heat Pipe Two-Phase Flow Systems For Vehicle Passesnger Cabin Cooling. *SAE Technical Paper.*

Henningsson, M. (2012). *Data-Rich Multivariable Control of Heavy-Duty Engines.* Lund: Department of Automatic Control, Lund University.

Heywood, J. B. (1988). *Internal combustion engine fundamentals.* New York: McGraw-Hill.

Hountalas D, K. C. (2007). Recovering energy from the diesel engine exhaust using mechanical and electrical turbocompounding. *SAE World Congress & Exhibition.* SAE Technical Paper.

Hountalas, D. M. (2007). Comparative Evaluation of EGR, Intake Water Injection and Fuel/Water Emulsion as NOx Reduction Techniques for Heavy Duty Diesel Engines. *SAE Technical Paper .*

Hung, T.-C. (2001). Waste heat recovery of organic Rankine cycle using dry fluids. *Energy Conversion and Management, 42*(5), 539-553.

Iwai, M. Y. (2011). *Study on Performance of Diesel Engine Applied with Emulsified Diesel Fuel: The Influence of Fuel Injection Timing and Water Contents.* SAE Technical Paper.

J.A. Gatowski, E. B. (1984). Heat release analysis of engine pressure data. *SAE Technical Paper.*

Jason Hill, E. N. (2006). Environmental, Economic, and Energetic Costs and Benefits of Biodiesel and Ethanol Biofuels. *Proceedings of the National Academy of Sciences of the United States of America, 103*(30), 11206-11210.

Kaneko, N. A. (2002). Expansion of the Operating Range with In-Cylinder Water Injection in a Premixed Charge Compression Ignition Engine. *SAE Technical Paper.* Kaneko, N., Ando, H., Ogawa, H., and Miyamoto, N., "Expansion of the Operating Range with In-Cylinder Water Injection in a Premixed Charge Compression Ignition Engine," SAE Technical Paper 2002-01-1743, 2002.

Kennedy, I. M. (2007). The health effects of combustion-generated aerosols. *Proceedings of the Combustion Institute*, (pp. 2757-2770).

Latz, G. A. (2012). Comparison of working fluids in both subcritical and supercritical Rankine cycles for waste-heat recovery systems in heavy-duty vehicles. *SAE Technical Paper.*

Leising, C. P. (1978). Waste Heat Recovery In Truck Engines. *SAE Technical Paper.*

Lemmon, E. H. (2011). *REFPROP Nist Standard Reference Database 23 (Version 9.0), Thermophysical Properties Division.* National Institute of Standards and Technology, Boulder, CO.

Lindquist, O. T. (2001). Humid Cycles A Novel Way to Increase Performance of Marine Propulsion Systems. *CIMAC, Hamburg.*

MAESC. (2009). Retrieved from Mid-South Annual Engineering and Sciences: http://www.maesc.org/maesc09/MAESC2009-CaoKengskool-Paper.pdf

Marie-Élise Parent1, M.-C. R. (2006). Exposure to Diesel and Gasoline Engine Emissions and the Risk of Lung Cancer. *American Jnl of Epidemiology*, 53-62.

Mignano, F. W. (1998). Development of a Diesel-Engine Piston by Incorporating Heat Pipe Technology - Experimental Simulation of Piston Crown. *SAE Technical Paper.*

Nadine Unger, D. T. (2009). Climate forcing by the on-road transportation and power generation sectors. *Atmospheric Environment, 43*(19), 3077-3085.

Nazha, M. R. (2001). The Use of Emulsion, Water Induction and EGR for Controlling Diesel Engine Emissions. *SAE Technical Paper* .

Nord, K. H. (2001). Particles And Emissions from a Diesel Engine Equipped with a Humid Air Motor System. *SAE Technical Paper.*

Nyberg. B. (2012). *Modelling of Wet Gas Turbine Cycles: Evaluation and Thermodynamic Calculation.* ISRN LUTMDN/TMHP--12/7081-SE/2012, Lund University .

Park, H. G. (2011). A Study on NOx Reduction of Marine 4-Stroke Diesel Engine Using Charge Air Humidification. *SAE Technical Paper* . Park, H., Ghal, S., Park, H., Choi, S. et al., "A Study on NOx Reduction of Marine 4-Stroke Diesel Engine Using Charge Air Humidification," SAE Technical Paper 2011-24-0172, 2011, doi:10.4271/2011-24-0172.

PrimeServ, M. (n.d.). *Humid Air Motor.Technology for green profits.* Retrieved May 03, 2016, from http://primeserv.man.eu/docs/librariesprovider5/primeserv-documents/ham.pdf?sfvrsn=2

Rahai, H. S. (2011). Investigation of the Effect of a Humid Air System on Diesel NOx and PM Emissions of a Small Diesel Engine. *SAE Technical Paper* .

Ringler, J. S. (2009). Rankine Cycle for Waste Heat Recovery of IC Engines. *SAE Technical Paper* .

Robin Hickman, D. B. (2007, September). Looking over the horizon: Transport and reduced CO2 emissions in the UK by 2030. *Transport Policy, Volume 14, Issue 5,*, pp. Pages 377-387.

Rosen, P. M. (1993). *Evaporative gas turbine cycles – A thermodynamic evaluation of their potential.* Lund University.

Rosen, P. M. (2000). *Evaporative Cycles - in Theory and in Practise.* Lund: Lund Institute of Technology.

Steve Sorrell, J. S. (2010). Global oil depletion: A review of the evidence. *Energy Policy, 38*(9), 5290-5295.

Steyr Engineering Center. (2016). Retrieved from Magna Powertrain: http://ecs.magna.com/Advanced-Combustion-Engines.3282.0.html?&L=2&cHash=e90d38c53ee5389f8bf648766511e4c8&tx_ttnews%5Btt_news%5D=1113

Strandh, P. (2006). *HCCI Operation—Closed Loop Combustion Control Using VVA or Dual Fuel.* Lund: Department of Energy Sciences, Lund University.

T.C. Hung, T. S. (1997). A review of organic rankine cycles (ORCs) for the recovery of low-grade waste heat. *Energy, 22*(7), 661-667.

Teng, H. K. (2011). A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines - Experimental Results. *SAE Technical Paper.*

Teng, H. R. (2007). Achieving High Engine Efficiency for Heavy-Duty Diesel Engines by Waste Heat Recovery Using Supercritical Organic- Fluid Rankine Cycle. *SAE Technical Paper .*

Thern. M. (2003). *"The Ethanol and Water Humidification Process in EvGT Cycles"'.* Lund: Lund University.

Tianyou Wang, Y. Z. (2011). A review of researches on thermal exhaust heat recovery with Rankine cycle. *Renewable and Sustainable Energy Reviews, 15*(6), 2862-2871.

Traverso, A. (2010). Humidification Tower for Humid Air Gas Turbine Cycles: Experimental Analysis. *Energy 35 , 35*, 894-901.

U. Asad, C. K. (2012). Effects of Intake Air Humidity on the NOx Emissions and Performance of a Light-duty Diesel Engine. *ASME Internal Combustion Engine Division, 2012 Fall Technical Conference.* Vancouver.

Udayakumar, R. S. (2003). Reduction of NOx Emissions by Water Injection in to the Inlet Manifold of a DI Diesel Engine. *SAE Technical Paper.*

V. Dolz, R. N. (2012). HD Diesel engine equipped with a bottoming Rankine cycle as a waste heat recovery system. Part 1: Study and analysis of the waste heat energy. *Applied Thermal Engineering, 36*, 269-278.

Woschni, G. (1967). Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine. *SAE Transactions, 76*.

Yiping Dai, J. W. (2009). Parametric optimization and comparative study of organic Rankine cycle (ORC) for low grade waste heat recovery. *Energy Conversion and Management, 50*(3), 576-582.

Yunshi Wang, J. T. (2011). China's soaring vehicle population: Even greater than forecasted? *Energy Policy, 39*(6), 3296-3306.

Paper I

SAE International

Waste Heat Recovery from Multiple Heat Sources in a HD Truck Diesel Engine Using a Rankine Cycle - A Theoretical Evaluation

2012-01-1602
Published
09/10/2012

Prakash Narayanan Arunachalam, Mengqin Shen, Martin Tuner, Per Tunestal and Marcus Thern
Lund University

doi:10.4271/2012-01-1602

ABSTRACT

Few previous publications investigate the possibility of combining multiple waste heat sources in a combustion engine waste heat recovery system. A waste heat recovery system for a HD truck diesel engine is evaluated for utilizing multiple heat sources found in a conventional HD diesel engine. In this type of engine more than 50 % of heat energy goes futile [1]. The majority of the heat energy is lost through engine exhaust and cooling devices such as EGRC (Exhaust gas recirculation cooler), CAC (Charge air cooler) and engine cooling. In this paper, the potential of usable heat recuperation from these devices using thermodynamic analysis were studied, and also an effort is made to recuperate most of the available heat energy that would otherwise be lost.

A well-known way of recuperating this heat energy is by employing a Rankine cycle circuit with these devices as heat sources (single loop or dual loop), and thus this study is focused on using a Rankine cycle for the heat recovery system. Furthermore, this paper investigates the possibilities and challenges involved in coupling these different sources in a single Rankine cycle and the selection of suitable working fluid for this Rankine cycle. The study shows that with recuperation from these multiple sources it is possible to recover 5-10 % of the otherwise wasted heat energy, which results in ~5 % power increase.

REFPROP was used for studying fluid properties, and the commercial software IPSEpro is used to build and simulate the Rankine cycle

INTRODUCTION

Political incentives and legislation combined with increased fuel prices force the automotive industry to develop more energy efficient engines. The stringent emission regulations restraints the modern engines to reach for higher thermal efficiency. The typicalthermal efficiency of a contemporary engine is ~42 % [2]. The rest is dissipated as heat_. This means that heat is dissipated through the exhaust gases which could have been used to produce more power. Much research has therefore focused on making use of this heat in the engine [2,3,4]. These studies focus on recovering the heat from just a single source, mainly the EGR. The use of the exhaust and the compressed air cooling, CAC, have been unheeded in the research.

This paper describes a theoretical evaluation of waste heat recovery from these different sources with five different configurations - 1. Only EGR, 2. Only Exhaust, 3. EGR and Exhaust combined, 4. EGR, Exhaust and CAC combined, and 5. EGR, Exhaust and CAC combined in dual loop Rankine cycle. A suitable working fluid is selected for the Rankine cycle. The Rankine cycle is simulated for these different configurations for a particular operating point C75 in 13 point European Stationary Cycle (ESC 13). In this investigated engine, the C75 operating point has more heat content “high temperature” EGR when compared to all other operating points in ESC13.This thermodynamically makes it simpler to extract the waste heat at this operating point. Hence C75 operating point is chosen to examine whether heat recovery is feasible or not and then extended the study to additional four load points namely B25, B50, B75, B100. The cycle is simulated in a virtual environment using commercial software IPSEpro, and the cycle is optimized for different configurations for maximum power.

RANKINE CYCLE

The Rankine cycle is a closed loop power cycle where the heat is transferred to usable power by means of a working fluid. The working principle of a Rankine cycle includes heating (isobaric), boiling (isobaric), superheating (isobaric), expanding (isentropic), condensation (isobaric), and compression (isentropic). The working fluid is heated and vaporized by absorbing heat from any external source, for example combustion of coal, nuclear and petroleum. This vaporized fluid is made to expand over an expander like turbine, piston expander and scroll expander to derive power. The vapor is then condensed to liquid form in a condenser using air or an external fluid circuit as in power plants. The condensed liquid is then compressed to increase the pressure and then the process is repeated. The process is illustrated with a T-S diagram for wet working fluid (water).

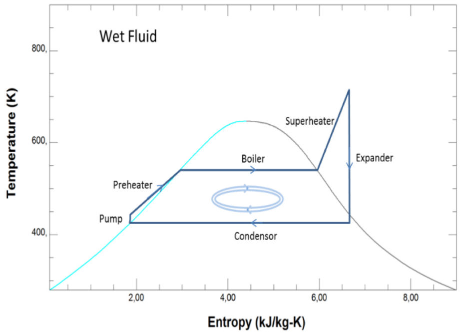


Figure 1. T-S diagram of water with typical Rankine cycle.

WORKING FLUIDS

Several studies have been carried out in finding a suitable working fluid for the Rankine cycle [3, 4, 5]. The latent heat of evaporation (evaporation enthalpy) of a fluid is a good indicator in choosing a working fluid for a given temperature level [4]; water has the highest latent heat of evaporation (~2250 kJ/kg) followed by organic working fluids and alcohols (methanol ~1100 kJ/kg, ethanol ~820 kJ/kg). Ethanol and methanol have maximum temperature limits of 650 K and 620 K, respectively, which is low when compared to the typical exhaust gas recirculation temperature (EGR ~750 K). Because of the higher latent heat of water, the flow rates required will be lesser in volume and thus smaller heat exchangers, which is an important factor in automotive applications.

Further, alcohols have flammability issues; ethanol and methanol have high flammability potential [3]. Water on the other hand has very high operating temperature limits making it ideal for EGR waste heat recovery and has no flammability issues. Since the engine model has the source temperatures ranging from 500 K to 750 K, water is chosen as a suitable working fluid. The drawback of using water as a working fluid is its freezing temperature of 273 K at ambient pressure, which has a negative effect when using this system in colder regions. This issue could be resolved by using anti-freezing agents. Also, studies have shown that mixed working fluids are appropriate for automotive waste heat recovery systems [3,8]. A mixture of water (80 % by mass) and methanol (20 % by mass) could improve the freezing temperature to 248K (−25 °C), allowing for performances comparable to water in a subcritical cycle [3]. Since the mixture (80 % water and 20 % methanol) has 80 % mass of water, it is expected to exhibit water like characteristics. REFPROP by the National Institute of Standards and Technology (NIST) is used to determine the fluid properties. Transport properties of this mixture cannot be estimated by REFPROP (v9.0) [6] at present, consequently water is selected and used in this evaluation. In the future the cycle could be extended for mixtures.

IPSEpro

The calculations in this paper have been performed in the program IPSEpro. IPSEpro is a simulating software developed by SimTech Technology. The program is constructed as a matrix solver connected to a graphical user interface. The governing equations representing models in IPSEpro is fully editable and based on one-dimensional heat and mass balance equations. This makes it possible to have full access to the equations describing the process and it thus prevents any black box scenarios during simulation. IPSEpro uses JANAF thermochemical tables [7] for calculating ideal gas properties and IAPWS IF97 [8] for calculating water and steam properties. This software is used to build the Rankine cycle to understand feasible coupling between these devices and the power output of the cycle. It has a built-in optimizer to optimize the cycle for maximum power output by varying pressures of the cycle.

The heat exchanger in the Rankine cycle transfers the hot waste heat from the engine to the low temperature Rankine cycle. The driving force for this heat transfer process is the temperature difference between the hot fluid and the cold fluid. The minimum temperature difference, here called the pinch-point, occurs at either the inlet or the outlet of the heat exchanger depending on the specific heat exchanging process. The pinch-point is a modeling parameter and it is set to 20 °C.

The models used in this paper make use of an isentropic efficiency to describe the expansion and compression processes. The isentropic efficiencies are of course dependent on several parameters such as mass flow, temperature and pressure levels of the working fluid and it is impossible to know the exact value without an aerodynamic design of the specific component. In this paper, values of the isentropic efficiencies have been chosen from the literature [9]. The isentropic efficiency of the pump is thus taken to be 60 % and the turbine isentropic efficiency is set to 78 % and they are kept constant throughout the evaluation. This values fall in between the lower range and the middle range in the literature and is thus considered to be conservative.

The condensers are modeled to condense the steam after expansion in the turbine and further subcool the vapor by 2 °C. By mass and heat balance equations the cycle power output is calculated at the turbine. The maximum moisture content in the vapor after expansion is limited to 10 %. Due to practical application issues the maximum pressure level is set to 60 bar. The pump power is also calculated in the IPSEpro, thus the power output from the turbine is the total recuperated power. Since the Turbine is expected to expand the vapor to near saturated, there is not much energy left to recover in a dedicated recuperator. A recuperator is efficient in case of dry or isentropic fluid but not for the wet fluid.

ENGINE MODEL

The engine model considered for waste heat recovery is a Volvo D-13, 13-liter heavy-duty truck diesel engine. The engine is operated in European stationary cycle ESC-13, except when idle, thus 12 operating points as shown in the table. The data used in the analysis are based on the results from the VolvoD-13, 13-liter heavy-duty truck diesel engine. Table 1 has the 12 operating points corresponding to the 13 point European Stationary Cycle (ESC 13) but for idling.

Table 1. Power, Torque and Speed for ESC 13 point cycle (excluding idle).

Operating point	Power	Torque	Speed
	kW	Nm	RPM
A25	76	600	1210
A50	152	1200	1210
A75	228	1801	1210
A100	305	2402	1210
B25	93	592	1500
B50	187	1185	1500
B75	280	1778	1500
B100	373	2369	1500
C25	92	486	1800
C50	183	974	1800
C75	275	1459	1800
C100	367	1948	1800

ENERGY BALANCE

Figure 2 shows the control volume of the engine, where $\dot{Q}$ is the heat flow, $\dot{m}$ is the mass flow, LHV is the Lower heating value of the fuel, and P_{brake} is the brake power. The reference state for the enthalpy calculation is taken as 25 °C and 1 atm. In the engine cycle the energy that goes in equals the energy coming out of the system.

$$\dot{m}_{fuel} = \dot{Q}_{Exhaust} + \dot{Q}_{EGR} + \dot{Q}_{CAC} + \dot{Q}_{coolant} + \dot{Q}_{catalyst} + \dot{Q}_{radiation} + P_{brake} + P_{misc-losses} \quad (1)$$

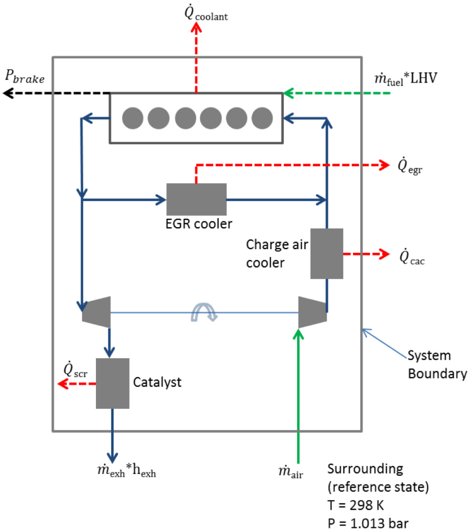


Figure 2. Energy flow (with system boundary).

In the above equation, $P_{misc-losses}$ includes the work done by water pumps, oil pumps, and air-conditioning, and $\dot{Q}_{radiation}$ is the radiation heat transfer losses. Both of the losses are not included in the calculation presented in Figure 3. To determine which source has the better recuperating potential a Carnot efficiency calculation is made. While calculating the Carnot efficiency the temperatures considered are the absolute engine operating temperatures, that is T_C is the exit temperature of the gases and T_H is the inlet temperature of the gases in the respective devices, and for exhaust T_C is considered as the ambient temperature. For this calculation only EGR, CAC and Exhaust sources are considered, while the coolant is not considered owing to its poor heat availability. For coolant typically the temperature difference is only 10-20 K. To recuperate this small heat energy from coolant requires a large surface area of heat exchangers compared to other sources and it would not be economical, thus the coolant source is not considered in this study.

$$\eta_{carnot} = 1 - \frac{T_C}{T_H} \tag{2}$$

Table 2. Carnot efficiency for EGR, Exhaust and CAC.

Device	η_{carnot}, @ C75 load
EGR	50.4 %
Exhaust	40.6 %
CAC	35 %

Heat Flow Across Devices

In the contemporary engines the heat energy from EGR, CAC and Exhaust is wasted through the EGR cooler, Charge air cooler and radiator. This otherwise wasted heat is recoverable and can be used as the source for Rankine cycle. This is calculated with average values of specific heat entering and leaving the device, temperature difference between the gases entering and leaving the device, and mass flow of the gases flowing across the device, eq. (3). The available heat energy for different sources considered in this study is presented in Figure 3.

$$\dot{Q} = \dot{m} * \frac{Cp_{in} + Cp_{out}}{2} * (T_{in} - T_{out}) \tag{3}$$

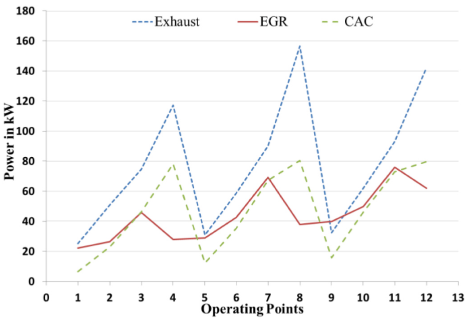


Figure 3. Energy available from three sources.

Temperature and Pressure Dependency

The efficiency of the Rankine cycle depends on the pressure and temperature levels available. For a given heat quantity the superheating temperature has to be higher, this could be achieved by increasing the pressure of the cycle. Attaining higher pressure levels greatly depends on the gas temperatures from the engine and the mechanical limitations of the boilers. A simple Temperature-Enthalpy analysis is made for water Figure 4, withconstant mass flow at different pressure levels.

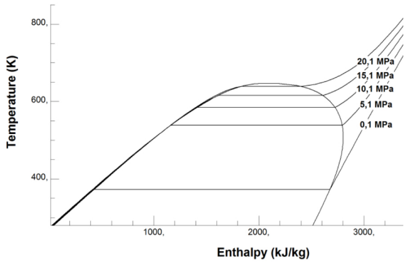


Figure 4. T-h diagram of water at five different pressure levels.

MULTIPLE SOURCES

Three sources, namely EGR gas, CAC gas and exhaust, have different temperature levels and different mass flow rates. In this study, Rankine cycles with single source (EGR or Exhaust) and multiple sources (all) were studied, and the power output and power improvement were analyzed and compared. Coupling these three sources in a Rankine cycle can be done in single loop or dual loop configuration, and both types are studied.

Table 3. Different Rankine configurations.

Loop	EGR	Exhaust	CAC
Single loop	✓	-	-
Single loop	-	✓	-
Single loop	✓	✓	-
Single loop	✓	✓	✓
Dual loop	✓	✓	✓

According to the order in Table 3, theconfigurations were simulated and operational methodology is presented in the following section. The Rankine cycle is simulated for these different configurations for the particular operating point C75. Thermal properties at load point C75 are shown in Table 4.

Table 4. Thermal properties of the sources.

Source	$\dot{m}$	$C_{p,avg}$	T_{in}	T_{out}	$\dot{Q}_{available}$
	g/s	j/kg-K	K	K	kW
EGR	182	1111	744	369	76
CAC	418	1035	482	314	73
Exhaust	432	1057	502	298	93

EGR as Source

The short route high pressure exhaust gas recirculation technique uses the hot burnt gases from the combustion into the cylinder with the inlet air. These gases have very high temperature levels, making it a suitable source for waste heat recovery [5]. The cycle has three heat exchangers-pre-heater, boiler and superheaterconnected in series as shown in Figure 5. The pump pressurizes the fluid to the desired pressure level of 60 bar, and the preheated water is sent into the boiler which makes the fluid boil at 549 K at the same pressure. The vapor then enters the superheater, where the temperature is further raised to 618 K. The superheated steam enters the expander where it expands to give the useful output power. After expansion the steam is condensed to water in the condenser, where the water is fed to the pump and the cycle continues.

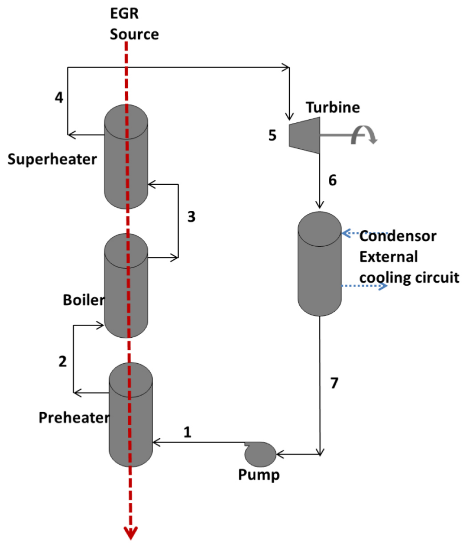


Figure 5. 1. Pressurized water; 2. Preheated water; 3. Boiled water vapor; 4. Superheated steam; 5. Expander; 6. Steam after expansion; 7. Condensed water.

Exhaust Gas as Source

Exhaust gas from the engine still carries energy, which is being targeted here for recuperation. The operating cycle is described in Figure 6. The cycle works similarly to the EGR cycle but without a superheater. Since the temperature is not high enough to work at higher pressures, the pressure in the cycle is reduced to 5 bar.

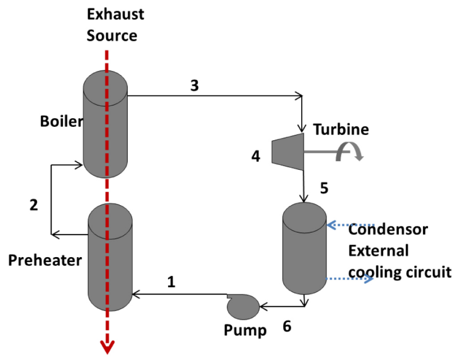


Figure 6. 1. Pressurized water; 2. Preheated water; 3.Boiled water vapor; 4. 4. Expander; 5. Steam after expansion; 6. Condensed water.

Exhaust and EGR as Sources

In the last two sections, the exhaust and EGR are utilized separately. In this cycle both of the heat sources are coupled as shown in Figure 7.

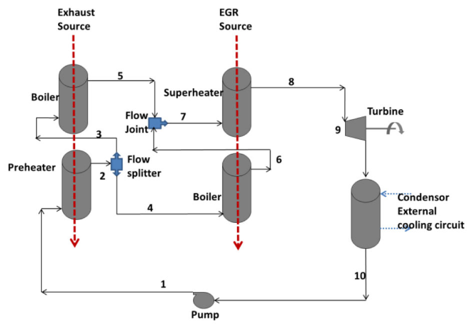


Figure 7. 1. Pressurized water; 2, 3 & 4. Preheated water; 5 & 6.Boiled water vapor; 7. Combined vapor to superheater; 8. Superheated steam; 9.Expander; 10.Steam after expansion and condensed water.

The water is pressurized in the pump to 11.12 bar and fed to the pre-heater, the preheated water is then divided into two flows: one flow enters the boiler in the exhaust stream and the other flow enters the boiler in the EGR stream. After boiling both flows convert into steam and are then combined into one flow, which flows into the superheater stream of the EGR source. The superheated steam then enters the turbine, where it expands to give useful work and then enters the condenser where it condenses to water, which is then fed to the pump to continue the cycle.

EGR, Exhaust and CAC as Sources

In the previous section, two sources, exhaust and EGR were coupled in the Rankine cycle. In heavy-duty engines the boost pressures are very high, and thus the temperature of the air after the compressor is high. Typically, an intercooler/aftercooler is used to let this heat to ambient. In this cycle, the CAC (charge air cooler) source is coupled with the other two sources as shown in Figure 8, thus recovering all of the possible sources of otherwise wasted heat energy.

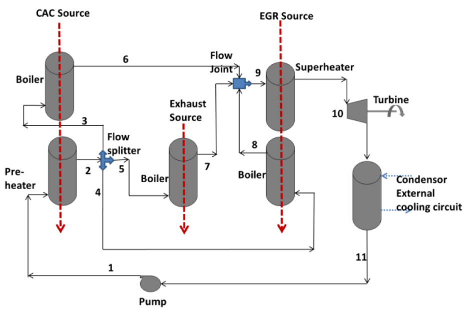


Figure 8. 1. Pressurized water; 2, 3, 4 & 5. Preheated water; 6, 7, 8 & 9. Boiled water vapor to superheater; 10. Superheated steam expansion; 11. Condensed water.

EGR, Exhaust, CAC in Dual Loops

When using the three sources in a single loop, cycle pressure is a limiting factor. The EGR source has a higher temperature, allowing us to work with higher pressures. On the other hand, the CAC and exhaust has lower temperatures when compared to EGR. A better way to recuperate the energy from these devices might be coupling these in a dual loop Rankine cycle (see Figure 9). In the dual loop Rankine cycle, the lower temperature exhaust and CAC work with a lower pressure cycle of 4.6 bar, and the high temperature EGR works with 60 bar pressure cycle. The cycle has two turbines: a high pressure turbine and a low pressure turbine. Two pumps were used to elevate the pressure levels of the two loops. In the high pressure loop, the pressurized fluid at 60 bar goes into the pre-heater and the preheated fluid goes into the boiler. After the fluid boils it enters the superheater, wherein the superheated vapor then does the expansion work in the high pressure expander and the vapor leaves the high pressure expander at 4.6 bar matching the low pressure cycle. In the low pressure loop the fluid is preheated with the heat source from the CAC stream and then divided into two flows: one flow passes across the boiler in the CAC stream and the other flow passes into the boiler in the exhaust stream. Both of the vapor flows are then combined with the high pressure turbine outlet vapor, and the combined mass flow[1] is expanded in the low pressure turbine.

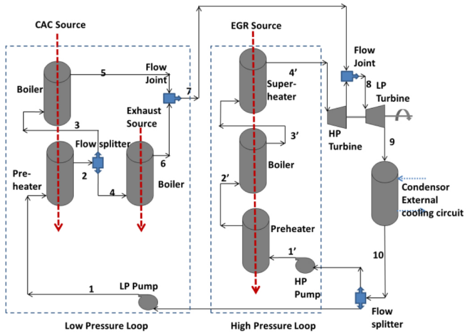


Figure 9. LP cycle (1.Pressurized water; 2, 3 & 4.Preheated water; 5, 6 & 7.Boiled water vapor to LP expander; 10.Condensed water); HP cycle (1'. Pressurized water; 2'. Preheated water; 3'. Boiled water vapor to superheater; 4'. Superheated steam expansion).

RESULTS

Different possible waste recovery sources using the Rankine cycle are listed. These different configurations are simulated and the results are presented below.

EGR as Source

When using the high temperature EGR as the source, the pressure of the cycle can be elevated to the design maximum of 60 bar. The pressurized water flows through the preheater, boiler and superheater which are connected in series, the resulting vapor expands in a expander to deliver the power. Table 5, shows the mass flow of the water and the output turbine power.

Table 5. Power from only EGR as source.

Cycle Presure	$\dot{m}$ $preheater$	$\dot{m}$ $boiler$	$\dot{m}$ $superheater$	$P_{turbine}$
bar	g/s	g/s	g/s	kW
60	20.1	20.1	20.1	11.7

Exhaust as Source

When using the relatively low temperature exhaust as the source, the pressure in the cycle is reduced to 5 bar. Table 6, shows the mass flow of the water and the output turbine power.

[1] The combined mass flow is not superheated, as only the high pressure steamis superheated.

Table 6. Power from only Exhaust as source.

Cycle Pressure	$\dot{m}$ $preheater$	$\dot{m}$ $boiler$	$\dot{m}$ $superheater$	$P_{turbine}$
bar	g/s	g/s	g/s	kW
5	12.8	12.8	12.8	2.7

EGR and Exhaust as Sources

When combining high temperature EGR and the exhaust as the source, the exhaust is used to preheat and boil the water and simultaneously the EGR source is used to boil and superheat the vapor. The amount of massflow flowing across different flows and the power delivered from the turbine are presented in Table 7.

Table 7. Power from EGR and Exhaust combined

Source	Cycle Pressure	$\dot{m}$ $preheater$	$\dot{m}$ $boiler$	$\dot{m}$ $superheater$	$P_{turbine}$
	bar	g/s	g/s	g/s	kW
EGR	11.1	-	24.5	30.3	10.8
Exhaust	11.1	30.3	5.8	-	

EGR, Exhaust and CAC, Combined in a Single Loop Rankine Cycle

Table 8, shows the amount of water mass flow through different devices across the three sources and the power delivered by the turbine. Since it is a single loop Rankine cycle the pressure is controlled by the low temperature sources.

Table 8. Power from EGR, Exhaust and CAC combined.

Source	Cycle Pressure	$\dot{m}$ $preheater$	$\dot{m}$ $boiler$	$\dot{m}$ $superheater$	$P_{turbine}$
	bar	g/s	g/s	g/s	kW
EGR	7.2	-	24.1	38.9	11.8
Exhaust	7.2	-	9.8	-	
CAC	7.2	38.9	4.9	-	

EGR, Exhaust and CAC, Combined in a Dual Loop Rankine Cycle

In a single loop Rankine cycle the pressure is controlled by the low temperature sources which hinder the efficiency of all the sources in the cycle. In dual loop Rankine cycle the high temperature EGR loop is operated with higher pressure than the Exhaust and CAC streams. Table 9, shows the amount of water mass flow through different devices across the three sources and the power delivered by the turbine.

Table 9. Power from EGR, Exhaust and CAC combined in a dual loop Rankine cycle.

Source	Cycle Pressure	$\dot{m}$ $preheater$	$\dot{m}$ $boiler$	$\dot{m}$ $superheater$	$P_{turbine}$
	bar	g/s	g/s	g/s	kW
EGR	60	21.5	21.5	21.5	16.3
Exhaust	4.6	-	13.5	-	
CAC	4.6	21.8	8.4	-	

Heat Utilized from Different Configurations

Heat utilized from these different configurations is presented in Table 10. The high temperature EGR source has higher heatto power conversion efficiency when compared to all other configurations, and the power output of the EGR is almost equal to the power delivered by combining all the sources in a single loop Rankine cycle. The heat to power conversion efficiency is not very high in a dual loop Rankine cycle with all the sources combined but since it has higher heat from all the sources, the power recuperated is higher.

Table 10. Utilized heat in different configurations[2].

Sources	$\dot{Q}_{in}$	$\dot{Q}_{out}$	$\dot{Q}_{used}$	$P_{turbine}$	$P_{turbine}/\dot{Q}_{used}$
	kW	kW	kW	kW	%
EGR	96	44	52,5	11.7	22,3
Exhaust	108	78.3	29.7	2.7	9.2
EGR, Exhaust	96 108	40.6 85.8	55.6 22.2 77.8	10.8	13.8
EGR, Exhaust, CAC	96.2 108 90.6	36.9 87.7 69.8	59.3 20.3 20.8 100,5	11.8	11.7
EGR, Exhaust, CAC, dual loop	96.2 108 90.6	42.7 79.3 68.4	53.6 28.7 22.3 104.5	16.3	15.6

[2] The reference for enthalpy calculation is 273 K at 1 atm in IPSEpro

Power Improvement for Operating Point C75 (274.81 kW)

It is seen from Table 11, the power improvement in the engine with bottomingRankine cycle is significant. With this investigated engine the increaseis ~4% for the operating point C75.

Table 11. Power improvement from different configurations.

Sources	Turbine Power, kW	Total power	% Power increase
EGR	11.7	286.6	4.0
Exhaust	2.7	277.6	1
EGR, Exhaust	10.8	285.6	3.8
EGR, Exhaust, CAC	11.8	286.6	4.1
EGR, Exhaust, CAC-Dual loop	16.3	291.1	5.6

Waste Heat Utilization in Other Load Points

The waste heat recovery proves to be significant from the above tables. The evaluation is extended for other load points to analyze the benefits. Analysis similar to what have been carried out for C75 are made for different load point in speed range B (1500 RPM) − 25 %, 50 %, 75 %, and 100 %. The results from the analysis are presented in the table below. In B25 operating point the CAC has low temperature gases making it unsuitable for recuperating. It can be noted in B50 operating point, the power recuperated from all the sources in the single loop Rankine cycle is lower than the power recuperated from combining the EGR and exhaust, and this is due to the heat lost from the saturated water to the low temperature CAC.

Table 12. Power recuperated for load points B100, B75, B50, B25

	EGR	Exha ust	EGR + Exhaust	EGR + Exhaust + CAC Single loop	EGR + Exhaust + CAC Dual loop
B100	7.9	11.2	18.2	20.9	19.6
B75	11.5	3.1	12.6	14.1	16.4
B50	5.6	3.1	6.2	5.2	7.3
B25	2.9	0.6	3.0	N/A[3]	N/A[3]

[3]*The heat in the charge air cooler is too low to be able to recuperate.*

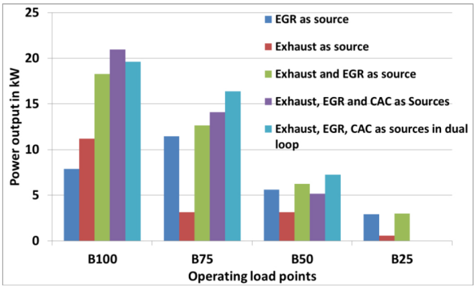


Figure 10. Comparison of different configuration of sources for Rankine cycle WHR for different operating load points

DISCUSSIONS

From Table 10 it is evident that even with more quantity of heat energy, it is the quality of the heat that determines the useful power output. When all of these sources are coupled in a single loop, the effectiveness of the high temperature EGR source is affected. This can be overcome by employing a dual loop Rankine cycle, which has the better power output among all these configurations. Although the power output is lower when only the EGR source is used, the cycle efficiency ($Power_{turbine}/\dot{Q}_{used}$) is higher. The low quality heat energy can be efficiently recuperated, if the pressures after turbine could be lower than ambient which is typicalin power plants. Then the challenge would be the condenser size for condensing the fluid with ambient air. In automotive applications the sizing is an important factor; however, there are other challenges including working with subatmospheric pressure, which could cause oil leakage in the turbine and end up mixing oil with working fluid.

To recuperate efficiently from the CAC and exhaust sources, a suitable way is to use a working fluid that has a condensation temperature close to ambient temperature at ambient pressure and the critical point temperature (85 % of critical [10]) close to the temperature of the Exhaust and CAC sources. In this case higher quantity of heat energy can be recuperated from the gas stream. Certain fluids that come close to these criteria have limited application potential either because of ozone depletion potential (ODP) or global warming potential (GWP) [3], for example HFC-152a, flammability limits, methanol or low latent heat of evaporation, or acetone. Hence, among the available fluids, water is a suitable and efficient working fluid for waste heat recovery at high temperatures (~700 K). For low temperatures (~500 K) water can still be used with lower thermal efficiency or at subatmospheric pressure levels to be thermally efficient provided that condensing is not difficult (perhaps useful for stationary power plants and boat/ferry engines).

CONCLUSION

For the engine investigated in this study it is economical to recuperate only the EGR source, which has cycle efficiency of 22.2 % and uses only four heat exchangers (including condensers) when compared to the dual loop Rankine cycle, where the cycle efficiency is 15.6 % and uses seven heat exchangers and an additional turbine. The additional devices increase the cost, weight and complexity of the system.

From Fig 10, It could be seen when running the engine close to full load the power output from all the sources combined is higher while for the typical HD truck application, the truck is operated between 25 % to 50 % load most of the time and for these applications, power recuperated from only the EGR is comparable to all the three sources combined. The Waste heat recovery using a Rankine cycle demonstrates to be resourceful. The source from which it has to be recuperated is highly reliant upon the application.

REFERENCES

1. Teng, H., Klaver, J., Park, T., Hunter, G. et al., “A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines - WHR System Development,” SAE Technical Paper 2011-01-0311, 2011, doi: 10.4271/2011-01-0311.

2. Park, T., Teng, H., Hunter, G., van der Velde, B. et al., “A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines - Experimental Results,” SAE Technical Paper 2011-01-1337, 2011, doi:10.4271/2011-01-1337.

3. Latz, G., Andersson, S., and Munch, K., “Comparison of Working Fluids in Both Subcritical and Supercritical Rankine Cycles for Waste-Heat Recovery Systems in Heavy-Duty Vehicles,” SAE Technical Paper 2012-01-1200, 2012, doi: 10.4271/2012-01-1200.

4. Ringler, J., Seifert, M., Guyotot, V., and Hübner, W., “Rankine Cycle for Waste Heat Recovery of IC Engines,” *SAE Int. J. Engines* 2(1):67-76, 2009, doi: 10.4271/2009-01-0174.

5. Teng, H., Regner, G., and Cowland, C., “Achieving High Engine Efficiency for Heavy-Duty Diesel Engines by Waste Heat Recovery Using Supercritical Organic-Fluid Rankine Cycle,” SAE Technical Paper 2006-01-3522, 2006, doi: 10.4271/2006-01-3522.

6. Lemmon, E.W., Huber, M.L. and McLinden, M.O., REFPROP Nist Standard Reference Database 23 (Version 9.0), Thermophysical Properties Division, National Institute of Standards and Technology, Boulder, CO, 2011

7. Stull, D, Thermochemical, J, et al. JANAF thermochemical tables. Clearinghouse for Federal Scientific and Technical Information; 1966.

8. Wagner, W., Cooper, J., Dittmann, A., Kijima, J., Kretzschmar, H., Kruse, A. et al. The IAPWS industrial formulation 1997 for the thermodynamic properties of water and steam, J Eng Gas Turb Power, 122 (2000), p. 150

9. Teng, H., Regner, G., and Cowland, C., “Waste Heat Recovery of Heavy-Duty Diesel Engines by Organic Rankine Cycle Part I: Hybrid Energy System of Diesel and Rankine Engines,” SAE Technical Paper 2007-01-0537, 2007, doi: 10.4271/2007-01-0537.

10. Teng, H., Regner, G., and Cowland, C., “Waste Heat Recovery of Heavy-Duty Diesel Engines by Organic Rankine Cycle Part II: Working Fluids for WHR-ORC,” SAE Technical Paper 2007-01-0543, 2007, doi: 10.4271/2007-01-0543.

11. David, J., and Nicholas, C.B., “Introduction to Turbomachinery”, Concepts ETI, Inc., 1997

12. Zhang, X., Zeng, K., Bai, S., Zhang, Y. et al., “Exhaust Recovery of Vehicle Gasoline Engine Based on Organic Rankine Cycle,” SAE Technical Paper 2011-01-1339, 2011, doi:10.4271/2011-01-1339.

13. Bae, S., Heo, H., Lee, H., Lee, D. et al., “Performance Characteristics of a Rankine Steam Cycle and Boiler for Engine Waste Heat Recovery,” SAE Technical Paper 2011-28-0055, 2011, doi:10.4271/2011-28-0055.

CONTACT INFORMATION

Prakash Narayanan A
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Prakash.narayanan@energy.lth.se

Martin Tuner
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Martin.tuner@energy.lth.se

Per Tunestal
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Per.tunestal@energy.lth.se

ACKNOWLEDGEMENTS

This study was conducted as a part of waste heat recovery project supported by the Swedish Energy Agency. The authors gratefully acknowledge them for the financial support. The authors would also like to thank OlofErlandsson of TitanX for his helpful discussions.

DEFINITIONS/ABBREVIATIONS

EGR - Exhaust Gas Recirculation
Exh - Exhaust gas source
CAC - Charge air cooler source
C_p - Specific heat, J/kg-K
Delta - Difference
GWP - Global warming potential
h - Enthalpy, J/kg
LHV - Lower heating value
$\dot{m}$ - Mass flow, g/s
η - Efficiency
ODP - Ozone depleting potential
$P_{turbine}$ - Turbine Power, kW
p, Pr - Pressure, bar
$\dot{Q}$ - Heat flow, kW
S - Entropy, J/kg-K
T_C - Sink temperature, K
T_H - Source temperature, K

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

ISSN 0148-7191

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE. The author is solely responsible for the content of the paper.

SAE Customer Service:
Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
SAE Web Address: http://www.sae.org
Printed in USA

Paper II

SAE International

A Comparative Analysis of WHR System in HD Engines Using Conventional Diesel Combustion and Partially-Premixed Combustion

2012-01-1930
Published
09/24/2012

Prakash Narayanan Arunachalam, Bjorn Nyberg, Martin Tuner and Per Tunestal
Lund University

doi:10.4271/2012-01-1930

ABSTRACT

In the truck industry there is a continuous demand to increase the efficiency and to decrease the emissions. To acknowledge both these issues a waste heat recovery system (WHR) is combined with a partially premixed combustion (PPC) engine to deliver an efficient engine system. Over the past decades numerous attempts to increase the thermal efficiency of the diesel engine has been made. One such attempt is the PPC concept that has demonstrated potential for substantially increased thermal efficiency combined with much reduced emission levels.

So far most work on increasing engine efficiency has been focused on improving the thermal efficiency of the engine while WHR, which has an excellent potential for another 1-5 % fuel consumption reduction, has not been researched that much yet. In this paper a WHR system using a Rankine cycle has been developed in a modeling environment using IPSEpro.

A comparative investigation of the WHR potential between the existing conventional diesel combustion and the novel PPC combustion is done. Even though the PPC is a low temperature combustion concept (LTC), implying that the exhaust temperatures are lower than for the traditional diesel combustion, the EGR quantity is higher which in total still offers improved WHR potential as that of conventional combustion. The EGR cooler offers higher quality heat when compared to exhaust gas and CAC, hence the WHR potential using only the EGR system is considered in this paper.

PARTIAL PREMIXED COMBUSTION (PPC)

In the conventional combustion the air is compressed and the fuel is directly injected at the end of compression stroke resulting in higher local temperatures which leads to NOx emissions. To avoid local temperature rise the mixture should be allowed to mix well and this could be attained by early injection of fuel, yet again the pressure rise rate would be enormous resulting in higher temperatures. To avoid NOx emissions, the local oxygen content has to be reduced during combustion and this can be achieved by introducing the already burnt gases in the engine cylinder (EGR) which has low oxygen content. Combination of early injection timing and higher EGR rates, allows the engine to operate with premixed combustion owing to longer ignition delay period and low combustion temperatures due to the cooling effect of EGR [1]. As secondary effect of PPC the HC emission increases as fuel increases in the quench area near the liner and top clearances [2]. Manente et al [3] demonstrated a 57 % indicated efficiency (~50% brake efficiency) for a HD engine without the need for an after treatment system. Yet more than 40% of the fuel energy still goes futile in the form of heat losses.

PPC ENGINE MODELLING

The experimental results and setup of the PPC engine are described in detail in [3]. Based on experimental results a single cylinder diesel engine with Partially-Premixed combustion is modeled in 1D using GT-Power [4], see Figure 2. The engine modeled is based on a Scania 13 liter Diesel engine, a number of simplifications were made in the model to represent the real experimental engine [5]. The burn rate of PPC combustion is simulated using DARS-ESM as described in [5], this burn rate is given as the input to perform the combustion in the 1D model. Pressure trace from the simulated engine is compared, with the experimental pressure trace to validate the model (Figure 1).

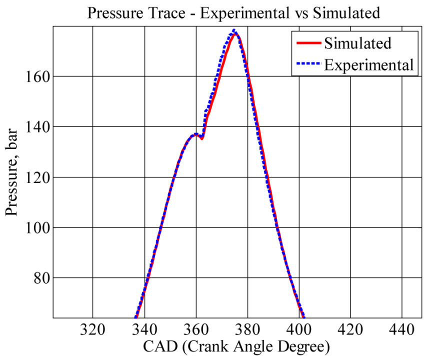


Figure 1. Pressure trace comparison between experimental and simulation

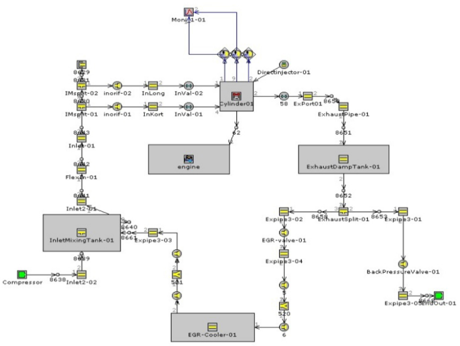


Figure 2. GT Power model of Scania D-13

RANKINE CYCLE

The Rankine cycle is a closed loop power cycle where the heat is transferred to usable power by means of a working fluid. The working principle of a Rankine cycle includes heating (isobaric), boiling (isobaric), superheating (isobaric), expanding (isentropic), condensation (isobaric) and compression (isentropic). The working fluid is heated and vaporized by absorbing heat from any external source for example combustion of coal, nuclear and petroleum. This vaporized fluid is made to expand over an expander like turbine, piston expander and scroll expander, to derive power. The vapor is then condensed to liquid form in a condenser using air or an external fluid circuit as in power plants. The condensed liquid is then compressed to increase the pressure and then the process continues. The process is explained with a T-S diagram for wet working fluid (water).

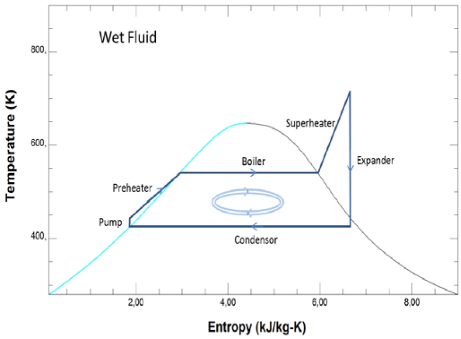


Figure 3. T-S diagram of water with typical Rankine cycle

WORKING FLUIDS

Several studies have been carried out in finding a suitable working fluid for the Rankine cycle [3, 4, 5]. The latent heat of evaporation (evaporation enthalpy) of a fluid is a good indicator in choosing a working fluid for a given temperature level [4]; water has the highest latent heat of evaporation (~2250 kJ/kg) followed by organic working fluids and alcohols (methanol ~1100 kJ/kg, ethanol ~820 kJ/kg). Ethanol and methanol have maximum temperature limits of 650 K and 620 K, respectively, which is low when compared to the typical exhaust gas recirculation temperature (EGR ~750 K). Because of the higher latent heat of water, the flow rates required will be lesser in volume and thus smaller heat exchangers, which is an important factor in automotive applications.

Further, alcohols have flammability issues; ethanol and methanol have high flammability potential [3]. Water on the other hand has very high operating temperature limits making it ideal for EGR waste heat recovery and has no flammability issues. Since the engine model has the source temperatures ranging from 500 K to 750 K, water is chosen as a suitable working fluid. The drawback of using water as a working fluid is its freezing temperature of 273 K at ambient pressure, which has a negative effect when using this system in colder regions. This issue could be resolved by using anti-freezing agents. Also, studies have shown that mixed working fluids are appropriate for automotive waste heat recovery systems [3,8]. A mixture of water (80 % by mass) and methanol (20 % by mass) could improve the freezing temperature to 248K (−25 °C), allowing for performances comparable to water in a subcritical cycle [3]. Since the mixture (80 % water and 20 % methanol) has 80 % mass of water, it is expected to exhibit water like characteristics. REFPROP by the National Institute of Standards and Technology (NIST) is used to determine the fluid properties. Transport properties of this mixture cannot be estimated by REFPROP (v9.0) [6] at present; consequently water is selected and used in this evaluation. In the future the cycle could be extended for mixtures.

IPSEpro

The calculations in this paper have been performed in the program IPSEpro. IPSEpro is a simulating software developed by SimTech Technology. The program is constructed as a matrix solver connected to a graphical user interface. The governing equations representing models in IPSEpro is fully editable and based on one-dimensional heat and mass balance equations. This makes it possible to have full access to the equations describing the process and it thus prevents any black box scenarios during simulation. IPSEpro uses JANAF thermochemical tables [7] for calculating ideal gas properties and IAPWS IF97 [8] for calculating water and steam properties. This software is used to build the Rankine cycle to understand feasible coupling between these devices and the power output of the cycle. It has a built-in optimizer to optimize the cycle for maximum power output by varying pressures of the cycle.

The heat exchanger in the Rankine cycle transfers the hot waste heat from the engine to the low temperature Rankine cycle. The driving force for this heat transfer process is the temperature difference between the hot fluid and the cold fluid. The minimum temperature difference, here called the pinch-point, occurs at either the inlet or the outlet of the heat exchanger depending on the specific heat exchanging process. The pinch-point is a modeling parameter and it is set to 20 °C.

The models used in this paper make use of an isentropic efficiency to describe the expansion and compression processes. The isentropic efficiencies are of course dependent on several parameters such as mass flow, temperature and pressure levels of the working fluid and it is impossible to know the exact value without an aerodynamic design of the specific component. In this paper, values of the isentropic efficiencies have been chosen from the literature [9]. The isentropic efficiency of the pump is thus taken to be 60 % and the turbine isentropic efficiency is set to 78 % and they are kept constant throughout the evaluation. This values fall in between the lower range and the middle range in the literature and is thus considered to be conservative.

The condensers are modeled to condense the steam after expansion in the turbine and further subcool the vapor by 2 °C. By mass and heat balance equations the cycle power output is calculated at the turbine. The maximum moisture content in the vapor after expansion is limited to 10 %. Due to practical application issues the maximum pressure level is set to 60 bar. The pump power is also calculated in the IPSEpro, thus the power output from the turbine is the total recuperated power. Since the Turbine is expected to expand the vapor to near saturated, there is not much energy left to recover in a dedicated recuperator. A recuperator is efficient in case of dry or isentropic fluid but not for the wet fluid.

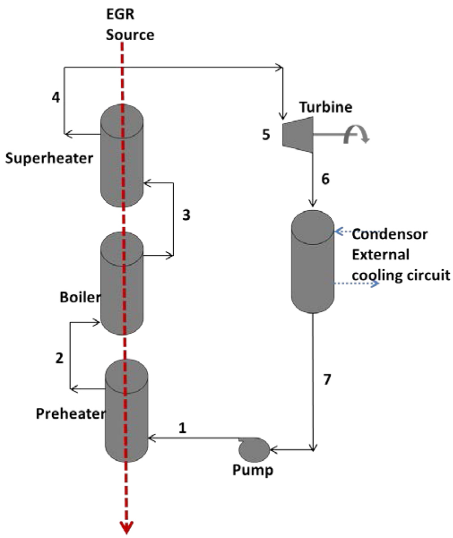


Figure 4. EGR as source: 1. Pressurized water; 2. Preheated water; 3. Boiled water vapor; 4. Super-heated steam; 5. Expander; 6. Steam after expansion; 7. Condensed water.

ENGINE MODEL

The engine model considered for waste heat recovery is a Volvo D-13, 13 liter heavy duty truck diesel engine; the data used in the analysis are based on the results from the same. The engine is operated in European stationary cycle ESC-13, except when idle, thus 12 operating points as shown in the Figure 6. PPC simulation is based on PPC combustion[1] experiments that were carried out [3] in a Scania 13 liter heavy duty truck engine.

[1] PPC engine is modeled as a single cylinder engine, the mass flows are corrected to compare it with the six- cylinder engine

ENERGY BALANCE

Figure 5 shows the control volume of the engine, where $\dot{Q}$ is the heat flow, $\dot{m}$ is the mass flow, LHV is the Lower heating value of the fuel and P_{brake} is the brake power. The reference state for the enthalpy calculation is stated as surrounding. In the engine cycle the energy which goes in equals the energy coming out of the system.

$$\dot{m}_{fuel} = \dot{Q}_{Exhaust} + \dot{Q}_{EGR} + \dot{Q}_{CAC} + \dot{Q}_{coolant} + \dot{Q}_{catalyst} + \dot{Q}_{radiation} + P_{brake} + P_{misc-losses} \quad (1)$$

In Eq. (1), $P_{misc\text{-}losses}$ include the work done by water pumps, oil pumps and air-conditioning, and $\dot{Q}_{radiation}$ is the radiation heat transfer losses. Both the losses are not included in the calculation. A relatively simple analysis of heat flow across devices gives the heat energy available for recuperation. Coolant is not considered owing to its poor heat availability.

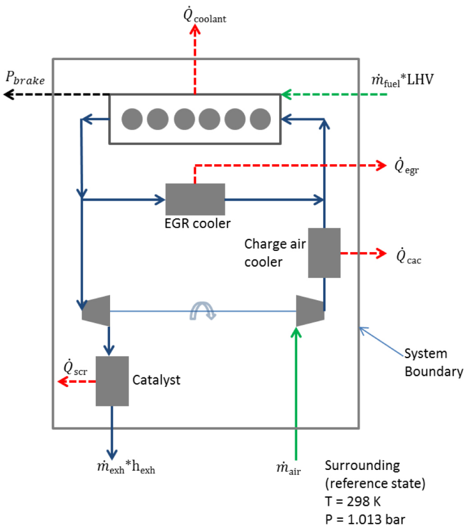


Figure 5. Energy flow (with system boundary)

Heat Flow across Devices

In the contemporary engines the heat energy from EGR, CAC and Exhaust is wasted through the EGR cooler, Charge air cooler and radiator. This otherwise wasted heat is recoverable and can be used as the source for Rankine cycle. This is calculated with average values of specific heat entering and leaving the device, temperature difference between the gases entering and leaving the device, and mass flow of the gases flowing across the device, eq. (2). The available heat energy for different sources considered in this study is presented in Figure 6.

$$\dot{Q} = \dot{m} * \frac{Cp_{in}+Cp_{out}}{2} * (T_{in} - T_{out}) \quad (2)$$

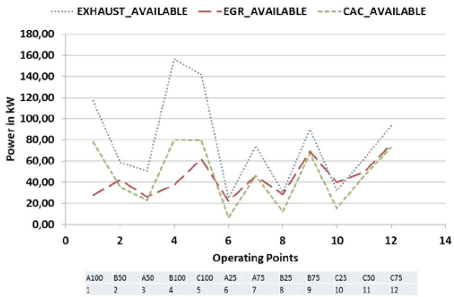


Figure 6. Energy availability in typical HD diesel engine.

TEMPERATURE AND PRESSURE DEPENDENCY

The efficiency of the Rankine cycle depends on the pressure and temperature levels available. For a given heat quantity the superheating temperature has to be higher, this could be achieved by increasing the pressure of the cycle. Attaining higher pressure levels greatly depends on the gas temperatures from the engine and the mechanical limitations of the boilers. Figure 7, Temperature-Enthalpy diagram for water demonstrates this. A simple Rankine cycle analysis is made with IPSEpro for constant mass flow of water over different temperatures. The mass flow doesn't influence in determining the percentage of useful power that could be recuperated from used heat energy[2]. The analysis is presented in Table 1, the maximum pressure level chosen is 60 bar.

---

[2]Used heat energy = Heat flow inlet - Heat flow exit

Table 1. Rankine cycle power outputs for different temperatures

Temperature	Heat energy flow[3]				Pressure Optimised for maximum power[4]	Useful Power	Power/Heat energy used
	In	Out	Used	Used heat to inlet heat %			%
K	kW	kW	kW		bar	kW	
450	9.6	7.9	1.7	18	2.5	0.09	5.7
500	12.4	8.9	3.5	28	4.7	0.31	8.9
550	15.2	9.8	5.4	35	8.2	0.66	12.2
600	18.1	10.5	7.5	42	13.4	1.14	15.1
650	21	11.3	9.7	46	22.6	1.73	18.0
700	23.9	11.5	12.3	51	32.2	2.46	20.0
750	26.8	11.9	14.9	55	48.9	3.32	22.3
800	29.8	11.7	18.2	61	60	4.31	23.7
850	32.8	10.9	21.9	67	60	5.35	24.4
900	35.9	10.2	25.7	72	60	6.46	25.1
950	39	9.6	29.4	75	60	7.61	25.9
1000	42.1	9	33.1	79	60	8.81	26.6

[3]Reference temperature in enthalpy calculation in IPSEpro is 273 K
[4]Maximum allowed pressure is 60 bar

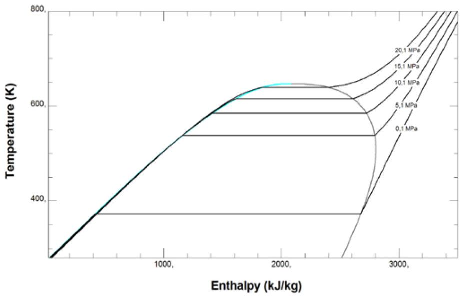


Figure 7. T-h diagram of water at five different pressure levels

It is known that the energy carrying capacity of a fluid increases with temperature, further from Table 1, it is evident that the effective energy used from the energy carrying fluid also depends upon these temperatures. This effective energy takes part in the Rankine cycle energy conversion. Thus, higher the temperatures are, higher will be the power output for the energy utilized, this is given in Figure 8. The heat from the other possible sources like charge air cooler (CAC), exhaust gas and engine coolant have been ignored since, they have low temperature sources and their potential with water as working fluid has been discussed already [10].

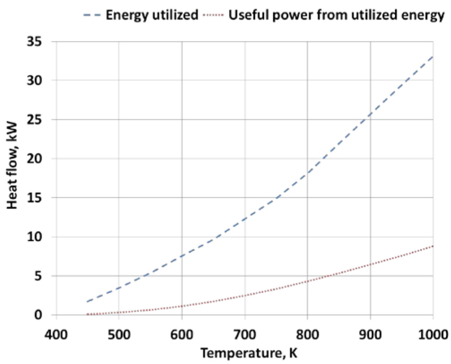


Figure 8. Useful power from utilized energy

Table 2. Operating characteristics of conventional and PPC combustion

Combustion type	IMEP (Gross)	Speed	Power	Indicated efficiency (Gross)	Mass flow (EGR)	Temperature inlet at EGR
	Bar	Rpm	kW		g/s	K
Conventional, C75	17.1	1798	275	45	182	744
Conventional, B75	19.9	1504	280	47.8	152	774
Conventional, A75	19.5	1211	228	47	96	795
PPC	20.5	1250	225	53.2	276	677.5

Table 3. Results of IPSEpro simulation for conventional and PPC combustion

Combustion	EGR inlet temperature	Cycle pressure	Heat energy used [5]	Mass flow of working fluid	Useful power out	Power to Used energy
	K	bar	kW	g/s	kW	%
Conventional C75	744	60	52.5	20.1	11.7	22.3
Conventional B75	774	60	49.8	17.2	11.5	23
Conventional A75	795	60	35.9	13.7	8.0	22.3
PPC	677.5	35	59	23.7	11.6	19.8

[5] *Reference temperature in enthalpy calculation in IPSEpro is 273 K*

PPC AND CONVENTIONAL COMBUSTION

The EGR from the conventional combustion have higher temperatures but reduced mass flow over the PPC combustion. For both combustion methods, Rankine cycle is employed to recuperate the heat energy from their EGR and an analysis is carried out to determine which of these two combustion types have more potential for waste heat recovery. To compare the WHR potential between the conventional combustion and PPC combustion, the engine is operated as shown in Table 2. EGR temperatures from both combustion types are given as input to the Rankine cycle. For the conventional combustion engine operating points A75, B75 and C75 is chosen as it has IMEP close to the PPC which could then be compared with each other. Using IPSEpro the Rankine cycle is constructed as shown in Figure 4, is simulated for both combustion types.

RESULTS

From Table 3 it is evident that the PPC combustion with combined cycle has power output comparable to that of conventional engine. The lower power to used energy is attributed to the lower ouput temperature of the PPC combustion.

CONCLUSION

Waste heat recovery potential using a Rankine cycle has been analyzed for two different types of combustion, conventional diesel combustion and partial premixed combustion. From the IPSEpro simulation it is evident that with both combustion types, the useful power output is analogous as shown in Table 3. But the advantages of PPC over conventional combustion are the indicated efficiency which is significantly higher and furthermore a much simpler aftertreatment system is sufficient compared to the conventional diesel engine. Conventional engine can recuperate 11.7 kW by using Rankine cycle WHR system and this quantifies from 3.5 % to 4.2 % of the brake power produced by the engine, on the other hand PPC combustion can recuperate 11.6 kW using the Rankine cycle WHR which quantifies to 5.17 % of the brake power produced by the engine. This additional power produced by WHR system corresponds to BSFC reduction of 3.4 % - 4.0 % in conventional combustion engine (75 % load) and 4.9 % reduction in PPC combustion engine. Even with the comparable power output from WHR system, the net waste heat recovery percentage is greater for PPC combustion than for a conventional combustion. Furthermore the thermal efficiency of the PPC combustion is higher than the conventional combustion, thus overall the engine efficiency is higher in PPC combustion with a combined Rankine cycle than a conventional combustion engine combined with a Rankine cycle.

REFERENCES

1. Noehre, C., Andersson, M., Johansson, B., and Hultqvist, A., "Characterization of Partially Premixed Combustion," SAE Technical Paper 2006-01-3412, 2006, doi: 10.4271/2006-01-3412.

2. Iwabuchi, Y., Kawai, K., Shoji, T., and Takeda, Y., "Trial of New Concept Diesel Combustion System - Premixed Compression-Ignited Combustion -," SAE Technical Paper 1999-01-0185, 1999, doi:10.4271/1999-01-0185.

3. Manente, V., Johansson, B., and Tunestal, P., "Partially Premixed Combustion at High Load using Gasoline and Ethanol, a Comparison with Diesel," SAE Technical Paper 2009-01-0944, 2009, doi:10.4271/2009-01-0944.

4. www.gtisoft.com

5. Tunér, M., Fröjd, K., Seidel, L., Mauss, F. et al., "Diesel-PPC engine: Predictive Full Cycle Modeling with Reduced and Detailed Chemistry," SAE Technical Paper 2011-01-1781, 2011, doi:10.4271/2011-01-1781.

6. Latz, G., Andersson, S., and Munch, K., "Comparison of Working Fluids in Both Subcritical and Supercritical Rankine Cycles for Waste-Heat Recovery Systems in Heavy-Duty Vehicles," SAE Technical Paper 2012-01-1200, 2012, doi: 10.4271/2012-01-1200.

7. Ringler, J., Seifert, M., Guyotot, V., and Hübner, W., "Rankine Cycle for Waste Heat Recovery of IC Engines," *SAE Int. J. Engines* 2(1):67-76, 2009, doi: 10.4271/2009-01-0174.

8. Teng, H., Regner, G., and Cowland, C., "Achieving High Engine Efficiency for Heavy-Duty Diesel Engines by Waste Heat Recovery Using Supercritical Organic-Fluid Rankine Cycle," SAE Technical Paper 2006-01-3522, 2006, doi: 10.4271/2006-01-3522.

9. Teng, H., Regner, G., and Cowland, C., "Waste Heat Recovery of Heavy-Duty Diesel Engines by Organic Rankine Cycle Part II: Working Fluids for WHR-ORC," SAE Technical Paper 2007-01-0543, 2007, doi: 10.4271/2007-01-0543.

10. Arunachalam, P., Shen, M., Tuner, M., Tunestal, P. et al., "Waste Heat Recovery from Multiple Heat Sources in a HD Truck Diesel Engine Using a Rankine Cycle - A Theoretical Evaluation," SAE Technical Paper 2012-01-1602, 2012, doi: 10.4271/2012-01-1602.

CONTACT INFORMATION

Prakash Narayanan A
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Prakash.narayanan@energy.lth.se

ACKNOWLEDGEMENTS

This study was conducted as a part of waste heat recovery project supported by the Swedish Energy Agency. The authors gratefully acknowledge them for the financial support. The authors would also like to thank Marcus Thern of Lund University for his helpful discussions.

DEFINITIONS/ABBREVIATIONS

EGR - Exhaust Gas Recirculation

Exh - Exhaust gas source

CAC - Charge air cooler source

C_p - Specific heat, J/kg-K

Delta - Difference

GWP - Global warming potential

h - Enthalpy, J/kg

LHV - Lower heating value

LTC - Low temperature combustion

$\dot{m}$ - Mass flow, g/s

η - Efficiency

ODP - Ozone depleting potential

$P_{turbine}$ - Turbine Power, kW

p, Pr - Pressure, bar

PPC - Partially Premixed Combustion

$\dot{Q}$ - Heat flow, kW

S - Entropy, J/kg-K

T_C - Sink temperature, K

T_H - Source temperature, K

WHR - Waste heat recovery

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

ISSN 0148-7191

SAE Customer Service:
Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
SAE Web Address: http://www.sae.org
Printed in USA

SAE International

Paper III

System Simulations to Evaluate the Potential Efficiency of Humid Air Motors

2013-01-2646
Published
10/14/2013

Prakash Narayanan Arunachalam, Martin Tuner, Per Tunestal, Bengt Johansson, and Marcus Thern
Lund University

doi:10.4271/2013-01-2646

ABSTRACT

In the quest for efficiency improvement in heavy duty truck engines, waste heat recovery could play a valuable role. The evaporative cycle is a waste heat recovery technology aimed at improving efficiency and decreasing emissions. A humid air motor (HAM) uses the waste heat from the exhaust of the engine to humidify the inlet air; this humid air, with higher specific heat, reduces NOx emission to a greater extent [1] [2]. Despite this benefit of emission reduction, the increase or decrease in efficiency of the humid air motor compared to the conventional engine is not discussed in the literature [3] [4] [5]. In this paper, an attempt is made to study the efficiency of the HAM using system model simulations of a 13-liter heavy duty Volvo engine with a humidifier. The commercial software GT-SUITE is used to build the system model and to perform the simulations. The efficiency improvement of the HAM comes from the expansion of the vapor mass flow produced as a result of humidification. An effort is also made to understand the relationship between the humidified engine and its efficiency.

INTRODUCTION

With stringent emission norms, the need for efficiency improvement and emission reduction becomes very important for engine makers. Water/steam injection is a promising technology which could be used for both emission reduction and efficiency improvement. While researchers have already studied emissions in water injection [6] [7] and water-fuel emulsification [8] [9] [10] technologies, there have been very few investigations of the efficiency improvement in water/steam injection. When the water is directly applied in the engine, the mixture becomes undesirably heterogeneous; by using evaporative cycles, the steam and air mixture become homogeneous [11]. One such steam injection evaporative cycle is the humid air motor (HAM). HAM is not a new concept; it was developed and patented by Per Rosen of Lund University over 20 years ago, with the idea of increasing the electrical efficiency of gas turbines and reducing emissions in reciprocating combustion engines [12]. Humid air cycles have been successfully demonstrated in gas turbines [11], and the HAM technology has been successfully tested in a marine application for emission reduction [1]. Although this technology seems likely to be able to reduce emissions similarly to other water/steam injection cycle techniques, the performance side of its application in heavy duty engines is uncertain. HAM is also a waste heat recovery technology, as it uses the waste heat in the engine cycle for steam production.

OBJECTIVE

The objective of this work is to theoretically analyze the potential efficiency improvements that a humid air cycle can create in reciprocating heavy duty diesel engines. This theoretical investigation also helps determine the boost levels required for HAM, and provides information that could be useful in dimensioning of the equipment for future HAM experiments.

SCOPE

Since the study is theoretical, the heat exchangers, condensers and humidification towers (HTs) are treated as 0-D thermo-dynamical models, and so no dimensioning has been performed for these equipment. The primary advantage of HAM is derived from reduction of pumping losses, although thermodynamic efficiency might also be improved. This study mainly focuses on the pumping aspect, and for the sake of simplicity the combustion parameters are kept constant.

The turbocharger map in the baseline Volvo model was originally developed for the requirements of the production engine, and so applying the same turbocharger model for HAM will affect the turbocharger performance. To cancel out this effect, the turbine and compressor work has been calculated with constant efficiency. For simplicity, the turbocharging analysis is performed with constant isentropic efficiency of 75% or 80% on both compressor and turbine respectively. It is not possible to attain this efficiency for all operating points in real application, but having a constant efficiency helps determine the future design point for turbomatching for HAM. An efficiency of 75% is in line with the peak efficiency of the production engine, while the choice of 80% efficiency reflects the assumption that the technology will be developed further in the future.

Working Principle of HAM Cycle

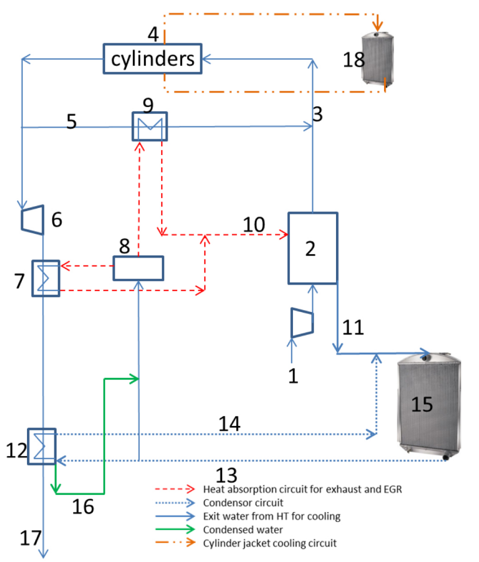


Figure 1. HAM engine circuit

The working principle of the HAM is shown in figure 2. The inlet air (1) enters the HT (2) where it makes contact with the hot water (10), and the resulting humid air (3) is fed as inlet to the engine (4). The engine exhaust is divided into two parts; a short-route EGR (5) and an exhaust which expands at the turbine (6). Downstream of the turbine is the first heat exchanger (7), through which water is sent from the tank (8) to take the exhaust heat. Heat is also taken off from the EGR cooler (9). Both water streams are heated to 90°C and sent to the HT. The exhaust stream, after flowing through the first heat exchanger (7), continues to flow through the second heat exchanger (12), which is a condenser; here, the exhaust is further cooled with a radiator/dedicated heat exchanger (15), and the condensed water (16) is again fed to the water circulation tank (8).

Hypothesis Behind the Concept

1. The vapor has higher specific heat capacity than air, as shown in figure 1, and so its presence reduces the temperature in the combustion process, resulting in lower peak temperatures and thus decreasing the emissions associated with high temperature combustion process. The specific heat capacity of vapor is almost twice that of air, and so this process is twice as effective as exhaust gas recirculation (EGR); this brings in the possibility of the removal of EGR and its associated components in the engine.

2. The vapor mass flow from the HT which has not been compressed in the compressor is expanded over the turbine, producing free additional work. This could be perceived as a Rankine cycle operated with in the engine cycle. (In this model, the HT is placed after the compressor)

3. The ratio of specific heat of vapor is lower than that of air, which makes it easier to compress in the cylinder than the EGR.

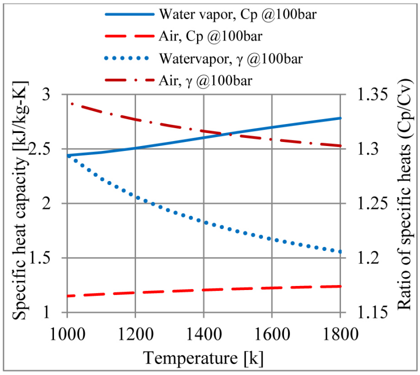


Figure 2. Specific heat capacities and ratios of specific heat capacities of air and water vapor

METHODOLOGY

The complete engine system is modeled with GT-POWER software, using as baseline model a validated and calibrated engine model provided by Volvo. The conventional engine is modified to simulate HAM, with the HAM vapor feed being added after the compressor. The vapor mass flows and temperature are calculated using a MATLAB model. For the sake of simplicity, the combustion parameters are left undisturbed and the lambda is kept constant with the model to make it relevant for comparison. Humidity is a function of temperature [13]; the higher the temperature of air, the higher the moisture content it can hold. In HAM, the factor controlling the inlet air temperature and the humidity is the amount of waste heat available in the exhaust. The water is

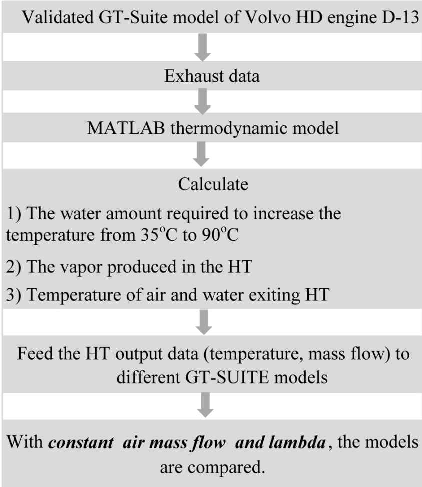


Figure 3. Methodology

heated to 90°C with this waste heat. The quantity of water heated depends on the exhaust energy, and thus the temperature and humidity of the exiting air vary for different operating points. The methodology is explained in figure 3.

The performances of the models are compared with the baseline case CDC40 (tables 1 and 2). The comparison includes two parameters: 1) the heat transfer losses, and 2) the turbine work (constant isentropic efficiency) and compressor work (constant isentropic efficiency). The sum of these two parameters gives the net benefit. Since the fuel energy and lambda are kept constant in all cases, the net benefit gives the potential additional power in each model.

Engine Models

Four different configurations are studied, as shown in tables 1 and 2. CDC40 is the reference/base model for comparing the other engine configurations; it models a conventional diesel combustion with a 40°C inlet air temperature, and is a validated engine model of a 13-litre Volvo truck diesel engine.

Table 1. Engine model index

Engine model	Index
Conventional engine with ~40°C inlet temperature	CDC40
Conventional engine with inlet temperature equal to HAM	CDC80
HAM without exhaust gas recirculation	HAM
HAM with exhaust gas recirculation	HAM+EGR

Table 2. Models for comparison

	Combustion parameters	Inlet temperature	Water vapor feed	EGR
CDC40	Validated	~40°C	No	Yes validated
CDC80	No change	~80°C	No	No change
HAM+EGR	No change	~80°C	Yes	No change
HAM	No change	~80°C	Yes	Nil

The Woschni [14] heat transfer model is used in the models to calculate the engine heat transfer, and the Chen-Flynn [15] engine friction model is used to calculate the friction in the cylinders. Without experiments, it is difficult to produce the combustion parameters using predictive models. The water content in the engine delays the start of ignition and thus could act like partially premixed combustion [16]. This makes it difficult to predict the combustion, and therefore the combustion parameters are kept constant. With humid air, the benefit is mainly derived from the in-cylinder pumping and turbine expansion.

Humidifier Modeling in MATLAB

An HT is used to humidify the inlet air to the engine. Unlike an evaporator, an HT produces vapor at temperatures lower than the boiling point of water. There are three driving mechanisms behind the water-to-vapor transformation [11]: 1) flashing caused by the higher temperature and pressure of the injected water in comparison to the conditions prevailing in the tower, 2) vaporization through cooling of the air, and 3) enthalpy exchange between water and air caused by the temperature drop between the water inlet and outlet of the tower. The enthalpy exchange is the main source of evaporation.

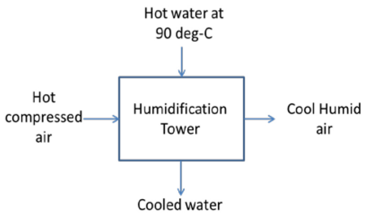


Figure 4. Humidification tower illustration

Thermodynamic Model of the Humidification Tower

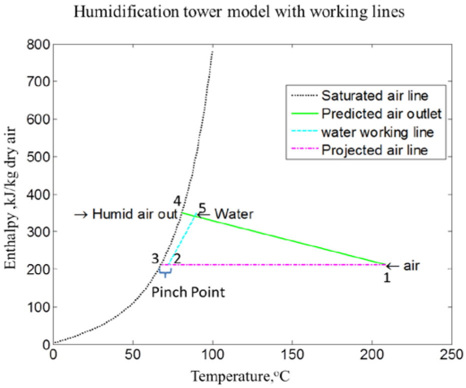


Figure 5. Humidification tower model with working lines

In the HT, heat and mass transfer take place between water and air as sensible heat and latent heat. The mass flows and temperatures of water vapor are predicted with a thermodynamic model of the HT, based on a model by Rosen [11] and built in MATLAB. Figure 5 shows the simplified humidification diagram for countercurrent operation. The temperature and mass flows of the inlet air and inlet water are known. Firstly, an assumption is made regarding the relative humidity of the vapor exiting the HT; on large towers this is typically close to 100% [11] [17], and so is assumed to be 100% for this work. Experiments conducted at Lund University have shown that the dynamic performance of the HT model is fast [18]. HT Models have also been verified by static experiments [13] [18]. Secondly, a relation connecting the exiting water temperature with the inlet air temperature is formed with a pinch point. This pinch point is the temperature between the lowest possible temperature of the exiting water (saturation temperature of incoming compressed air) and the incoming compressed air (between points 2 and 3 in figure 5) [11]. A pinch point of 5°C is set to account for the inefficiencies connected with the limitation of physical properties [11] [13].

The heat and mass balance in the HT model is explained in figure 6.

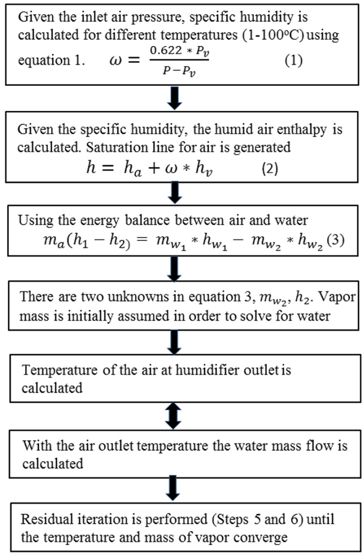


Figure 6. Humidification tower model working procedure

Operating Points

The engine is simulated in the 13-point European stationary cycle (ESC-13) with the exception of the idle stage, thus giving 12 operating points. These operating points are considered as reference points.

Table 3. The 12 ESC operating points

Operating point		**Power**	**Torque**	**Speed**
		kW	Nm	RPM
1	A25	76	600	1210
2	A50	152	1200	1210
3	A75	228	1801	1210
4	A100	305	2402	1210
5	B25	93	592	1500
6	B50	187	1185	1500
7	B75	280	1778	1500
8	B100	373	2369	1500
9	C25	92	486	1800
10	C50	183	974	1800
11	C75	275	1459	1800
12	C100	367	1948	1800

Turbocharging

The conventional engine has a variable geometry turbine coupled with a compressor to provide the required boost. As explained in the methodology section, the exit temperature of the HT (~80°C) is higher than the inlet temperature of the CDC40 (~40°C). This brings in the requirement of additional boost when compared to the conventional engine, to equate the air mass flow. The turbocharger in the model is incapable of boosting to the higher levels required by HAM, and so a new turbo matching would have to be performed for HAM. Instead, to simplify things, the turbine and compressor have been decoupled and run with an external drive. To compare the potential benefits of HAM, the compressor work and turbine work have been studied theoretically using equations 1 and 2 with the two constant efficiencies as explained in the scope section (75% or 80% respectively). The same assumed efficiencies have been employed for each of the four cases investigated.

$$Turbine\ Work = \eta_t * C_p * T * \left[1 - TPR^{((\gamma-1)/\gamma)}\right] \tag{4}$$

$$Compressor\ Work = \frac{C_p * T * \left[CPR^{((\gamma-1)/\gamma)} - 1\right]}{\eta_c} \tag{5}$$

RESULTS

Humidity, mass flow of vapor and mass flow of air at the inlet are shown in figure 7. Moisture at the inlet is dependent on the inlet air temperature, which in turn depends on the exhaust heat. EGR fraction is also presented for all the operating points.

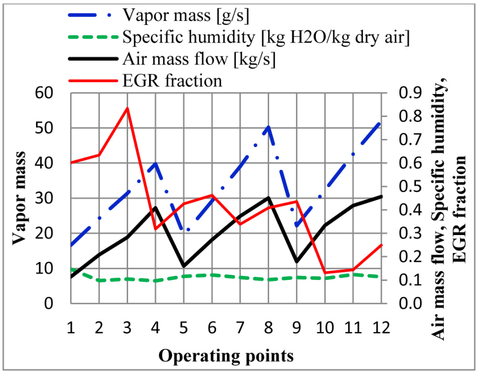


Figure 7. Inlet charge air at different operating points

Figure 8 shows the comparison of in-cylinder pressures for different models at operating point A100. The HAM and HAM+EGR models have additional water vapor mass flow to pump in along with the air mass. HAM+EGR has higher inlet temperature compared to the baseline condition (CDC40); this requires additional boost pressure at the intake, which results in higher pressures than the rest of the models. The HAM model does not need to pump the EGR, giving it the advantage over higher boost temperature. It can be seen that the pressure of HAM closely follows the baseline condition. CDC80 has a higher temperature at the inlet compared to baseline, and thus requires a higher boost pressure at the inlet for the same mass flow. Hence, the pressure level of CDC80 is higher compared to baseline.

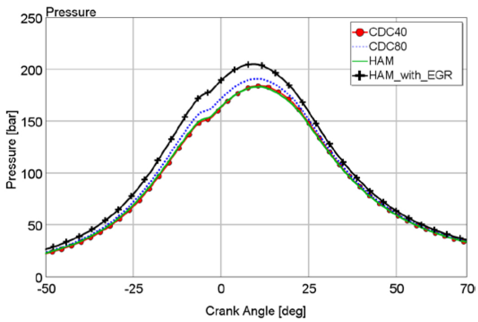


Figure 8. In-cylinder pressure

Figure 9 shows the global in-cylinder temperature of different models at operating point A100. CDC80 has a higher peak temperature compared to the other models, as it has higher inlet temperature but the same EGR mass as baseline. The non-EGR HAM model with water vapor (higher specific heat) has almost the same peak temperature as the baseline case (which has EGR); this brings in the possibility of EGR removal. HAM+EGR has both the EGR and water vapor, and hence lower temperatures than the other models.

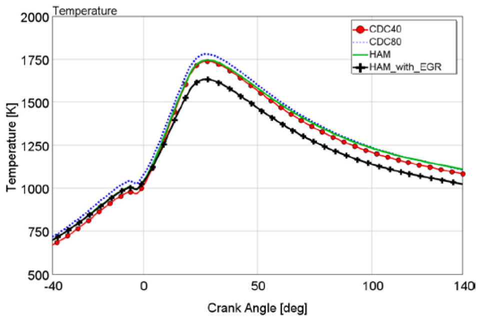


Figure 9. In-cylinder temperature

Power and Efficiency Improvements

The improvements in power and efficiency have two main sources:

1. Power from the cylinder: This is the additional power derived from the cylinder because of better heat transfer and reduction of pumping losses.

2. Power from the turbine: This is the additional enthalpy contained by the exhaust, reflecting the power work produced in the turbine in excess of the need of compressor work.

Brake Power

The brake power is affected by the in-cylinder combustion, reduced heat transfer and reduced pumping losses. Higher back pressure affects the pumping loop of the engine. Figure 10 presents the brake power in comparison to the baseline case. It is notable that in the case of HAM with EGR, the in-cylinder compression work is higher than the baseline case, as is evident from the logP-logV diagram in figure 11. Along with higher back pressures, this model produces lower power output than CDC40. The HAM engine omits the EGR loop, and so the back pressure was controlled to produce the same power output as the baseline case. However, even at low speeds, where the baseline case is operated with a greater fraction of EGR, the heat transfer is better than HAM. This results in lower power output than CDC40. Ultimately, this disparity would be the difference in heat content at the exhaust. The power in CDC80 is also affected by the back pressure. The different inlet and exit pressures of the engine are given in table 4 for different loads at engine speed A.

Table 4. Pressure at the inlet and exhaust for speed A

Model	Pressure	A25	A50	A75	A100
CDC40	In	1.5	2.2	3.1	3.8
	Out	1.6	2.3	3.2	3.7
CDC80	In	1.6	2.4	3.4	4.1
	Out	1.6	2.4	3.4	4.0
HAM	In	1.2	2.0	2.8	4.0
	Out	1.4	1.7	2.4	3.2
HAM+EGR	In	1.8	2.7	3.8	4.7
	Out	1.8	2.7	3.8	4.5

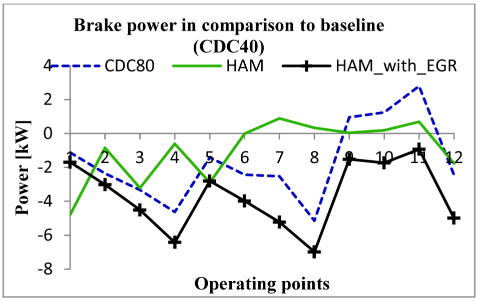


Figure 10. Brake power of different models in comparison with baseline (CDC40)

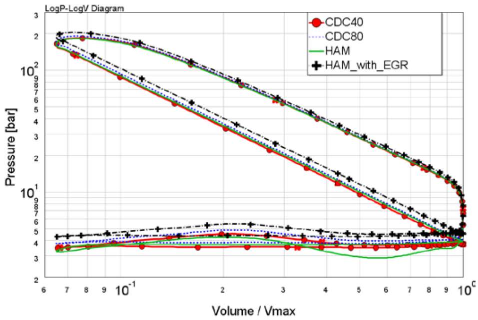


Figure 11. LogP-LogV diagram

Turbocharging

The turbocharging analysis is performed with constant isentropic efficiency of 75% and 80% on both compressor and turbine. The results are presented in figures 12 and 13 (bars absent indicates no change). The compressor work and turbine work is calculated using equations 4 and 5. The required boost (table 4) is calculated with 75% and 80% efficiency. Similarly, with exhaust data the potential expansion is calculated with 75% and 80% efficiency. The difference in the work gives the potential benefit from turbocharging. It is clear from the results that HAM+EGR has higher potential with the available exhaust heat.

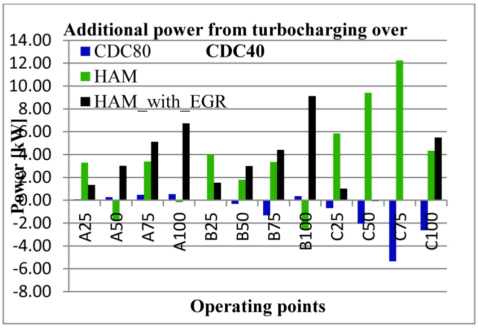


Figure 12. Potential turbine benefit with 75% constant efficiency

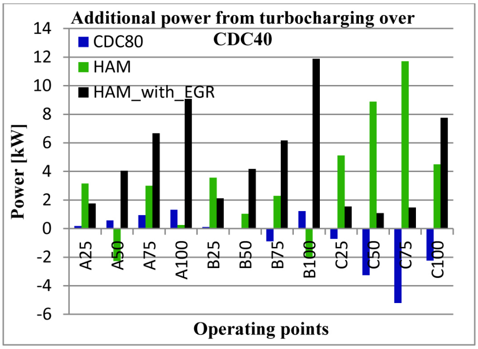


Figure 13. Potential turbine benefit with 80% constant efficiency

Heat Transfer

Heat transfer is mainly affected by the in-cylinder temperature. Figure 14 shows the heat transfer comparison of different models. HAM+EGR has lower heat transfer losses owing to the lower in-cylinder temperature, as shown in figure 15. Conversely, CDC80 has higher heat transfer because of the higher in-cylinder temperature.

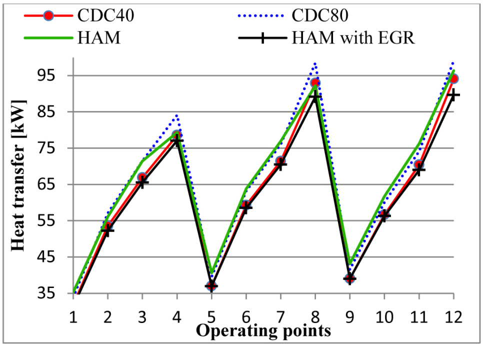


Figure 14. Heat transfer comparison

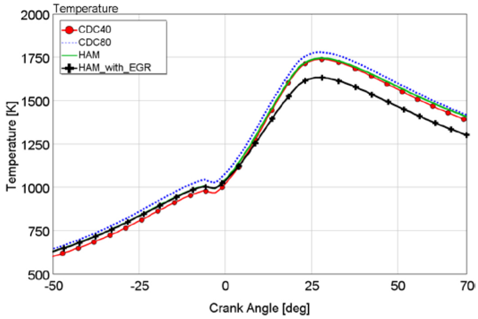


Figure 15. In-cylinder temperature

Net Benefit

The net benefit is given by summing the in-cylinder power, compression work and turbine expansion; this can be used to compare the models. Figure 16 shows the net benefit of all the models over baseline (CDC40). It is clear that HAM and HAM+EGR both have very minimal benefits. Additional effort is required to pump in the same mass of air as CDC40 at higher temperatures. It is therefore reasonable to compare CDC80, HAM and HAM+EGR to see the potential benefits; see figure 17.

Considering operating points with the same in-cylinder power output (C25 and C50), as shown in figure 10, HAM has higher potential benefit over CDC40 (figure 16). Figure 17 shows the potential benefit of HAM and HAM+EGR over CDC80. At lower speed and load (A25 and A50), as shown in figure 17, CDC80 is operated with a higher EGR amount which gives heat transfer advantage over HAM; hence HAM has lower benefits at these operating points. At higher speeds, HAM delivers greater benefit compared to CDC80. HAM +EGR is affected in the pumping of additional mass, and hence delivers lower benefit compared to HAM in general.

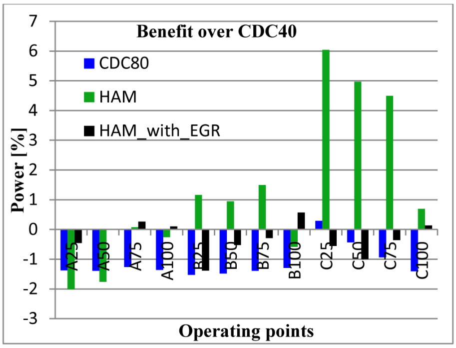


Figure 16. Potential benefit over CDC40

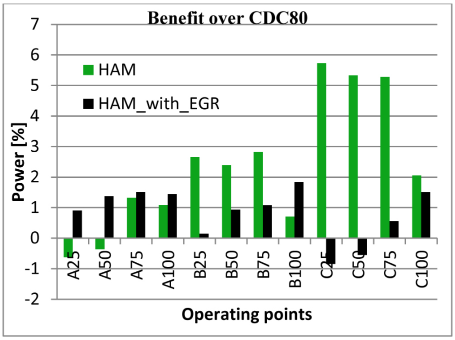


Figure 17. Potential benefit over CDC80

DISCUSSION

The HAM model produces pressures and temperatures identical to the baseline model, CDC40. Its peak temperatures are lower than those of CDC80 (which has same inlet temperature as HAM), offering the possibility of EGR removal. Performance improvement also depends on the EGR strategy; at low speeds the EGR fraction used in this engine is high, enabling lower heat losses in the cylinder and higher efficiency. Potential benefit is derived from both the heat transfer and the pumping (in-cylinder and turbocharging). The ignition could be delayed because of humid air in the cylinder, allowing more time for mixing; and hence the combustion could act like partially premixed combustion. This type of combustion increases the in-cylinder expansion by having shorter combustion duration, which would still increase the HAM efficiency. These results could also be helpful in determining the thermodynamic performance requirement of the condenser and HT. The primary challenge in HAM is utilizing its potential. HAM enables us to have more potential in the exhaust, but recovering the heat is the biggest challenge posed. There are a few options available for recovery, including 1) turbo-compounding, 2) an additional power turbine producing electricity downstream of the conventional turbocharger, and 3) a positive pumping loop or downsizing/down-speeding with the additional boost. The best form of work conversion remains to be further investigated.

FUTURE WORK

The HAM work will be experimentally carried out in the heavy duty truck. The sizing of HT and condenser will be carried out for the same.

CONCLUSIONS

A humid air system (HAM) was studied using system model simulation. The results show clear potential for improving efficiency.

1. The HAM system effectively reduced the in-cylinder temperatures compared to EGR.

2. At moderate and high speed operation, where EGR mass is not too high, HAM is beneficial over CDC80.

3. The increment in efficiency is brought by a combination of factors: 1) lower heat losses, 2) reduction of in-cylinder pumping losses, and 3) free vapor expansion over the turbine. It is difficult to isolate the key factor for improving the performance.

REFERENCES

1. Entec UK Ltd (2005a) Report “Service Contract on Ship Emissions: Assignment, Abatement and Market-based Instruments, Task 2b NOx Abatement” for the European Commission, DG Environment.

2. Rahai, H., Shamloo, E., and Bonifacio, J., “Investigation of the Effect of a Humid Air System on Diesel NOx and PM Emissions of a Small Diesel Engine,” SAE Technical Paper 2011-01-0692, 2011, doi:10.4271/2011-01-0692.

3. Nord, K., Haupt, D., and Egebäck, K., “Particles And Emissions from a Diesel Engine Equipped with a Humid Air Motor System,” SAE Technical Paper 2001-01-3616, 2001, doi:10.4271/2001-01-3616.

4. Park, H., Ghal, S., Park, H., Choi, S. et al., “A Study on NO_x Reduction of Marine 4-Stroke Diesel Engine Using Charge Air Humidification,” SAE Technical Paper 2011-24-0172, 2011, doi:10.4271/2011-24-0172.

5. Asad U., Kelly C., Tjong J., “Effects of Intake Air Humidity on the NOx Emissions and Performance of a Light-duty Diesel Engine”, ASME Internal Combustion Engine Division, 2012 Fall Technical Conference, 2012, Vancouver, Canada, ICEF2012-92027.

6. Bedford, F., Rutland, C., Dittrich, P., Raab, A. et al., “Effects of Direct Water Injection on DI Diesel Engine Combustion,” SAE Technical Paper 2000-01-2938, 2000, doi:10.4271/2000-01-2938.

7. Udayakumar, R., Sundaram, S., and Sriram, R., “Reduction of NOx Emissions by Water Injection in to the Inlet Manifold of a DI Diesel Engine,” SAE Technical Paper 2003-01-0264, 2003, doi:10.4271/2003-01-0264.

8. Nazha, M., Rajakaruna, H., and Wagstaff, S., “The Use of Emulsion, Water Induction and EGR for Controlling Diesel Engine Emissions,” SAE Technical Paper 2001-01-1941, 2001, doi:10.4271/2001-01-1941.

9. Hountalas, D., Mavropoulos, G., and Zannis, T., “Comparative Evaluation of EGR, Intake Water Injection and Fuel/Water Emulsion as NOx Reduction Techniques for Heavy Duty Diesel Engines,” SAE Technical Paper 2007-01-0120, 2007, doi:10.4271/2007-01-0120.

10. Iwai, M., Yoshida, K., Iijima, A., and Shoji, H., “Study on Performance of Diesel Engine Applied with Emulsified Diesel Fuel: The Influence of Fuel Injection Timing and Water Contents,” 2011.

11. Rosen, P. M., "Evaporative gas turbine cycles - A thermodynamic evaluation of their potential", ISRN LUTMDN/TMKV-7010-SE/1993.

12. Lindquist, O., T., Rosen, P., Olsson, L., "Humid Cycles A Novel Way to Increase Performance of Marine Propulsion Systems", 2001, CIMAC, Hamburg.

13. Nyberg, B., "Modelling of Wet Gas Turbine Cycles: Evaluation and Thermodynamic Calculation", ISRN LUTMDN/TMHP-12/7081-SE/2012.

14. Woschni, G., "A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine," SAE Technical Paper 670931, 1967, doi:10.4271/670931.

15. Chen, S. and Flynn, P., "Development of a Single Cylinder Compression Ignition Research Engine," SAE Technical Paper 650733, 1965, doi:10.4271/650733.

16. Kaneko, N., Ando, H., Ogawa, H., and Miyamoto, N., "Expansion of the Operating Range with In-Cylinder Water Injection in a Premixed Charge Compression Ignition Engine," SAE Technical Paper 2002-01-1743, 2002, doi: 10.4271/2002-01-1743.

17. Traverso A. "Humidification Tower for Humid Air Gas Turbine Cycles: Experimental Analysis". Energy 35 (2010) 894-901. Elsevier Science.

18. Thern, M., "The Ethanol and Water Humidification Process in EvGT Cycles", ISRN LUTMDN/TMHP-03/7015-SE/2003.

19. Manuals and tutorials, Gamma Technologies, 2013.

20. Cengel Yunus A. and Boles Michael A., "Thermodynamics An Engineering Approach" 7th Edition, McGraw Hill, ISBN: 978-0-07-352932-5.

CONTACT INFORMATION

Prakash Narayanan A
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Prakash.narayanan@energy.lth.se

ACKNOWLEDGMENTS

This study was conducted as a part of waste heat recovery project supported by the Swedish Energy Agency. The authors gratefully acknowledge the financial support of this agency.

DEFINITIONS/ABBREVIATIONS

HAM - Humid air motor

HT - Humidification tower

TPR - Turbine pressure ratio

CPR - Compressor pressure ratio

CDC - Conventional diesel combustion

EGR - Exhaust gas recirculation

SYMBOLS

η_t - Turbine efficiency

η_c - Compressor efficiency

C_p - Specific heat, J/kg-K

γ - Ratio of specific heats

ω - Specific humidity kg H2O/ kg dry air

h - Enthalpy, kJ/kg

m - Mass flow, kg/s

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

ISSN 0148-7191

SAE Customer Service:
Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
SAE Web Address: http://www.sae.org
Printed in USA

Paper IV

Experimental Analysis of Humid Air Motor

Prakash Narayanan Arunachalam, Per Tunestal, Marcus Thern, Martin Tuner
Lund University

Abstract

Humid air motor (HAM) is an engine operated with humidified inlet charge. System simulations study on HAM showed the waste heat recovery potential over a conventional system. An HAM setup was constructed, to comprehend the potential benefits in real-time, the HAM setup was built around a 13-litre six cylinder Volvo diesel engine. The HAM engine process is explained in detail in this paper. Emission analysis is also performed for all three modes of operation. The experiments were carried out at part load operating point of the engine to understand the effects of humidified charge on combustion, efficiency and emissions. Experiments were conducted without EGR, with EGR, and with humidified inlet charge. These three modes of operation provided the potential benefits of each system. Exhaust heat is used for partial humidification process. Results show that HAM operation without compromising on efficiency reduces NOx and soot significantly over the EGR operated the engine.

Keywords: Humidification, humid air, waste heat, multi-cylinder experiment, emission, NOx, Soot

Introduction

Internal combustion engines are used in road and maritime transport sector for over a century. There are two ever refining aspects present in I.C engines, namely fuel economy and emission reduction. Approximately one-third of the fuel energy is converted to useful work in an I.C engine while remaining 2/3 of the fuel energy is wasted in the form of heat. Utilizing this heat, termed as “waste heat recovery”, can play a significant role in addressing the issue of improving the thermal efficiency of engines and thereby improving the fuel economy [1,2]. In the past, several technologies were developed to harness the wasted heat [1,3,4]. Humid Air Motor (HAM) is one such waste heat recovery technology, which not only has the potential to increase the fuel economy but also has an additional benefit of reducing emissions.

Humid air cycles were first developed in 1990's by Lund University researchers with the primary objective of increasing efficiency in gas turbines [5,6]. They also applied this technique to ship engines with the focus of bringing down the NOx emissions, which is also a major emission concern for diesel engines [7]. A large part of the road transport is operated by trucks and buses with diesel engines. Hence, there is a huge interest in improving diesel engines.

In this study, the HAM concept is incorporated into a multi-cylinder truck diesel engine. Most research in the field of humidification in engines has only focused on emission reduction and not on efficiency improvement. Previous studies of HAM have reported a significant decrease in NOx emissions, but the HAM was not extensively studied as a means of waste heat recovery to improve the efficiency [8, 9, 10].

HAM uses the heat to humidify the inlet air; this is done after the compressor thus it has additional enthalpy in the form of vapor. The humidification process also brings down the temperature of the compressed inlet charge making the charge air cooler dispensable and helps in reducing the cooling losses. The additional vapor enthalpy along with lesser cooling losses has the potential to increase the thermal efficiency.

Scope

HAM experimental study shows, in what manner the HAM can be employed as waste heat recovery method along with the experimental results. Low speed and a low load of ESC13 point cycle are experimented to understand the process.

Hypothesis behind the concept

1. The vapor has higher specific heat capacity than air, as shown in figure 1, and so its presence reduces the temperature in the combustion process, resulting in lower peak temperatures and thus decreasing the NOx emissions associated with high-temperature combustion. The specific heat capacity of vapor is almost twice that of air, and so this method is twice as effective as exhaust gas recirculation (EGR); this brings in the possibility of the removal of EGR and its associated components in the engine.

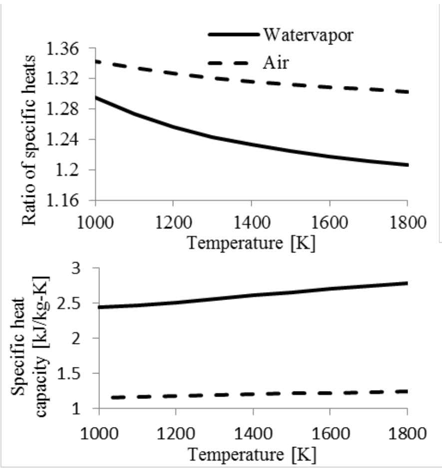


Figure 1. Specific heat capacities and ratios of specific heat capacities of air and water vapor

2. The vapor mass flow from the humidification tower (HT), which has not been compressed in the compressor, is expanded over the turbine, producing free additional work. This cycle could be perceived as a Rankine cycle operated within the engine cycle. (In this model, the HT is placed after the compressor)
3. Apart from increased specific heat capacity to bring down the temperatures, the concentration of the atomic oxygen by reaction also decreases the NO significantly [12].

Water injection strategies

There are a couple of ways to feed water into the engines. Three standard methods are

1. *Direct water injection:* In this processes high-pressure water is directly sprayed into the engine combustion chamber to reduce the NOx and soot [12]. One of the disadvantages of this process is the homogeneity of the mixture formation.

2. *Fuel water emulsion:* Several studies are performed on the fuel-water emulsion [11,12]. One of the disadvantages of this method is lower water to fuel ratio, and hence, the effects it brings are also low. Hardware changes are to be made to avoid corrosion, cavitation, and boiling in the fuel system.

3. *Humidification of the inlet charge:* Humidification of the inlet charge can be done by injecting water either before or after the turbo-compressor. To raise the humidity level (specific humidity) the inlet charge temperature has to be raised, and hence injection after the compressor is a viable solution. Although one has to be careful in using an intercooler since it could condense the charge and form water droplets. The homogeneity of the mix depends on the water pressure, mist formation and the engine speed …etc.

Another method of humidification process uses a humidification tower, where the water and gas can come in contact and because of heat and mass transfer the humidification process takes place. The mixture is homogenous in this process. Also, this method allows the fuel to water ratio to go below 1.

Selection of the injection strategy

Because of the homogeneity of the mixture and higher water to fuel ratio, humidification of the inlet charge by humidification tower seems plausible. Also for humidification purpose the exhaust heat can be utilized, thus recovering otherwise wasted heat.

Humidifier

An HT is used to humidify the inlet air to the engine. Unlike an evaporator, an HT produces vapor at temperatures lower than the boiling point of water. There are three driving mechanisms behind the water-to-vapor transformation [5]: 1) flashing caused by the higher temperature and pressure of the injected water in comparison to the conditions prevailing in the tower, 2) vaporization through cooling of the air, and 3) enthalpy exchange between water and air caused by the temperature drop between the water inlet and outlet of the tower. The enthalpy exchange is the largest source of evaporation. The water into the humidification tower is always kept at 90 °C which in turn governs the maximum mass flow of water circulated, and that determines the inlet temperature and humidity it carries with it.

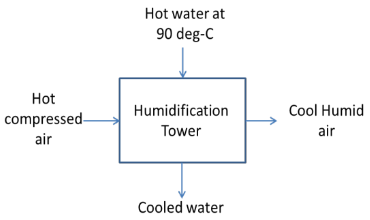


Figure 2. Humidification tower; temperature overview

For the humid air motor simulations, the humidification tower was modeled to predict the amount of water vapor that can be added to the system. In the figure, the point 1 is the compressed inlet air, which upon contact with water in the tower, 5, exchanges the enthalpy. Because of the heat and mass transfer the air temperature drops down to the temperature closer to the inlet water temperature 4. To account for the thermal losses in the system a pinch point is introduced 2-3 [5].

$$h_{air} * m_{air} + h_{water\_in} * m_{water\_in} = h_{humidair} * m_{air} + h_{water\_out} * m_{water\_out}$$

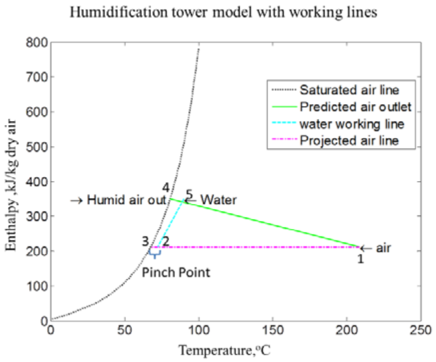


Figure 3. h-T diagram of humidification process

HAM Simulation Results

Before the HAM experiments were conducted, the HAM system is modeled with commercial gas exchange software (1-D), GT-Suite, to evaluate the potential efficiency benefits of HAM over conventional engine operation [15]. Several assumptions were made for the simulations. The operating point chosen was A25 (from ESC 13 point cycle). The assumptions and simulation settings used in the study can be found in detail in Arunachalam et.al [15]

The primary losses in HAM simulations were found to be the exhaust losses. The HAM mixture had lower specific heat ratio than the conventional engine which brings down the thermal efficiency of the engine. Figure 4 shows the benefit of HAM system simulations over conventional diesel engine operation. HAM system is beneficial from mid speeds to higher speeds.

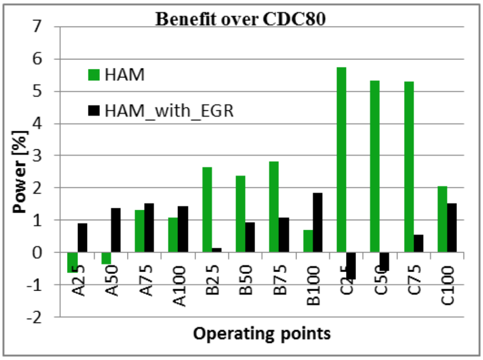


Figure 4. The benefit of HAM over conventional diesel compared with similar Inlet temperature (80 °C).

Experimental Layout

To corroborate the HAM simulations which were made with assumptions mentioned earlier experiments were carried out at the same operating point as reference A25. Despite the results suggested by simulation, operating point A25 was selected to understand the HAM. With experiments, it was also possible to analyze the combustion parameters and emissions of the HAM operation. In the simulations, the lambda was kept constant, and the additional or reduction in power was compared to the HAM's and conventional. In experiments, the inlet pressure and fuel duration are held constant to be able to perform the analysis at approximately at the same load; This is done because 1. Turbocharger control is difficult to have same air mass flow in the HAM cycle as much as in the conventional engine cycle. 2. Lambda control would be difficult with different inlet pressures. The engine has a unit injector, which might inject different amounts of fuel for the same fuel duration if the in-cylinder pressure is varied because of the inlet pressure. For this reason, we tried to have same inlet pressure for conventional and HAM engine.

Methodology

The engine cylinder pressure is recorded with an in-cylinder pressure sensor. Using the cylinder pressure data and the cylinder volume, the gas temperature, rate of heat release (RoHR) and mean effective pressures are calculated. Experiments are conducted with the A25 operating point of the ESC 13 point cycle as a reference. EGR percentage obtained from Volvo calibration experiments were used in our experiments. Injection duration and injection pressure were kept constant. Long route EGR system was used along with an upsized turbocharger (of a 16L engine). HAM mode doesn't utilize EGR. To understand the performance of HAM the engine were operated with three modes namely, HAM, EGR, and W/O EGR. Hydrocarbon (HC), Nitrogen oxide (NOx), carbon monoxide (CO) are analyzed with AVL AMA i60 measurement system and Soot (PM) emissions are measured with AVL micro soot sensor

Construction

The HAM construction differs from the standard engine setup in some aspects, to explain, the conventional engine setup is shown in figure 5. The standard engine setup has long route EGR path and thus the EGR cooler is placed after the Turbocharger. The HAM setup is shown in figure 6. As noticeable the heat exchanger after the turbo was replaced with the humidification tower and instead of EGR cooler a waste heat recovery cooler was installed. The waste heat recovery cooler uses the exhaust enthalpy to heat up the water from the reservoir, and the pumps are used to circulate.

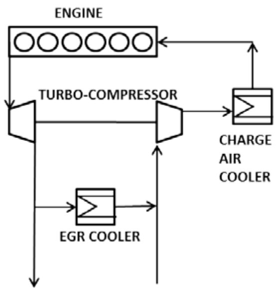


Figure 5. Conventional engine layout

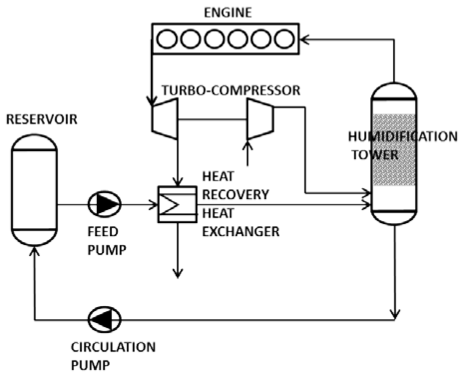


Figure 6. Humid air motor layout

Experimental setup

Engine	Volvo HD diesel engine
Fuel	MK1 Diesel
Compression ratio	16:1
SOI	1 CAD ATDC
Injection Duration / Injection Pressure	Constant / Constant
Speed (RPM) / Reference Torque (Nm)	1200 / 600 - Operating condition A25
Operating modes	Without EGR, with EGR, HAM

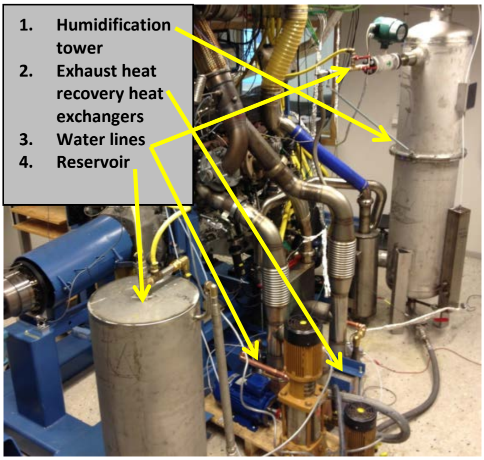


Figure 7. Installation of components on HD engine

Process path

Figure 8 shows the process layout of HAM. Water from the reservoir is fed using a pump overcoming boost pressure. This water is then fed through the heat recovery heat exchanger, where it absorbs the exhaust heat. Hot water (90 °C) is sprayed in the humidification tower via a set of nozzles arranged circularly in the tower (appendix). Inlet air after the compressor is fed to the tower from the lower side. Air and hot water come into contact with the structured packing inside the tower (appendix). After heat and mass transfer the air along with the vapor (humidified charge) is let in the engine. Water remaining in the tower is then pumped to the reservoir tank. Approximately 10% of the water is vaporized and fed in as humidified charge.

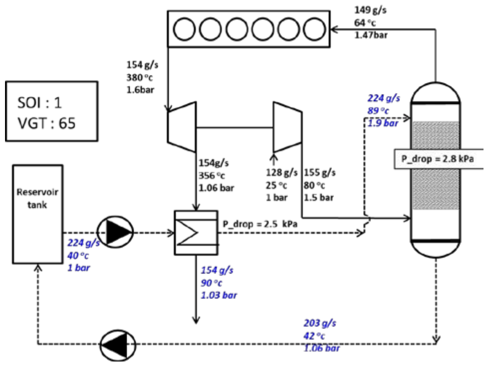


Figure 8. HAM layout with flows, temperatures, and pressures

Inlet conditions

Table 1. Inlet conditions during the experiments

Mode	W/O EGR	EGR	HAM
Inlet pressure, bar	1.47	1.49	1.47
Inlet air temperature, °C	63.2	60.2	63.8
Lambda[1]	2.32	1.46	1.97

Results and Discussion

Rate of Heat Release

Figure 9 shows the heat release rate for the three different modes of operation. For EGR and HAM cases, the lower specific heat of the mixtures increases the ignition delay. Longer ignition delay aids in better mixing and hence the combustion duration could be reduced, achieving higher thermal efficiency.

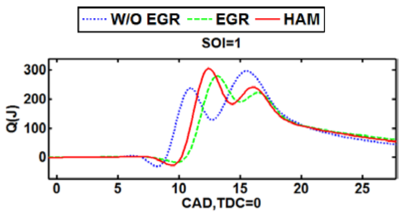


Figure 9. The rate of heat release

[1] is calculated from the emissions.

Heat Transfer

Figure 10 shows the in-cylinder gas temperatures. The temperature profile is calculated from the measured cylinder pressure. Woschni[13] heat transfer equation is then used to calculate the cylinder heat transfer. Gatowski model [14] is used to calculate the specific heat capacity of the gas for different temperature. From figure 10 it can be seen that HAM and EGR bring down the peak temperature of the gas compared to the engine without EGR. The total heat transfer, shown in Table 2 gives us the benefit produced by heat transfer reduction.

Table 2. Heat transfer comparison

Mode	W/O EGR	EGR	HAM
Heat transfer in kW	33.6	32.9	32.8
% Benefit	-	2.2%	2.3%

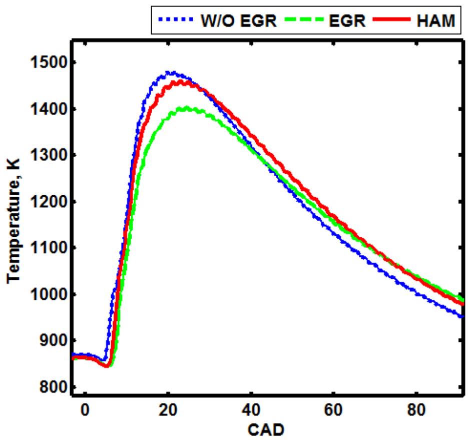


Figure 10. In-cylinder temperature profile

Pumping mean effective pressure (PMEP)

PMEP is a measure of the pumping performance of an engine and hence a good indicator of gas exchange efficiency. The HAM cycle is fed with humid air after the compressor which can be expanded over the turbine. Thus, HAM cycle has higher mass flow over the exhaust compared to the other modes of operation. This means, work done by the exhaust turbine is more than it requires compressing the inlet charge. Hence, the HAM cycle can have better pumping loop; this can be seen in figure 11. It is important to note that pressure drop for the HAM tower is lower than the charge air cooler (lower by ~3 kPa). Pressure drop for exhaust heat exchangers for HAM system (~2 kPa) is introduced because of the heat recovery heat exchanger.

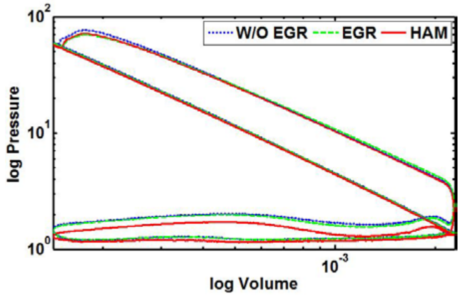


Figure 11. LogP-logV diagram

Table 3. Pumping mean effective pressure comparison

Mode	W/O EGR	EGR	HAM
PMEP, bar	0.53	0.47	0.26
% Benefit	-	11.3%	51%

Turbocharger efficiency

The turbocharger is not optimized for the engine or for the HAM operation. The efficiency of the turbocharger is affected with both EGR and HAM, notably the turbine efficiency is affected significantly. The lower efficiency is brought down by operating the turbine in off-design operating point.

$$Turbine\ efficiency^2 = \frac{H_{in} - H_{out}}{H_{in} - H_{out,S}}$$

[2] It is the ratio of the actual enthalpy drop to the isentropic enthalpy drop. Compressor efficiency is also calculated in the same way.

Table 4. Turbine and compressor efficiency comparison

Mode	W/O EGR	EGR	HAM
Turbine efficiency %	52.3	38.5	37.5
Compressor efficiency %	63.8	50.9	67.9

Brake specific fuel consumption (BSFC)

Tables 5 shows the brake specific fuel consumption (BSFC) indicating that HAM has higher efficiency than the conventional process with EGR, but lower than the standard operation without EGR. BSFC improvement could be because of the start of injection (SOI = +1). Since the modes have a late start of injection (in the expansion stroke), some of the useful energy could not be extracted. Also, the lower specific heat of EGR and HAM makes it hard to extract as much energy as with only air. Although the Heat transfer and pumping losses are reduced in HAM and EGR cases, the useful work transfer is better in the case without EGR or HAM. BSFC improvement can partly improve with the varied start of injection.

Table 5. Brake specific fuel consumption comparison

Mode	W/O EGR	EGR	HAM
BSFC, g/kWh	229	234	231
% Benefit	-	-2.2%	-1.2%

Emissions

The previous section presented an overview of fuel consumption of the three modes. Even though the fuel consumption results show that it is beneficial to run with only air, the emissions from this type of combustion are detected. Table 7 provides the overview on emissions.

Table 6. Brake specific emission results

Mode	W/O EGR	EGR	HAM
BSCO, g/kWh	0.57	4.6	1.25
BSNOx, g/kWh	7.14	0.19	0.44
BSTHC, g/kWh	0.15	0.10	0.12
BS Soot, g/kWh	0.003	0.45	0.01

When the engine is operated without EGR, CO, HC, and soot emissions are within the range of the other modes except for NOx. Current diesel engines us EGR to suppress NOx. With EGR or HAM, the NOx levels are reduced to a great extent.

HAM has the benefit of combined low NOx and soot. CO is also lower than for the EGR case. HC is similar for all the cases.

Conclusion

It is proved that introducing the humid air brings down the peak combustion temperature and along with the lesser concentration of oxygen to form NO, brings down the NOx emissions significantly. The combustion in HAM closely follows conventional combustion with EGR. Furthermore, the fuel consumption is not affected very much by the three operating modes in this operating point. The cooling requirement for HAM process would be much lower than the conventional operating modes as it involves no EGR cooler or CAC cooler.

Future work will include experiments performed with different injection timing, Variable geometry position, different load and speed to the evaluate the potential benefits.

References

1. Ringler, J., Seifert, M., Guyotot, V. and Hübner, W., “Rankine Cycle for Waste Heat Recovery of IC Engines”, SAE Technical Paper 2009-01-0174, 2009

2. Teng, H., Klaver, J., Park, T., Hunter, G., and Velde, B., “A Rankine Cycle System for Recovering Waste Heat from HD Diesel Engines - WHR System Development”, SAE Technical Paper 2011-01-0537, 2011

3. Zhang, C., Shu, G., Tian, H., Wei, H. et al., "Theoretical Analysis of a Combined Thermoelectric Generator (TEG) and Dual-loop Organic Rankine Cycle (DORC) System Using for Engines' Exhaust Waste Heat Recovery," SAE Technical Paper 2014-01-0670, 2014, doi:10.4271/2014-01-0670.

4. Briggs, I., McCullough, G., Spence, S., Douglas, R. et al., "Waste Heat Recovery on a Diesel-Electric Hybrid Bus Using a Turbogenerator," SAE Technical Paper 2012-01-1945, 2012, doi:10.4271/2012-01-1945.

5. Rosen, P. M., “Evaporative gas turbine cycles – A thermodynamic evaluation of their potential”, ISRN LUTMDN/TMKV-7010-SE/1993.

6. Traverso A. “Humidification Tower for Humid Air Gas Turbine Cycles: Experimental Analysis”. Energy 35 (2010) 894-901. Elsevier Science

7. Lindquist, O., T., Rosen, P., Olsson, L., “Humid Cycles A Novel Way to Increase Performance of Marine Propulsion Systems”, 2001, CIMAC, Hamburg.

8. Park, H., Ghal, S., Park, H., Choi, S. et al., “A Study on NOx Reduction of Marine 4-Stroke Diesel Engine Using Charge Air Humidification,” SAE Technical Paper 2011-24-0172, 2011, doi: 10.4271/2011-24-0172.

9. Rahai, H., Shamloo, E., and Bonifacio, J., “Investigation of the Effect of a Humid Air System on Diesel NOx and PM Emissions of a Small Diesel Engine,” SAE Technical Paper 2011-01-0692, 2011, doi:10.4271/2011-01-0692.

10. Nord, K., Haupt, D., and Egebäck, K., “Particles And Emissions from a Diesel Engine Equipped with a Humid Air Motor System,” SAE Technical Paper 2001-01-3616, 2001, doi:10.4271/2001-01-3616

11. Iwai, M., Yoshida, K., Iijima, A., Shoji, H., “Study on Performance of Diesel Engine Applied with Emulsified Diesel Fuel: The Influence of Fuel Injection Timing and Water Contents”, SAE Technical Paper, 2011-32-0606, DOI:10.4271/2011-32-0606

12. Chybowski, L., Laskowski, R., Gawdzińska, K., “An overview of systems supplying water into the combustion chamber of diesel engines to decrease the amount of nitrogen oxides in exhaust gas” Journal of Marine Science and Technology September 2015, Volume 20, Issue 3, pp 393-405

13. Woschni, G., "A Universally Applicable Equation for the Instantaneous Heat Transfer Coefficient in the Internal Combustion Engine," SAE Technical Paper 670931, 1967, doi:10.4271/670931

14. Gatowski JA, Balles EN, Chun KM, Nelson Fe, Ekchian JA, Heywood FB. A heat release analysis of engine pressure data. SAE paper NO:841359, 1984

15. Arunachalam, P., Tuner, M., Tunestal, P., Johansson, B. et al., "System Simulations to Evaluate the Potential Efficiency of Humid Air Motors," SAE Technical Paper 2013-01-2646, 2013, doi:10.4271/2013-01-2646.

Contact Information

Martin Tuner
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Martin.Tuner@energy.lth.se

Acknowledgments

This study was conducted as a part of waste heat recovery project supported by the Swedish Energy Agency. The authors gratefully acknowledge the financial support of the agency. Also, we would like to thank Mr. Kalev Kuklane of Aerosol Laboratory at Lund University for his contributions.

Appendix

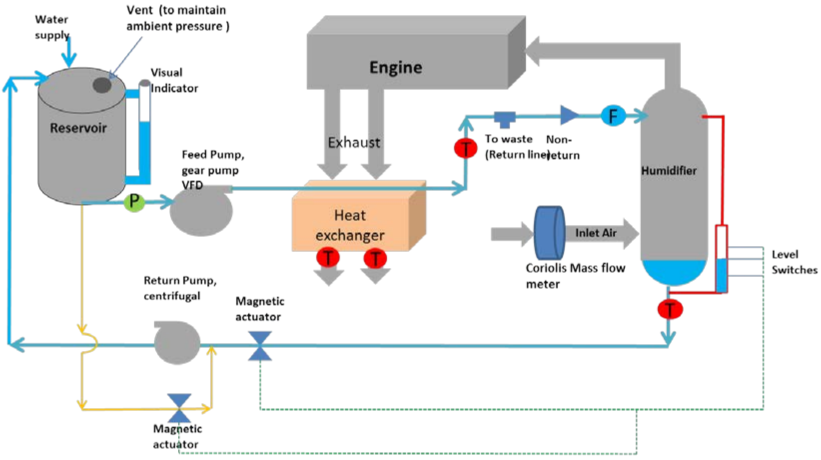


Figure 12. Graphical layout of the HAM system

Figure 13. Humidification tower shell (upside down) with provisions for nozzles

Figure 14. Nozzle type used in the tower

Figure 15. Structured packing used to enhance the surface area of contact between air and water

Paper V

Emission and Efficiency Potential of Humidified Air Motor (HAM)

Prakash Narayanan A, Erik Svensson, Per Tunestal, Marcus Thern, and Martin Tuner
Lund University, Sweden

Abstract

Humidification of the inlet charge has been done previously to reduce the NOx emissions in a diesel engine. In this study humid air motor (HAM) concept was used as a way of waste heat recovery technique to improve efficiency and to decrease the emissions alongside. The study was performed on a 13-litre multi-cylinder Volvo diesel engine with a long route EGR system. The experiments were carried out at three different operating points from European stationary cycle (ESC 13). In each operating point, the engine is operated with three different modes 1.Conventional engine operation without EGR, 2.Conventional engine operation with EGR and 3.Engine operated with humid air cycle. The exhaust heat is recovered by using an additional heat exchanger (plate-type) in the exhaust path. The recovered heat is then fed to humidification tower where it gets contacted with incoming compressed air resulting in humidification of the inlet charge. The humidification process takes place in a humidification tower, which has structured packing material filled inside to increase the surface area of contact between air and water. The results show that NOx emissions can be brought down significantly along with soot emissions without affecting the efficiency while CO and HC increase marginally. With the HAM engine, there is no need for charge air cooler, as the HAM tower acts as a cooler. The results presented in the paper discuss the benefits of waste heat recovered HAM engine operation over the conventional engine operation.

Introduction

Humid air cycles (HAT) was invented in early 90's to enhance the performance of the gas turbines [1]. This technology then has transpired to internal combustion engines as Humid Air Motor (HAM) [2]. This HAM cycle was used to decrease the NOx emissions, which is usually associated with diesel engines.

Current engines mostly use exhaust gas recirculation (EGR) to reduce in-cylinder NOx production by reducing the peak temperatures and by stemming the availability of oxygen. This comes with a penalty in the form of increased cooling loss in EGR cooler [3]. The thermal efficiency of the engine is also affected since the mixture of air and exhaust has lower specific heat ratio (gamma) compared to only air.

An alternative to the EGR could be to use water/water vapor [4,5,6,7]. Water/water vapor usage helps in reducing the bulk gas temperatures [8,9,10] and aids in better mixing by increasing the ignition delay [11]. The reduced oxygen paves the way for reduced NOx [12]. But water/water vapor doesn't help in increasing thermal efficiency directly as its gamma value would be further lower, but it could assist in reducing the NOx to a greater extent [13] without aforementioned cooling loss. The humid air cycle is a solution to compensate the thermal efficiency loss. In humid air cycle water vapor is produced and fed to the engine after the compressor, meaning, less work has to be done compressing the additional vapor mass flow in the compressor. This vapor mass could be extracted over the exhaust turbine to produce useful work.

The humid air cycle doesn't require any additional source of cooling/heating. With the humid air cycle, the humidification tower cools the inlet charge and decides the inlet temperature, eliminating the necessity for charge air cooler. For humidification, the exhaust (otherwise waste) heat is utilized.

Hypothesis behind the concept

1. The vapor has higher specific heat capacity than air, as shown in Figure 1, and so its presence reduces the temperature in the combustion process, resulting in lower peak

temperatures and thus decreasing the emissions associated with high-temperature combustion process. The specific heat capacity of vapor is almost twice that of air, and so this process is twice as efficient as exhaust gas recirculation (EGR); this brings in the possibility of the removal of EGR and its associated components in the engine.

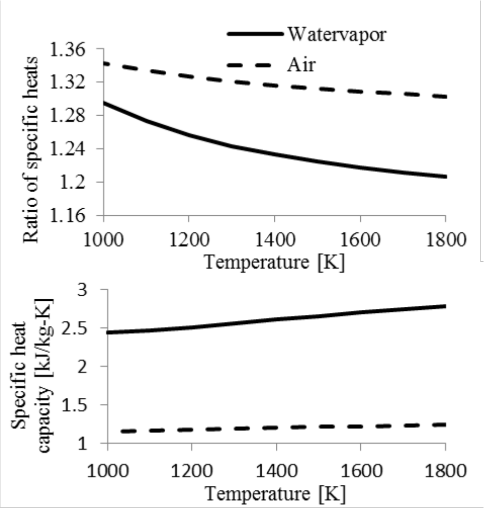


Figure 1. Specific heat capacities and ratios of specific heat capacities of air and water vapor

2. The vapor mass flow from the humidification tower (HT) which has not been compressed in the compressor is expanded over the turbine, producing additional work. This could be perceived as a Rankine cycle operated within the engine cycle. (In this model, the HT is placed after the compressor)
3. Apart from increased specific heat capacity to bring down the temperatures, the concentration of the atomic oxygen by reaction also decreases the NO significantly [12].

Humidifier

An HT is used to humidify the inlet air to the engine. Unlike an evaporator, an HT produces vapor at temperatures lower than the boiling point of water. There are three driving mechanisms behind the water-to-vapor transformation [5]: 1) flashing caused by the higher temperature and pressure of the injected water in comparison to the conditions prevailing in the tower, 2) vaporization through cooling of the air, and 3) enthalpy exchange between water and air caused by the temperature drop between the water inlet and outlet of the tower. The enthalpy exchange is the primary source of evaporation. The water into the humidification tower is always kept at 90 deg-C which in turn governs how much mass flow of water should be circulated, and that determines the inlet temperature and humidity it carries with it.

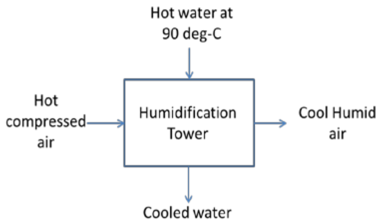


Figure 2. Humidification tower simplified

Figure 3 shows the operation of such humidification process. Point 1 is the compressed inlet air, which upon contact with water in the tower, 5, exchanges the enthalpy. Because of the heat and mass transfer the air temperature drops down to the temperature closer to the inlet water temperature 4. To account for the thermal losses in the system a pinch point is introduced 2-3.

$$h_{air} * m_{air} + h_{water\_in} * m_{water\_in} = h_{humidair} * m_{air} + h_{water\_out} * m_{water\_out}$$

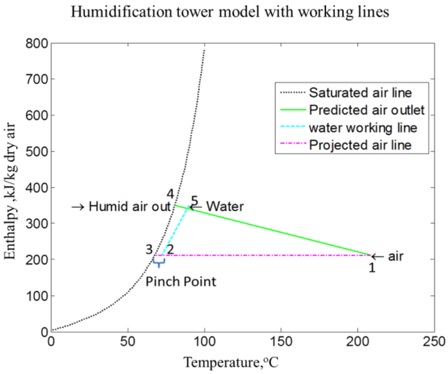


Figure 3. Enthalpy-Temperature diagram of humidification process

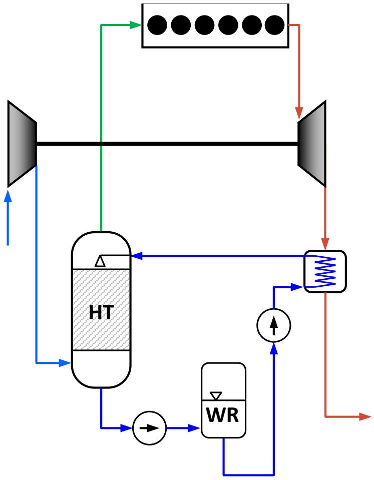


Figure 5. Humid air motor layout

Methodology

Construction

The HAM construction differs from the conventional engine setup in some aspects, to explain, the standard engine setup is shown in Figure 4. The standard engine setup has long route EGR path and thus the EGR cooler is placed after the turbine and before the compressor. The HAM setup is shown in Figure 5. As noticeable the heat exchanger after the turbo was replaced with the humidification tower and instead of EGR cooler a waste heat recovery heat exchanger was installed. The waste heat recovery cooler uses the exhaust enthalpy to heat up the water from the reservoir, and the pumps are used to circulate.

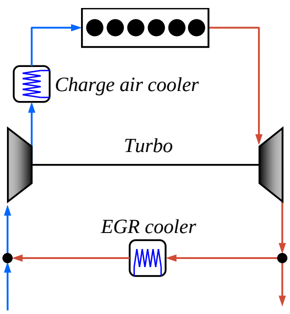


Figure 4. Conventional engine layout

Experiments

HAM pilot tests carried out earlier, at the operating point A25, showed the emission potential without losses in the efficiency [14]. It also showed that the ignition delay is slightly increased with the HAM technique. To see the potential benefits and to understand the HAM process precisely the engine is operated with the different start of injections at different loads and speeds. With these experiments, it is also feasible to analyze the best point in three different modes and compare those with each other. Furthermore, an effort is made to extract the additional vapor energy by using the VGT turbo. In the experiments, the inlet pressure and fuel duration are kept constant to be able to perform the analysis approximately at the same load. This is done because 1. Performing experiments with constant lambda is difficult, as control of the mass flow of air in the turbocharger, is demanding. 2. Lambda control would be difficult with different inlet pressures. The engine has unit injectors, which injects different amounts of fuel for the same fuel duration if the in-cylinder pressure is varied with the inlet pressure. For this reason, the same inlet pressure for both conventional and HAM engine operation was targeted.

Emission measurement

Emissions were measured with AVL AMA i60 Exhaust Measurement System. NOx and HC emissions are measured wet, and the CO, CO2, and O2 are measured dry. For soot measurement, AVL Micro Soot Sensor is used. It is a system for continuous measurement soot concentrations in the exhaust gas from internal combustion engines.

Experimental setup

Table 1. Experimental setup

Engine	Volvo HD truck engine
Fuel	Regular MK1 Diesel
Compression ratio	16:1
SOI	-7:1:+1 CAD ATDC
Injection Duration / Injection Pressure	Constant / Constant
Speed (RPM) , Load%	1200, 25% 1200, 50% 1500, 25%
Modes	Without EGR, with EGR, HAM

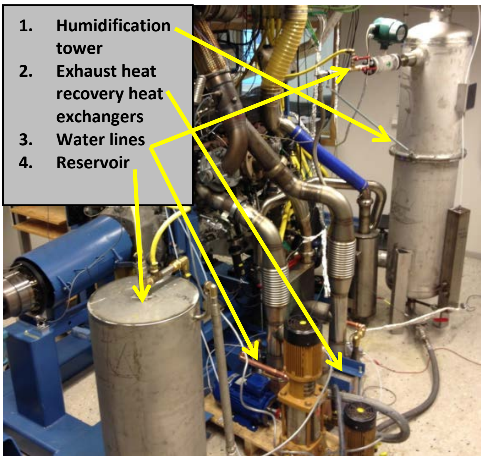


Figure 6. Installation of components on HD engine

Results and Discussions

Cases: The engine modes of operation are given in the table.

Table 2. Modes of engine operation

Conventional engine with EGR	EGR case
Conventional engine operation without EGR	Without EGR case
Humid air cycle operation	HAM

Inlet Conditions

The inlet temperature is controlled with a charge air cooler for the cases with and without EGR. Figure 7 shows the inlet temperature profile of the three different modes of operation. For the HAM case, the inlet temperature is controlled by the mass and heat transfer between the hot water and inlet compressed air *(section: humidifier, figure 2)*. It is worth to mention here, at high load cases the temperature of the inlet air is approximately the same as the inlet temperature of the hot water. This is because the enthalpy of the water mass will be very high compared to the inlet air enthalpy. As mentioned earlier, is the inlet pressure kept constant, hence for the vapor addition and EGR addition the lambda is varied as shown in the Figure 7.

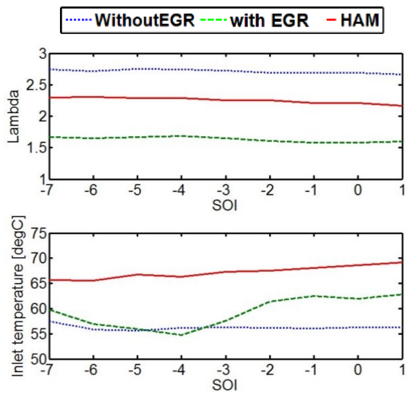


Figure 7. Temperature and Lambda of three different modes (Operating point: B25)

Emission

NOx emission

To comprehend how the peak flame temperature changes a temperature-equivalence (T-ϕ) ratio diagram is constructed. Figure 8 shows a T-ϕ plot for diesel like combustion. The time interval considered for this quasi-static study is 2ms. Pressure is kept at 60 bar, which approximates the cylinder pressure during the onset of combustion. The vertical cloud (ϕ>2) is the soot cloud and the horizontal cloud (ϕ<2) is the NOx cloud. The lines, $T_{flame,}$ shows the flame propagation path for the conventional operation (base), 10% EGR, and 10% water vapor. From the plots, it is clear that when the water vapor is mixed with air, the flame temperature limits entrance to the NOx and soot formation zone better than with EGR .

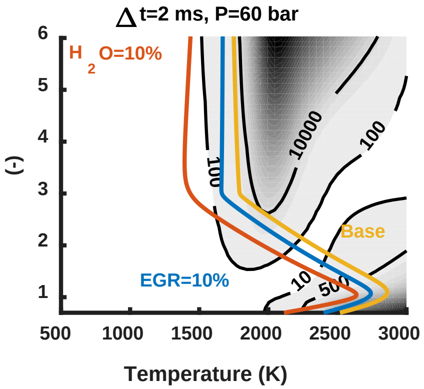


Figure 8.Temperature-Equivalence ratio plot

From the experiments, we see in Figure 9 that the NOx reductions were significant, both, with EGR and with HAM. At the lowest speed and load, the NOx produced from HAM is higher than EGR case. When the speed is increased the NOx became almost equal to the EGR case and with increased load, the NOx reduction potential was even higher than the EGR; this is brought by both vapor introduction and reduced ignition delay (Appendix). This is attributed to the lower instantaneous flame temperature brought by, the higher specific heat capacity of the vapor mixture.

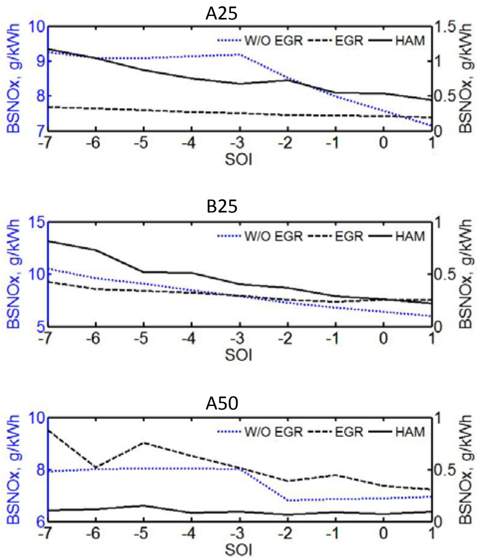


Figure 9. BSNOx vs. SOI. Top A25, Middle B25, Bottom A50

CO Emission

It is clearly seen that CO is much lower at low loads compared to the engine operation with EGR case. From Figure 10 as the speed is increased, the CO increased marginally. But as the load is increased, CO increases significantly comparable to the EGR case. The lower flame temperature caused by the vapor injection reduces the oxidation rate for CO and hence there is an increase in CO.

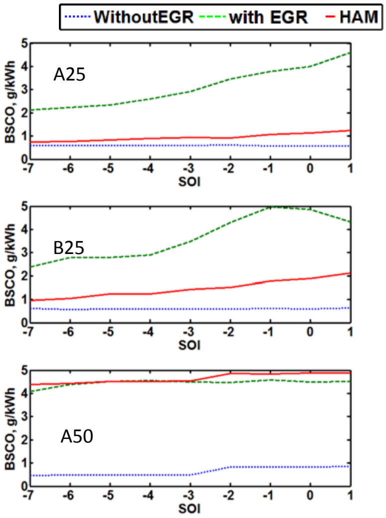


Figure 10. BSCO vs. SOI. Top A25, Middle B25, Bottom A50

HC emission

HC emissions were noted to increase marginally compared to the EGR case in all speeds and loads (Figure 11).

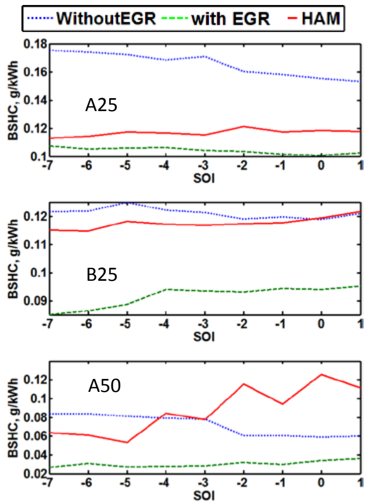


Figure 11. BSHC vs. SOI. Top A25, Middle B25, Bottom A50

Soot Emissions

Figure 12 shows the soot emissions from the experiment. Soot emissions have reduced significantly at low loads. At high load soot is seen to increase, although still much lower than the case with EGR. Soot emission is dependent on the oxidation rate and thus, when the peak flame temperature goes down, the rate of oxidation reduces increasing the Soot emissions.

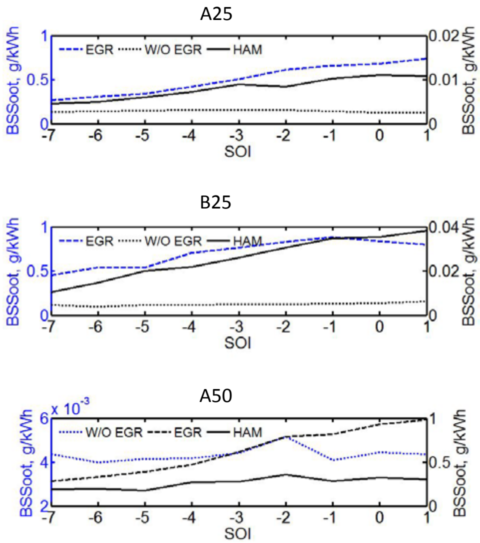


Figure 12. BSSoot vs. SOI. Top A25, Middle B25, Bottom A50

Efficiency

Efficiency is affected by numerous factors. Although some of the main parameters which influence the brake efficiency are combustion (timing), in-cylinder heat transfer, friction, pumping losses, and cooling losses in the engine. Figure 13 shows the efficiency plots of all three operating points in three modes. Point A25 indicates that HAM mode of operation has better efficiency for most of the SOI except for the late injection timing. For operating point B25 the efficiency of HAM engine is clearly superior to the other modes for any SOI. At point A50, the HAM efficiency is slightly lower than either, with or without EGR mode of operation.

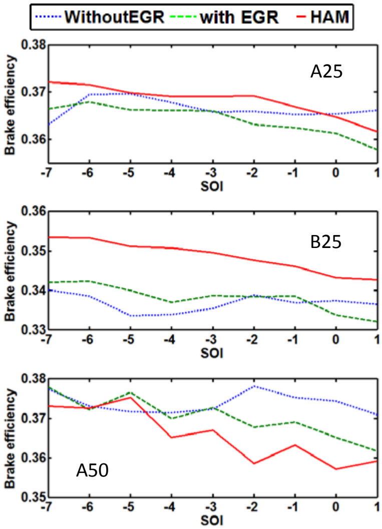


Figure 13. Brake efficiency vs. SOI. Top A25, Middle B25, Bottom A50

The energy distributions for all the three modes of operation at three different operating points are shown in the pie charts (figures 14,15,16). The comparison is made for different modes and operating points at their best brake efficiency point as a function of SOI.

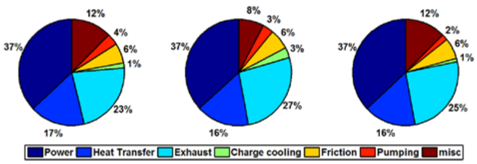


Figure 14. Energy Split for operating point A25 (Left: Without EGR, middle: With EGR, Right: HAM)

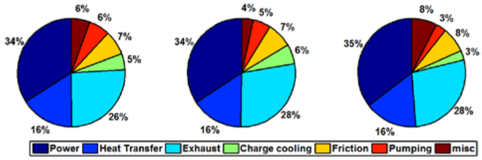


Figure 15. Energy Split for operating point B25 (Left: Without EGR, middle: With EGR, Right: HAM)

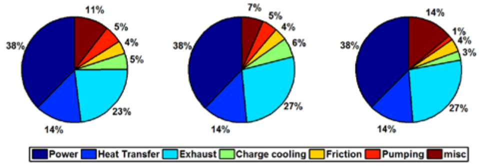


Figure 16. Energy Split for operating point A50 (Left: Without EGR, middle: With EGR, Right: HAM)

Brake efficiency is not affected at 1200 rpm (A25, A50), but it increases by around 3% for 1500 rpm (B25). The improvement in the effectiveness is brought by reduced pumping losses and reduced charge air cooling losses in HAM compared to the cases with and without EGR. (Sample pumping loop is given in Appendix)

Turbine and compressor efficiency

Isentropic turbine and compressor efficiencies are calculated from the inlet and exit temperatures. The turbocharger is a 16 liter Volvo HD engine’s turbo, fitted on a 13-liter diesel engine. At low load engine operation (25-50% max load), the turbocharger is operated at low speeds and thus operated out of range of the design point.

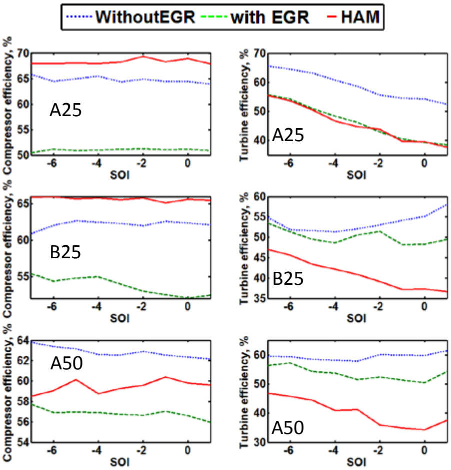


Figure 17. Turbine and compressor efficiency. Top A25, Middle B25, Bottom A50

Variable geometry study

During the experiments, the inlet pressure was kept constant to compare at the same loads. By this, the variable geometry turbine (VGT) position was opened to let the exhaust pass through quickly and thus, not extracting the work from additional vapor. This could have limited the potential of the variable geometry turbine (VGT). To check if the efficiency can be improved by using the VGT. The operating point A25 was chosen, and a VGT sweep was performed at SOI -6 (best brake efficiency for the EGR case). The brake efficiency results are presented in Figure 18. Varying the VGT position doesn't yield significant improvements.

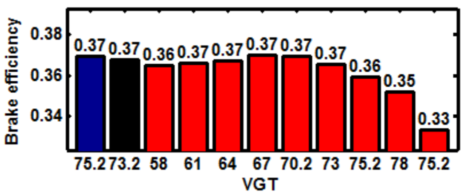


Figure 18. Brake efficiency vs. VGT at SOI: -6 (*Blue: Without EGR, Black: With EGR, Red: HAM* VGT sweep)

Conclusion

The humid air cycle is tested in an HD truck diesel engine. The engine is operated in three different modes, 1. Without EGR, 2.with EGR as in conventional diesel operation, and 3.Humid air cycle to compare the potential benefits. In each mode, the engine was operated with different SOI to select the best point and then the results were used for comparison. From the results, it can be seen that HAM cycle can be used to reduce the NOx and soot without affecting the efficiency as opposed to EGR. At higher speed (1500rpm) it is also possible to improve brake efficiency by about 3% by reducing the pumping losses and cooling loss. For HAM operation, Shifting variable geometry position for better utilization of vapor does not help in increasing efficiency. The experiments were performed without optimizing the turbocharger for humid air cycle and hence there is scope for further improvement. Furthermore, no EGR cooler and charge air cooler are required in this concept, and hence, the cooling requirements for a vehicle would be reduced.

Future Work

The experiments are performed only at low to half load because of the HAM construction limitation. Earlier HAM simulations have predicted the efficiency to be greater at higher engine speeds and predicted negative impact at lower speeds. It is evident from the current experiments, even at lower speeds the performance is not affected, and hence, the potential at higher speeds could, therefore, be greater than predicted.

References

1. Rosen, P. M., "Evaporative gas turbine cycles – A thermodynamic evaluation of their potential", ISRN LUTMDN/TMKV-7010-SE/1993.
2. Lindquist, O., T., Rosen, P., Olsson, L., "Humid Cycles A Novel Way to Increase

Performance of Marine Propulsion Systems”, 2001, CIMAC, Hamburg.
3. Arunachalam, P., Shen, M., Tuner, M., Tunestal, P. et al.,“Waste Heat Recovery from Multiple Heat Sources in a HD Truck Diesel Engine Using a Rankine Cycle - A Theoretical Evaluation,” SAE Technical Paper 2012-01-1602, 2012, doi:10.4271/2012-01-1602.
4. Kegl B. and Pehan S., “Reduction of diesel engine emissions by water injection”, SAE Paper No 2001-01-3259, 2001.
5. Udayakumar, R., Sundaram, S., and Sriram, R., "Reduction of NOx Emissions by Water Injection into the Inlet Manifold of a DI Diesel Engine," SAE Technical Paper 2003-01-0264, 2003, doi:10.4271/2003-01-0264
6. Mello, J., and Mellor, A., "NOx Emissions from Direct Injection Diesel Engines with Water/Steam Dilution," SAE Technical Paper 1999-01-0836, 1999, doi:10.4271/1999-01-0836.The performance of Marine Propulsion Systems”, 2001, CIMAC, Hamburg.
7. Park, H., Ghal, S., Park, H., Choi, S. et al., “A Study on NOx Reduction of Marine 4-Stroke Diesel Engine Using Charge Air Humidification,” SAE Technical Paper 2011-24-0172, 2011, doi: 10.4271/2011-24-0172.
8. Rahai, H., Shamloo, E., and Bonifacio, J., “Investigation of the Effect of a Humid Air System on Diesel NOx and PM Emissions of a Small Diesel Engine,” SAE Technical Paper 2011-01-0692, 2011, doi:10.4271/2011-01-0692.
9. Iwai, M., Yoshida, K., Iijima, A., Shoji, H., “Study on Performance of Diesel Engine Applied with Emulsified Diesel Fuel: The Influence of Fuel Injection Timing and Water Contents”, SAE Technical Paper, 2011-32-0606,DOI:10.4271/2011-32-0606
10. Hountalas, D., Mavropoulos, G., Zannis, T., and Mamalis, S., "Use of Water Emulsion and Intake Water Injection as NOx Reduction Techniques for Heavy Duty Diesel Engines," SAE Technical Paper 2006-01-1414, 2006, doi:10.4271/2006-01-1414.
11. Zaidi, K., Andrews, G., and Greenhaugh, J., "Reduction of NOx with Superheated Steam in a DI Diesel Engine," SAE Technical Paper 2002-01-1157, 2002, doi:10.4271/2002-01-1157.
12. Chybowski, L., Laskowski, R., Gawdzińska, K., “An overview of systems supplying water into the combustion chamber of diesel engines to decrease the amount of nitrogen oxides in exhaust gas” Journal of Marine Science and Technology September 2015, Volume 20, Issue 3, pp 393-405
13. Nord, K., Haupt, D., and Egebäck, K., “Particles And Emissions from a Diesel Engine Equipped with a Humid Air Motor System,” SAE Technical Paper 2001-01-3616, 2001, doi:10.4271/2001-01-3616
14. Arunachalam, P.N., Thern, M., Tunestal, P., Tuner, M., “Experimental Analysis of Humid Air Motor” SAE Technical Paper, 2016. (Submitted)

Contact Information

Martin Tuner
Division of Combustion Engines
Department of Energy Sciences
Lund University, Sweden
Martin.tuner@energy.lth.se

Acknowledgements

This study was conducted as a part of waste heat recovery project supported by the Swedish Energy Agency. The authors gratefully acknowledge the financial support of the agency. Also, we would like to thank Mr. Kalev Kuklane of Aerosol Laboratory at Lund University for his contributions.

Appendix

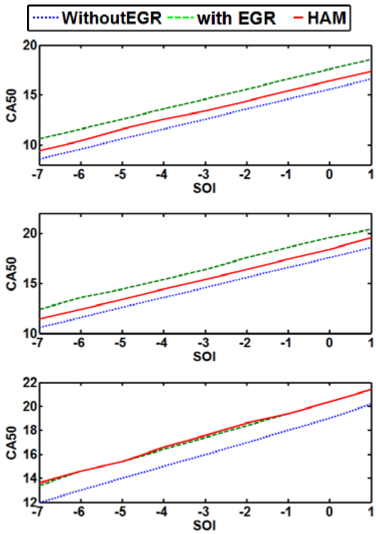


Figure 19. CA50 vs SOI. Top: A25, Middle: B25, Bottom: A50

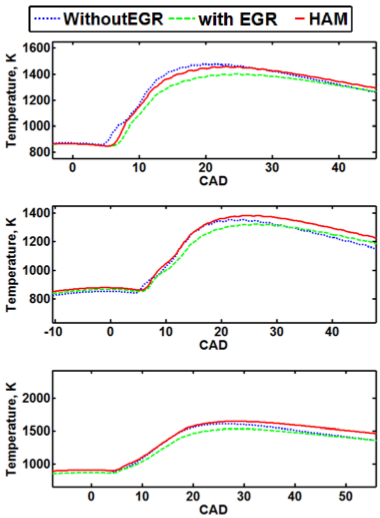


Figure 20. Cylinder temperature vs CAD (at SOI: +1). Top: A25, Middle: B25, Bottom: A

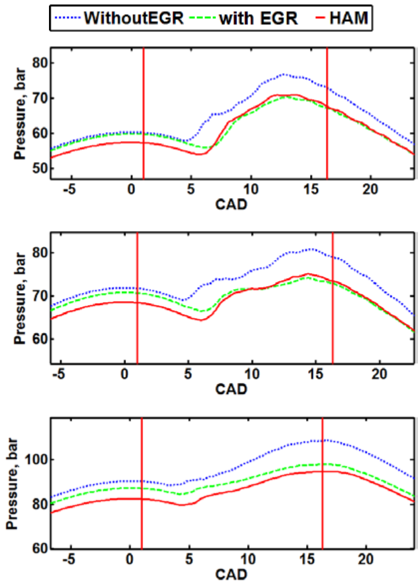


Figure 21. Cylinder Pressure vs CAD (at SOI: +1). Top: A25, Middle: B25, Bottom:A50

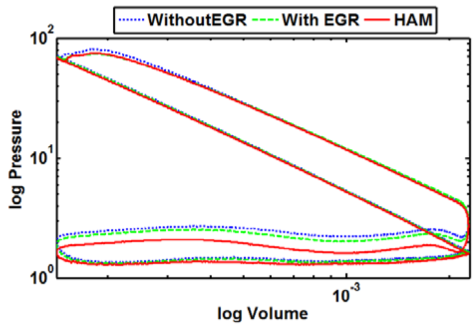


Figure 22. LogP-logV diagram (B25, SOI:+1)

Printed by Media-Tryck, Lund University 2016 Nordic Ecolabel 341903

LUND UNIVERSITY

Faculty of Engineering
Division of Combustion Engines
ISRN LUTMDN/TMHP-16/1119-SE
ISSN 0282-1990
ISBN 978-91-7623-872-1