BACHELOR THESIS & COLLOQUIUM – ME184841

Qualitative and Quantitative Determination of the Effects of Slow Steaming on Engine Operation

Andari Wisnuwardhani Sungging Rakhmamurti
NRP. 04211541000022

SUPERVISOR :
Prof. Dr. –Ing. Frank Bernhardt
Dr. –Ing. Wolfgang Busse

DOUBLE DEGREE PROGRAM
DEPARTMENT OF MARINE ENGINEERING
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SKRIPSI – ME184841

Penentuan Kualitatif dan Kuantitatif Dari Efek 'Slow Steaming' Pada Operasi Mesin

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APPROVAL FORM

QUALITATIVE AND QUANTITATIVE DETERMINATION OF THE EFFECTS OF SLOW STEAMING ON ENGINE OPERATION

BACHELOR THESIS

Submitted to Comply One of the Requirements to Obtain Bachelor Engineering Degree

on

Bachelor Program Department of Marine Engineering
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember

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DECLARATION OF HONOR

I hereby who signed below declare that:

This bachelor thesis has written and developed independently without any plagiarism act. All contents and ideas drawn directly from internal and external sources are indicated such as cited sources, literatures and other professional sources.

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Bachelor Thesis Title	: Qualitative and Quantitative Determination of the Effects of Slow Steaming on Engine Operation
Department	: Marine Engineering

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Andari Wisnuwardhani Sungging Rakhammurti

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QUALITATIVE AND QUANTITATIVE DETERMINATION OF THE EFFECTS OF SLOW STEAMING ON ENGINE OPERATION

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ABSTRACT

Slow steaming or the practice of operating cargo ships much below their maximum speed has been around the shipping industry since 2008 due to economic crisis. Nowadays slow steaming are popular as one of the strategy of shipping companies to reduce their operating cost as implementing slow steaming means cutting off their fleets' fuel consumption. However, operating the engine below the design load range for a long period can evoke some problems for the engine itself. The discussion is finding out what effects the implementation of slow steaming brought to the engine what is causing those effects and also making model from one of the found effects. From the discussion of this bachelor thesis, there are seven problems on the engine caused by implementation of slow steaming and one of them is corrosive impact on the cylinder and piston cause by cold corrosion. Modelling from the bachelor thesis concluded that cold corrosion start to occur in the cylinder when the engine is operated below 30% of load.

Keywords: Ship Engine, Slow Steaming

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PENENTUAN KUALITATIF DAN KUANTITATIF DARI EFEK ‘SLOW STEAMING’ PADA OPERASI MESIN

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ABSTRAK

Slow steaming atau praktik pengoperasian kapal kargo jauh di bawah kecepatan maksimumnya telah ada di sekitar industri pelayaran sejak 2008 karena krisis ekonomi. Saat ini *slow steaming* populer sebagai salah satu strategi perusahaan pelayaran untuk mengurangi biaya operasinya karena menerapkan *slow steaming* berarti memotong konsumsi bahan bakar armada mereka. Namun, mengoperasikan mesin di bawah kisaran beban desain untuk waktu yang lama dapat menimbulkan beberapa masalah bagi mesin itu sendiri. Diskusi dalam tugas akhir ini adalah mencari tahu apa saja efek dari implementasi slow steaming terhadap mesin dan juga membuat model dari salah satu efek yang ditemukan. Dari pembahasan skripsi ini, ada tujuh masalah pada mesin yang disebabkan oleh penerapan *slow steaming* dan salah satunya adalah dampak korosif pada silinder dan piston yang disebabkan oleh korosi dingin. Pemodelan dari tesis sarjana ini menyimpulkan bahwa korosi dingin mulai terjadi dalam silinder ketika mesin dioperasikan di bawah 30% dari beban.

Kata kunci: Mesin Kapal, Slow Steaming

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Preface

Praise to Allah SWT for his grace and mercy this Bachelor Thesis entitled "Qualitative and Quantitative Determination of The Effects of Slow Steaming on Engine Operation" can be completed.

This thesis is made to attain Double Degree Bachelor of Marine Engineering from Faculty of Marine Technology of Hochschule Wismar and Institut Teknologi Sepuluh Nopember Surabaya.

During the process of completing this thesis, I received numerous assistance, support, and prayers. I would like to dedicate my gratefulness to these names below:

1. Prof. Dr. –Ing. Frank Bernhardt and Dr. –Ing. Wolfgang Busse as my supervisor and assistant supervisor who had guide me through all the process of completing this Bachelor Thesis
2. Mama and Papa who had provided me with the resources so that I can have the opportunity to write my Bachelor Thesis in Rostock, Germany and always pray for my wellbeing and success
3. Rachell, Gea, Akbar, Bisma, and Rifram, my fellow Bachelor Thesis buddy in Rostock who had support each other during the hard times of completing this Bachelor Thesis
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5. All DDME 2015 friends, who had struggle together to attain this Bachelor degree
6. Raisha Bani Pramadhani, my sister from another mother and my personal diary
7. Mas Eky and Mas Iji, who has been like big brothers to me and my source of laugh
8. Tatyana Ibrahim, bless you for always keeping my sanity and my expectation as rational as possible

Hopefully, this Bachelor Thesis can provide for information and benefits for any parties and any future studies in engineering, especially marine engineering. Also, this Bachelor Thesis is far from perfect and any feedbacks and corrections will be very much appreciated. Author can be reach by email andariwisnu22@gmail.com.

Sincerely,

Andari Wisnuwardhani S R.

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List of Abbreviations

$CaCO_3$	Calcium carbonate
CIMAC	International Council on Combustion Engines
CO	Carbon Monoxide
CO_2	Carbon Dioxide
H_2O	Water
H_2SO_4	Sulphuric Acid
Pcomb	Combustion Pressure
Pmax	Maximum Pressure
SLOC	Specific Lubricating Oil Consumption
SO_2	Sulphur dioxide
SO_3	Sulphur trioxide
Tcomb	Combustion Temperature
Tcw	Cylinder Wall Temperature
Tmax	Maximum Temperature
Tmean	Mean Temperature

Chapter 1
Introduction

1.1. Background

Maritime transport has important and continuous role to play vital role in advancing international trade flows. Hence, international trade is highly dependent on Maritime Transport. In 2017, total global volumes of seaborne trade reached 10.7 billion tons (United Nations Conference on Trade and Development, 2018) Global containerized trade alone increased by 6.4 per cent from two previous following years and the prospect for seaborne trade are promising (United Nations Conference on Trade and Development, 2018).

Fuel cost account for by far the highest proportion of ship's operational expenses. It made up around 50-70% of the total operational cost (Zanna, Pocuca, & Bajec, 2013). Low load operation, also well known as "slow steaming", has been very commonly practiced in shipping industry nowadays, especially in container shipping. This practice of operating cargo ships much below their maximum speed became known in 2008 due to economic crisis and rising of bunker price. However, in the beginning of mid-2010, slow steaming has become shipping company's strategy to minimize their operational costs.

Since the required propulsion power scales over-proportionally with the vessel speed, significant fuel cost reductions can be achieved with only moderate reductions in service speed. The vessel service speed, therefore, provides a useful control variable for ship-owners to maximize income as a response to fuel prices and freight rates. In figure 1.1 we can see ship operation at different speed, reducing the speed give significant reduction in fuel cost, however reducing ship speed means prolong voyage time.

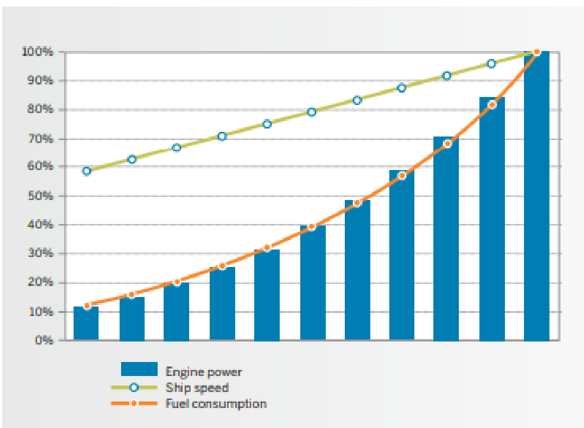

Figure 0.1 Correlation between ship speed, required engine power and fuel consumption
Source: Wartsila Technical Journal

Slow steaming also benefit the emissions produced by ships for environmental side. It lowers CO2 emissions that are proportional to the amount of fuel burned. Reducing ship's speed by 10% decreases emissions by at least 10-15% (Psaraftis & Kontovas, 2010).

However, operating the engine below their designed range can evoke some problems for the main engine itself, for example increase of carbon deposit formation from incomplete combustion. Carbon deposit formation may occur in the diesel engine due to prolonged periods of running at low loads. Running diesel engine under low loads causes lower cylinder pressures and consequent poor piston ring sealing since this relies on the gas pressure to force them against the oil film on the bores to form the seal. Poor combustion caused by low cylinder pressure, it results in lower combustion pressures and temperatures. This poor combustion leads to carbon deposit formation and unburned fuel residue. These intermediate products contribute to fouling, for example in the exhaust gas ducting. Consequences could be important reduction of the turbocharger efficiency as well as clogging of the exhaust gas boilers (Holtbecker, 2014).

There are some more problems to the engine that is reported related to the implementation of slow steaming. This thesis will discuss about effects of slow steaming on engine operation by knowing the expected qualitative changes on the engine parameters as well as making model of the engine processes in order to understand more about the cause-effects relation from engine operation parameter.

1.2. Research Objectives

Research objectives of this thesis are:

1. Research on experiences and peculiarities of the application of slow steaming in ship engine operation;
2. Determine qualitative changes of diesel engine process parameters in slow steaming compared to nominal load operation and describe effects of slow steaming on internal engine damage processes;
3. Model (e.g., Simu-Link) of the diesel engine processes at the required depth;
4. Variation of the process parameters and presentation of the calculation results.

1.3. Research Limitation

Research limitation of this thesis are:

1. Experiences and peculiarities during the slow steaming operation are based on engine manufacturer's service letters and chief engineer's experience.
2. Data for modelling is only coming from literature.

1.4. Research Benefits

Research objectives of this thesis are:

1. Know the experiences and peculiarities of the application of slow steaming in engine operation;
2. Know qualitative changes of diesel engine process parameters in slow steaming compared to nominal load operation and describe effects of slow steaming on internal engine damage processes;
3. Modelling (e.g., Simu-Link) of the diesel engine processes at the required depth;
4. Variation of the process parameters and presentation of the calculation results.

Chapter 2
Research on Effects of Slow Steaming in Ship Engine Operation

This chapter focused on discussing the effects of slow steaming that has been reported based on literature research. Tables and flow chart are made in this chapter to simplify the list of problems or peculiarities in engine operation during slow steaming implementation. Description of internal engine damage is also discussed in this chapter to deepen the knowledge of the slow steaming effects to engine operation.

2.1 Cause and Effects of Slow Steaming Based on Report

Table 0.1 Effects of slow steaming and its causes

No	Effects	Cause
1.	Corrosive impact on cylinder liners and piston rings (CIMAC Working Group 8 'Marine Lubricants', 2017)	Condensation of sulphuric acid due to low cylinder liner temperature
		Sulphur content of Fuel Oil
		Underlubricated cylinder
2.	Exhaust valve burn-away (MAN B&W, 2013)	Increase on exhaust gas temperature on load area of A/B switch point
3.	Broken scavenge air reciever valve (MAN B&W, 2011)	Pressure pulsation in load area of A/B switch point
4.	Accumulated incomplete combustion by-products (Holtbecker, 2014)	Low combustion temperature
		Reduced exhaust gas massflow
5.	Abrassive wear of liner and piston rings (Kolawak, 2015)	Cylinder oil overdosing
6.	Oil flows down to scavenge spaces in form of black sludge (Kolawak, 2015)	
7.	Slow Minor developing fatigue on central pad in lower crosshead bearing shells (MAN B&W, 2014)	Increased gas pressure at low RPM due to T/C cut-out

2.2 low Chart of Expected Qualitative Changes on Engine Parameters

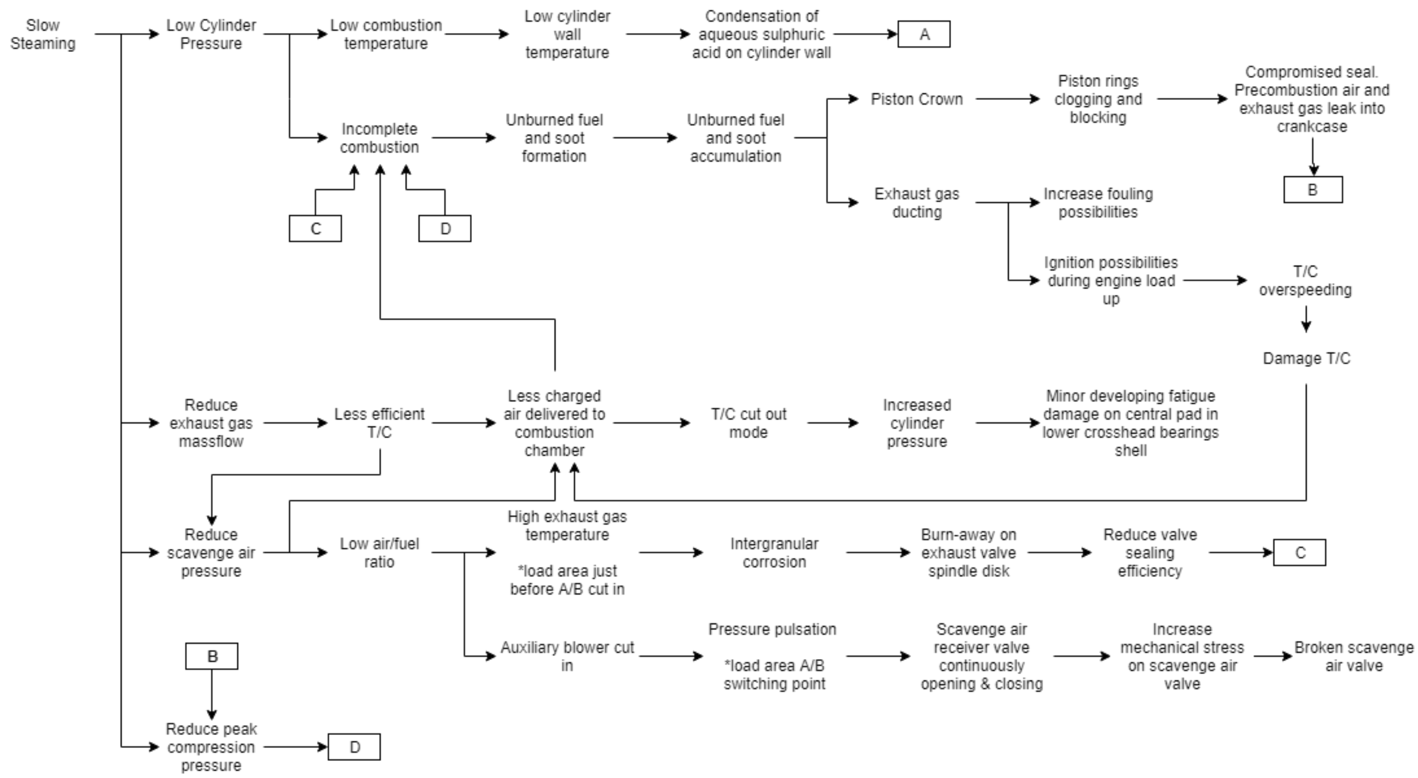

Figure 0.1 Flow chart of slow steaming effects (a)

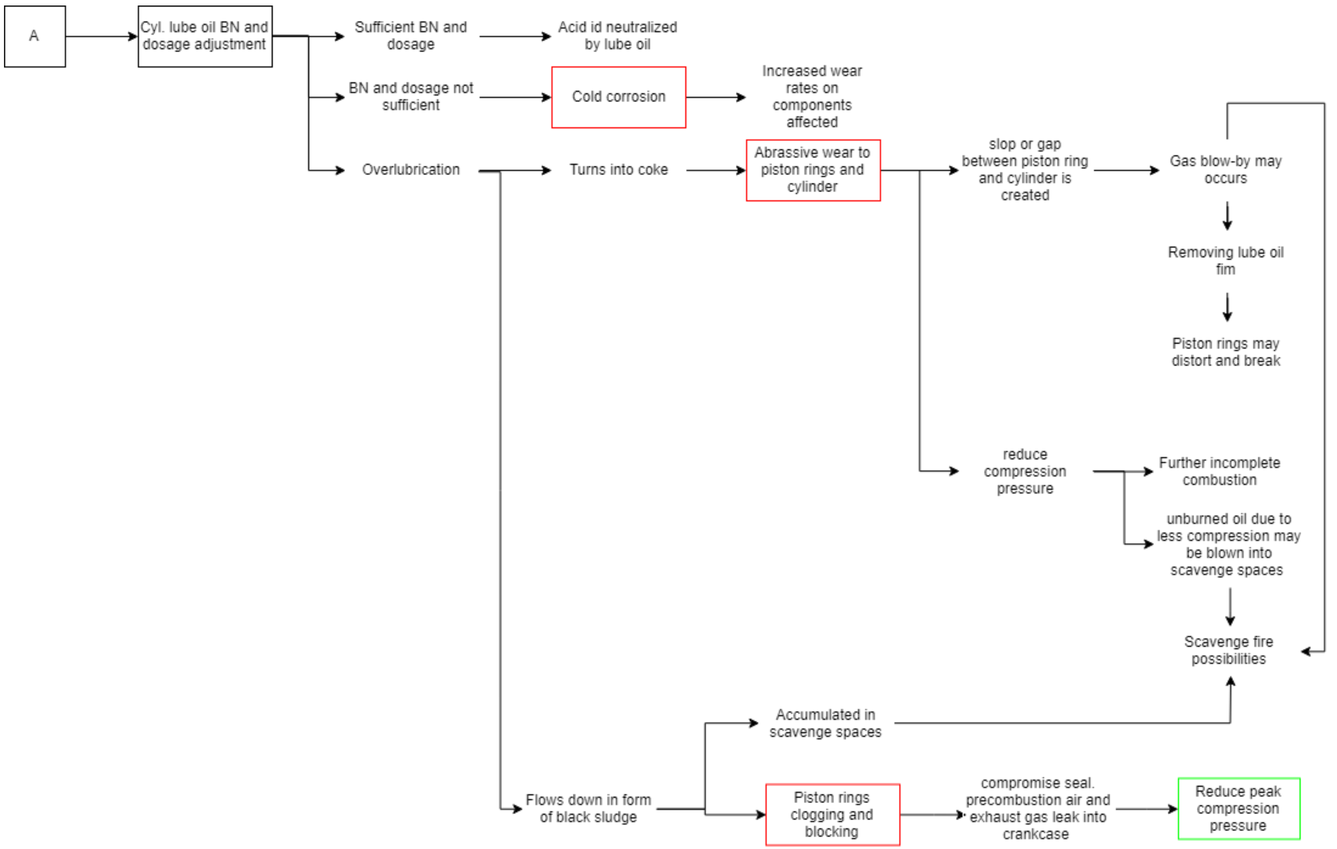

Figure 0.2Flow chart of slow steaming effects (b)

2.3 Description of Effects on Internal Engine Damage

2.3.1 Corrosive Impact of Cylinder Liners and Piston Rings

Cold corrosion has become a regular occurrence along with the trends of operating two-stroke marine engines at low load or better known as slow steaming operation. Corrosive wear or cold corrosion occurs when there is a combination of a wear situation (abrasive and adhesive) and a corrosive environment. Abrasive wear can be caused by hard asperities in the components (cylinder liner and piston rings) rubbing again each other, abrasive wear occur due to loss of oil film between piston / ring and liner, and corrosive environment can be caused by sulphur content of the fuel.

Figure 0.3 Cylinder liner surface with signs of extreme corrosion attack

Source: (Weber, 2016)

Figure 0.4 Piston ring running surface with signs of extreme corrosive attack

Source: (Weber, 2016)

There are three stages in the cold corrosion process inside the cylinder (CIMAC Working Group 8 'Marine Lubricants', 2017):

1. Formation of SO_3
   Sulphur content in the fuel is oxidized to Sulphur dioxide (SO2) during the combustion, however a small amount SO2 is further oxidized to Sulphur trioxide (SO3). The formation of SO3 is driven by the dynamic equilibrium between SO_2, O_2, and SO_3, which changes with concentration of these chemical components, pressure and temperature according to Le Chatelier's Principle. Depending on the engine design, tuning, and operating conditions (load, fuel Sulphur, ambient conditions etc.) up 10% of SO2 are oxidized to SO3 during combustion (CIMAC working group, 2007)

$$S_{(s)} + O_{2(g)} \rightarrow SO_{2(g)}$$
$$SO_{2(g)} + ½O_{2(g)} \rightarrow SO_{3(g)}$$
$$SO_{3(g)} + H_2O_{(l)} \leftrightarrow H_2SO_{4(aq)}$$

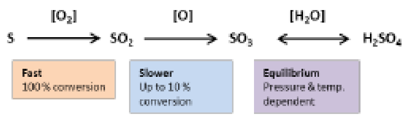

Figure 0.5 Reaction mechanism of the formation of sulphuric acid
Source: CIMAC

   During the combustion process gas mixture formed in a vapour-liquid-equilibrium system for SO3, H2SO4, and H2O. The vapour phase and the liquid phase is in equilibrium below the dew point temperature of the gas mixture, which consists of equeous sulphuric acid.

2. Condensation of aqueous sulphuric acid
   Temperature and pressure conditions inside the combustion chamber (and exhaust gas system) determine whether the equilibrium vapour and liquid phase of gas mixture will condense as sulphuric acid (and water) at the surfaces of combustion chamber components (Garcia, Gehle, & Schakel, 2014).

   The bulk gas temperatures in the cylinder are substantially higher than the dew point temperature of aqueous sulphuric acid. Hence, no condensation occur in the gas volume. However, since cylinder liner temperature always has the lowest temperature (lower cylinder liner temperature are associated with slow steaming operation), condensation of the aqueous sulphuric acid on the liner surface occurs.

   Experience show that the sulphuric acid corrodes the softer phases of the cast iron at a faster rate than the hard cementite, and also that it attacks in the boundary between the different phases (CIMAC Working Group 8 'Marine Lubricants', 2017). This is causing the hard cementite protruding from the matrix, and also the cementite grains loosened from the matrix as figure 2.4 shown below.

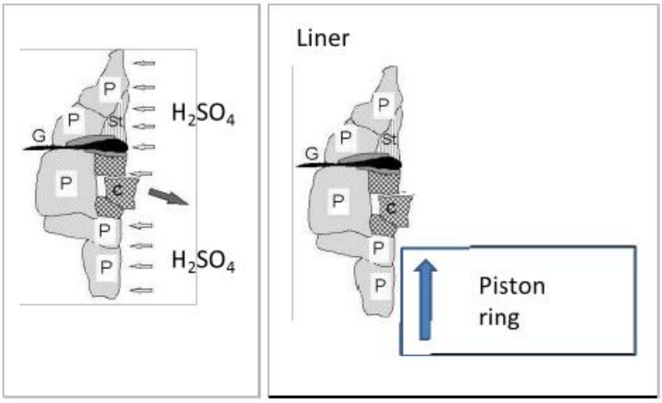

Figure 0.6 Schematics of H2SO4 acid corroding grey cast iron (left) and piston ring hitting and breaking the hard cementite grain during operation (right). P: Pearlite, St: Steadite, G: Graphite, C: Cementite
Source: CIMAC

Such corroded surfaces are extremely vulnerable to mechanical load. Depending on whether the remaining hard phase protrudes from the base matrix or not, the piston ring, which is passing over it, can chop it off or not. Such chopped-off hard particles will disturb the lubrication between ring and liner and can easily embed in, for example, the piston ring base material to start acting abrasively.

3. Neutralization reaction in the cylinder lubrication oil
   As explained before, lower cylinder liner wall temperatures during slow steaming operation might promote increased acid condensation. Therefore, cylinder oil feed rates and maintenance must be adjusted to ensure the necessary protection for the engine. Minimum feed rate must be obtained for pure lubrication, and when the Sulphur content in the fuel increases more neutralizing ability from the lubrication oil is necessary to protect the liner wall against corrosion as more Sulphuric acid condense on the cylinder wall.

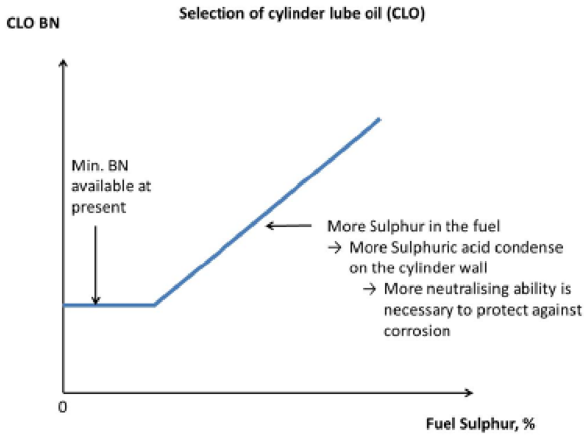

Figure 0.7 General recommendation for selection of cylinder lube oil (CLO) depending on Sulphur content
Source: CIMAC

High alkaline lubricants are used to neutralize the sulphuric acid to avoid cold corrosion of piston rings and cylinder liner surfaces. Calcium carbonate (CaCO3) is commonly used as additive in cylinder lube oil. It neutralises the acid according to the following reaction:

$$H_2SO_4(aq) + CaCO_3(s) \rightarrow CaSO_4(s) + H_2O(l) + CO_2(g)$$

If the sulphuric acid is not neutralized, corrosion of iron occurs according to the following reaction:

$$Fe(s) + H_2SO_4(aq) \rightarrow FeSO_4(s) + H_2(g)$$

Loose corrosion products from cylinder liner and piston rings are easily removed by wear to continually reveal fresh metal beneath, which in turn another corrosion can attack quickly. Hence, material loss rate on the combustion chamber components can be very high, much higher than the sum of the individual contribution of wear and corrosion (CIMAC Working Group 8 'Marine Lubricants', 2017).

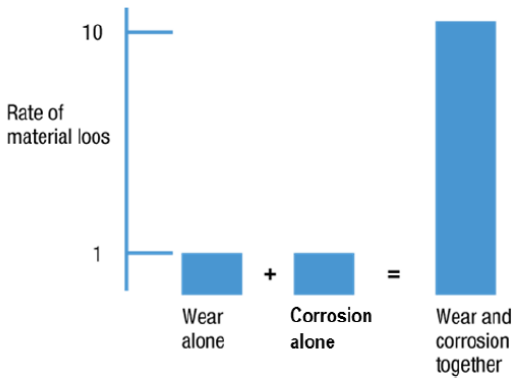

Figure 0.8 Wear rate caused by corrosion
Source: CIMAC

2.3.2 Faster Exhaust Gas Valve Burn-Away

For some types of engine, low load operation in the 30-45% load increases the exhaust gas valve temperature. Operating in this load range is just above cutting-in of the auxiliary blowers, exhaust valve spindle temperatures are known to be rather high, causing faster burn-away rate (MAN B&W, 2013). This condition can be caused by the extreme low air/fuel ration which cause higher exhaust gas temperature, affecting the exhaust gas valve spindle temperatures.

Exhaust valves have surfaces that is exposed to thermal overstress due to exposure of both high temperature combustion flame and high temperature exhaust gas that flows past the exhaust port during the exhaust process. Unlike intake valves that cooled by the incoming air, exhaust valves have no medium to cool them. Therefore, exhaust valves are much more prone to erosion and burning than intake valves.

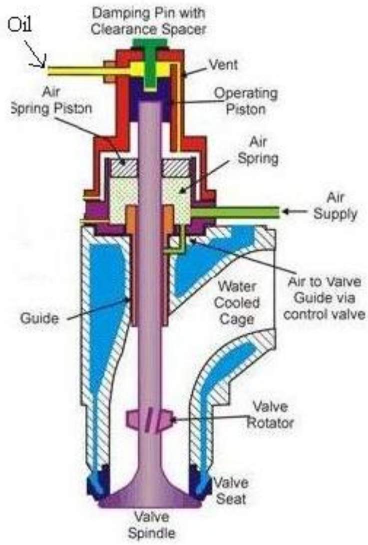

Figure 0.9 Diagram of exhaust valve
Source: www.brighthubengineering.com

Mechanism that leads to burn-away on valve face is called intergranular corrosion (Vardar & Ekerim, 2009). It started by the formation of Sulphur dioxide, sulphur trioxide, sodium oxide and vanadium pentoxide during combustion process from the oxidation of Sulphur, vanadium, and sodium content of the fuel. These oxides react with each other and the calcium content in the lubricating oil to form low-melting point salts. Particles of these salts deposit themselves on the valve surfaces in their molten state and get cooled sufficiently to adhere to valve spindle, preventing them from being carried away by the exhaust gases. At temperatures of 550^{o} C and upwards, the deposited salts change into their molten state as respective melting points are reached. In a molten state these salts flow along the grain boundaries and, depending on the valve material, dissolve the protective oxides on the grain boundaries, thereby exposing the less resistant grain to these corrosive salts.

Faster burn-away may reduce the time between overhaul for the exhaust gas spindle and, since overhaul (rewelding) is recommended up to two times only, this will also reduce the lifetime of the exhaust gas spindles.

Figure 2.10 illustrates some examples of burn-away rates observed during low-load operation in relation to the previous standard for overhaul.

Engine type	Burn away	Running (hrs.)	Burn away rate	Spindle lifetime
K98ME	9 mm	14,000	0.64	61,000 hrs.
S60MC-C	11 mm	26,000	0.43	64,000 hrs.
K98MC-C	7.5 mm	15,000	0.50	78,000 hrs.
S90MC-C8	14 mm	15,000	0.93	39,000 hrs.

Burn away rate = mm / 1,000 hrs.
Normal lifetime = 100,000 hrs. including reconditioning of spindle

Figure 0.10 Example of previous spindle lifetime
Source: MAN Service Letter SL2013-573/JAG

Figure 2.11 shows an exhaust valve on an S60MC-C8.1 engine inspected after 26,000 hours after long-time low-load operation. The burn-away rate is more than 11 mm and according to the instruction manual, the maximum burn-away for rewelding on this engine type is 9 mm. In view of low load, this means more frequent inspection interval and planning of maintenance required.

Figure 0.11 Exhaust valve spindle disk of MAN B&W Engine S60MC-C8.1
Source: MAN Service Letter SL2013-573/JAG

When the burn-away has reach the set-up limit rewelding must be done, otherwise the metal loss from the valve spindle will cause valve sealing loss. This can lead to reduce compression pressure that effects the combustion process, eventually reducing or even total loss of generated power from the particular cylinder.

2.3.3 Broken Scavenge Air Reciever Valve

During continuous slow steaming operation the engine is running on low rpm and low load, causing less energy in the exhaust gas to be supplied to the turbo charger turbine compared to normal operation. This condition causes less turbocharger rpm which lead to surging. Therefore during slow steaming operation the operation of auxiliary blowers are substantially needed.

Based on MAN B&W service letter (MAN B&W, 2011), there have been several reports of broken scavenge air receiver flap valve in connection with low load operation. At some point in the load area around the switch point of the auxiliary blower, the flap valves (non-return valves) in the scavenge air receiver port will be continuously opening and closing due to the pressure pulsations in the scavenge air receiver, increasing mechanical load on the flap valve, causing it to break.

This load area can be recognised by distinct hammering noise from the flap valves. The hammering of the valves will damage and/or break the valves after a short period.

Figure 0.12 Broken flap valves
Source: MAN B&W Service Letter SL11-544/MTS

Broken and missing valves will have a negative impact on the engine performance when the auxiliary blowers are running, since the broken valve will disturb the turbine operation in the auxiliary valve that drew the air in.

2.3.4 Accumulated Incomplete Combustion Products

Diesel fuel is constructed from carbon and hydrogen, complete combustion is understood as combustion where all the fuel hydrocarbons are burn to build CO2 and H2O. However, operating diesel engine at low load or slow steaming cause lower cylinder pressure, hence lowering the combustion temperature causing incomplete combustion to occur. If the combustion is incomplete the intermediate products in such as CO (carbon monoxide), pure carbon (particulates partly surrounded by liquid components or soot), or intermediate smaller hydrocarbon (unburned fuel or lube oil) may be generated and accumulated on piston crowns or exhaust gas ducting.

Accumulated unburned fuel and soot in the exhaust gas ducting increases of fouling in the exhaust gas. Consequences could be as important reduction of turbo charger efficiency, clogging of exhaust gas boilers, also ignition possibilities after engine load up in the exhaust gas ducting causing damage to the turbocharger due to sudden overspeeding.

2.3.4.1 Accumulation on the Exhaust Gas Ducting

Accumulation of unburned fuel and soot in the exhaust gas ducting can lead to increase fouling possibilities and fires in exhaust gas ducting during engine load up, both have consequences in the performance of turbocharger.

Fouling inside the exhaust gas ducting can reduce the amount of exhaust gas that will drive the turbocharger turbine, leading to less efficient turbocharger. Fire in exhaust gas ducting during engine load up can be dangerous for the turbocharger because the sudden burst of air from the fire can cause the turbine in the turbo charger to over-speed, hence damaging the turbocharger. Less efficient and damage turbocharger both can cause less efficient combustion leading to more incomplete combustion, causing more formation of soot and unburned fuel.

2.3.4.2 Accumulation on Piston

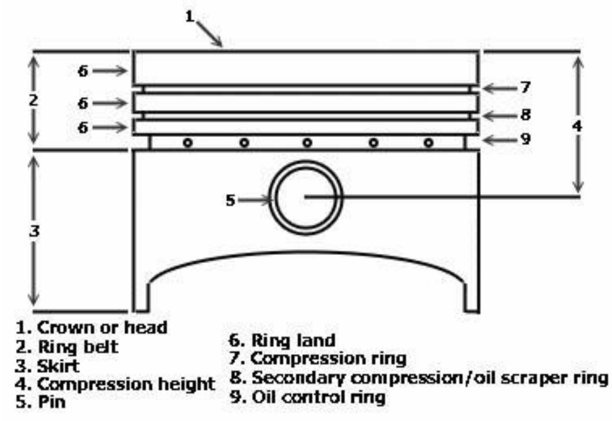

Figure 0.13 Piston crown cross section
Source: google.com

Accumulated unburned fuel and soot can cause clogged piston rings. Piston compression rings are made from an especially hard, often chrome-plated spring-steel alloy, and the role they play is critical, acting as a super-tough gasket between fast-moving pistons and stationary cylinder walls. Their C shape and springiness allow them to expand and contract with temperature changes, while still maintaining good contact with the piston groove or land in which they reside and the cylinder wall. This function of compression rings can be compromised, however, if the land and ring gap become clogged with soot. When this occurs, the seal is compromised, and pre-combustion compressed air and exhaust gases, which include soot and water, leak into the crankcase. None of this is good for the engine's power, fuel efficiency or longevity.

Figure 0.14 Clogged ring groove due to carbon deposits

Source: www.diesel-engine-motor-service.com

Pistons are equipped with oil control ring, which are used to control oil distribution to cylinder walls. They rely on fine “cross-hatch” grooves to retain lubricating oil on the walls’ otherwise smooth surface. If the oil remained there during the combustion process, it would burn or clog the compression rings with coke. The very hot rings cause the oil to burn, leaving behind carbon or coke. This is where oil-control rings come into play. Oil is often deposited or sprayed onto cylinder walls beneath the piston as it moves up the cylinder and then scrapped away, squeegee-like, by the oil-control rings as the piston descends. If these rings also become clogged with soot, they begin to allow oil to slip by and further clog the compression rings, which in turn allows greater loss of compression.

2.3.5 Abrasive Wear of Cylinder Liner and Piston Rings

Abrasive wear on cylinder liner can be cause by many reasons, but one of the highlighted reason of abrasive wear during the slow steaming operation is because of cylinder liner over lubrication. Slow steaming operation is prone to ‘miss-match’ cylinder oil dosing. Since more condensation of sulphuric acid occur, adjustment of the cylinder oil dosage must be carefully done. Under-lubrication can lead to corrosive wear as explained before, while over-lubrication can lead to abrasive wear on cylinder liner.

When overlubrication occurs, excess cylinder oil and additives from the cylinder oil will burn off and form layer of deposit on top land of the piston. The deposit build up on the piston top land appears to restrict the free movement of the piston, possibly affecting the normal alignment of the piston. In these circumstances there is a scraping action of the piston skirt (figure 2.15) against the liner wall disrupting the lubricant film and causing abrasive wear (McGeary, Chew, & Fogh, 2004).

(2.15) (2.16)

Figure 0.15 Intial wear on copper band of the piston skirt
Figure 0.16 Deposits on piston top land abrading the liner wall
Source: (McGeary, Chew, & Fogh, 2004)

Wear, whether it is friction, abrasive, corrosive, or adhesive, cause the same problem to the engine:

1. If the cylinder is operated with excessive wear, the rate of wear will rapidly increase.
2. Wear scrapped cylinder bore and piston, causing it to become bigger and smaller respectively. This condition will create ‘slop’ or gap between cylinder and piston. When the gap become significant compressed gases are free to go around the piston and blow-by occurs. Blow-by can remove cylinder lubrication oil film, causing metal to metal contact known as scuffing, eventually piston ring may distort and break.
3. The gap or ‘slop’ that was created also reduced compression pressure, causing less efficient combustion. This condition can promote fouling of exhaust gas system and rise further problems like turbocharger surging and loss of engine power. Also, by the reduction of compression pressure there will be unburned oil that may be blown to scavenge spaces, risking the possibilities of scavenge fire.

2.3.6 Oil Flows Down to Scavenge Spaces in Form of Black Sludge

When there’s abundant supply of lubricating oil sprayed to the combustion chamber, the excess lubricating oil will be scrapped down by the oil-control rings on the pistons. This condition is reported by chief engineer (Kolawak, 2015) caused piston rings clogging and increasing risk of scavenge fire.

a. Piston rings clogging and blocking

If too much cylinder oil is supplied, the loss of fresh, unused oil in the scavenge ports will be high, and the piston rings might be prevented from moving in their grooves by the so called “hydraulic lock” (Christensen, 2010).

Also when the amount of carbon (a mixture of unburned hydro-carbons, residues of burned lube oil and ashes) is too high, it will clog the ring grooves. Due to the heat transfer from the piston crown, a compound will be formed and the piston ring will stick to its groove

Figure 0.17 Stuck rings due to heavy carbon deposit
Source: www.diesel-engine-motor-service.com

b. Increase risk of scavenge fire

Scavenge fire can occur when there’s enough source of oxygen, heat, and fuel to support the combustion. The excess lubricating oil that flows down to the scavenge spaces provides the combustion material (fuel), scavenge air which is supplied for the operation of the engines provides the oxygen needed for the combustion, and blow-by may provide the heat to ignite the combustion process.

2.3.7 Slow Minor Developing Fatigue on Central Pad in Lower Crosshead Bearing Shells

Experience of engine manufacturer MAN B&W on K98 engine optimised for low load engine with turbocharger cut out has resulted in minor slow-developing fatigue damage on the central pad in the lower crosshead bearing shell. This slow-developing fatigue is caused by the changed balance between inertia force and gas force, which resulted in an increased load on the crosshead bearing shells in Turbocharger cut out mode with the increased gas pressure at low rpm (MAN B&W, 2014).

MAN is currently testing and developing modified design of crosshead bearing shells. However, they have not yet reach the conclusion since the development of fatigue damage typically takes two years.

Chapter 3
Modelling

This chapter will focus on the process of making the model of the effects of slow steaming on engine operation based on literature and from the qualitative flow chart that has been made in the previous chapter. The chosen effects of slow steaming that will be modelled is the occurrence of cold corrosion inside the engine cylinder and modelling is done using MATLAB Simulink.

3.1. Simulink

Simulink is a MATLAB application which provides a way to implement differential equations in an intuitive way, where blocks are used to represent everything from simple additions to integration of time derivative variables. Pre-defined blocks can be used just as well as self-defined functions where bigger block systems preferably are put together in bigger blocks as sub-systems.

3.2. Model Overview

To build model of the occurrence of cold corrosion due to slow steaming, different parts of engine operation parameters calculations are built into blocks and sub-blocks. Figure 3.2 below gives view of what the modelling look like. The flow direction started from the desired of engine load, then calculation of the in-cylinder pressure, after that in-cylinder mean temperature, followed by calculation of cylinder wall temperature and dew point temperature of aqueous sulphuric acid, and finally the cold corrosion consideration.

The aim of this modelling is to make a rough estimation at which load the engine started to have cold corrosion occur as the effect of slow steaming implementation.

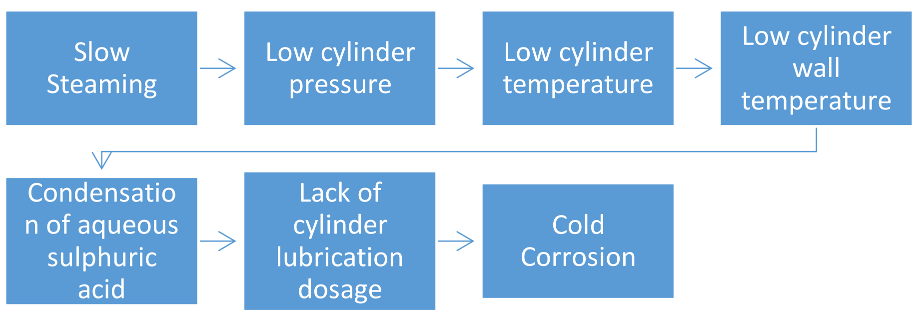

Figure 0.1 Cold Corrosion Process

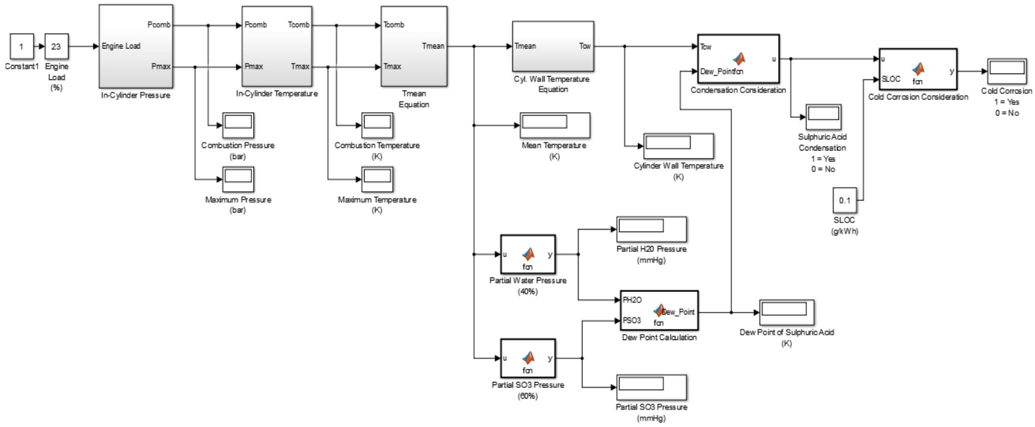

Figure 0.2 Model in Simulink

3.3. In-Cylinder Pressure

From the previous qualitative effects flow chart, cold corrosion inside cylinder liners started by the low pressure generated by the engine due to low load operation. Therefore, determining the pressure inside the cylinder at various load is important.

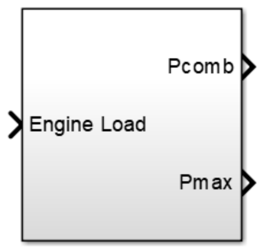

Figure 0.3 In-cylinder pressure calculation block

The input of this block is the desired engine load that wanted to be modelled. After that combustion pressure and maximum pressure inside the cylinder, which is the output, will be determined. The cylinder pressure data used for the modelling is from MAN B&W 7K98MC with specification below

Table 0.1 MAN B&W 7K98MC specification

Bore	980 mm
Stroke	2660 mm
Number of Cylinders	7
Brake Power at MCR	40,100 kW
Engine Speed at MCR	94 rpm
BMEP at MCR	18,2 bar

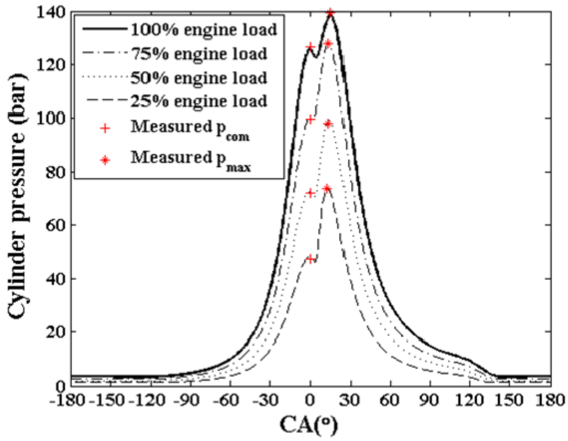

Figure 0.4 In-cylinder pressure graph of MAN B&W 7K98MC

From the graph above, the in-cylinder pressure at various engine load can be noted.

Table 0.2 In-cylinder pressure of MAN B&W 7K98MC from graph

Load (%)	P0 (bar)	Pcomb (bar)	Pmax (bar)
25	2	45	74
50	2	70	96
75	2	98	126
100	2	127	140

The known cylinder combustion and maximum pressure above is used to make a graph to acquire equation for cylinder combustion pressure and maximum pressure estimation for desired engine load.

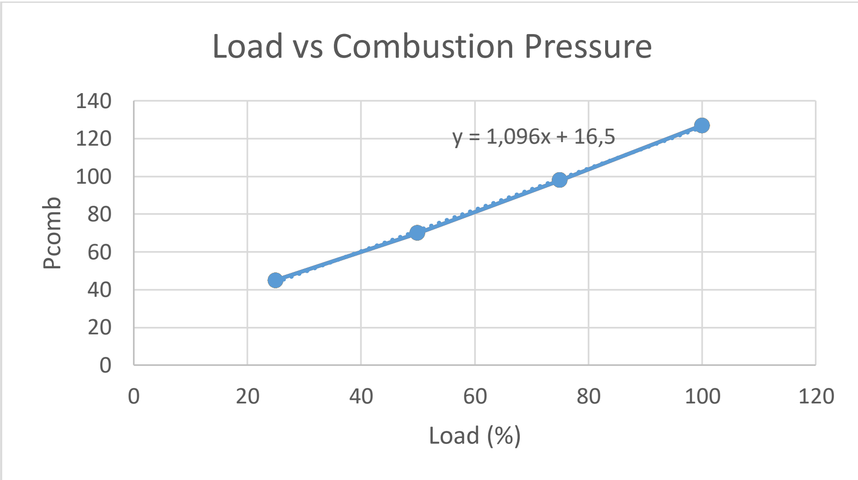

Figure 0.5 Load vs Combustion Pressure Graph

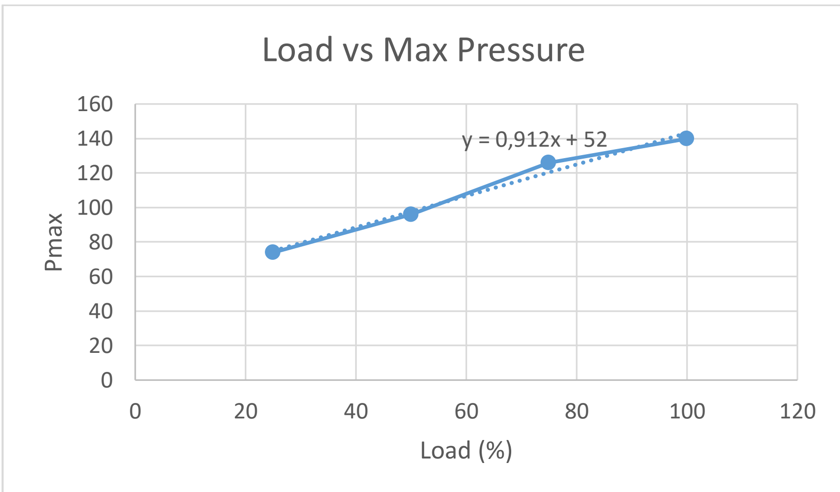

Figure 0.6 Load vs Maximum Pressure Graph

From the two graph above, 2 equations are acquired to determine combustion pressure and maximum pressure of the desired engine load from the chosen diesel engine. Equation y = 1,096x + 16,5 from Figure 3.5 can be used to determine the combustion pressure and equation y = 0,912x + 52 from Figure 3.6 can be used to determine the maximum pressure, where "x" is the desired engine load in percent (0-100).

3.4. In-Cylinder Temperature

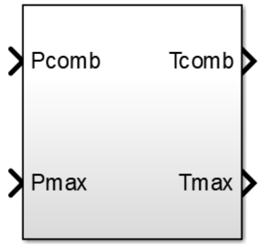

Figure 0.7 Pressure calculation block

The inputs of this calculation block is combustion pressure (Pcomb) and maximum pressure (Pmax). Then combustion temperature (Tcomb) and maximum temperature (Tmax) of the cylinder liner or the outputs are calculated based on equation from ideal gas law such below,

$$T = \frac{To \ x \ P \ x \ Vcyltop}{Po \ x \ Vo} \tag{1}$$

Where,

T = Cylinder Temperature (Combustion/Maximum)
To = Initial Temperature of Cylinder (assumed constant at 400K)
Po = Initial Cylinder Pressure (assumed constant at 2 bar)
Vo = Volume of Cylinder (2005,4 L)
Vcyltop = Volume of Cylinder at TDC (159,16 L)
P =In-Cylinder Pressure Acquired from Pressure Graph (Combustion/Maximum)

3.5. Cylinder Wall Temperature

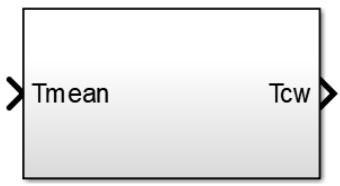

Figure 0.8 Cylinder wall temperature calculation block

In determining the cylinder wall temperature, graph of temperature declination from temperature inside the cylinder to cooling water temperature at desired engine load is used. The graph consist of average temperature inside the cylinder (Tmean) at the centre of cylinder bore and the average temperature of cooling water at the centre of cooling water space. (add info on where you calculate the cyl temperature, take half the length of the cylinder)

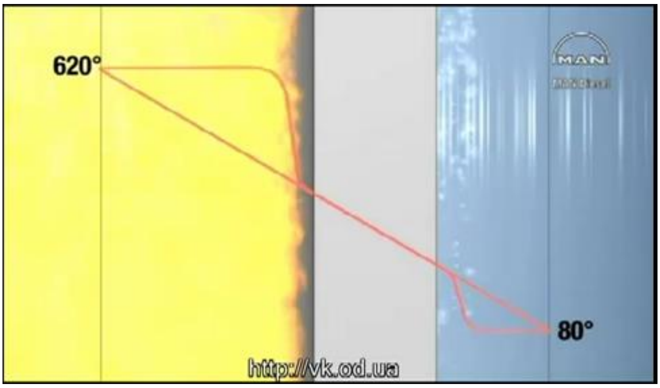

Figure 0.9 Illustration of temperature declination from cylinder to the cooling water

Source: youtube.com/watch?v=OATBbeEd90c

From the illustration above, a graph of temperature declination from the cylinder mean temperature to the cooling water temperature can be made on each desired engine load.

To make the graph, mean or average temperature inside the cylinder is first calculated to represent the temperature inside the cylinder. The average temperature of the cylinder (Tmean) is calculated by,

$$Tmean = \frac{To + Tcomb + Tmax}{3} \tag{2}$$

Where,

Tmean = Average Temperature in Cylinder (K)
To = Intial temperature of cylinder (K)
Tcomb = Combustion temperature in cylinder (K)
Tmax = Maximum temperature in cylinder (K)

Then average temperature of cooling water at the center of cooling water space is taken at 80^{o}C or 353 K.

Cylinder wall thickness and cooling water space thickness are also calculated to define the x-axis, they are calculated by (Khairhar, 2019),

$$t = 0{,}045D + 1{,}6mm \tag{3}$$

$$cws = 0{,}08D + 6{,}5mm \tag{4}$$

Where,

t = Cylinder wall thickness (mm)
cws = Cooling water space (mm)
D = Cylinder bore (mm)

The temperature calculation for the cylinder wall is assumed at the middle of cylinder height to represent average, since the other temperature parameters (cylinder temperature and cooling water temperature) to build the graph are average temperatures. After that a graph of temperature declination is made as below,

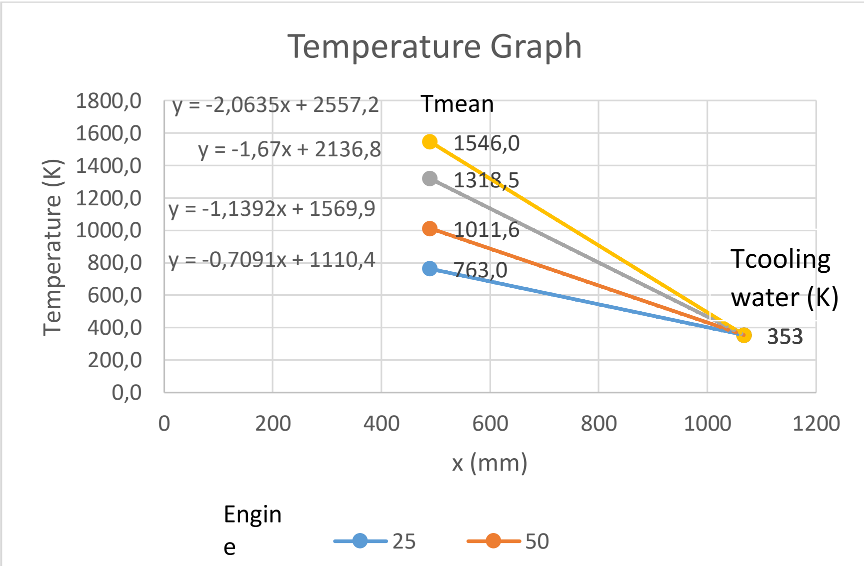

Figure 0.10 Temperature declination graph

The equations acquired from each represented load on the graph above are used to find temperature at the inner cylinder wall (x = 980 mm) at desired engine load.

Table 0.3 Result of cylinder wall temperature at desired load

Load (%)	Xinnercylwall at mid cylinder height (mm)	Tmean (K)	Tcw (K)
25	980	763,0	415,5
50	980	1011,6	453,5
75	980	1329,1	501,9
100	980	1546,0	535,0

After the temperature of the cylinder wall (Tcw) at desired load is known, relation graph between mean temperature of the cylinder (Tmean) and the cylinder wall temperature (Tcw) is made.

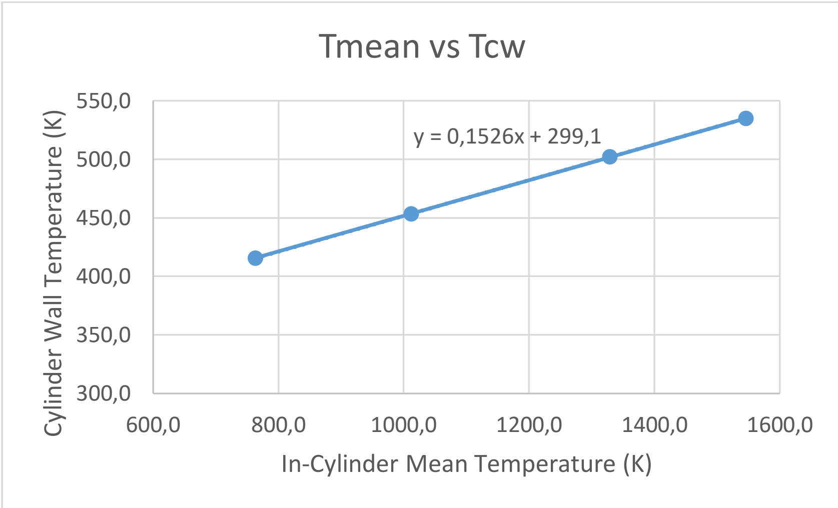

Figure 0.11 Tmean vs Tcw graph

The equation y = 0,1526x + 299,1 acquired from the graph in figure 3.10 is used to determine cylinder wall temperature (Tcw) at desired engine load, where "x" is the average or mean temperature (Tmean) inside the cylinder at desired engine load.

3.6. Dew Point of Aqueous Sulphuric Acid

Acid dew point is the temperature, at a given pressure, where any gaseous acid will start to condense into liquid acid. Dew point of aqueous sulphuric acid is determined using the equation used to predict dew point of sulphuric acid that commonly occur in gases as below (Kiang, 1981),

$$\frac{1000}{T} = 2{,}276 + 0{,}02943 \ln\ (pH2O) - 0{,}0858 \ln(pSO3) + 0{,}0062 \ln(pH2O) \ln(pSO3) \qquad (5)$$

Where,

T = Dew Point of Aqueous Sulphuric Acid (K)
pH2O = Partial Pressure of H2O (mmHg)
pSO3 = Partial Pressure of Sulphur Trioxide (mmHg)

Partial pressure of H2O and SO3 depends on the temperature where the gas occur, but temperature fluctuation inside the cylinder is very high. Therefore, in the modelling, partial pressure is determined according to the cylinder mean temperature as the assumption of the temperature where the gas occur.

Proportion of H_2O and SO_3 contained in the aqueous sulphuric acid also affects their partial pressure. In this modelling the actual proportion of H_2O and SO_3 is not known, hence assumption are made by varying the H_2O and SO_3 proportion. Variation of H_2O and SO_3 proportion in this modelling is made to also find out whether those proportions affects the starting point of cold corrosion occurrence. Below are example of graph of partial pressure or H_2O (80%) and SO_3 (20%) proportion contained in the aqueous sulphuric acid that is used for this modelling. Partial pressure data to build the graph can be found in Perry's Chemical Engineering Handbook (1934).

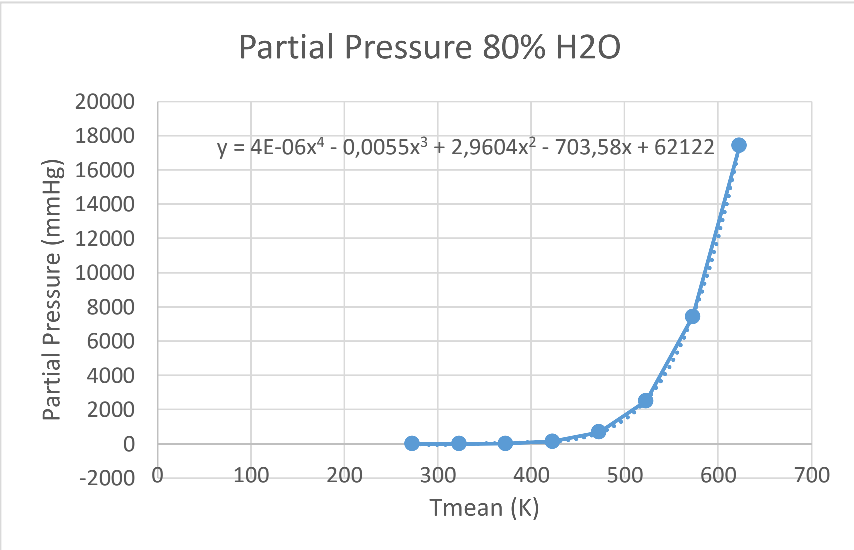

Figure 0.12 Partial pressure graph of 80% H2O

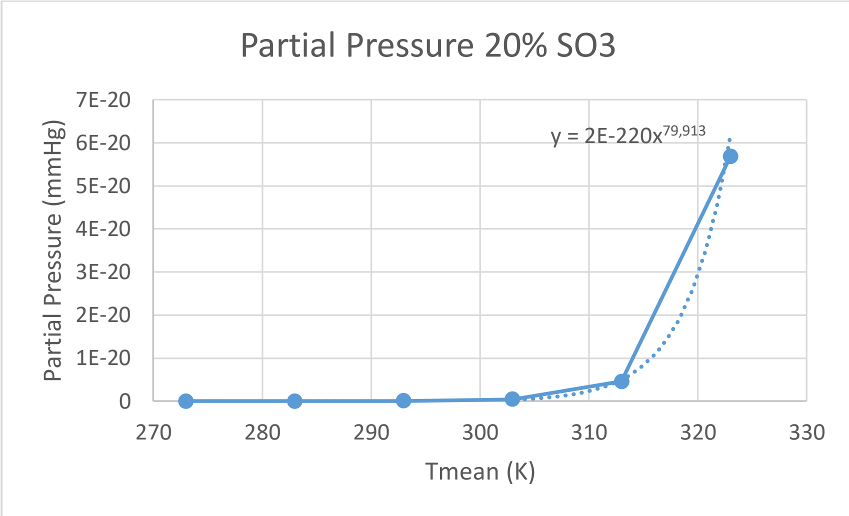

Figure 0.13 Partial pressure of 20% SO3

Two equations acquired from the graph above is used to determine partial pressure of each H2O and SO3 with the proportion above, where “x” is the mean temperature inside the cylinder (depends on the engine load running). After each partial pressure at desired engine load is determined, then the dew point of aqueous sulphuric acid with 80% H2O and 20% SO3 can be calculated using the prediction formula (equation 5). Graph for other variations of H2O and SO3 proportion can be found in the attachment.

3.7. Aqueous Sulphuric Acid Condensation and Cold Corrosion Consideration

Towards the final output of this modelling, which is statement if any cold corrosion occurs, there are two considerations to define whether cold corrosion will occur or not. First is consideration of the occurrence of sulphuric acid condensation and second is the consideration of the occurrence of cold corrosion itself.

3.7.1. Aqueous Sulphuric Acid Condensation

As stated in the previous chapter, the condensation of aqueous sulphuric acid depends on its dew point. When the aqueous sulphuric acid temperature reach below its dew point then it will start to condense. The bulk gas temperatures in the cylinder are substantially higher than the dew point temperature of aqueous sulphuric acid. Hence, no condensation is expected in the gas volume. However, since the cylinder wall always has the lowest temperature, condensation of aqueous sulphuric acid on the liner surface occurs.

Therefore, in order for aqueous sulphuric acid to condense on the cylinder wall the cylinder wall temperature must be lower than the dew point temperature. If the cylinder wall temperature is higher than the aqueous sulphuric acid dew point, then the condensation will not occur.

Figure 0.14 Condensation consideration block

There are two inputs in this condensation consideration block, cylinder wall temperature (Tcw) and dew point of aqueous sulphuric acid. If the cylinder wall temperature is higher than the dew point the output, u, will display the number "0" which means no condensation occur. Otherwise, if the cylinder wall temperature is lower than the dew point the output, u, will display the number "1" which mean condensation occur.

3.7.2. Cold Corrosion

Cold corrosion occurs on the cylinder liner walls when there are film or layer of sulphuric acid condensate form on its surface and there's not enough alkali (lubricating oil) to neutralise the acid. Hence, sufficient cylinder lubricating dosage is important to avoid or minimize the effect of cold corrosion.

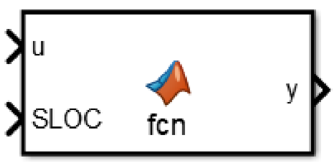

Figure 0.15 Cold corrosion consideration block

The cold corrosion consideration block has two inputs, u which is the determination whether condensation of aqueous sulphuric acid occurs and Specific Lubricating Oil Consumption (SLOC) which represent the cylinder oil dosage of the engine. If the input u is "1" and SLOC is less than or equal 0.1 g/kWh (engine SLOC based on project guide), output y will display the number "1" which means cold corrosion occur. Otherwise, output y will display the number “0” which means no cold corrosion occur.

Chapter 4
Simulation Result

This chapter will focus on the result of the calculation for the engine process parameters and simulation from the modelling that has been made, which is at which load of slow steaming cold corrosion started to occur. Results of the simulation are presented at each variations of water (H2O) and sulphur trioxide (SO3) proportion contained in the aqueous sulphuric acid as mentioned in the previous chapter.

4.1. Results

4.1.1. Engine Process Parameter Calculation from Modelling

Below are the calculation result of the engine parameter process calculation that has been made in chapter 3.

Table 0.1 Engine process parameter calculation result

Load (%)	**Pcomb (bar)**	**Pmax (bar)**	**Tcomb (K)**	**Tmax (K)**	**Tmean (K)**	**Tcw (K)**
60	82,26	106,72	1305,71	1693,97	1133,23	472,03
50	71,3	97,6	1131,75	1549,21	1026,98	455,82
40	60,34	88,48	957,78	1404,44	920,74	439,61
30	49,38	79,36	783,81	1259,68	814,50	423,39
20	38,42	70,24	609,84	1114,92	708,25	407,18
10	27,46	61,12	435,87	970,16	602,01	390,97

As seen from the table, all the parameters value are decreasing as the engine load is decreased. This condition fit the qualitative chart that has been made to define the process of cold corrosion.

4.1.2. Scenario 1

This first scenario is made by proportion of 80% H2O and 20% SO3 to calculate dew point of aqueous sulphuric acid. Below are the result of the modelling with H2O and SO3 proportion as mentioned.

Table 0.2 Modelling result of scenario 1

Load (%)	Cylinder Wall Temperature (K)	Dew Point of Aqueous Sulphuric Acid (K)	Cold Corrosion
60	472	348,81	NO

50	455,8	375,94	NO
40	439,6	401,90	NO
30	423,4	421,61	NO
29	421,8	422,93	YES
28	420,2	424,10	YES
27	418,5	425,09	YES
26	416,9	425,91	YES
25	415,3	426,53	YES
24	413,7	426,96	YES
23	412	427,16	YES
22	410,4	427,14	YES
21	408,8	426,88	YES
20	407,2	426,37	YES
10	391	405,41	YES

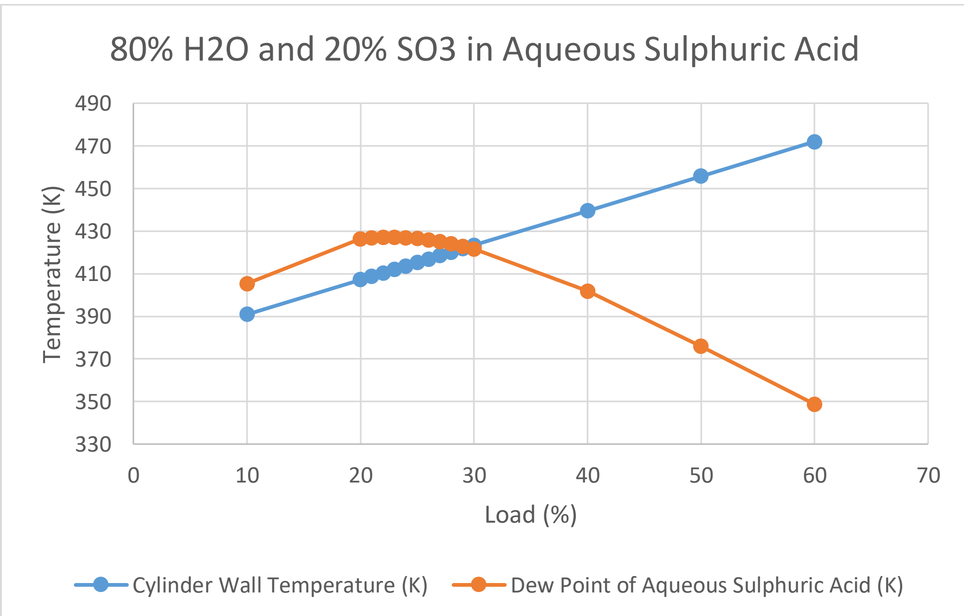

Figure 0.1 Cylinder wall temperature and dew point to load graph (scenario 1)

As seen on the table and graph above, cylinder wall temperatures decreases as the engine load decrease and the dew point of aqueous sulphuric acid increase as the load decrease. At certain point of load, temperature of the cylinder wall become lower than the dew point of aqueous sulphuric acid and that's when cold corrosion started to occur, in this scenario it started occur when the engine is running at 29% of load.

4.1.3. Scenario 2

The second scenario is made by proportion of 60% H2O and 40% SO3 to calculate dew point of aqueous sulphuric acid. Below are the result of the modelling with H2O and SO3 proportion as mentioned.

Table 0.3 Modelling results of scenario 2

Load (%)	Cylinder Wall Temperature (K)	Dew Point of Aqueous Sulphuric Acid (K)	Cold Corrosion
60	472	352,2	NO
50	455,8	372,1	NO
40	439,6	393,2	NO
30	423,4	409,6	NO
29	421,8	410,9	NO
28	420,2	412	NO
27	418,5	413,1	NO
26	416,9	414	NO
25	415,3	414,8	NO
24	413,7	415,5	YES
23	412	416,1	YES
22	410,4	416,5	YES
21	408,8	416,7	YES
20	407,2	416,8	YES
10	391	405,4	YES

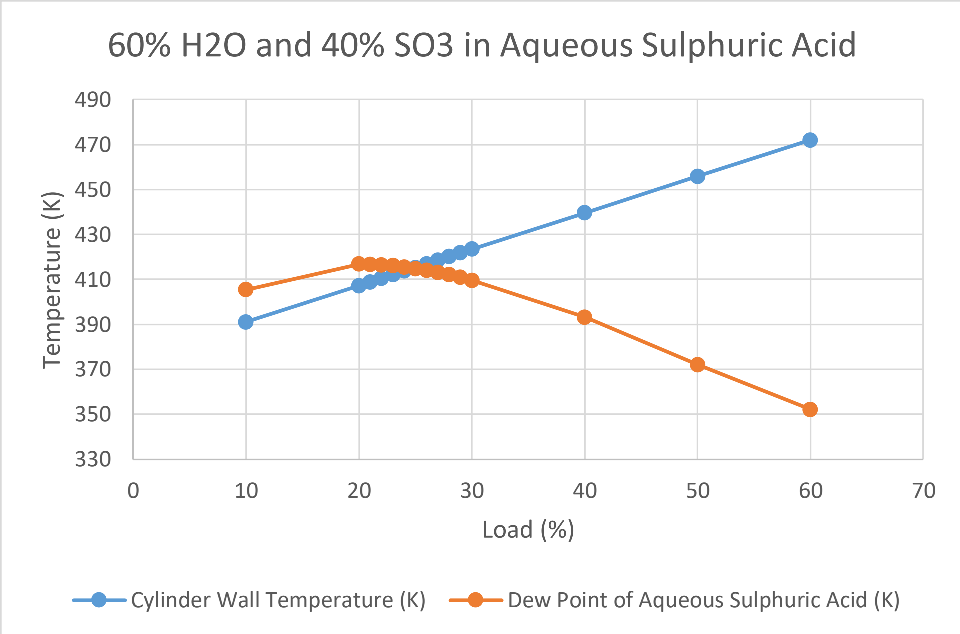

Figure 0.2 Cylinder wall temperature and dew point to load graph (scenario 2)

As seen on the table and graph above, cylinder wall temperatures decreases as the engine load decrease and the dew point of aqueous sulphuric acid increase as the load decrease. At certain point of load, temperature of the cylinder wall become lower than the dew point of aqueous sulphuric acid and that's when cold corrosion started to occur, in this scenario it start occur when the engine is running at 24% of load.

4.1.4. Scenario 3

The third scenario is made by proportion of 50% H2O and 50% SO3 to calculate dew point of aqueous sulphuric acid. Below are the result of the modelling with H2O and SO3 proportion as mentioned.

Table 0.4 Modelling results of scenario 3

Load (%)	Cylinder Wall Temperature (K)	Dew Point of Aqueous Sulphuric Acid (K)	Cold Corrosion
60	472	349,2	NO
50	455,8	369,4	NO
40	439,6	389,1	NO

30	423,4	405,8	NO
29	421,8	407,1	NO
28	420,2	408,4	NO
27	418,5	409,5	NO
26	416,9	410,6	NO
25	415,3	411,5	NO
24	413,7	412,4	NO
23	412	413,1	YES
22	410,4	413,7	YES
21	408,8	414,1	YES
20	407,2	414,4	YES
10	391	406,5	YES

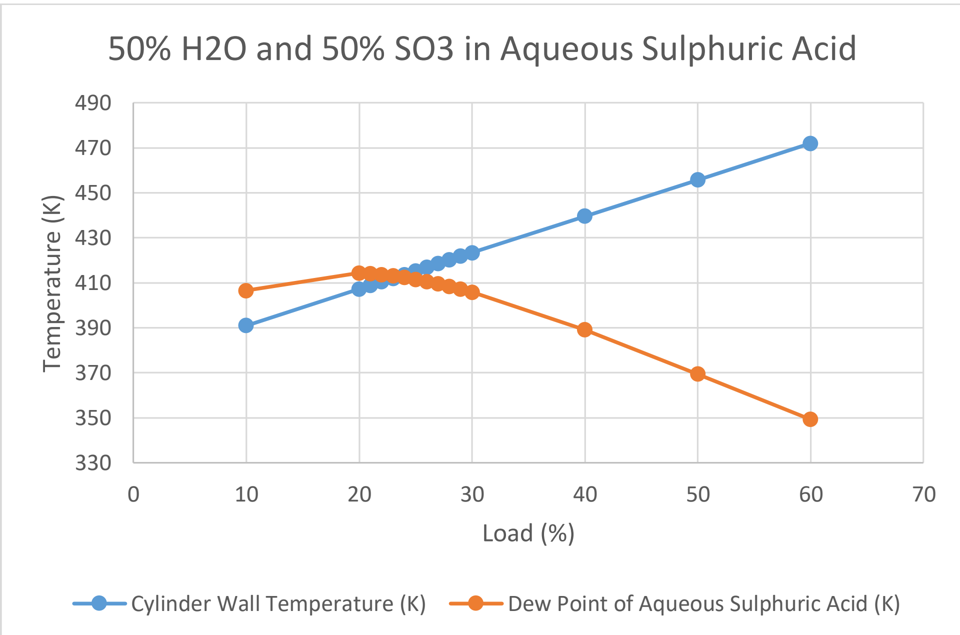

Figure 0.3 Cylinder wall temperature and dew point to load graph (scenario 2)

As seen on the table and graph above, cylinder wall temperatures decreases as the engine load decrease and the dew point of aqueous sulphuric acid increase as the load decrease. At certain point of load, temperature of the cylinder wall become lower than the dew point of aqueous sulphuric acid and that's when cold corrosion started to occur, in this scenario it start occur when the engine is running at 23% of load.

4.1.5. Scenario 4

The fourth scenario is made by proportion of 40% H2O and 60% SO3 to calculate dew point of aqueous sulphuric acid. Below are the result of the modelling with H2O and SO3 proportion as mentioned.

Table 0.5 Modelling result of scenario 4

Load (%)	Cylinder Wall Temperature (K)	Dew Point of Aqueous Sulphuric Acid (K)	Cold Corrosion
60	472	349,2	NO
50	455,8	368,6	NO
40	439,6	387,6	NO
30	423,4	404,1	NO
29	421,8	405,4	NO
28	420,2	406,7	NO
27	418,5	407,9	NO
26	416,9	409	NO
25	415,3	410	NO
24	413,7	411	NO
23	412	411,8	NO
22	410,4	412,5	YES
21	408,8	413,1	YES
20	407,2	413,5	YES
10	391	408,3	YES

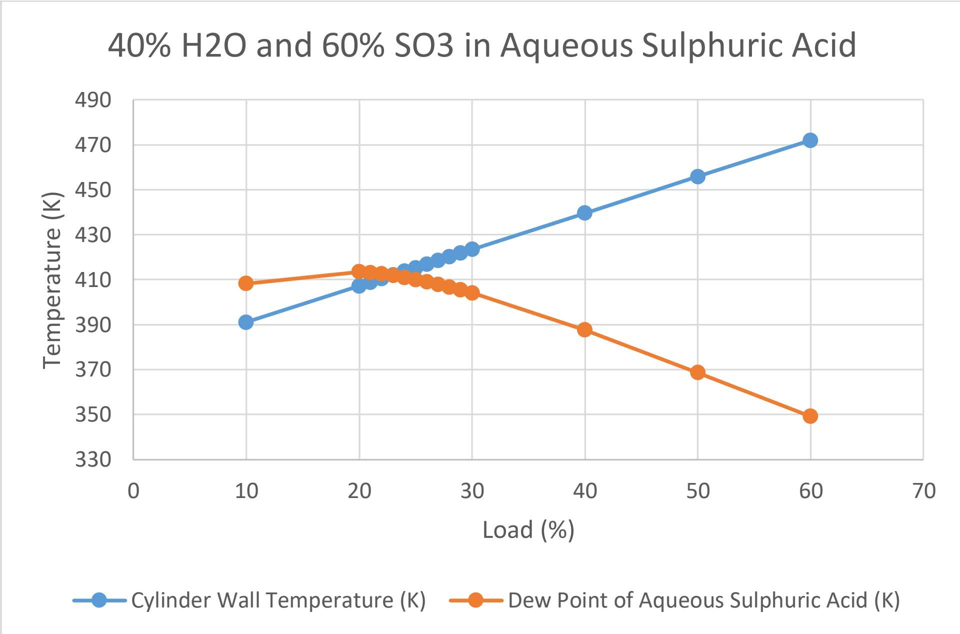

Figure 0.4 Cylinder wall temperature and dew point to load graph (scenario 4)

As seen on the table and graph above, cylinder wall temperatures decreases as the engine load decrease and the dew point of aqueous sulphuric acid increase as the load decrease. At certain point of load, temperature of the cylinder wall become lower than the dew point of aqueous sulphuric acid and that's when cold corrosion started to occur, in this scenario it start occur when the engine is running at 22% of load.

4.1.6. Scenario 5

The last scenario is made by proportion of 20% H2O and 80% SO3 to calculate dew point of aqueous sulphuric acid. Below are the result of the modelling with H2O and SO3 proportion as mentioned.

Table 0.6 Modelling results of scenario 5

Load (%)	Cylinder Wall Temperature (K)	Dew Point of Aqueous Sulphuric Acid (K)	Cold Corrosion
60	472	351	NO
50	455,8	368,6	NO
40	439,6	386,2	NO

30	423,4	402,3	NO
29	421,8	403,7	NO
28	420,2	405,1	NO
27	418,5	406,5	NO
26	416,9	407,7	NO
25	415,3	409	NO
24	413,7	410,1	NO
23	412	411,2	NO
22	410,4	412,2	YES
21	408,8	413,1	YES
20	407,2	413,9	YES
10	391	415,2	YES

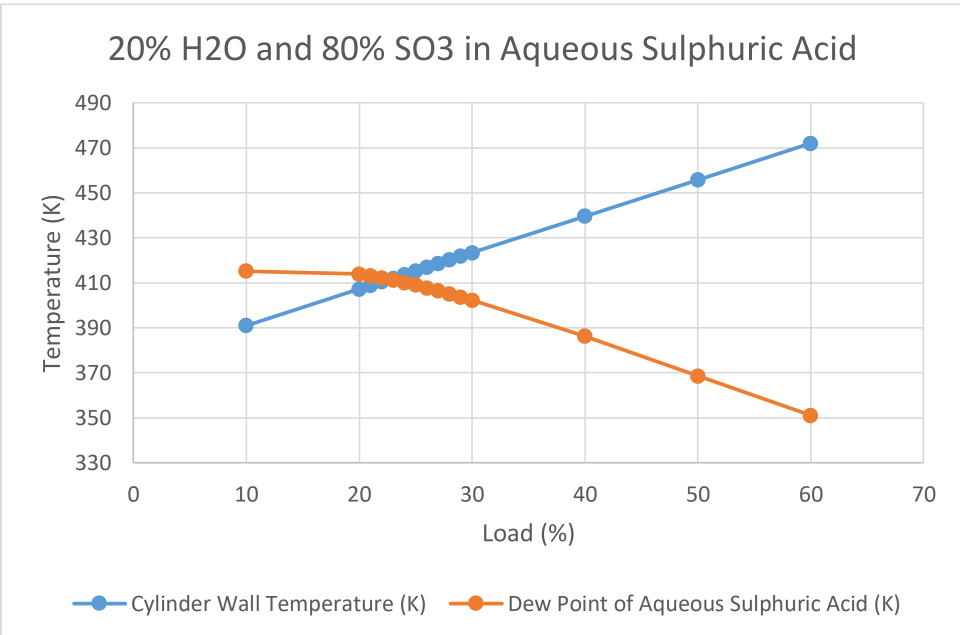

Figure 0.5 Cylinder wall temperature and dew point to load graph (scenario 5)

As seen on the table and graph above, cylinder wall temperatures decreases as the engine load decrease and the dew point of aqueous sulphuric acid increase as the load decrease. At certain point of load, temperature of the cylinder wall become lower than the dew point of aqueous sulphuric acid and that's when cold corrosion started to occur, in this scenario it start occur when the engine is running at 24% of load.

4.2. Result Conclusion

Result of the simulation from the modelling shows that cold corrosion start to occur fastest on the first scenario (80% H_2O and 20% SO_3) which is when the engine is running at 29% of load. On the scenario two (60% H_2O and 40% SO_3) and three (50% H_2O and 50% SO_3), cold corrosion start to occur when the engine is running at 24% and 23% of load respectively. On the last two scenario (40% H_2O and 60% SO_3, 20% H_2O and 80% SO_3), cold corrosion start to occur at 22% of engine load.

Results from the simulation shows that variations of the H_2O and SO_3 proportion contained in the aqueous sulphuric acid does not significantly effects the starting point of cold corrosion occurrence, although it can be seen from the table of simulation results that the dew point of aqueous sulphuric acid decrease as the H_2O proportion decreased and SO_3 proportion increased.

4.3. Implementation of The Modelling For Other Slow Steaming Effects

Cold corrosion model that has been made in this bachelor thesis can only be used to make a rough estimation about at which load the engine start to experience cold corrosion. Although, with some alteration and addition in the calculation this model might be applicable to make models of accumulation of combustion by-products on piston and exhaust ducting from incomplete combustion. Amount of fuel burned data at certain pressure inside the cylinder and calculation of combustion by-products might be needed to perform the modelling of accumulation rate of the by-products.

For the other slow steaming effects this modelling is not applicable, because it has a very different root of cause from cold corrosion. For Faster exhaust gas valve burn-away ,data of exhaust gas and exhaust valve temperature, scavenge air pressure, air/fuel ratio, as well as burn-away rate of the spindle disk at each operating condition might help to make model of estimation of when is the next maintenance or inspection should be done. For abrasive wear and accumulation of black sludge in scavenge spaces due to over-lubrication, results from piston underside inspection (lube oil inspection), engine manufacturer recommendation regarding cylinder oil dosage, data of engine cylinder oil dosage might be needed to define whether over-lubrication happen.

Chapter 5
Conclusion

The practice of slow steaming has proven to reduce ship's fuel cost, which is the biggest cost proportion in ship's operating costs. However operating the engine much below their designed load range for prolonged time can evoke problems to the engine operation.

From the discussion and research on the effects of slow steaming, there are seven reported problems or peculiarities occur to the engine due to slow steaming. The problems are:

a. Corrosive impact (cold corrosion) on cylinder liners and piston rings
   Cold corrosion inside the cylinder is caused by condensation of aqueous sulphuric acid on the surface of cylinder walls. This problem cause damage to the engine because loose corrosion products from cylinder liner and piston rings are easily removed by wear to continually reveal fresh metal beneath, which in turn another corrosion can attack quickly. Hence, material loss rate on the combustion chamber components can be very high, much higher than the sum of the individual contribution of wear and corrosion.
b. Faster exhaust valve burn away
   Burn-away on the exhaust valve spindle disk is caused by a phenomena called intergranular corrosion.Faster burn-away may reduce the time between overhaul for the exhaust gas spindle. When the burn-away has reach the set-up limit re-welding must be done, otherwise the metal loss from the valve spindle will cause valve sealing loss. This can lead to reduce compression pressure that effects the combustion process, eventually reducing or even total loss of generated power from the particular cylinder.
c. Broken scavenge air receiver valve
   This damage is caused by increased mechanical load on the scavenge air valve the auxiliary blower is continuously on-off on when the engine is operating on the load area of switch point of auxiliary blower cut-in. Broken and missing valves will have a negative impact on the engine performance when the auxiliary blowers are running, since the broken valve will disturb the turbine operation in the auxiliary valve that drew the air in.
d. Accumulation incomplete combustion by-products
   Accumulated unburned fuel and soot in the exhaust gas ducting increases of fouling in the exhaust gas. Consequences could be as important reduction of turbo charger efficiency, clogging of exhaust gas boilers, also ignition possibilities after engine load up in the exhaust gas

ducting causing damage to the turbocharger due to sudden over-speeding.

e. Abrasive wear of cylinder liner and piston rings
   When over-lubrication occurs, excess cylinder oil and additives from the cylinder oil will burn off and form layer of deposit on top land of the piston. The deposit build up on the piston top land appears to restrict the free movement of the piston, possibly affecting the normal alignment of the piston, causing wear on the piston skirt. Wear scrapped cylinder bore and piston, causing it to become bigger and smaller respectively, this can cause gas blow-by to occur and reduce compression pressure.
f. Accumulation of black sludge in scavenge spaces
   When there’s abundant supply of lubricating oil sprayed to the combustion chamber, the excess lubricating oil will be scrapped down by the oil-control rings on the pistons. This condition caused piston rings clogging and increasing risk of scavenge fire.
g. Minor developing fatigue on central pad in lower crosshead bearing shells.
   Turbocharger cut out has resulted in minor slow-developing fatigue damage on the central pad in the lower crosshead bearing shell.

From the modelling of cold corrosion occurrence, it can be concluded that cold corrosion started to occur when the engine is running below 30% of load. Cold corrosion occur fastest on the first scenario (80% H2O and 20% SO3), which is at 29% of load. Variations of H2O and SO3 proportion contained in the aqueous sulphuric acid does not give significant effects on the starting point of cold corrosion occurrence.

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Attachment

Due to page limitation, supporting materials of this research such as Simulink files of the modelling, excel files of the calculation material, supporting data and paper of study case can be found in the CD-ROM attached on the last page of this report.

Author's Biography

The author named Andari Wisnuwardhani Sungging Rakhammurti was born in Jakarta, 25 March 1997. The author studied at SDK Ign. Slamet Riyadi II Elementary School, 49 State Junior High School, and 28 State Junior High School. Then the author continue the education at Double Degree of Marine Engineering Department, Institut Teknology Sepuluh Nopember – Hochschule Wismar in 2015 with registered number 04211541000022. During the study, the author was active as the treasurer of ITS Marine Solar Boat Team (2017-2018) and also active as a member of ITS Society of Petroleum Engineer (SPE) (2018). The author also had on job training experience at PT. Citra Shipyard Batam and Pertamina Trans Kontinental. In the fourth year of the study the author went to Rostock, Germany to work and finish the Bachelor Thesis at Hochschule Wismar.