

DESIGN, THEORY, DEVELOPMENT AND TESTING OF A NEW FORM OF STEAM
POWER GENERATION – THE PRACTICAL STEAM ENGINE

by

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ABSTRACT

A new form of steam power generation, a Practical Steam Engine (PSE), is analyzed. This product focuses on applications of power generation from steam with insufficient flow and/or pressure reduction for efficient steam turbine operation. This text considers the theory, design and development of the product as well as testing and performance review. An application of the PSE in parallel to a pressure reducing valve (PRV) is analytically considered. The theoretical performance in this application is predicted and then experimentally tested. Results of this experiment show the PSE can reach isentropic efficiencies of more than 80% and, after optimizing friction, net efficiencies can reach more than 65%. Furthermore, the analytical model is proven to very accurately represent the experimental data with average error of 2.35% for most flow ranges. Additionally, a case study of the PSE in a commercial PRV parallel application is considered, demonstrating significant economic benefit of use of the PSE in district energy applications.

DEDICATION

This dissertation is dedicated to my loving wife, Mary. You encourage me when the going gets tough and support me in tough times. You are my soulmate and I couldn't imagine doing this without you. To my brother Ben, who is always there when I need you and I truly appreciate it. You have lessened my burden during this endeavor. To my mother, Cindy, for always taking the time and effort to read with me and make sure I did my homework (which I regularly did not do). You always took a deep interest in my education and I owe you more than you could know. To my father, Robby, for teaching me engineering principles and teaching me that you get out of this life what you put in – nothing is for free. I love you all and thank you from the bottom of my heart.

LIST OF ABBREVIATIONS AND SYMBOLS

ABBREVIATIONS

BBC	Big Block Chevrolet
BDC	Bottom Dead Center
CHP	Combined Heat and Power (Co-generation)
HP	Horsepower
ICE	Internal Combustion Engine
kW	Kilowatt
kWe	Kilowatt – Electrical Power
MEP	Mean Effective Pressure
PMG	Permanent Magnet Generator
PRV	Pressure Reducing Valve
PSE	Practical Steam Engine
PV	Pressure-Volume
TDC	Top Dead Center

SYMBOLS

A – area

a – acceleration

c_p – the isobaric heat capacity

D – diameter

f – friction factor

g – acceleration due to gravity

h – convective heat transfer coefficient or enthalpy

h_{fg} – heat of vaporization

k – thermal conductivity

M – mach number

m – mass

$\overline{Nu}$ – Nusslet number

P – pressure

Pr – Prandtl number

Q – thermal energy

R – gas constant

Ra – Rayleigh number

Re – Reynolds number

S or s – entropy

T – temperature

t – time

U or u – flow velocity, internal energy, or uncertainty

V – volume or velocity

v – specific volume

V_d – engine displacement

W – work

X – position or quality

Z – compressibility factor

α – thermal diffusivity

β – volumetric thermal expansion coefficient

γ – ratio of specific heats

θ – crankshaft degrees from TDC

μ – viscosity

ρ – density

σ – surface tension

ν – kinematic viscosity

τ - Torque

ψ – Exergy of fluid

ω – engine speed

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Lastly, and most importantly, I want to thank God. There are many gifts He has given me, but even those gifts are not enough without Him. I thank Him for giving me an aptitude for science. We live in a world where a great many people believe that you

have to respect God OR science. I disagree. Having belief in God is not contradictory to science. I suggest that God is complimentary to science. Science is simply the study of the existence of all earthly things. Science develops by creating hypotheses for what we observe. In my opinion, it requires more faith to believe the things we observe exist and function without belief of an intelligent creator than it does to believe in such a being.

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1. BACKGROUND/MOTIVATION

a. INTRODUCTION

Despite continual changes in the U.S. energy market, nothing has changed the fact that energy costs remain high enough to force end users to seek cost savings as well as the fact that the easiest savings are from increased energy efficiency. While recent energy market focus has been on increased production of fossil fuels and renewable energy, no developments in this area will change the fact that reducing energy waste is both the most attractive traditional energy strategy and the premiere energy opportunity.

There is a tremendous market for energy efficiency applications, with demand driven by energy prices and by governmental policy with incentives for production and use of “green energy”. Energy efficiency projects may include several types including: co-generation, waste heat recovery, waste steam recovery and others.

Many states have furthered the push for energy efficiency by implementing mandates for percentage of energy production to be in form of renewable energy. Each of these mandates are called a Renewable Portfolio Standard (RPS) and are shown in Figure 1-1. These RPS policies for renewable energy and energy efficiency projects focus primarily on the industrial and commercial sector. While these RPS policies may, at first glance, appear to only affect renewable energy applications, they have significant impact on energy efficiency projects. As a substitute for renewable energy production, utilities may improve energy efficiency of the distributed power. In most cases, the cost

of end user increased efficiency is significantly less than the cost to produce the offset energy with renewable energy methods. Therefore, many utilities are offering incentives to end users for energy efficiency improvements.

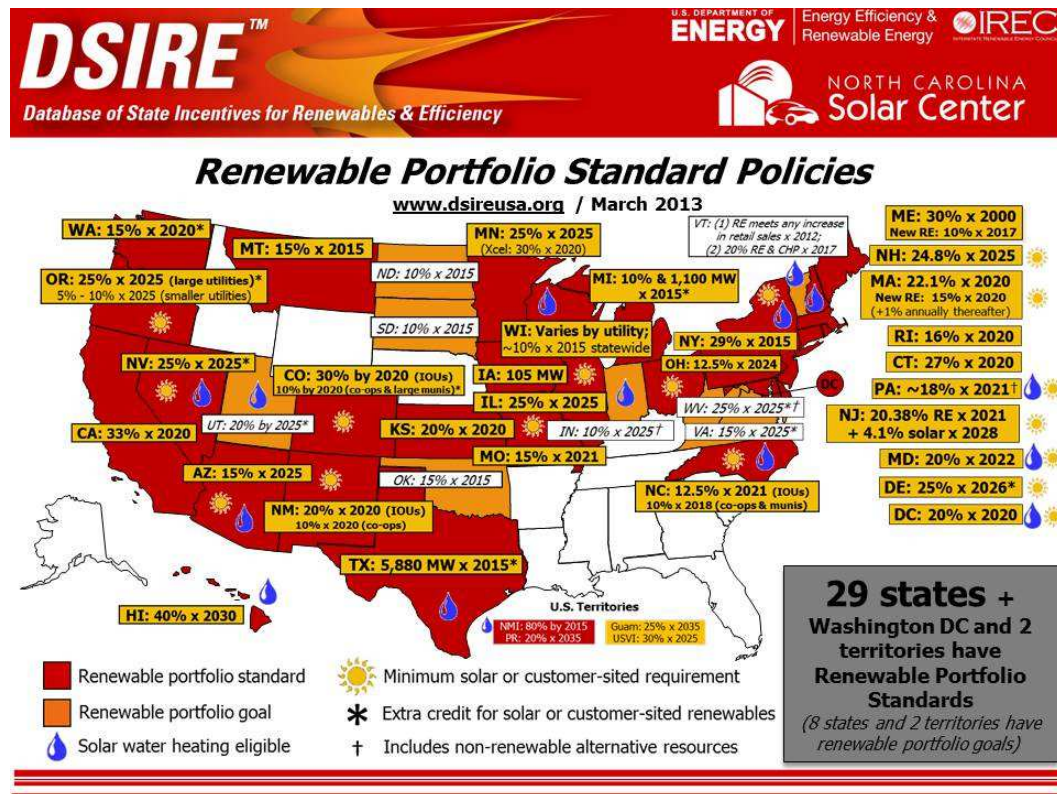


Figure 1-1. Renewable Portfolio Standard Policies (Database of State Incentives for Renewables & Efficiency, 2013)

These energy efficiency and renewable energy projects may be in various forms, over various industries in various applications. However, a vast majority of these projects utilize some form of power generation, often in the form of steam power generation, thus creating the demand and need for flexible form of steam power generation.

b. OVERVIEW OF CATEGORIES WITH OPPORTUNITIES FOR ENERGY
EFFICIENCY IMPROVEMENTS AND RENEWABLE ENERGY
PRODUCTION

There are seven primary categories that this text will focus on as areas of opportunity for energy efficiency improvement and renewable energy production. Of these seven categories, four of which focus on energy efficiency improvements while the other three focus on renewable energy production. These categories are as follows:

A. Energy Efficiency Improvements

- a. PRV Parallel
- b. Co-generation (Combined Heat and Power)
 - i. Also Tri-generation (Combined Cooling, Heat and Power)
- c. Industrial Waste Steam
- d. Industrial Waste Heat

B. Renewable Energy Production

- a. Biogas
- b. Biomass
- c. Solar

Brief overviews of the seven above categories are given in the following text.

i. Identified Markets with Energy Efficiency Improvement Opportunities

Presented here are categories that offer opportunities for energy efficiency improvements as applicable to the product studied in this text. In addition to description of the category, obstacles in improvements are discussed.

a. PRV Parallel-

Commercial and industrial steam systems primarily consist of district energy systems and distributed multi-pressure systems, respectively. The commercial district energy systems are systems in which steam is used for space heating and small process use in commercial buildings. There are 800+ district heating systems in the United States as shown in Figure 1-2.

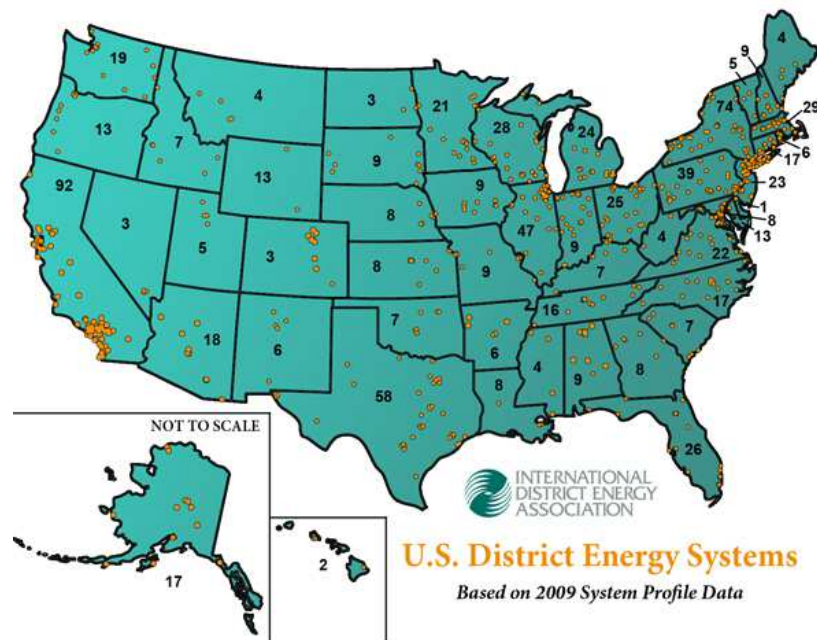


Figure 1-2. U.S. District Energy Systems Map (International District Energy Association, 2009)

These district energy systems cover all 50 states and include colleges and universities, community utilities, healthcare installations, military/government installations. There are 4.7 million commercial buildings in the U.S. and more than 580,000 (12 percent) are served by boilers (Oak Ridge National Laboratory, 2005).

In most district energy systems, there is a central boiler(s) that produces medium to high pressure steam (>100 psi). This steam is distributed to surrounding commercial/industrial buildings through pipes in the district energy network. This steam is distributed at an elevated pressure (>100 psi) in order to increase density and thus reduce volumetric flow rate required for a given mass flow rate. This reduced volumetric flow rate results in the allowable use of smaller, less expensive, piping systems. However, most radiators and process steam systems in the commercial buildings use the steam at a relatively low pressure (<50 psi). Therefore, in order to use the steam, the pressure must be reduced. This is commonly done by means of a Pressure Reducing Valve (PRV). These PRVs act as a throttling valve to throttle the high pressure steam in the supply pipes down to low pressure steam for use by the steam systems in the buildings. At first glance, this may seem like a very efficient method to reduce the steam due to the fact that a throttling valve is isenthalpic. However, there are energy losses associated with implementation of a PRV. First, as with any device, the PRV includes inherent friction and inefficiencies. Furthermore, in most district energy systems the steam enters the PRV as a saturated vapor, thus reducing heat transfer coefficient and consequently effectiveness of heat exchangers. Therefore, any isenthalpic pressure reduction results in a superheated vapor leaving the PRV. Furthermore, while an isenthalpic PRV is constant energy, it is not constant exergy. The

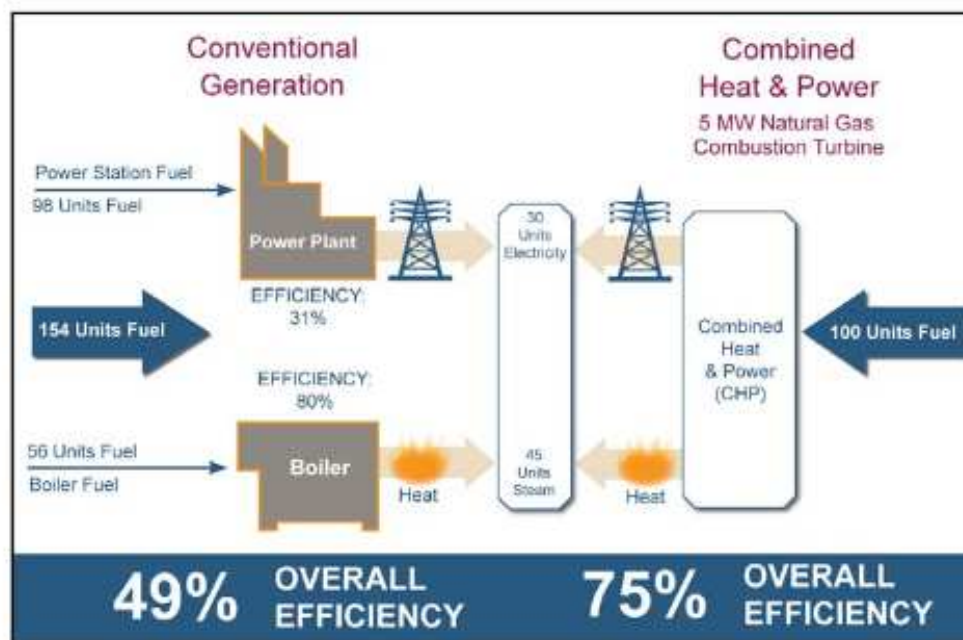
reduction in exergy, or ability to do work, through a PRV is a practical loss of energy and therefore, inefficiency. A common method of reducing this inefficiency is by use of a backpressure steam power generation product in which a steam power generation produce is installed in parallel to a PRV. The high pressure steam enters the backpressure steam power generation equipment and the pressure is reduced while a by-product of electricity is produced. The low pressure steam leaves the steam power generator and is used by the facility. This is a prime example of how energy efficiency can be improved in district energy systems.

Similar to commercial district energy systems, industrial steam systems use steam at a variety of pressures, commonly utilizing PRVs to reduce pressure from one location to another. As with the district energy applications, use of a backpressure steam power generation device can improve the efficiency of many industrial steam system applications. However, in both district energy systems, as well as industrial steam systems, the steam flow rate and pressures may be insufficient for efficiency power generation with available products today. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. This solution, and application of the product of focus, is discussed in a future portion of this text.

b. Commercial Co-Generation (Combined Heat and Power) Systems –

Co-generation systems, also known as CHP, generate electricity and useful thermal energy in a single, integrated system. CHP is not a technology but an approach to applying technologies. Heat that is normally wasted in conventional power generation is

recovered as useful energy, which avoids the losses that would otherwise be incurred from separate generation of heat and power. An example of a CHP application is shown in Figure 1-3. While the conventional method of production useable heat and power separately has a typical combined efficiency of 45-50 percent, CHP systems can operate at levels as high as 80 percent (American Council for and Energy-Efficient Economy, 2009). Figure 1-3 illustrates a typical application of CHP when compared to that of conventional, separate heat and power generation.



Note: Assumes national averages for grid electricity and incorporates electricity transmission losses.

Figure 1-3. Efficiency Gains of CHP (U.S. Environmental Protection Agency - Combined Heat and Power Partnership, 2008).

CHP is widely used in steel, chemical, paper, and petroleum-refining industries, and at large institutional campuses such as universities. In recent years, smaller CHP systems have begun to make advancement in the food, pharmaceutical, and light

manufacturing industries, in commercial buildings; and at smaller institutions such as hospitals. The CHP industry, U.S. Department of Energy (DOE), and the U.S. Environmental Protection Agency (EPA) set an ambitious but achievable goal to double U.S. CHP capacity between 1999 and 2010, using 1998 as a baseline, by adding 92 GW of new capacity (American Council for and Energy-Efficient Economy, 2009). A recent report, co-funded by DOE, suggests that CHP could represent as much as 20% of U.S. electric generation capacity by 2030, or 241 GW of total capacity (U.S. Environmental Protection Agency, 2008). CHP is used in every state and in the District of Columbia at over 4,100 facilities including factories, commercial buildings, hospitals and universities (U.S. Department of Energy & U.S. Environmental Protection Agency, 2012). Approximately 12 percent of the total electricity generated in the United States comes from CHP (ICF International, 2013).

CHP, a proven technology that has been used for decades, can replace on-site boiler use and grid supplied electricity (U.S. Environmental Protection Agency - Combined Heat and Power Partnership, 2013). Replacement of on-site boiler use and installation with on-site boiler use, are two great examples of how CHP can improve the efficiency of the energy market. In these two examples, steam is either purchased from a utility or produced on site with a boiler. The steam is then supplied to the CHP system in which electricity is produced and low pressure steam is supplied for heating, cooling or both (in tri-generation). The electricity produced by the CHP system can be used to offset some or all of the electricity purchased from the grid. This results in improved energy efficiency and resistance to brownouts and blackouts. Application of the CHP system as replacement of on-site boiler is similar to the aforementioned PRV bypass

application in district heating systems in that district steam is supplied to the steam power generation device, in which power is produced and low pressure steam is delivered for heating. These are examples of how CHP systems can improve energy efficiency of the industrial and commercial sectors. However, many of the potential applications of CHP are small scale in which no steam power generation device on the market can efficiently make power. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. A potential solution is the focus of this text.

c. Industrial Waste Steam –

Many industrial and commercial facilities utilize steam for a variety of reasons including drying processes, heating processes, cleaning processes, etc. Over 226,000 domestic manufacturing/industrial facilities have on-site boilers while there are more than 163,000 commercial and industrial boilers in the United States (Energy Information Administration, 1998) (Oak Ridge National Laboratory, 2005). These boilers consume about 8,100 TBtu per year, accounting for about 40 percent of all energy assumed in steam-consuming industrial sectors (Oak Ridge National Laboratory, 2005). The majority of the boilers and the largest boilers are located at facilities in the food, paper, chemicals, refining and primary metals industries as shown in Figure 1-4. These industries account for 71 percent of industrial boiler units and 82 percent of the industrial boiler capacity (Oak Ridge National Laboratory, 2005). The following figure

illustrates the percentage of sector energy consumption that is utilized for steam production in a few well-known sectors.

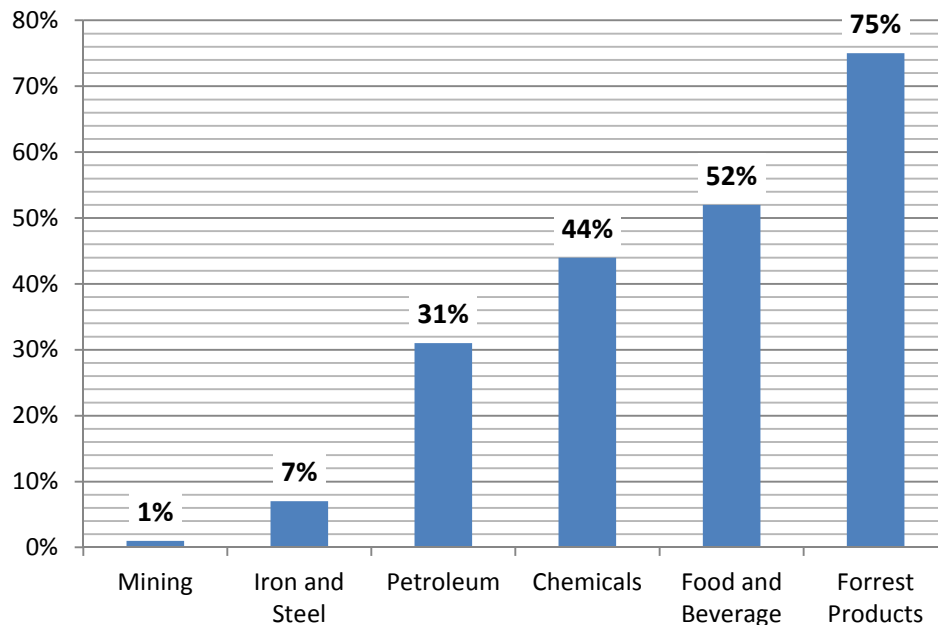


Figure 1-4. Steam System Energy Use as a Percentage of Total Sector Energy Use (Thekdi, 2011).

Despite the staggering amount of energy consumption these sectors use for steam production, steam systems are often times one of the most inefficient systems in an industrial plant. Inefficiencies in a steam can include leaks, heat losses, piping restrictions, improper steam rates, etc. One of the most significant of these is improper steam rates. In many steam consuming industries, steam demand can vary substantially throughout the year. When steam demand decreases, often times, this results in substantial quantities of excess steam produced due to the fact that steam production cannot always easily be reduced to match demand for a variety of reasons. If this excess steam cannot be utilized, it is disposed of, usually vented to atmosphere.

Not only is this process noisy and inconvenient, it is extremely inefficient. This is inefficient for a variety of reasons. First, all energy in the form of steam enthalpy is wasted. Second, the water is wasted. In many industrial sites, water is extremely valuable and often times hard to come by. Lastly, the wasted water usually contains expensive anti-corrosion, water softening and similar chemicals. As the water is wasted, so are the chemicals.

A common method that is utilized to reduce this inefficiency is to use some form of condensing steam power generator system to extract the enthalpy from the steam and produce power while condensing the steam and chemicals such that they can be utilized as hot water in the plant and/or returned to the boiler feed water system. However, many of the potential applications of waste steam recovery are small scale in which no steam power generation device on the market can efficiently make power. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. A potential solution is the focus of this text.

d. Industrial Waste Heat –

Nearly all industrial facilities utilize heat for a variety of reasons such as steam production, drying, melting, forming, burning, etc. Industrial heating processes (not including steam generation) consume about 5.2 quads (quadrillion Btu), which is nearly 17 percent of all energy consumed by industry (Office of Energy Efficiency and Renewable Energy, 2010). Heat derived from combustion of fossil fuels accounts for 92 percent of this energy; electricity use accounts for the remaining eight percent (Office of

Energy Efficiency and Renewable Energy, 2010). Figure 1-5 illustrates the percentage of sector energy consumption that is utilized for process heating (not including steam production) in a few well-known sectors.

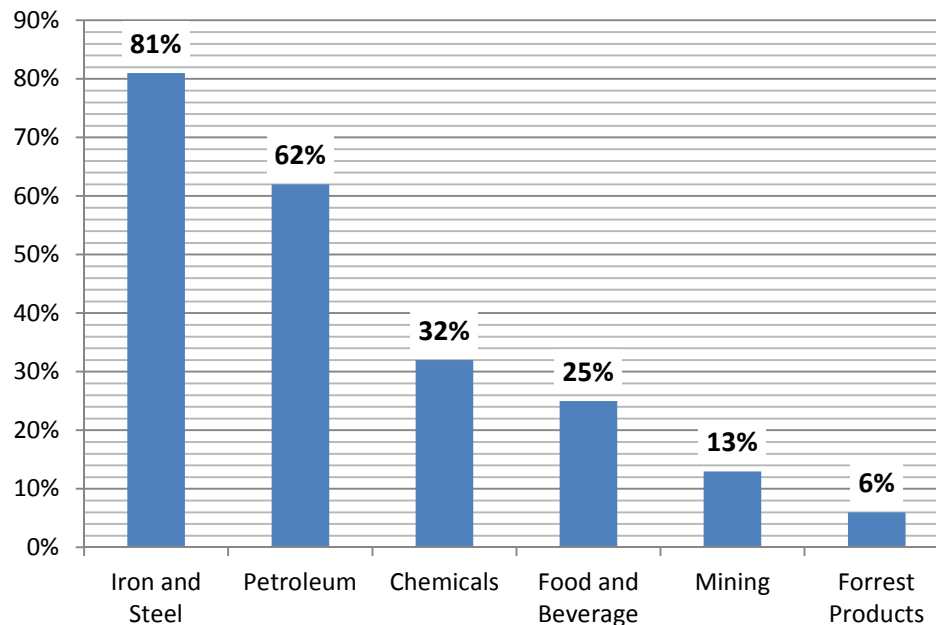


Figure 1-5. Process Heating Systems Energy Use as a Percentage of Total Sector Energy Use (Thekdi, 2011).

Today, overall thermal efficiency of process equipment varies from 15 percent to 80 percent, compared to the thermal efficiency of steam generation, which varies from 65 percent to 85 percent (Office of Energy Efficiency and Renewable Energy, 2010). Much effort has been taken by the DOE and other groups to determine energy improvement opportunities for these process heating systems. Some commonly identified areas of improvement opportunities are heat generation control, heat transfer improvement, heat containment, enabling technology and heat recovery. The DOE estimates approximately 330 million MWh annually (23.6 million households) could be

recovered from industrial waste heat with available energy efficiency improvement techniques and technologies (Thekdi, 2011).

A common method that is utilized to reduce this inefficiency is to recover waste heat in order to generate steam for power production and/or use in the plant. The DOE suggests that 20% of all energy used for process heating systems could be recovered by use of this effective technique (Office of Energy Efficiency and Renewable Energy, Industrial Heating Equipment Association, 2007). In most waste heat recovery applications, waste heat is used to generate steam which is provided to some form of steam power generator system to extract the enthalpy from the steam and produce power while condensing the steam and chemicals such that they can be utilized as hot water in the plant and/or returned to the boiler feed water system. However, many of the potential applications of waste steam recovery are small scale in which no steam power generation device on the market can efficiently make power. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. A potential solution is the focus of this text.

ii. Identified Markets with Renewable Energy Production Opportunities

Presented here are categories that offer opportunities for renewable energy production as applicable to the product studied in this text. In addition to description of the category, obstacles in improvements are discussed.

a. Biogas

In recent years, much traction has been gained in the push for biogas as a fuel for energy production. Effective utilization of biogas for power production is a practical source of renewable energy production. One of the largest sources of biogas is landfills. Landfills produce vast amounts of methane, commonly referred to as landfill gas (LFG). Landfills were the third largest human-made source of methane in the United States in 2011, accounting for 17.5 percent generated (Landfill Methane Outreach Program, 2013). There are currently 621 operational LFG power production projects across 48 states in the United States with a current annual production rate of 102 billion cubic feet of LFG generating 16 billion kilowatt-hours of electricity – enough to power 1.18 million homes and heat 746,000 homes. An additional 450 EPA-identified candidates are estimated to annually produce 155 billion cubic feet of LFG generating seven million megawatt-hours of electricity – enough to power 500,000 homes or heat 1.1 million homes (Landfill Methane Outreach Program, 2013).

LFG is eligible as a renewable resource in 37 states as well as the District of Columbia, Guam, Northern Mariana Islands, Puerto Rico, and the U.S. Virgin Islands. Therefore, in these locations, LFG power production may be used towards the RPS policies shown in Figure 1-1. To utilize the LFG, it is simply burned for heat to produce power. This combustion process is sometimes used for an internal combustion engine, a boiler to produce steam, a combined cycle, or some other method to generate useful energy, with an internal combustion engine being the most common. However, the LFG is often quite impure and can create significant maintenance difficulties for the engines. For this reason, among others, a common method of utilizing LFG is by use a boiler

such to produce clean steam. Clean steam is then supplied to a steam power generation device to produce power while only burning LFG in an easy to maintain boiler.

However, many of the potential applications of biogas power production are small scale in which no steam power generation device on the market can efficiently make power. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. A potential solution is the focus of this text.

b. Biomass

Another form of renewable energy that utilized at an increasing rate is biomass. Biomass includes any substance derived from living, or recently living organisms. The most common forms of biomass include forest-derived products such as wood and pulp, municipal solid waste, ethanol, and biodiesel. Biomass power production accounted for approximately 5.5 percent of the United States power production in 2013 (United States Energy Information Administration, 2014). It is projected that the biomass consumption will increase by more than 50% by 2030 (Oak Ridge National Laboratory, 2011). This projected sharp increase in biomass consumption is easily supported by current biomass supply. There are 423 million tons per year of biomass resources available in the United States (Milbrandt, 2005). Due to these facts, it is expected that biomass will play an increasingly large role in the United States' power production in future years.

Energy production from biomass is a simple process, very similar to biogas power production. The various forms of biomass are combusted to generate heat for power production. This heat can be used in a boiler to produce steam, a combined cycle, an organic Rankine cycle, or similar. Most forms of biomass power production are eligible as a renewable energy source and can be applied towards the RPS policies. Biomass is often used in boilers to produce steam which is supplied to a steam power generation device to produce power

However, many of the potential applications of biomass power production are small scale in which no steam power generation device on the market can efficiently make power. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. A potential solution is the focus of this text.

c. Solar

One of the most common forms of renewable energy that can be utilized is solar power. While still a minor source of the United States produced electricity, 0.4% in 2013 (United States Energy Information Administration, 2014), solar power is an extremely popular form of renewable energy power production in residential and small scale power production. Solar power includes photovoltaic (PV) and other forms. For the purpose of this discussion and application to the product of focus of this text, we will focus on one of the other form of solar power – solar power steam generation for steam power generation. The concept is that solar power (light energy from the sun) is focused onto a mass of water in order to boil the water. This water is then used to produce power

through typical steam power generation methods. This solar powered steam generation is not a new concept. The first recorded application was in 1865 when Auguste Mouchout designed and patented the first machine that generated electricity using the sun and the steam was fed to a steam engine. However, solar steam power plants have not remained novelties.

Near the Nevada-California border lays a 400MW solar powered steam power plant. The plant utilizes nearly 350,000 computer-controlled mirrors to focus sunlight to boilers atop 459 foot tall towers. The power plant can generate enough power for 140,000 homes (Huge solar-thermal plant opens near Nevada-California Border, 2014). This power plant goes to show the validity of potential utilization of solar power for large scale power production. This form of power generation would qualify as a renewable energy source and count toward the RPS policies.

However, many of the potential applications of solar power production by steam generation are small scale in which no steam power generation device on the market can efficiently make power. Therefore, development of a steam power generation device to utilize these reduced steam flows and pressure could substantially advance energy efficiency projects in these systems. A potential solution is the focus of this text.

c. STEAM POWER GENERATION

i. Introduction

Four categories with energy efficiency improvement opportunities were summarized as well as two categories with sources of renewable energy production have been discussed. For each of these categories, a brief description was given for methods in which the energy efficiency improvements could be made or how a renewable energy

source can be converted into usable power. In all six of these categories, one of the methods that power can be produced is with some form of steam generation. However, simply producing steam is not always the end source of usable power. In most cases, the steam is utilized to produce usable power, usually mechanical power. This mechanical power is often times converted to electrical power by means of an electrical generator. While the ultimate form of energy, mechanical or electrical, is important to the end user, it has little impact on steam power generation – the conversion of steam to mechanical power. The history of steam power generation is presented here.

ii. History

Steam power generation dates back as far as the 1st century AD; the first recorded rudimentary steam power device being the Aeolipile, shown in Figure 1-6, described by Hero of Alexandria. This was a rudimentary steam turbine in which steam was ejected tangentially from a sphere through nozzles. The outlet flow of steam produced a reaction force and caused the device to rotate. While this device was not capable of performing successful work, it proves the concept of utilizing steam pressure to produce mechanical work was known in Roman Egypt in the 1st century.

Over the next 1500 years, several steam power generation devices were constructed. These included an organ powered by “heated water” and a basic impulse turbine in which steam was projected against the periphery of a wheel to make it rotate, among others. However, it was not until the 15th century that effective use of steam was utilized to perform successful work in commercial applications. A number of scientists began to

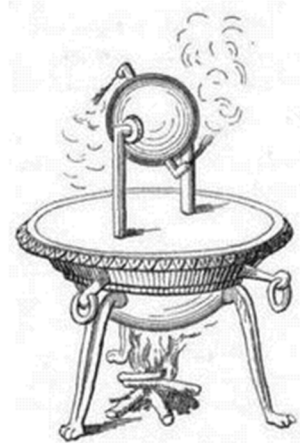


Figure 1-6. The Aeolipile (Melson, 2006).

harness the force that could be developed with a vacuum formed from the condensation of steam. This concept was utilized to successfully pump water out of wells and mines. The first steam engine to accomplish this useful purpose was Thomas Savery's pumping engine, a reciprocating engine.

An image of Savery's engine is shown in Figure 1-7. In Savery's engine (1698),

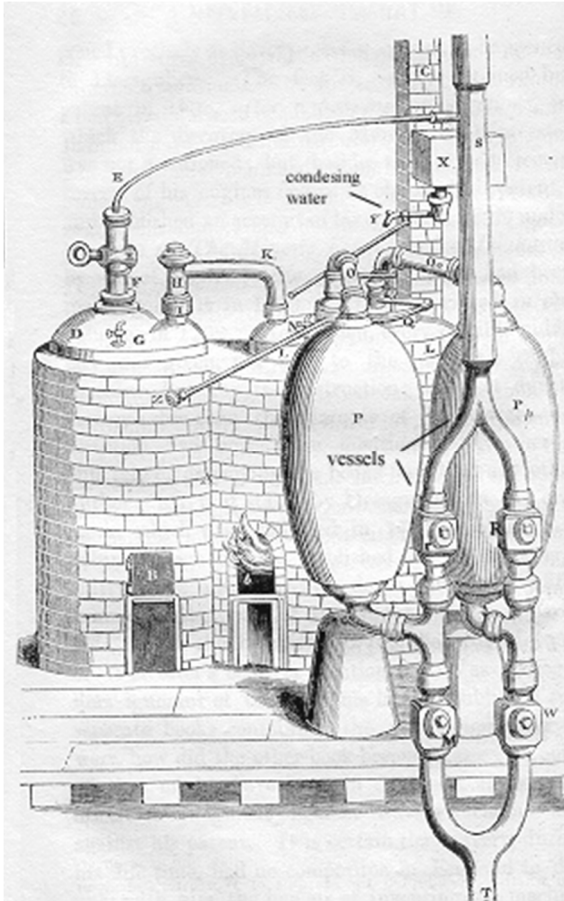


Figure 1-7. Steam pump by Thomas Savery, 1698.
(Lira, 2013)

steam pressure acted on the surface of water inside a chamber. The steam pressure forced the water out of the chamber through a discharge line. When the chamber was evacuated, the steam supply was shut off. Next, cold water passing over the chamber caused the steam to condense, forming a vacuum in the chamber. This vacuum pulled water into the chamber. Once the chamber was full of water, the steam supply was opened and the process started again. While effective, Savery's engine was very inefficient due to heat transfer losses between the hot steam and the cold water, among other things. Additionally, relief valves

were not utilized and explosions were not uncommon (Faires, 1962).

Denis Papin, a contemporary of Savery, Boyle, Huygens, Leibnitz, and other scientists, is credited with the invention of the safety relief valve. In 1707, Papin also outlined an improvement to the Savery engine in which a piston would separate the

steam from the water, reducing direct contact heat transfer losses, among other things. However, the Papin engine was never built.

Thomas Newcomen, another contemporary of Savery, did build a steam engine with a piston between 1707 and 1712. In the Newcomen engine, steam was supplied to one side of a piston, forcing the piston up. Once the piston reached the end of the stroke, cold water was injected into the cylinder, thus condensing the steam and causing a vacuum. This vacuum pulled the piston back to the original position and the process could begin again. The piston was connected to one end of a rotating beam, while the other end was connected to the piston of a water pump. With this arrangement, the ratio of piston areas could be designed such to develop any desired water pressure, regardless of steam pressure. Additionally, as the engine developed, the Newcomen engine was fitted with strings or linkages to automatically operate the steam and water valves. This is the first known instance of automatically controlled valves (Faires, 1962).

The Newcomen engine was a commercial success. About 60 years later, in 1765, a man named James Watt came along and brought along substantial improvements to the engine. Features he developed include: separate condenser (such that the cylinder remain hot to prevent heat losses), the double-acting engine, the expansion of steam (to get work from its internal energy), the flyball governor and the conversion of reciprocating motion into rotary motion (utilizing the crank-slider mechanism). The action of the steam cylinder was similar to the most modern of steam engines but continued to utilize a rotating beam like the Newcomen engine (Faires, 1962). In 1781, Jonathan Hornblower patented a two-cylinder “compound” in which the exhaust steam from the first cylinder is delivered to the second cylinder such that much of the

remaining energy in the exhaust steam from each cylinder could be recovered. This new engine offered significant efficiency improvements for use of high pressure steam due to a reduction in condensation. By 1804, Richard Trevithick had built the first steam-driven locomotive (Hossli, 1969).

The next major improvement to the steam engine was in 1849 when George Henry Corliss developed the Corliss valves and valve gear, which provided quick closing of the intake valves, thus reducing the throttling and friction of flow during the closing periods. About three decades later, in 1881, Alexander C. Kirk designed the first practical triple expansion steam engine (triple-cylinder compound engine). This triple-expansion style engine was widely used well into the 20th century. The latest significant advancement to commercial steam engines, designed to further reduce condensation losses, was made by Johannes Stumpf in 1908. The engine was called the unaflow, to distinguish from the counterflow engine. In the Watt and Corliss counterflow engines, steam was admitted and exhausted in the same end of the cylinder. Stumpf realized that the hot steam was then entering in the same end as the cooler exhausting steam, and thus causing condensation when the hot steam contacted the cooled engine components. Stumpf created an engine in which steam enters at each end of the cylinder (double-acting), but exhausts in the middle of the cylinder, thus improving the maintaining of hot cylinder components where the new steam is admitted. Additionally, this caused the percent of stroke in which the cylinder was compressing to increase, resulting in elevated temperatures and pre-heating of the hot end of the cylinder. These components reduced condensation issues and improved the overall efficiency of the steam engine (Faires, 1962).

During the later stages of steam engine development, a new steam generation product was being developed – the steam turbine. As mentioned, rudimentary impulse steam turbines were used to produce effective work as early as the 15th century. These rudimentary impulse steam turbines advanced over the next 200 years, seeing the pinnacle of the impulse turbine developed by Carl Gustav Patrik de Laval in 1878. This simple impulse turbine is used to separate cream from milk, using a convergent-divergent (supersonic) nozzle which ran the turbine to a maximum speed of 100,000 rpm. However, the first modern-day expansion steam turbine design was developed by Charles A. Parsons in 1884. This turbine design is a multi-stage axial flow reaction turbine using brass blades on a steel wheel (Nag, 2002). It was this Parson design that was licensed to George Westinghouse in order to scale the design into turbines 10,000 times as large as Parson's original 7.5 kilowatt unit (Parsons, 1900). Turbines with these designs were not only employed to operate electric generators in thermal and nuclear power plants to produce electricity, but they were also used to propel large ships and to drive power absorbing machines like compressors, blowers, fans and pumps (Nag, 2002). However, early turbines were not efficient gained a reputation for guzzling steam. By 1888, the steam consumption for a 32 kilowatt generator, which originally had been 200 lbs. per kilowatt-hours, was reduced to 34.5 lbs. per kilowatt-hours, when working non-condensing with steam at 90 psig (Hills, 1989).

The development of the Parson's steam turbine was the beginning of the end for commercial steam engine use. Steam turbines, utilizing the new Parson's design, could produce power greater than that of the most advanced steam engines while being smaller, lighter and less expensive. The turbines operated at much greater speeds than

did the engines, which was one of the chief advantages of the turbines. To produce a given amount of power, the required torque is inversely proportional to speed. Therefore, these turbines could be much smaller and lighter than equivalent power steam engines due to the fact that they were required to withstand and output significantly less torque. As an example, in the mid-1920's, two separate cotton mills were constructed. In one, a compound steam engine was installed; in the other, a Parson's turbine. The engine weighed 90 tons, operated at 66 rpm, and produced 2,500 horsepower with the output pulley 24 feet in diameter. The engine had 32 cylinders with a 70 inch bore and a stroke of 5 feet. The turbine produced 2,600 horsepower, operated at 5,000 rpm and was coupled to a gearbox to slow output speed to 333 rpm. The output pulley out of the gearbox was 5 feet in diameter. The total length of the turbine-gearbox combination was 16 feet, less than width of 3 of the engine's 32 cylinders (Hills, 1989).

Since the original scaling of the Parson's design, steam turbines have undergone few substantial changes. Primary improvements in the last 100 years include better materials and slight modifications to blade geometry design. Other turbine design efforts have been focused on expanding the range of applications for steam turbines to include efficient performance in low power applications.

Developments since the steam turbine primarily include Organic Rankine Cycles (ORCs). An ORC is a version of the thermodynamic Rankine cycle in which some organic working fluid is evaporated then expanded (producing power), then condensed and then returned to the evaporator to start again. The input heat can be in the form of steam. ORCs have gained popularity due to the fact that they can utilize low grade (low

temperature) heat. The first ORC was developed and exhibited in 1961 by solar engineers Harry Zvi Tabor and Lucien Bronicki. Improvements to the ORC since initial development include primarily changes in working fluid and more efficient equipment for expanding the fluid.

iii. Overview of Steam Power Generation Applications

In general, there are two primary categories of steam power generation: backpressure and condensing. Backpressure applications are applications in which inlet steam enters the steam power generation device at an elevated pressure and leaves the device a lower pressure to be used by some steam process while electrical power is produced. This is the type of application typically seen in CHP systems. Condensing applications are applications in which steam enters the steam power generation device at an elevated pressure and the exhausted steam is condensed into hot water to be used by a hot water system and/or returned to the feedwater system for the boiler. This is the type of application typically seen in waste heat/waste steam applications as well as renewable energy production applications. Additionally, in each of these applications, the inlet steam conditions can vary substantially. In order to improve efficiency, the inlet steam is often times superheated as compared to incoming as a saturated vapor such to reduce condensate losses. While this improves system performance, in many of the aforementioned categories, superheated inlet steam is not economically feasible.

iv. Current Technologies

The development of steam power generation throughout history has included many substantial improvements. However, there are areas of potential improvement even still today. Steam turbines dominate the steam power generation market today, but they are not without flaws. Steam turbines perform well in large, electric utility applications with

electrical generating efficiency of 36 percent¹ HHV². However, steam turbines in smaller backpressure applications, which make electricity as a byproduct of delivering steam to industrial processes or district heating systems for colleges, industrial parks and building complexes, operate at lower efficiency levels, often times less than 10 percent HHV (Energy and Environmental Analysis (an ICF International Company), 2008). Most steam turbine manufacturers focus on large turbines and do not offer turbines for steam flow less than 10,000 lbs per hour.

The product that is the focus of this text is designed to primarily operate in applications less than 500 kilowatts and 20,000 lbs per hour. Therefore, the focus of the discussion of current technologies will focus on these applications. In past years, steam turbine generators under 500 kilowatt commonly have turbine efficiencies in the 35-50% range (Valencia, 2006) and are expensive due to similar cost and construction of larger steam turbines. This low efficiency has resulted in low power production for a given load which has resulted in a high installed cost per kilowatt produced when combined with poor efficiency. Therefore, until recently, there have been very few products for this market. The following products aim to tackle this market.

Microsteam® Turbine Generator – Carrier Corporation

¹ This efficiency includes all system efficiencies including: boiler efficiency, piping efficiency, turbine efficiency, gearbox efficiency, generator efficiency, etc.

² All turbine and engine manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. However, the usable energy content of fuels is typically measured on a higher heating value basis (HHV). In addition, electric utilities measure power plant heat rates in terms of HHV. For natural gas, the average heat content of natural gas is 1,030 Btu/scf on an HHV basis and 930 Btu/scf on an LHV basis – or about a 10% difference (Energy and Environmental Analysis (an ICF International Company), 2008).

Recently, a new product has been produced by Carrier Corporation to target the backpressure market of 275 kilowatt and below. This product, the Microsteam® turbine generator, is a vertically mounted small steam turbine generator that includes a gearbox, generator and all required accessories. Carrier Corporation advertises electrical power output up to 275 kilowatt with turbine efficiencies up to 75 percent (Carrier Corporation, 2014). Case studies have been performed on the Microsteam®, demonstrating maximum turbine efficiency of 74 percent (Valencia, 2006). The gearbox is recorded to operate at an efficiency of 97 percent and the generator can perform at a maximum efficiency of 95.4 percent (Valencia, 2006). Considering all of these efficiencies, the Microsteam® turbine generator can operate at a maximum efficiency of 68.5 percent efficiency¹. As with all steam turbines, the Microsteam® turbine generator efficiency is heavily dependent upon steam inlet and outlet conditions and load percentage. Minor disturbances in inlet or outlet conditions can have significant impacts on performance. Additionally, as with all steam turbines, steam quality must remain at a very high quality. Steam quality below 95 percent can damage the turbine blades (Carrier Corporation, 2011). Furthermore, the Microsteam® is relatively costly, with an installed price near \$600,000 (Carrier Corporation, 2011).



Figure 1-8. Microsteam® Turbine Generator System (Energent, 2014).

¹ Microsteam® Turbine Generator efficiency calculated based upon maximum commercial demonstrated turbine efficiency of 74 percent, multiplied by gearbox efficiency of 97 percent and by generator efficiency of 95.4 percent.

Voith Piston Expander – Voith Turbo GmbH & Co. KG.

Voith Turbo has recently produced a product called the piston expander as part of their SteamTrac and SteamDrive Systems. This product is a piston engine in which steam is expanded to produce from 15 kilowatt up to 160 kilowatt of power in a condensing application. The piston expander requires superheated steam at a pressure between 145 psia and 870 psia. The steam must be provided at a temperature between 572 °F and 752 °F (Voith Turbo, 2014), which is not common in typical industrial/commercial systems. The SteamTrac and SteamDrive Systems are primarily marketed as a heat recovery system for internal combustion engines on mobile motorized equipment.



Figure 1-10. S2E Turbine Generator (Technopa Engineering, 2014)

S2E Steam To Energy Transformer – Technopa GmbH

Technopa has recently produced a product called the Steam to Energy Transformer (S2E). This product is a backpressure steam turbine generator that can provide outlet pressures as low as 16 psia. The S2E can operate on saturated or superheated inlet steam between 58 psia and 508 psia (Technopa Engineering, 2014). The S2E turbine utilizes bristles instead of blades, thus allowing lower qualities of input steam. The total efficiency of the S2E module depends on input steam conditions but varies



Figure 1-9. Voith Piston Expander (Clean Thermodynamic Energy Conversion, 2014)

between 55 percent, at the minimum allowable input steam pressures, and 60 percent at the maximum allowable steam conditions (Green Energy Engineering, 2014).

Green Turbine™ - Green Turbine Inc.

Green Turbine, Inc. has produced a new product called Green Turbine™. The Green Turbine™ is an ultra-small (slightly larger than the size of a football) condensing turbine and can produce power up to 15 kilowatts. The turbine requires superheated inlet steam and produces outlet steam at 1.5 psia and 104 °F (Green Turbine, Inc, 2014). The efficiency of the Green Turbine™ is only 13.6 percent¹ and power production is not sufficient for most industrial/commercial applications.



Figure 1-11. Green Turbine (Green Turbine, Inc, 2014).

SHAPE Engine – Exoes

Exoes has recently developed a piston engine for use with steam and organic fluids. The product is primarily designed for small scale solar power plants and exhaust waste heat recovery on mobile equipment. The engine is designed for applications under 20kW (Daccord, 2011). The engine is a self-lubricated, oil-free piston expansion engine. The engine is designed for relatively high pressure input of organic fluids (>500 psia.) and power production is not sufficient for most industrial/commercial applications.



Figure 1-12. SHAPE Engine (Daccord, 2011).

Table 1-1 shows a summary of all mentioned current technologies in the steam power generation market less than 500 kilowatts.

¹ Calculated as power output/enthalpy change of steam through turbine. Inlet steam condition of advertised maximum condition of 12 bar abs (175 psia) and 428 °F.

Table 1-1. Summary of all mentioned current technologies.

Product	Type	Application	Inlet Steam	Efficiency	Power Output
Microsteam®	Turbine	Backpressure	115-265 psia	68.5%	<275 kWe
Piston Expander	Piston Engine	Condensing	Superheated 145-870 psia	N/A	<160 kWe
S2E Transformer	Turbine	Backpressure	58-508 psia	55-60%	<600 kWe
Green Turbine™	Turbine	Condensing	Superheated 75 psia	13.6%	<15 kWe
SHAPE Engine	Piston Engine	Condensing	N/A	N/A	<20 kWe

v. Limitations of Current Technology

While each of the mentioned products may operate in the market of interest, none of which can meet the needs of all of the previously discussed applications. A brief examination of each aforementioned application will be given and the limitations of the current technology in that application.

a. PRV Parallel –

Requirements:

PRV parallel applications require a cost-effective backpressure steam power generation device with high efficiency that can handle moderate pressure saturated vapor steam input.

Limitations of each current technology:

Microsteam® - High Cost, moderate efficiency. Only cost effective in markets with extreme energy costs such as Manhattan, New York.

Piston Expander – Requires superheated steam input and extreme pressure. Condensing only.

S2E Transformer - High Cost, moderate efficiency. Only cost effective in markets with extreme energy costs such as Manhattan, New York.

Green Turbine – Insufficient power output, insufficient efficiency and condensing only.

SHAPE Engine – Insufficient power output.

b. Commercial Co-generation (Combined Heat and Power)

Requirements:

CHP system market applications require a cost-effective backpressure steam power generation device with high efficiency that can handle moderate pressure saturated vapor steam input.

Limitations of each current technology:

Microsteam® - High Cost and moderate efficiency. Only cost effective in markets with extreme energy costs such as Manhattan, New York.

Piston Expander – Requires superheated steam input and extreme pressure. Condensing only.

S2E Transformer - High Cost, moderate efficiency. Only cost effective in markets with extreme energy costs such as Manhattan, New York.

Green Turbine – Insufficient power output, insufficient efficiency. Condensing only.

SHAPE Engine – Insufficient power output.

c. Industrial Waste Steam

Requirements:

Industrial waste steam market applications in reference to waste steam recovery and energy production require a condensing steam power generation device with low cost that can handle poor steam quality.

Limitations of each current technology:

Microsteam® - High Cost, moderate efficiency, backpressure only and sensitive to steam quality.

Piston Expander – Requires superheated steam input and extreme pressure.

S2E Transformer - High Cost, moderate efficiency and backpressure only

Green Turbine – Insufficient power output, insufficient efficiency.

SHAPE Engine – Insufficient power output.

d. Industrial Waste Heat

Requirements:

Industrial waste heat market applications in reference to heat recovery and energy production require a condensing steam power generation device with low cost that can handle moderate pressures.

Limitations of each current technology:

Microsteam® - High Cost, moderate efficiency and backpressure only.

Piston Expander – Requires superheated steam input and extreme pressure.

S2E Transformer - High Cost, moderate efficiency and backpressure only

Green Turbine – Insufficient power output, insufficient efficiency.

SHAPE Engine – Insufficient power output.

e. Biogas

Requirements:

Biogas renewable energy market applications in reference to biogas steam production and power generation require a condensing steam power generation device with low cost that can handle moderate pressures.

Limitations of each current technology:

Microsteam® - High Cost, moderate efficiency and backpressure only.

Piston Expander – Requires superheated steam input and extreme pressure.

S2E Transformer - High Cost, moderate efficiency and backpressure only

Green Turbine – Insufficient power output, insufficient efficiency.

SHAPE Engine – Insufficient power output.

f. Biomass

Requirements:

Biomass renewable energy market applications in reference to biomass steam production and power generation require a condensing steam power generation device with low cost that can handle moderate pressures.

Limitations of each current technology:

Microsteam® - High Cost, moderate efficiency and backpressure only.

Piston Expander – Requires superheated steam input and extreme pressure.

S2E Transformer - High Cost, moderate efficiency and backpressure only

Green Turbine – Insufficient power output, insufficient efficiency.

SHAPE Engine – Insufficient power output.

g. Solar

Requirements:

Solar renewable energy market applications in reference to solar steam production and power generation require a condensing steam power generation device with low cost that can handle moderate pressures.

Limitations of each current technology:

Microsteam® - High Cost, moderate efficiency and backpressure only.

Piston Expander – Requires superheated steam input and extreme pressure.

S2E Transformer - High Cost, moderate efficiency and backpressure only

Green Turbine – Possible Application.

SHAPE Engine – Possible Application.

In summary, while specific applications may exist that would allow effective use of these products, the need for a cost effective, low output steam power generation device still exists. The purpose of this text is to study the theory, development, design and testing of such a device. This device is introduced in following text.

vi. Introduction to a New Technology: Practical Steam Engine (PSE)

With the understanding of the need of a product to meet the need of the aforementioned applications, a company, Practical Steam, has developed the Practical Steam Engine (PSE) technology, which is the focus of this text. The PSE is a high-speed, cost effective, reciprocating piston steam engine that is designed to work in both condensing and backpressure applications without product modification. The PSE is of similar design to reciprocating steam engines of past. However; the PSE has several distinct characteristics and improvements to provide significantly better performance than past steam engines for applications previously outlined. In a later section of the text, these improvements are presented, as is the theory, development, design and testing of the PSE. Prior to this presentation, a literature review is provided in an effort to understand similar and recent studies in the field and to better understand the ramifications of the information provided in this text.

d. OBJECTIVES OF THIS RESEARCH

The objective of this research is to determine validity and feasibility of use of the PSE for industrial/commercial steam power generation. Furthermore, this research is intended to examine the validity of the model used for performance predictions. Finally, review of case studies is designed to substantiate both the practical and economic feasibility of the product in industrial settings. Each subsequent chapter is intended to provide supporting information such to meet the specified objectives.

Chapter 2 (Literature Review) outlines previous research performed in this area. Chapter 3 (Steam Engine Operation, Design and Performance) describes the technical history of steam engine operation and identifies goals and challenges for progressing the field. Chapter 3 also discusses specific design concepts such to meet goals and overcome challenges laid out in chapter 3. Chapter 4 (Theory/Modeling) provides a technical description of the science behind the modelling of engine operation. Chapter 5 (Parametric Studies) explores modeled effects of changes in various parameters. Chapter 6 (Modeling of Experimental Application) describes the theoretical projected performance of a PSE in a specific application. Chapter 7 (Testing/Experimentation) specifies experimentation procedure of full-scale testing of the engine in an industrial setting comparable to the application analyzed in Chapter 6. Chapter 8 (Experiment Conclusions & Model Validation) outlines the results of the experimentation in reference to model projections presented in chapter 6 as well as drawing general conclusions regarding experimental results. Chapter 9 (Field Application Case Study) studies the economic and practical feasibility of the engine tested according to the procedure given in chapter 7. Chapter 10 (Conclusions) summarizes all findings from all previous chapters and frames the conclusions in reference to the objectives of the research

stated here. Chapter 10 also recommends future expansions to work detailed in this research.

2. LITERATURE REVIEW

The PSE, and study thereof, which are the focus of this paper, are significant in the development of energy efficiency in the industrial and commercial market. The product described in this text allows use of many waste energy recovery opportunities as well several renewable energy applications that have been previously difficult to grasp. This section of the text presents a literature review of current and recent research in the field pertaining to development and performance of the product of this text, a sub 500 kilowatt steam power generation devices and systems.

1. Model of a small steam engine for renewable domestic CHP (combined heat and power) system. – G. Ferrara, G. Manfrida, A. Pescioni (Ferrara, Manfrida, & Pescioni, 2013).

Summary with conclusions:

Ferrara, Manfrida and Pescioni (2013) performed computational studies of a small counterflow reciprocating steam engine for application of renewable domestic CHP systems. Their thermodynamic analysis uses real-fluid and real-cycle assumptions. It includes a heat transfer model for non-adiabatic compression and expansion, losses through admission/discharge valves and the effects of clearance volume. These results are also compared with those of a dynamic model, working under perfect-gas assumptions. Their analysis found that a non-stationary model of the engine

provided similar results confirming that un-steady flow and wave propagation effects do not alter substantially the results of the Quasi-Stationary model. Their theoretical analysis showed the engine to perform with sufficient efficiency and power output to result in effective use in a domestic CHP system.

Comments:

Ferrara, Manfrida and Pescioni (2013) performed studies that lead to very informative conclusions and results; specifically that wave propagation does not play a major role in engine performance at these speeds and can likely be neglected. Additionally, the studies show the validity of such a system and provide much encouragement for a system to be developed and tested and compared to such results. However, this model is purely thermodynamic and does not consider mechanical efficiencies and other factors that must be considered. Furthermore, engine performance was not modeled at a variety of inlet and outlet conditions which are critical parameters to study. Finally, a few factors that were neglected in the thermodynamic model were differences between steam flow and ideal gas flow and heat transfer coefficient as a function of steam quality. These parameters will be implemented in the model given in this text.

2. Design and performance evaluation of an innovative small scale combine cycle cogeneration system – M. Badami, M. Mura, P. Campanile, F. Anzioso (Badami, Mura, Campanile, & Anzioso, 2008).

Summary with conclusions:

Badami, Mura, Campanile, Anzioso (2008) designed and performed experimental studies of the performance of a bottoming cycle of a natural gas internal combustion engine (ICE) CHP system. The bottoming cycle utilized a small three cylinder counterflow steam engine to serve as the heat engine producing power from the waste heat of the natural gas engine. The results of the test showed an increase of more than 10 percent of the maximum efficiency of the single ICE CHP system without the bottoming cycle. The steam engine produced nearly 16% of the total power of the system. The results of this study are significant in that the validity of such a system is confirmed.

Comments:

Badami, Mura, Campanile, Anzioso (2008) performed studies that yielded significant results. This application is very similar to potential applications of the PSE. Furthermore, the steam engine control method in this study is a variable speed method instead of variable cutoff. For a steam engine direct coupled to a generator, variable speed control is not a very practical method as the generator must operate at an approximately constant speed. The PSE is designed to utilize cutoff control such that constant speed can be achieved.

3. Preliminary design and controlling strategies of a small-scale wood waste Rankine Cycle (RC) with a reciprocating steam engine (SE) – M. Badami, M. Mura (Badami & Mura, Preliminary design and controlling strategies of a small-scale wood waste Rankine Cycle (RC) with a reciprocating steam engine (SE), 2009).

Summary with conclusions:

Badami and Mura (2009) performed a study of a wood-fueled biomass steam CHP system plant utilizing a reciprocating counterflow steam engine. The total efficiency of the CHP system was found to be 78%. The analysis has shown that the steam engine could be an interesting technology for small-scale energy plants and offers a good solution when fed with wood waste. Another focus of the paper was to explore control strategies. The strategies studied were speed control, cutoff control and governing control. The cutoff control was found to be the most efficient method of control but quite difficult to mechanically achieve. The speed control was found to be easily obtained using an electronic power system but exhibited lower efficiency than the cutoff control. Lastly, the governing control was found to be the most simple and easiest to accomplish but the least efficient.

Comments:

Badami and Mura (2009) showed a steam engine to be an applicable product for CHP systems with wood-fueled biomass steam generation. This study not only shows that the use of a steam engine is applicable for a wood-fueled biomass plant, but also would imply that a steam engine could be a good application for any low flow and low pressure steam power generation plant. Additionally, the study showed cutoff control to be the most efficient and effective method of control for the steam engine. This agrees with conventional thinking and is the method of control that the PSE utilizes.

4. Thermo-economic analysis of a micro-cogeneration system based on a rotary steam engine – Kari Alanne, Kari Saari, Maunu, Juha Jokisalo, Andrew R. Martin (Alanne, Saari, Kuosa, Jokisalo, & Martin, 2012).

Summary with conclusions:

Alanne, Saari, Kuosa, Jokisalo and Martin (2012) performed an analysis on a micro-cogeneration steam plant for residential application. In this application, it was found that a 17 kilowatt household pellet-fueled boiler can obtain an electrical efficiency of approximately nine percent and a thermal efficiency of 77 percent¹ and overall cogeneration efficiency of 86 percent¹. This design is based upon 150 °C (302 °F), which allows the design pressure to be kept acceptable in terms of safety and cost-effectiveness. The amount of electricity produced would meet approximately 30 percent of a household's electrical annual demand and 40 percent of produced power was exported. The economic viability analysis suggests that the incremental price of integrated boilers no more than € 1500 (\$2500) above standard boilers is acceptable and results in payback periods under five years.

Comments:

Alanne, Saari, Kuosa, Jokisalo and Martin (2012) performed studies that consider the overall efficiency and economic impact of a residential cogeneration plant. While this application is not the primary scope of the PSE, the conclusions of this study are extremely interesting. This is just another example of the application of the PSE in a biomass application. The conclusions of a similar study with the PSE should be even more compelling due to the superior efficiency.

Each of the discussed papers provides specific insight pertaining to use of a steam engine for small-scale energy efficiency and renewable energy applications. However, none of which utilize a small-size, high speed and efficient steam engine to perform the

¹ In terms of LHV

work. Furthermore, each design utilizes poppet valves or some variety of such, resulting in significant performance limitations. For these reasons, significant information should be ascertained from the research, both theoretical and experimental, presented in this text.

3. STEAM ENGINE OPERATION, DESIGN AND PERFORMANCE

a. INTRODUCTION

Understanding the basics behind steam engine operation, design and performance is essential in allowing effective and efficient development of a new, more advance form of steam engine. For this reason, an overview of these parameters is given in this portion of text.

b. OPERATION

i. General Overview of Operation

Steam engine operation is based upon the Rankine cycle utilizing boundary work. The process begins with opening of admission valve before top dead center (TDC). The cylinder begins to fill with steam up to TDC and continues until the admission valve closes. The percentage of the stroke in which the piston has traveled at point of admission valve closing is defined as cutoff and described in the following equation:

$$Cutoff = \frac{\text{Distance Piston Moves from TDC to Point of Cutoff}}{\text{Length of Stroke}} \quad (3.1)$$

From cutoff, the steam expands until the exhaust valves are opened. At this point, steam continues to expand as steam begins to flow out of the cylinder down while the piston travels to and past to bottom dead center (BDC). The piston travels back up the cylinder, while the exhaust valve closes, and the process begins again. The work output

of the system is in form of boundary work and is equal to the enclosed area of the PV diagram and is described by the following thermodynamic equation:

$$W = \int P dV \quad (3.2)$$

Where:

W is work

P is pressure

V is volume

An ideal engine Rankine cycle would include infinitely large valves and a lack of fluid friction and throttling losses. Also, in the ideal engine Rankine cycle, expansion and compression from would be isentropic and therefore adiabatic.

ii. Losses in the Steam Engine

There are a number of losses in any actual engine. Many of the losses can be reduced or eliminated and many cannot. For this reason, each of the losses is described to better understand the economic and practical limitations of trying to reduce them. Primary sources of losses in steam engines provided here (Faires, 1962):

- Mechanical Friction:

As with any moving machinery, the steam engine is inclusive of mechanical friction. However, this loss, defined in mechanical efficiency, is a minor loss, especially in well-built modern engines and can be kept low by a proper design of bearings and of the lubrication system.

- Heat Loss of Conduction and Radiation:

In most steam engines, heat losses from conduction and radiation heat transfer are on a lower order of magnitude than that of the mechanical friction and are kept at a minimum by proper insulation.

- Steam Leakage:

Steam leakage around valves and piston rings is a loss that is relatively minor and can be reduced by better fitting of parts and better maintenance.

- Incomplete Expansion:

Losses from a lack of complete expansion are, again, relatively minor. These losses are not easily eliminated as increases in expansion would reduce the power density of an engine and increase condensation heat loss.

- Fluid Friction and Throttling Losses:

Losses due to fluid friction and throttling are relatively large. As described previously, the shape of the lines are affected by throttling losses and therefore reduce the enclosed area of the PV diagram and such, the work output. These losses can be reduced by increasing valve and port sizes and increasing the rate in which the valves open and close.

- Initial Condensation.

Losses due to condensation are the single largest loss in the steam engine. The cause of the condensation is heat transfer. Any condensation formation due to heat transfer is an energy loss and therefore an efficiency loss. The primary mechanism causing the heat transfer is expansion. This can be explained by considering a T-S diagram. If the engine receives saturated steam, the temperature range in the cylinder is from the saturation temperature at the inlet pressure, t_1 , to the saturation

temperature at the bottom dead center (BDC) pressure, t_2 . The greater the expansion ratio, the greater the temperature range. Now consider a mean wall temperature that is the average of the two temperature, t_m . Of course this mean temperature will not be one single steady temperature, because the walls will alternately be heated and cooled as the adjacent steam is hot or cold; but for the purposes of explanation, the mean temperature is assumed to be steady, and in any case, it varies through a relatively small range. Therefore, when hot steam enters at a high temperature, t_1 , heat is transferred to the cylinder wall at the lower temperature, t_m . This heat transfer causes condensation. The greater the expansion ratio, the greater the temperature range resulting in greater difference between t_1 and t_m , and therefore, the greater the heat transfer and condensation formation. The expansion ratio cannot be reduced without suffering efficiency losses as a high level of expansion is necessary to extract most of the internal energy of the steam. In an effort to maintain a high level of expansion, while limiting condensation, the compound engine was developed. The theory is that if a specific expansion ratio is desired, say 16, two cylinders could be utilized in which the first cylinder expanded by the square root of the desired expansion ratio, four ($\sqrt{16} = 4$), and the exhaust be delivered to a second cylinder which has expansion ratio of the same value, four. With this design, each individual cylinder would have a much smaller temperature range and therefore a much lesser amount of heat transfer and condensation formation. These compound designs resulted in significantly improved efficiencies. The most modern commercial steam engines produced were triple expansion in

which the same compound concept was utilized across three cylinders instead of two in an effort to further reduce the temperature range.

Another cause of the heat transfer is relatively cool exhaust steam passing over the hot walls of the cylinder and cooling the walls. With the walls cooled, significant heat transfer occurred between the entering hot steam and the cooler cylinder walls, resulting in condensation. To solve this problem, the uniflow steam engine was developed. The uniflow, initially unaflow, steam engine was typically a double acting cylinder in which the steam entered in the ends of the cylinder and the steam exhausted through ports in the cylinder walls in the middle of the cylinder. Therefore, the cool, expanded steam did not pass the hot end of the cylinder when exhausting. Additionally, the remaining steam in the cylinder was compressed and preheated the hot end of the cylinder. These two physical mechanisms resulted in a lesser temperature difference between the inlet steam and the cylinder walls, manifesting in a reduction of condensation formation.

An additional method of reducing in-cylinder heat transfer, and thus condensation, is steam jacketing in which “live” steam was supplied to the cavities around the cylinder walls. With this design, the cylinder walls were held at a high temperature and condensation was primarily limited to that in the steam jackets. However, this method primarily improved efficiency due to the absence of reevaporation of the condensation and to the lower surface conductance of the dry cylinder walls.

The simplest method of reducing condensation is by using superheated steam. This allowed heat transfer to the cylinder walls without condensation formation.

iii. Operational Characteristics

In addition to tackling the losses found in steam engines, when considering a new steam engine, it is important to recall what proved to be the downfall of steam engines – steam turbines. But more than understanding what, we must know why. As described in chapter 1, the ultimate cause of steam turbine success – and thus steam engine failure – was the relative size and cost of an equivalent power turbine. The key to these advantages was the speed of the turbine. The high operating speed, in comparison to an engine, allowed substantially smaller units and thus lesser costs to match power outputs of steam engines.

However, despite the cost and size disadvantages, steam engines often maintained better turndown characteristics and controllability than did turbines. Furthermore, steam turbines often times must be designed specifically for a given steam pressure and/or steam flow. Variations in pressure and/or flow could substantially impact turbine efficiency. For these reasons, if a steam engine could be produced to take advantage of the high speed operation of a turbine, while maintaining turndown characteristics, controllability, efficiency and variability in application, it seems there would be great success in the industrial marketplace for such a product to take advantage of steam power generation.

With this in mind, it is expected that a proper steam engine can be developed and efficiency can be maximized by accomplishing the goals of reducing the presented losses while overcoming the challenges of operating at high speed in a compact package without sacrificing turndown characteristics or controllability.

c. DESIGN AND DEVELOPMENT

d. INTRODUCTION

The PSE is intended for use of improved energy efficiency and renewable energy production in practical industrial and commercial applications. Therefore, the product must meet specific operational requirements outlined in the previous chapter such as (1) a high degree of functional control, (2) be able to handle a wide range of inlet and outlet steam conditions in both condensing and backpressure applications, (3) operate at synchronous speed of 1800 rpm for direct connection to generator and (4) perform with a high level of efficiency while (5) being a cost effective product and (6) maintaining small size and simplicity of installation and use. In order to meet these operational requirements, a proper design must be developed. However, prior to development of an appropriate design, the theory of the operation of the engine must be well understood. With that in mind, a summary of the operation of the PSE is presented in this portion of the text.

e. PSE CONCEPT

i. Prior Art

Before a specific model of the PSE can be developed, a conceptual understanding must be realized, resulting in a physical description of the product. To do this, the history, development and improvement of the steam engine is examined. A brief history of the steam engine is given in Chapter 1. As the steam engine entered the 20th century, designs had evolved through more than 300 years of improvement and advancement. The latest commercial, and most efficient, steam engines produced had

two specific design qualities that can be learned from, (1) compound design (multiple expansions) and (2) uniflow design. These two design qualities yielded high efficiencies in optimal operating conditions. These two designs were utilized for specific and similar reasons, the chief efficiency loss with steam engines: steam condensation (Faires, 1962).

The compound and uniflow engine designs resulted in improved thermal efficiencies however, these designs did not come without a price. For the compound design, the multiple cylinders required for expansion from the inlet pressure to the outlet pressure results in increased costs and size. For the uniflow design, due to (1) the lack of expansion after the cylinder ports are exposed and (2) power required compressing the steam during the compression stroke results in a decreased power density. Therefore, to produce the same power as a similar counterflow engine, a uniflow engine must be larger in size, resulting in additional costs. Furthermore, due to the compression stroke, the outlet pressure must be extremely low to prevent over-compression (compression to a pressure higher than the admission pressure which could result in negative power output). Therefore, uniflow engines are not optimal for high backpressure applications. However, as a solution to this problem, exhaust valves can be placed into the hot end of the cylinder to prevent over-compression. This configuration is called semi-uniflow since it is partially uniflow and partially counterflow.

ii. Design Selection Based on Differentiating Conditions from Prior Art

In addition to these fundamental understandings of the most advanced commercial steam engines, differentiating factors from the prior art must be understood as well. There are two primary differences between the design requirements of the current art

and those of the prior art. First, the current art is required to operate in both backpressure and condensing applications without design changes. Therefore, a uniflow design is not practical, but rather a counterflow or semi-uniflow design should be chosen. A semi-uniflow offers many the advantages of a uniflow with the flexibility of a counterflow but presents significant design challenges including selection/design/construction of cylinder block, robust piston rings to prevent water/steam in the crankcase and oil in the exhaust steam, amongst others. For those reasons, a counterflow configuration is utilized. The second fundamental difference is the current art is to operate at a speed of 1800 rpm, a multiple of the operating speeds of the prior art engines. In the time of the prior art, an engine operating at several hundred rpm was considered a high speed steam engine as some of the most modern steam engines operated at a maximum of 400 rpm (Faires, 1962). This speed variation has several impacts on the design, but primarily one impact on the conceptual design selection: heat transfer. The primary mechanism of heat transfer between the steam and the walls is convection heat transfer. The mechanism for this heat transfer is given in the equation below:

$$\frac{dQ}{dt} = h * A * (T(t) - T_s) \quad (3.3)$$

Where:

Q is thermal energy [Btu]

h is the convective heat transfer coefficient $\left[\frac{\text{Btu}}{\text{ft}^2 * \text{s}} \right]$

A is the surface area of heat being transferred $[\text{ft}^2]$

$T(t)$ is the temperature of the steam at the given point in time $[^\circ\text{F}]$

T_s is the wall surface temperature [$^{\circ}\text{F}$]

t is time [s]

Further, the time required for a single cycle of the engine is a function of engine speed and given below:

$$t_{cycle} = \frac{60}{\omega} \quad (3.4)$$

where:

ω is engine speed [rpm]

Therefore, heat transfer for a cycle of the engine is equal to:

$$Q_{cycle} = \sum_{n=1}^{cycle} Q_n = \left(\sum_{n=1}^{cycle} h_n * A_n * (T(t)_n - T_{wall_n}) \frac{t_{cycle}}{n} \right) \quad (3.5)$$

or:

$$Q_{cycle} = \sum_{n=1}^{cycle} Q_n = \left(\sum_{n=1}^{cycle} h_n * A_n * (T(t)_n - T_{wall_n}) \frac{60}{\omega n} \right) \quad (3.6)$$

Therefore, the heat transfer per cycle has an inverse relationship to engine speed, given a constant heat transfer coefficient. While the heat transfer coefficient would increase with engine speed, it would increase at a rate less than direction proportional relationship with speed. The reduced relationship with speed is for two reasons: (1) for basic heat transfer mechanism, while the convective coefficient is a function of Reynolds number, and thus velocity, it is not a direct relationship and (2) for condensation and evaporation, the convective coefficient is not a function of velocity. As a result, the heat transfer per cycle is expected to be significantly less than that of a

similar sized steam engine of most recently produced commercial engine designs, when operating in similar inlet and outlet conditions and similar work per cycle but at lesser speeds.

It is also noteworthy that the current art is expected to work primarily with saturated vapor inlet steam compared to superheated steam on which historic commercial engines typically operated. As a result, advantages offered by use of a compound engine are likely to be offset by the added complication partially due to the fact that the temperature range expected in the current art is much less than the range experienced in prior art. Therefore, the reduction in heat transfer by utilization of a compound engine is unlikely to out-weigh the disadvantages of substantially increased size, cost and complication. Furthermore, with this in mind as well as the fact that the engine must operate both as a backpressure and condensing engine, the conceptual design of the current art is a counterflow steam engine in which admission valves, along with exhaust valves, are placed in the cylinder head. Additionally, it is expected that steam jacketing will offer little advantage at the elevated operating speeds of the current art but present technical difficulties in constructing steam jackets to withstand the pressure. Therefore, steam jacketing will not be utilized.

Furthermore, and possibly most important, is a need for special attention to be given to valve design. In order to operate efficiently at high speed requires quick opening and closing and large valve area in order to accomplish the high rate of gas exchange. For this, a proprietary valve system was developed.

iii. Component Selection

Due to the fact that the current are must be cost-effective, it is important to construct the engine from the most common and inexpensive components available. The main components to be selected/designed are the cylinder block and crankshaft as these would likely be extremely expensive to construct from scratch. While constructing these components from scratch is certainly possible, it would likely be much more cost effective to retrofit available blocks, if possible. The counterflow design lends itself to use a most production cylinder blocks. Also, all steam engines must operate as a two-stroke engine since the exhaust/compression strokes are one in the same as are the intake/power strokes. Therefore, an engine with a crankshaft designed for two stroke operation is desired. However, very few production engines in the desired size range were manufactured as a two stroke. Obviously, the crankshaft must be of appropriate size, bore spacing, stoke, etc. for use with the selected block. Essentially, a block-crankshaft combination must be found that are a pair. The engine should be readily available and of significant displacement (> 7 liters) such to facilitate desired power levels. Furthermore, the components must be capable of operating at the power and speed levels required for the current art. A v-configuration engine is preferred due to reduced valve length as a result of shorter block for given displacement. After an extensive search, several engines met the requirements, but the Big Block Chevrolet and Small Block Chevrolet were short-listed.

Both the Small Block Chevrolet (SBC) and the Big Block Chevrolet (BBC) are some of the most produced engines of all time, resulting in tremendous part availability. In fact, the 100 Millionth SBC was produced in 2011. The BBC, designed to meet greater power demands than the SBC, was also produced in great quantities. This engine was

used in many General Motors vehicles from 1958 all the way into the 21st century. Additionally, there is a tremendous aftermarket part supply such to support high performance needs and custom applications of the BBC. The BBC is primarily used for heavy duty trucks and high performance vehicles, resulting in a rugged design, likely to withstand the demands of the PSE. For these reasons, along with greater displacement than the SBC, the BBC was chosen as the base platform for the PSE.

The BBC was rated for speeds to approximately 6000 rpm and operated with an approximate mean effective pressure (MEP) of 150 psi in stock applications and more than 250 psi in high performance applications. These speed and MEP ratings fit well with the requirements of the steam engine: 1800 rpm and 200 psi respectively.

iv. Big Block Chevrolet Modifications

While the BBC series engine is a great fit to be modified for steam engine use, some modification is required as may be imagined; it was designed as a gasoline-fueled internal combustion engine after all. The most significant of modifications is the cylinder heads and valvetrain. For a high speed steam engine, the ability to effectively get steam into and out of the cylinder is key for efficient operation. To accomplish this, a valvetrain must be designed that can accommodate large valve area, operate at a high speed and open/close quickly. Equally important for efficient operation is the ability to control the cutoff of the engine for speed/power control. While conventional throttling valves may be effective, they significantly impair the efficiency of the engine. The combination of these requirements, large valve area, high speed operation, open/close quickly and throttle control, are the guidelines which are used in order develop a proper valvetrain and corresponding cylinder head. This cylinder head/valvetrain modification and

construction constitutes the majority of the engine modification for steam engine use. The new head/valvetrain design is proprietary in nature (patent-pending) and will not be discussed in great detail. The new head/valvetrain design does not utilize a camshaft but rather crankshaft driven rotary valves. The use of rotary valves offers the advantages of quick opening and closing, thus reducing fluid friction and throttling losses (Faires, 1962). In the current design, a relatively simplistic, but effective, method of controlling cutoff is provided. With the implementation of the new head design, the gas exchange process is fairly straight-forward. Steam is supplied to the intake valve/port through an intake manifold. The rotary intake valve and cutoff control mechanism allows timed delivery of steam to the cylinder. As the piston travels upwards towards TDC, the exhaust valves in the cylinder head are opened to evacuate steam and the process begins again.

The only other noteworthy modification to the engine is the lubrication system. The means of engine lubrication is not specifically changed, but rather oil coalescers have been used to separate condensate out of the oil. Additionally, proprietary coatings have been used on critical wear surfaces due to compromised lubrication.

With exception to these presented modifications, the BBC engine is utilized as-is in a steam engine application of the current art.

4. THEORY/MODELING

a. INTRODUCTION

In this section of the text, a model is composed in order to predict engine performance in a variety of conditions as well as assist in the cylinder head design described in Chapter 3. The model encompasses several components of the engine operation in order to provide a comprehensive model. Those components include:

1. Engine Geometry, Kinematics and Dynamics
2. Valve and Port Geometry, Kinematics and Dynamics
3. Fluid Flow Through Valves and Ports
4. Heat Transfer
5. Thermodynamics of Engine Cycle

Analyses of each of these sections are described in detail and are used to generate the computer model.

The analysis is an iterative analysis with a numerical solution process in which the cycle is divided into a large number of steps (e.g. 0.001 degrees per step yielding 360,000 steps per cycle). Engine geometry, port geometry, flow analysis, heat transfer and thermodynamic analysis are performed at each given step to develop the conditions to calculate the next step. Once the geometries have been calculated, the solution is

performed by starting with an initial guess for TDC conditions (i.e. temperature, pressure, quality, etc.) and the cycle is calculated through the 360 degrees of rotation. Once the cycle is completed, the error between the resulting parameters at the 360th degree and the initial guess is measured. If the error is greater than an acceptable level, the guess is modified and the process is performed again. For simplicity, the initial guess of TDC conditions is equal to that of the inlet conditions. The series of calculations to be performed is as follows:

Fluid Flow through Valves and Ports

Heat Transfer

Thermodynamics

To begin the analysis, the list of inputs is defined:

- Engine Speed – Speed of operation of engine.
- Bore – Inside diameter of cylinder.
- Stroke – Total length of piston travel from TDC to BDC.
- Connecting Rod Length – Length of connecting rod from center of rod journal to center piston pin.
- Combustion Chamber Volume (terminology used from ICE) – Volume of cavity in cylinder head adjacent to cylinder.
- Head Gasket Thickness – Thickness of head gasket.
- Piston Dome Volume – Volume of dome on top of piston (negative value is dish volume).

- Piston to Deck Clearance – Distance from top of piston to top of cylinder block deck when at TDC (negative value is piston sticking out past deck)
- Valves and ports physical dimensions and locations
- Input Pressure – Inlet steam pressure.
- Exhaust Pressure – Steam pressure of exhaust reservoir.
- Superheat – Amount of superheat temperature of inlet steam (temperature in excess of saturation temperature).
- Inlet Quality – Quality of inlet steam.
- Material thermal properties and dimensions – thermal conductivity and thickness of solids through which heat is conducted.

As a note, all steam thermodynamic properties were calculated using a previously constructed MATLAB® program that includes functions to provide thermodynamic properties of water and steam according to IAPWS-IF97¹. Thermal conductivities and fluid viscosities in this program are per Revised release on the IAPS Formulation 1985 for the Thermal Conductivity of ordinary water substance, released 1998² and from the Revised Release on the IAPS Formulation 1985 for the Viscosity of Ordinary Water Substance, released in 2003³.

¹ This is the Revised Release on the IAPWS Industrial Formulation 1997 for the Thermodynamic Properties of Water and Steam provided by The International Association for the Properties of Water and Steam (IAPWS). This revision may be found on the IAPWS website: www.iapws.org. Direct access to the IAPWS-IF97 Revised Release may be found at <http://www.iapws.org/relguide/IF97-Rev.pdf>. The properties in this equation are accurate from 0-1000 Bar and from 0-2000 °C. The MATLAB® program was written by Magnus Holmgren, www.x-eng.com.

² Can be found here: http://www.15icpws.de/IAPWS_documents/Releases/thcond.pdf

³ Can be found here: <http://www.iapws.org/relguide/visc.pdf>

b. ENGINE GEOMETRY, KINEMATICS AND DYNAMICS

This section of the analysis defines the mechanical description of the action of the piston as required for calculations in following sections. In this section, the goal is to construct numerical descriptions for the piston location and movement for instances in time or crankshaft rotation, such that this information may be used in following analyses.

With these inputs defined, the analysis can begin. First, displacement per cylinder is defined:

$$V_d = \frac{Bore^2}{4} * \pi * Stroke \quad (4.1)$$

Where:

V_d is cylinder displacement.

Now piston motion is to be described. The piston location, from TDC, is defined as follows:

$$X_{piston}(\theta) = Stroke + Connecting\ Rod\ Length - \left[\frac{Stroke}{2} * \cos(\theta) + \sqrt{Connecting\ Rod\ Length^2 - \left(\frac{Stroke}{2} \right)^2 * \sin^2(\theta)} \right] \quad (4.2)$$

Where:

θ is crankshaft degrees from TDC

From this, piston velocity and acceleration can be defined as follows, respectively:

$$u_{piston}(\theta) = \frac{dX_{piston}(\theta)}{dt} \quad (4.3)$$

And

$$a_{piston}(\theta) = \frac{du_{piston}(\theta)}{dt} \quad (4.4)$$

The change in time is dependent upon the engine speed and step size of calculations. Recall the time to complete a cycle is given in equation (3.4). The time step to be used to determine velocity and acceleration is this cycle time divided by number of steps per cycle.

Additionally, the cylinder volume can be calculated from piston position as follows:

$$V(\theta) = (X_{piston}(\theta) + \text{Piston to Deck Clearance} + \text{Head Gasket Thickness}) * \frac{Bore^2}{4} * \pi + \text{Combustion Chamber Volume} \quad (4.5)$$

Where:

V is cylinder volume

Finally, compression ratio can be established as follows:

$$CR = \frac{\max(V)}{\min(V)} \quad (4.6)$$

Where:

CR is compression ratio.

c. VALVE AND PORT GEOMETRY, KINEMATICS AND DYNAMICS

This section of the analysis defines the mechanical description of the opening and closing of ports and valves required for calculations in following sections. In this section, the goal is to construct numerical descriptions for the intake and exhaust opening areas for instances in time or crankshaft rotation, such that this information may be used in following analyses. The mathematical analysis and computational components of this section are straight-forward and are not of tremendous technical interest. However, a significant amount of logic is required in order to determine when each port is open and how much. Therefore, details of the valvetrain geometry are not presented. The calculated output of this section describes a profile for intake and exhaust valve area as a function of crankshaft angle.

d. FLUID FLOW THROUGH VALVES AND PORTS

This section of the analysis defines the compressible flow analysis of steam into, and out of, the cylinder. In this section, the goal is to construct numerical descriptions for the steam flow in and out of the cylinder for instances in time or crankshaft rotation, such that this information may be used in following analyses.

For this analysis, flow into the cylinder will first be considered. The flow is assumed to be isentropic flow as the process is expected to be adiabatic due to the high mass flow rate and steady state operation.

Compressible flow is well defined for ideal gases. The relationship between the pressure ratio and flow velocity in the subsonic region is given below:

$$u = \sqrt{\gamma R T_0 \frac{2}{\gamma-1} \left[1 - \left(\frac{P}{P_0} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (4.7)$$

Where:

U is flow velocity

γ is the ratio of specific heats

R is the gas constant

P_0 is the stagnation pressure

P is the outlet pressure

However, in this analysis the fluid is steam, a non-ideal fluid. Equation (4.7) may be modified to consider non-ideal fluids, such as steam, by incorporating a compressibility factor as given below:

$$u = \sqrt{Z_0 \gamma R T_0 \frac{2}{\gamma-1} \left[1 - \left(\frac{P}{P_0} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (4.8)$$

Where:

Z_0 is compressibility factor at the stagnation conditions

The compressibility factor is a measure of deviation from ideal gas behavior and is described below:

$$Z_0 = \frac{Pv}{RT} \quad (4.9)$$

Where:

v is specific volume.

The compressibility factor is unity for ideal gasses and deviates for non-ideal gasses. Figure 4-1 illustrates the compressibility factor for 150 PSI steam at a variety of qualities.

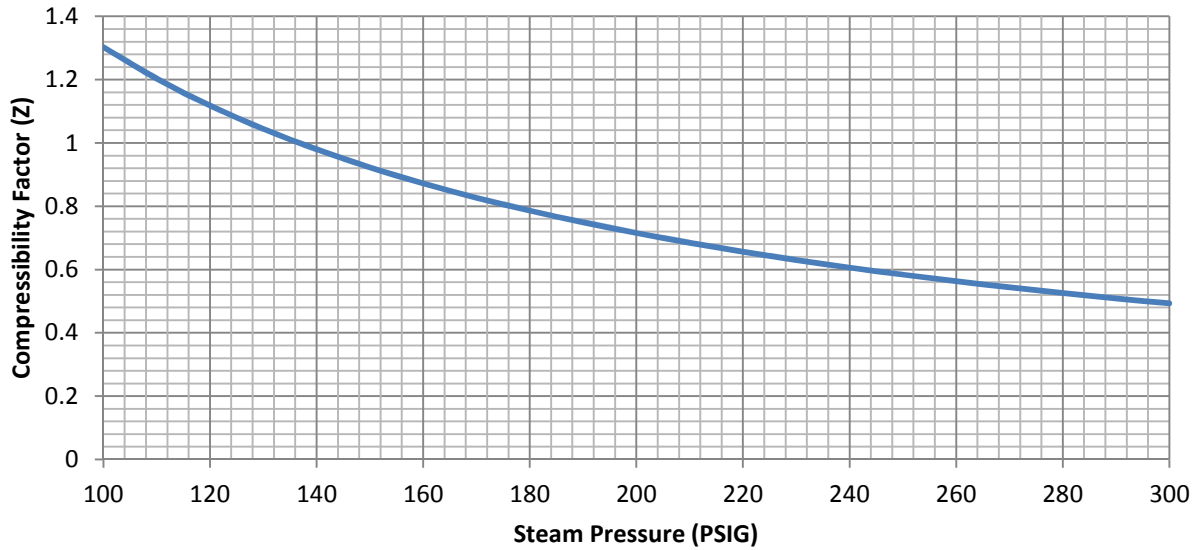


Figure 4-1. Compressibility Factor for saturated steam at a range of pressures.

Furthermore, with the understanding of the relationship between velocity, speed of sound and mach number, mach number may be derived from equation (4.8) as follows:

$$M = \sqrt{Z_0 \frac{2}{\gamma-1} \left[1 - \left(\frac{P}{P_0} \right)^{\frac{\gamma-1}{\gamma}} \right]} \quad (4.10)$$

Where:

M is mach number

From equation (4.10), the critical pressure ratio may be determined by setting mach number equal to unity. Below this pressure ratio, the flow is choked and therefore the mach number is equal to unity. This critical pressure ratio is defined as follows:

$$\left(\frac{P}{P_0}\right)_{cr} = \left(1 - \frac{\gamma-1}{2Z_0}\right)^{\frac{\gamma}{\gamma-1}} \quad (4.11)$$

With the velocity determined from equation (4.8), while checking for choked flow, mass flow rate into the cylinder is easily deduced with the following equation:

$$\dot{m} = u_2 A_{intake} \rho_2 \quad (4.12)$$

Where:

$\dot{m}$ is mass flow rate

A_{intake} is intake port area exposed as calculated in valve and port geometry section

ρ_2 is density at point two

Next, outlet flow is analyzed. The process to determine outlet flow is similar in nature to determining inlet flow as presented. Therefore, outlet flow determination will not be demonstrated. However, it is noteworthy to point out that the fluid in the cylinder will now become point one and the exhaust ports and valves will become point two.

The presented method for flow analysis is meaningful because the analysis may be adjusted for changes in flow medium by simply altering the compressibility component. However, as a point of proof of analysis, the analysis may be performed differently for analysis specifically for steam as the flow medium. This analysis is given here and compared to that of the compressible flow analysis performed above. For the steam analysis, first consider the 1st law of thermodynamics.

$$KE_2 + PE_2 + h_2 = KE_1 + PE_1 + h_1 - W + Q \quad (4.13)$$

Where:

KE is kinetic energy

PE is potential energy

h is enthalpy

W is work performed by fluid from point one to point two

Q is heat transfer to the fluid

Potential energy changes as well as work performed by the fluid and heat transfer are considered to be zero in the isentropic flow. Therefore, the equation simplifies further.

$$KE_2 + h_2 = KE_1 + h_1 \quad (4.14)$$

Kinetic energy can be defined as a function of fluid velocity.

$$KE = \frac{u^2}{2} \quad (4.15)$$

Combination and rearrangement of equations (4.14) and (4.15), along with the understanding that the initial velocity into the port is approximated to be zero (inlet conditions are stagnation conditions), the velocity in the port may be calculated as a function of the enthalpy conditions.

$$u_2 = \sqrt{(h_1 - h_2)2g_c} \quad (4.16)$$

Furthermore, the enthalpy at point one may simply be determined from the steam tables as a specific steam parameter.

$$h_1 = h(P_1 \text{ or } T_1, X_1) - \text{for saturated mixture}$$

$$h_1 = h(P_1, T_1) - \text{for superheated steam}$$

(4.17)

Additionally, the entropy may easily be determined from the steam tables by knowing enthalpy and any other condition. With the condition fully known at point one, while keeping in mind this is analysis of isentropic flow, enthalpy at point two may be easily found as a function of entropy at point one and pressure at point two.

$$h_2 = h(P_2, s_1) \quad (4.18)$$

With knowledge of enthalpy at point one and point two, equation (4.16) is used to determine flow velocity. This flow velocity is used with equation (4.12) to determine mass flow rate. As with compressible flow analysis, the procedure for outlet flow analysis is virtually identical to that for inlet flow analysis and therefore is not demonstrated.

For the sake of proof of validity of these two methods of flow analysis, analysis of a single adiabatic cycle of the engine was performed based upon methods given later in this chapter. Figure 4-2 shows the striking similarity of results between compressible isentropic flow with compressibility factor to account for non-ideal gas and that of isentropic steam analysis.

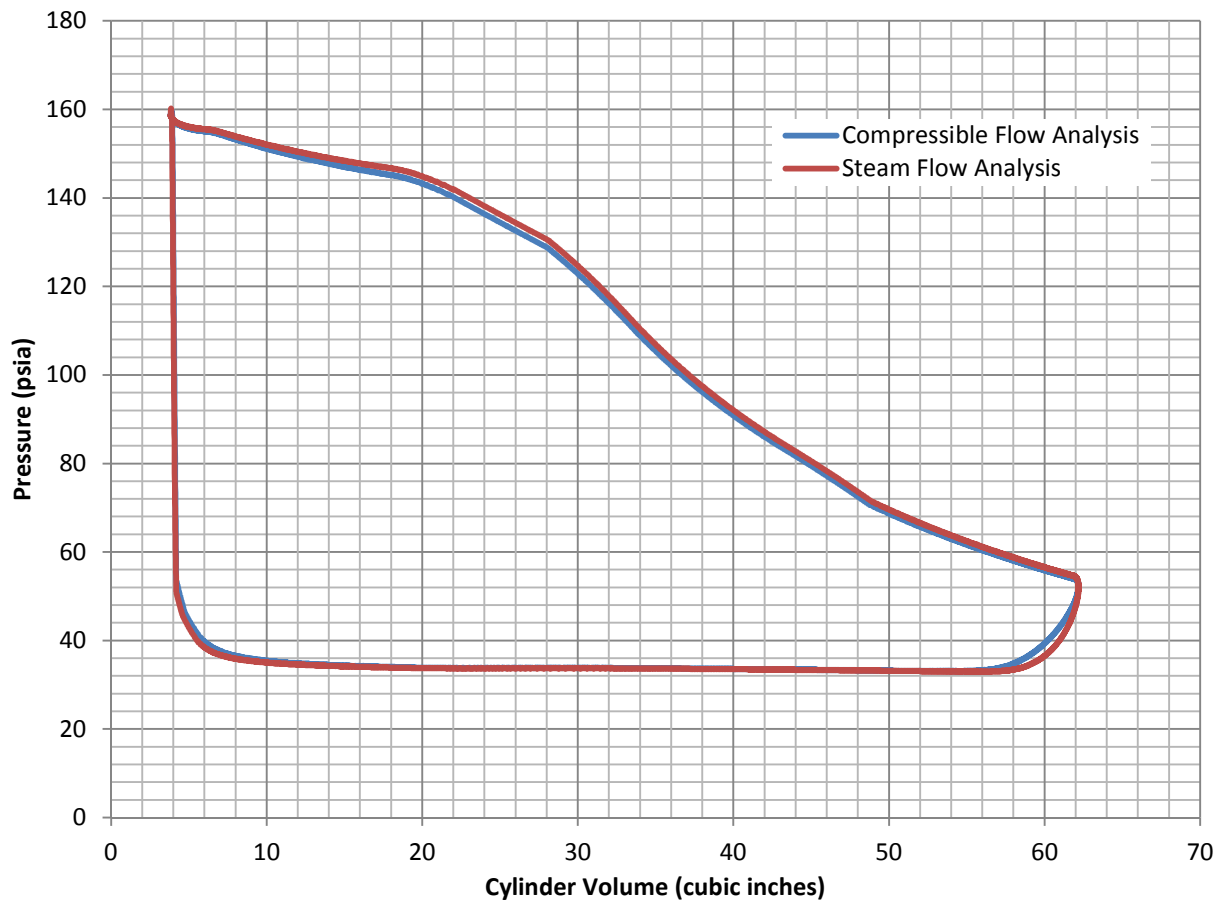


Figure 4-2. Comparison of compressible flow analysis and steam analysis for adiabatic cycle of engine.

e. HEAT TRANSFER

For this analysis, heat transfer from the steam to the cylinder wall, and vice-versa, is considered. In this section, the goal is to construct numerical descriptions for the heat transfer from steam to cylinder wall based on cylinder conditions and wall temperature (calculated based upon cylinder conditions, engine materials and insulation) for instances in time or crankshaft rotation, such that this information may be used in following analyses. The primary mechanism of heat transfer is convection and

therefore, it is considered. While conduction and radiation heat transfer may contribute, these mechanisms of heat transfer are relatively small when compared to that of convection heat transfer (except for radiation heat transfer in film evaporation as discussed in later text).

The basic convection heat transfer equation, Newton's law of cooling, was given previously in equation (3.3) is recalled here:

$$\frac{dQ}{dt} = q = h * A * (T(t) - T_s) \quad (4.19)$$

Where:

Q is thermal energy [Btu]

h is the convective heat transfer coefficient $\left[\frac{Btu}{ft^2 * s} \right]$

A is the surface area of heat being transferred $[ft^2]$

$T(t)$ is the temperature of the steam at the given point in time $[^{\circ}F]$

T_s is the wall surface temperature $[^{\circ}F]$

t is time [s]

The steam temperature and surface area are known values. Therefore, the convective heat transfer coefficient and wall temperature are to be determined. To begin, consider the analysis of the convective heat transfer coefficient. This coefficient shares a relationship with the temperature gradient at the surface of the cylinder. The temperature gradient is reflective of the conduction heat transfer. The ratio of convection heat transfer to conduction heat transfer is defined in a parameter called the Nusselt number, Nu . The average Nusselt number is described below:

$$\overline{Nu} = \frac{\bar{h}D}{k} \quad (4.20)$$

Where:

k is the thermal conductivity of the fluid

D is the cylinder diameter

The Nusselt number is a function of Reynolds number (the ratio of the inertia and viscous forces), Re_D , and the Prandtl number (the ratio of the momentum and thermal diffusivities), Pr . Each of these is defined below:

$$Re_D = \frac{u_m D}{\nu} = \frac{\rho u_m D}{\mu} \quad (4.21)$$

And

$$Pr = \frac{c_p \mu}{k} = \frac{\nu}{\alpha} \quad (4.22)$$

Where:

u_m is mean fluid velocity

ν is kinematic viscosity

c_p is the isobaric heat capacity (at constant pressure)

μ is viscosity

α is thermal diffusivity

ρ is density

The Nusselt number, while a function of Reynolds number and Prandtl number, is also function of the type of flow (e.g. internal flow, external flow, etc.) and geometry (e.g. flat plate, circular duct, bank of tubes, etc.). In the application of heat transfer inside the cylinder of a steam engine, a pair of special cases, in addition to simple heat

transfer, may arise in that condensation forms as well as boiling occurs when the condensation evaporates. Which case is applicable at each point in time is easily determined as listed in the following table:

Table 4-1. Determination of mechanism of heat transfer.

Case	Fluid Temperature	Wall Temperature
Basic Heat transfer	$T \neq T_{sat}$	<i>Any</i>
Condensation	$T = T_{sat}$	$T_s < T_{sat}$
Evaporation	$T = T_{sat}$	$T_s > T_{sat}$

For each of these cases, the physical mechanisms of heat transfer are examined.

i. Basic Heat Transfer

The simplest case is heat transfer with the absence of condensation or evaporation. In such a case, the Nusselt number is simply a correlation defined for internal pipe flow depending upon if the flow is laminar or turbulent. The distinction between these two is determined by the Reynolds number. For Reynolds number up to 2300, flow is considered laminar while turbulent flow occurs above 2300. As given in equation (4.21), Reynolds number is a function of pipe size, fluid properties and mean fluid velocity. For flow in the cylinder, the mean velocity is taken to be the velocity of the center of volume of the cylinder. This velocity is approximated to be one half of piston velocity and is given below:

$$u_m = \frac{u_{piston}}{2} \quad (4.23)$$

Such that Reynolds number is given as:

$$Re_D = \frac{u_{piston} D}{2\nu} = \frac{\rho u_{piston} D}{2\mu} \quad (4.24)$$

From equation (4.24), the Reynolds number can be determined. This calculation has been performed for a variety of conditions and has been determined that the flow is well within the entry region of the laminar regime for parts of the cycle and turbulent for others. Therefore, the most appropriate correlations for Nusselt number are given below (Incropera, Dewitt, Bergman, & Lavine, 2007) (Gnielinski, 1976):

$$Nu_D = 1.86 \left(\frac{Re_D Pr}{L/D} \right)^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14} [Laminar]$$

$$Nu_D = \frac{\left(\frac{f}{8} \right) (Re_D - 1000) Pr}{1 + 12.7 \left(\frac{f}{8} \right)^{1/2} \left(Pr^{2/3} - 1 \right)} [Turbulent]$$

(4.25)

Where:

Pr is Prandtl number

f is friction factor

Prandtl number is the ratio of the momentum and thermal diffusivities as here described

$$Pr = \frac{c_p \mu}{k} \quad (4.26)$$

Equation (4.26) can be combined with equations (4.25), (4.24) and (4.20) to solve for the convective heat transfer coefficient for simple heat transfer in the absence

of condensation and evaporation. The results of that combination can be inserted into equation (4.19) to solve for the heat transfer rate in such application. The next scenario to be examined is that of heat transfer in the presence of condensation.

ii. Condensation

Condensation occurs when the temperature of a vapor is reduced below its saturation temperature. The latent energy of the vapor is released, heat is transferred to the surface, and the condensate is formed. Condensation occurs in one of two ways, depending on the condition of the surface. The dominant form of condensation is one in which a liquid film covers the entire condensing surface, and under the action of cavities the film flows continuously from the surface. Film condensation is generally characteristic of clean, uncontaminated surfaces. However, if the surface is coated with a substance that inhibits wetting, it is possible to maintain dropwise condensation. The drops form in cracks, pits, and cavities on the surface and may grow. Typically, more than 90% of the surface is covered by drops, ranging from a few micrometers in diameter to agglomerations visible to the naked eye. The droplets flow from the surface due to the action of gravity (Incropera, Dewitt, Bergman, & Lavine, 2007).

Regardless of the method of condensation, the condensate provides a resistance to heat transfer between the vapor and the surface. Droplet formation provides significantly less resistance to heat transfer than does film formation. Heat transfer rates with dropwise condensation are typically on an order of magnitude greater than those of film condensation. In practical applications, it is difficult to maintain the condition of dropwise contamination (Incropera, Dewitt, Bergman, & Lavine, 2007). Furthermore, it is proposed that the primary mechanism of condensation in a steam engine is film

formation (Snow & Leland, 1908). For film condensation on a vertical surface, the equation for the heat transfer coefficient number is given (Incropera, Dewitt, Bergman, & Lavine, 2007):

$$\bar{h} = \frac{Re_{\delta} h_{fg} \mu}{4L\Delta T} \quad (4.27)$$

Where:

Re_{δ} is the Reynolds number as described in equation (4.28)

L is the length of the cylinder wall exposed

ΔT is difference in surface temperature and saturation temperature

Reynolds number is calculated from equation (4.28) (Incropera, Dewitt, Bergman, & Lavine, 2007).

$$Re_{\delta} = 3.78 \left[\frac{k_l L \Delta T}{\mu_l h_{fg} * \left(\frac{v_l^2}{g} \right)^{1/3}} \right]^{3/4} \quad Re_{\delta} \leq 30$$

$$Re_{\delta} = \left[\frac{3.70 k_l L \Delta T}{\mu_l h_{fg} * \left(\frac{v_l^2}{g} \right)^{1/3}} + 4.8 \right]^{0.82} \quad 30 \leq Re_{\delta} \leq 1800$$

$$Re_{\delta} = \left[\frac{0.069 k_l L \Delta T}{\mu_l h_{fg} * \left(\frac{v_l^2}{g} \right)^{1/3}} Pr_l^{0.5} - 151 Pr_l^{0.5} + 253 \right]^{4/3} \quad 1800 \leq Re_{\delta}$$

(4.28)

This equation adequately describes the heat transfer coefficient for all condensation formation throughout the cycle.

iii. Evaporation

The last case to consider is evaporation. Evaporation is broken down into four primary mechanism regions as shown in the boiling curve as shown below

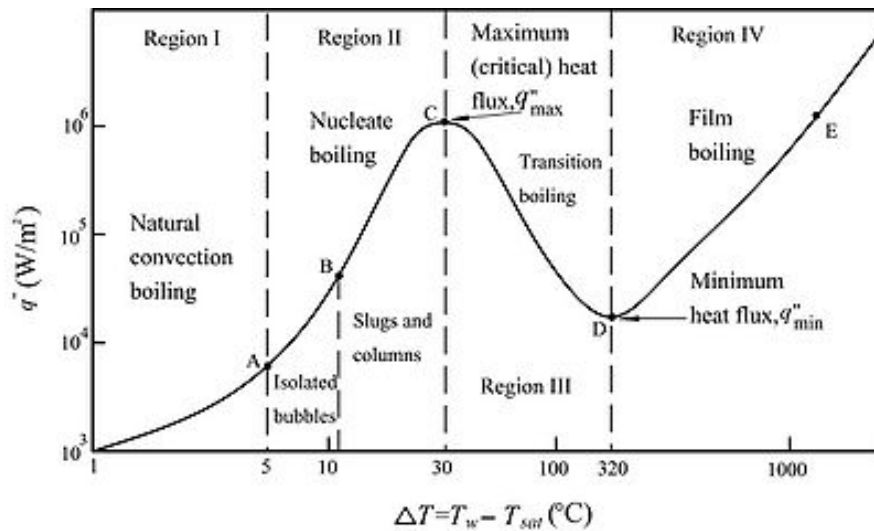


Figure 4-3. Boiling Curve. (Thermal-Fluids Central)

The regions are natural convection, nucleate, transition and film. The region of application is determined by the difference in surface temperature and saturation temperature. Heat transfer correlations have previously established for each region. These correlations are examined here.

Natural Convection Evaporation: $(T_s - T_{sat}) \leq 9^\circ\text{F}$

For natural convection evaporation, the Nusselt number is defined as follows:

$$\overline{Nu} = 0.15Ra^{\frac{1}{3}} \quad (4.28)$$

Where:

Ra is the Rayleigh number

The Rayleigh number is defined as follows:

$$Ra = \frac{g\beta(T_s - T_{sat})D^3\rho}{\mu\alpha} \quad (4.30)$$

Where:

β is the volumetric thermal expansion coefficient

α is the thermal diffusivity

Thermal diffusivity is defined as follows

$$\alpha = \frac{k}{\rho c_p} \quad (4.29)$$

This volumetric thermal expansion coefficient is a thermodynamic property which describes a measure of the amount by which the density changes in response to a change in temperature at constant pressure and defined as follows:

$$\beta = -\frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_p \quad (4.30)$$

Therefore, by solving equation (4.28) and inserting into equation (4.20), the convective heat transfer coefficient is as follows:

$$\bar{h} = \frac{0.15 \left\{ \frac{g\beta(T_s - T_{sat})D^3\rho}{\mu\alpha} \right\}^{\frac{1}{3}} k}{D} \quad (4.31)$$

Nucleate Evaporation: $9^\circ\text{F} \leq (T_s - T_{sat}) \leq 54^\circ\text{F}$

For nucleate evaporation, the Nusselt number has the general form as follows:

$$\overline{Nu} = C_{fc} Re^{m_{fc}} Pr^{n_{fc}} \quad (4.32)$$

Where

C_{fc} , m_{fc} and n_{fc} are experimentally determined constants

These constants have been previously experimentally determined for water and established to be $C_{fc} = 0.0132$, $m_{fc} = 0.33$, and $n_{fc} = 0$ (Vachon, Nix, & Tanger, 1968).

The most common arrangement of the formula for nucleate boiling is as follows:

$$h = \frac{\mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{\sigma} \right]^{\frac{1}{2}} \left(\frac{c_{p,l}(T_s - T_{sat})}{0.0132 h_{fg} Pr_l} \right)^3}{(T_s - T_{sat})} \quad (4.33)$$

Where:

σ is surface tension

Transition Evaporation: $54^\circ\text{F} \leq (T_s - T_{sat}) \leq 216^\circ\text{F}$

Transition evaporation is merely a transition between nucleate and film evaporation. For the transition region, the convective heat transfer is taken to be an interpolated value between that of nucleate evaporation and film evaporation.

Film Evaporation: $216^\circ\text{F} \leq (T_s - T_{sat})$

In film evaporation, a continuous vapor film blankets the surface and there is no contact between the condensate and the surface. For such a case, the convective heat transfer coefficient is as follows:

$$\bar{h}_{conv} = 0.62 \left[\frac{g(\rho_l - \rho_v) h'_{fg} D^3 \rho_v}{\mu_v k_v (T_s - T_{sat})} \right]^{\frac{1}{4}} \frac{k_l}{D} \quad (4.34)$$

Furthermore, radiation heat transfer may be significant with the elevated surface temperature. For this, an effective radiation coefficient has been developed as follows:

$$\bar{h}_{rad} = \frac{\epsilon \sigma (T_s^4 - T_{sat}^4)}{(T_s - T_{sat})} \quad (4.35)$$

Where:

ϵ is the emissivity of the solid (approximately 0.25 for this application)

σ is the Stefan-Boltzman constant

This effective radiation coefficient may be combined with the convective heat transfer coefficient to determine a combined coefficient. The equation for this combined coefficient is given below:

$$\bar{h} = (\bar{h}_{conv})^{\frac{4}{3}} + (\bar{h}_{conv})^{\frac{1}{3}} \bar{h}_{rad} \quad (4.36)$$

If $\bar{h}_{rad} < \bar{h}_{conv}$, equation (4.36) may be replaced with the simplified equation below:

$$\bar{h} = \bar{h}_{conv} + \frac{3}{4} \bar{h}_{rad} \quad (4.37)$$

iv. Cylinder Wall Surface Temperature Determination

For calculation of heat transfer, the temperature difference between the steam and the cylinder wall must be known. Therefore, we must calculate cylinder wall surface temperature. A steady-state wall temperature profile is assumed. That is to say it is assumed that a wall temperature at any point on the wall does not change throughout the cycle. An initial guess is made as to the wall temperature profile. To find the correct wall temperature, an analysis of an adiabatic cycle of the engine is performed to approximate the steam temperature profile. Then heat transfer characteristics are calculated as a function of the interaction between the calculated steam temperature profile and the initial guess of the wall temperature. The results of this heat transfer analysis are used to refine the wall temperature profile guess. This process was

repeated until a proper wall temperature profile was calculated. The method of using heat transfer to predict wall temperature is one of thermal-resistance model.

By use of the thermal resistance model, if two temperatures are known, and all material and geometry properties are known between said temperatures, then all intermediate temperatures may be calculated. Heat transfer mechanisms between the two temperatures are given associated resistances. The assumption is made that geometry dictates a radial system. For convection heat transfer from the steam to the wall, thermal resistance is defined as such:

$$R_{conv} = \frac{1}{2\pi h r} \quad (4.40)$$

Where:

r is radius of the cylinder bore

For conduction heat transfer, the thermal resistance is defined as such:

$$R_{cond} = \frac{\ln\left(\frac{r_{out}}{r_{in}}\right)}{2\pi k} \quad (4.38)$$

Where:

r_{out} is the radius of the outer surface of the solid

r_{in} is the radius of the inner surface of the solid

While, there is theoretically contact resistance between the mechanical interfaces of solids, the interfaces between most of the solids in this application are smooth, machined surfaces and contact resistance is expected to be minimal. Therefore, contact resistances are neglected. The resulting total thermal resistance between the steam and the ambient air is that of the convection heat transfer from the steam to the cylinder

wall surface, the conduction heat transfer through the cylinder liner, the conduction heat transfer through the engine block, the conduction heat transfer through the insulation on the exterior of the engine block and finally, the convection heat transfer from the insulation to the ambient air. Figure 4-4 shows this arrangement.

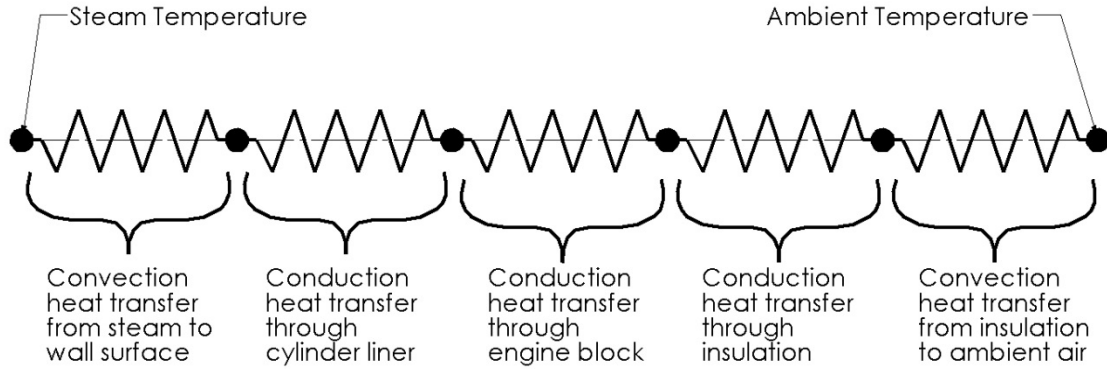


Figure 4-4. Thermal resistance diagram.

As a note, the convection heat transfer coefficient for the insulation to the ambient air is similar form to that for free convection and is as follows:

$$\bar{h} = \frac{0.54 \left\{ \frac{g\beta(T_{ins}-T_{\infty})D^3\rho}{\mu\alpha} \right\}^{\frac{1}{4}} k}{D} \quad 10^4 \leq Ra \leq 10^7 \quad (4.39)$$

or

$$\bar{h} = \frac{0.15 \left\{ \frac{g\beta(T_{ins}-T_{\infty})D^3\rho}{\mu\alpha} \right\}^{\frac{1}{3}} k}{D} \quad 10^7 \leq Ra \leq 10^{11} \quad (4.40)$$

Where:

T_{ins} is the temperature of the outer surface of the insulation

T_{∞} is the ambient temperature

To solve for the mean steady-state wall surface temperature, the steam temperatures, one from each time (or crankshaft rotation) step, are time-averaged and used in the thermal resistance model. It must be kept in mind that a wall temperature may be developed twice at a location, once with the piston on the way down and once on the way up. The resulting temperatures will likely not be the same and therefore will be averaged. With this understanding, the surface temperatures may be approximately determined with known values of the convection coefficient. However, the convection coefficient is a function of the surface temperature. Therefore, the solution may be optimized through an iterative process.

f. THERMODYNAMICS

For this analysis, thermodynamic properties of the steam are considered. In this section, the goal is to construct numerical descriptions for the thermodynamic properties of the steam based on cylinder conditions (as calculated by engine geometry section), mass flow in and out of the cylinder (as calculated in fluid flow through ports and valves section) and heat transfer (as calculated in heat transfer section) for instances in time or crankshaft rotation, such that this information may be used in following steps of the iteration process. The primary function of the thermodynamic analysis is to develop values for entropy change due to mass and heat transfer is calculated in order to predict the resulting entropy and thus, the next step conditions (i.e. temperature, pressure, quality, etc.).

Upon completion of engine geometry and port geometry calculations, the process begins with an initial guess for TDC conditions (e.g. temperature, pressure, quality, etc.). This is followed by calculations for fluid flow and heat transfer as described

previously. From the fluid flow calculations, mass transfer is determined, and from the heat transfer calculations, heat transfer rate is calculated. Using the first law, change in internal energy may be found:

$$\Delta U = (\dot{m}h)_{in} - (\dot{m}h)_{out} - W + Q \quad (4.41)$$

Where:

ΔU is change in internal energy

$\dot{m}$ is mass flow rate

W is work out from the system

Q is heat transfer into the system.

And:

$$W = \int P dV \quad (4.42)$$

Where:

P is cylinder pressure

V is cylinder volume

For a small finite step, work can be calculated as the product of the average pressure and the change in volume.

$$W = \int P dV = \bar{P} * \Delta V \quad (4.43)$$

Where:

$\bar{P}$ is average cylinder pressure for the step

ΔV is change in cylinder volume for the step

This calculation allows a formulation for internal energy for the next time step (or crankshaft rotation step) of the calculation:

$$U_{i+1} = U_i + \Delta U = (\dot{m}h)_{in} - (\dot{m}h)_{out} - W + Q = (\dot{m}h)_{in} - (\dot{m}h)_{out} - \bar{P} * \Delta V + Q \quad (4.44)$$

Where:

U_{i+1} is internal energy of the steam in the cylinder at the next step

U_i is internal energy of the steam in the cylinder at the current step

Additionally, with the mass flow rate determined and time step known, the mass in the cylinder for the next step can be determined:

$$M_{i+1} = M_i + \dot{m} \quad (4.45)$$

Where:

M_{i+1} is mass in the cylinder at the next step

M_i is mass in the cylinder at the current step

Furthermore, with engine geometry fully defined, the volume of the cylinder of the next step is known. With this volume known, as well as the mass found in equation (4.45), the density of the next step may be determined as follows:

$$\rho_{i+1} = \frac{M_{i+1}}{V_{i+1}} \quad (4.46)$$

Where:

ρ_{i+1} is cylinder steam density at the next step

V_{i+1} is cylinder volume at the next step

By taking a guess for pressure at the next step, work out of the system can be calculated and as a result internal energy is calculated. With two independent thermodynamic parameters known, internal energy and density, all thermodynamic parameters may be found from the steam tables. The pressure is compared to the pressure guess and iterations are completed until there is an acceptable level of error. This allows a full understanding of the steam conditions at the next step and the process may begin again. As described in the introduction to this chapter, this process is performed for a full revolution of the engine. The conditions for the steam at 360 degrees of crankshaft rotation are compared to those of the initial guess of TDC conditions to check for unacceptable error. If the unacceptable error exists, the new initial guess is modified and the process begins again. Additionally, due to the fact that heat transfer is based upon an initial guess, an error calculation is performed at the end of every cycle and the surface temperature initial guess is modified accordingly. Once a cycle is completed and each of these aforementioned errors are within an acceptable level, the simulation is complete and engine performance data is delivered to an output file.

g. ENGINE FRICTION

Internal engine friction must be overcome by the power produced by the engine and therefore reduces the work output of the engine thus reducing efficiency. The internal friction can have major impacts, especially at lesser output levels. Therefore, it is important to predict friction. The primary sources of friction in the PSE are from the crankshaft, reciprocating components, and piston/piston rings. There are many correlations provided for these factors in spark-ignited and compression ignited internal

combustion engines. Based on the limited compression in the PSE and use of spark ignited engine components, spark ignited engine correlations are expected to be more properly suited for use. For this analysis, correlations provided in a widely used textbook (Heywood, 1988) are used. Due to differences in the PSE and spark-ignited engines, as well as due to excess friction experienced from a trial piston skirt coating, an experimentally determined coefficient is used to modify the equation. This modified correlation is provided is as follows:

$$MEP_{friction} = \left(0.97 + 0.15 * \frac{\omega}{1000} + .05 * \left(\frac{\omega}{1000} \right)^2 \right) + C_1 \quad (4.50)$$

Where:

$MEP_{friction}$ is friction mean effective pressure in Bar.

ω is engine speed in rpm.

C_1 is an experimentally determined coefficient.

The experimentally determined coefficient is determined by measuring friction torque. The coefficient can then be calculated by the following formula.

$$C_1 = \frac{\frac{\bar{\tau}_{friction} * 12 * 2\pi}{V_d}}{14.5} - .97 \quad (4.47)$$

Where:

$\bar{\tau}_{friction}$ is friction torque in ft-lbs

V_d is engine displacement in cubic inches.

This correlation may be used to calculate power and torque requirements to overcome friction. The work per cycle may be calculated as follows:

$$W_{friction} = (MEP_{friction}) * \frac{14.5 \text{ psi}}{\text{Bar}} * V_d \quad (4.48)$$

Where:

$W_{friction}$ is work per cycle required to overcome friction in in-lbs.

V_d is engine displacement in cubic inches.

From equation, and understanding between the relationship of work and power as well as the definition of horsepower, power to overcome friction may easily be determined. With knowledge of time per cycle as defined in equation (3.4), power to overcome friction is given in equation (4.48).

$$P_{friction} = \frac{W_{friction}}{t_{cycle}} * \frac{1 \text{ in}}{12 \text{ ft}} * \frac{1 \text{ HP}}{550 \frac{\text{ft} * \text{lbs}}{\text{s}}} \quad (4.49)$$

With this information, along with all knowledge described in previous chapters, a comprehensive model analysis of the operation of a Practical Steam Engine can be performed.

5. PARAMETRIC STUDIES

a. INTRODUCTION

One of the primary benefits of proper modeling allows testing of different operating conditions and different design parameters without requiring empirical testing. Testing at under various conditions often times results in a more thorough understanding of the process. It is precisely this purpose of the current chapter in which various parametric studies will be conducted.

b. CUTOFF

The general effects of cutoff in a steam engine have been historically well understood. However, the innovative valvetrain of the PSE could yield unique results with changes in cutoff. For this reason, the effect of cutoff is modeled and explored utilizing the current valvetrain. Additionally, effect of cutoff is compared for multiple inlet/outlet conditions.

Figure 5-1 shows the effect of cutoff on power and efficiency for 150 psig inlet and 20 psig outlet conditions. While the power curve is somewhat similar to what would be expected historically, the efficiency is not. Historically, steam engines exhibited greater efficiency at lesser cutoff points, within reason, due to more complete expansion of steam and less steam enthalpy remaining in steam and end of expansion. However, at the reduced power output due to the relatively low pressure differential, the friction power consumption plays a major role. For this reason, at lesser cutoff, while the indicated efficiency is improved, the total efficiency decreases as shown. By observing the efficiency curve, at greater cutoff and subsequently greater power levels, the friction becomes less important and efficiency appears to have a tendency to level off or

decrease with further increased cutoff. As expected, the power output is directly proportional to cutoff. It should be noted, that there does not appear to be significant point of diminishing returns in terms of power increase with cutoff. In typical steam engines, this is not the case. Historically, steam engines use relatively small area valves with slow opening/closing times. This, in combination with high piston velocities at increased cutoff points, results in relatively small increases in power with increases in cutoff. The benefits of the large area, fast-acting proprietary valves in the PSE are emphasized in high cutoff applications.

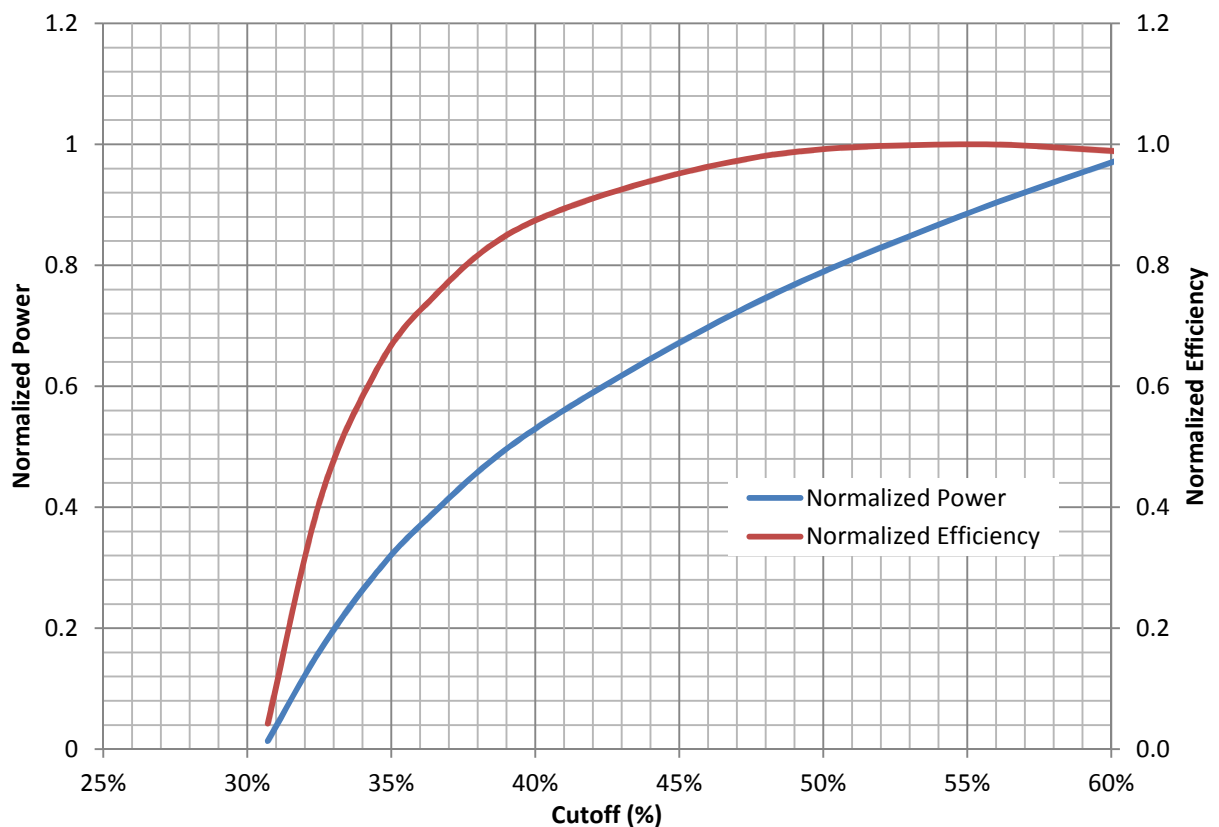


Figure 5-1. Effects of cutoff on power and efficiency for 150 psig inlet to 20 psig outlet.

In comparison, Figure 5-2 shows the effects of cutoff on power and efficiency for 250 psig inlet to 5 psia outlet. Both curves are similar to what is expected historically. The efficiency increases with reductions in cutoff until the cutoff is low enough that

friction becomes significant in comparison to power output. Again, the power curve follows historical understanding and increases with increases in cutoff. As with the previous cutoff analysis, there is not an obvious point of diminishing return for the power as a result to large area, fast-acting proprietary valves of the PSE.

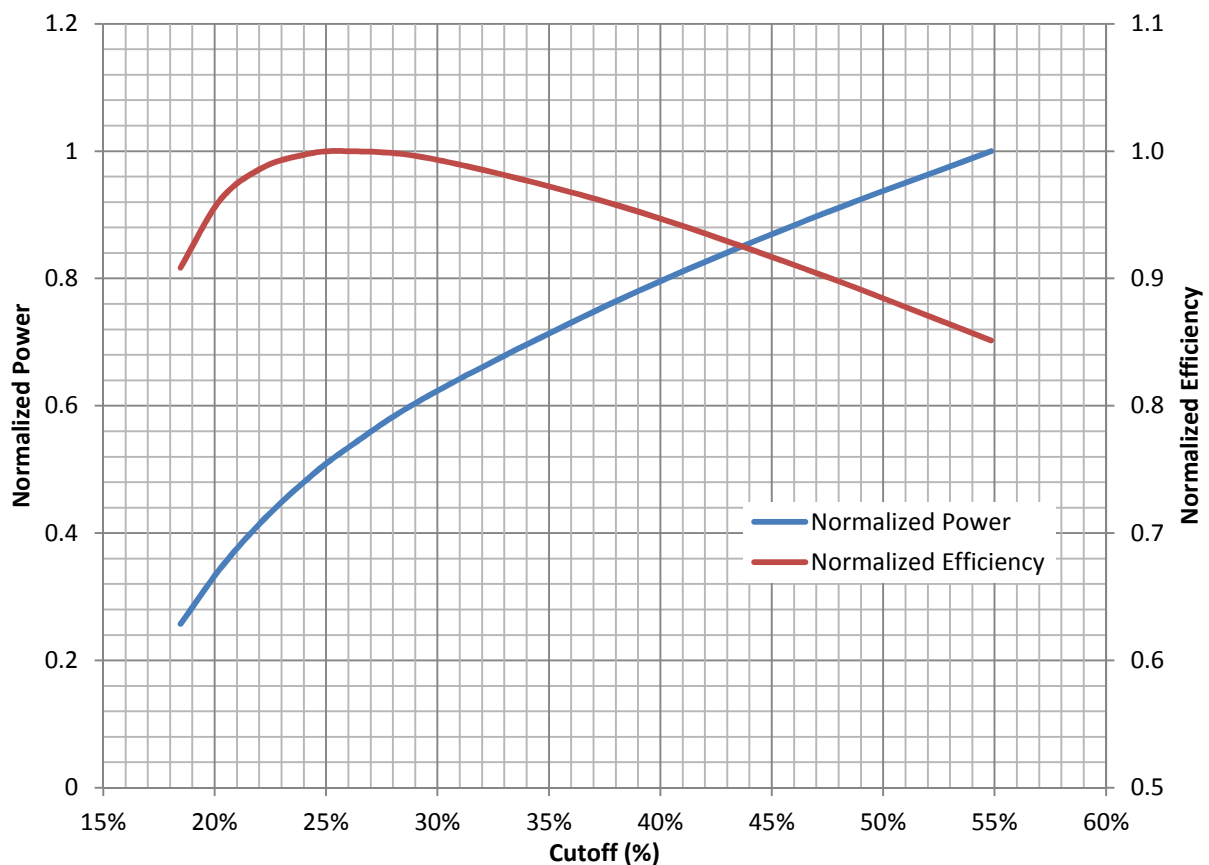


Figure 5-2. Effects of cutoff on power and efficiency for 250 psig inlet to 5 psia outlet.

c. SUPERHEAT

Utilizing inlet superheat is a common procedure for steam turbines and, historically, steam engines. This is primarily due to reduction in initial condensation and heat transfer. However, in the typical application of the PSE, modification of the steam system to produce superheated steam out of the boiler is not practical. As an alternative, a throttling valve could be used for isenthalpic expansion such to produce

superheat. While this would reduce supply pressure to the PSE, it is worth exploring if the improved efficiency from superheat would be substantial enough to overcome this pressure reduction. Figure 5-3 shows the PV diagram for some varying levels of throttling.

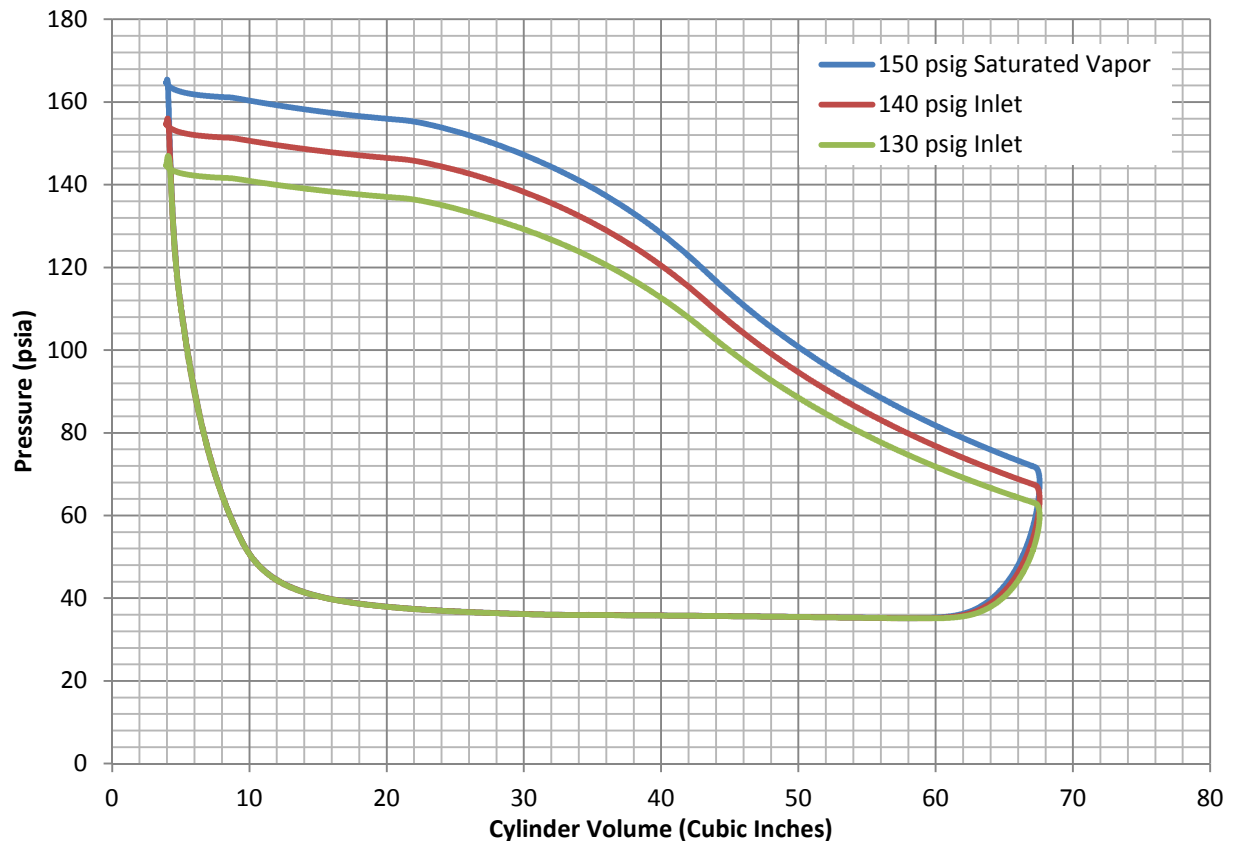


Figure 5-3. PV diagrams for various levels of throttling.

The superheat did prove to reduce heat transfer losses for each case with superheat. As a result, the isentropic efficiency of the engine with superheat was greater than that for saturated vapor. The isentropic efficiency increased by 0.8 efficiency points (1.4% improvement) for the 140 psig case and 1.6 efficiency points (2.6% improvement) for the 130 psig case. However, the indicated efficiency of the total

process was less (i.e. more steam was required to be provided to the engine to produce an indicated unit of power) for the cases with superheat than for saturated vapor. Furthermore, due to reduced power output while considering constant friction losses, the superheated cases included had reduced mechanical efficiency and therefore suffered further reduced efficiencies. In summary, throttling to produce superheat seems to not be a viable method of improving efficiency in most practical applications.

d. VALVETRAIN DESIGN

The chief area of research in the development of the Practical Steam Engine (PSE) centered on valvetrain design. The high speed operation of the PSE required fast gas exchange process and therefore significant valvetrain requirements. To meet these requirements, as well as maintain high efficiency, the valvetrain needs to have large valve area and fast acting operation. The most common type of valve used in an engine is a cam driven poppet valve. However, after analyzing mechanical and physical limitations of the poppet valve, it was determined that an appropriate opening time and valve area could not be achieved. A proprietary, patent-pending, valve train was developed to meet these requirements. Figure 5-4 illustrates a comparison of PV diagrams for the PSE operating with the current valve design vs both practical and radical poppet valves.

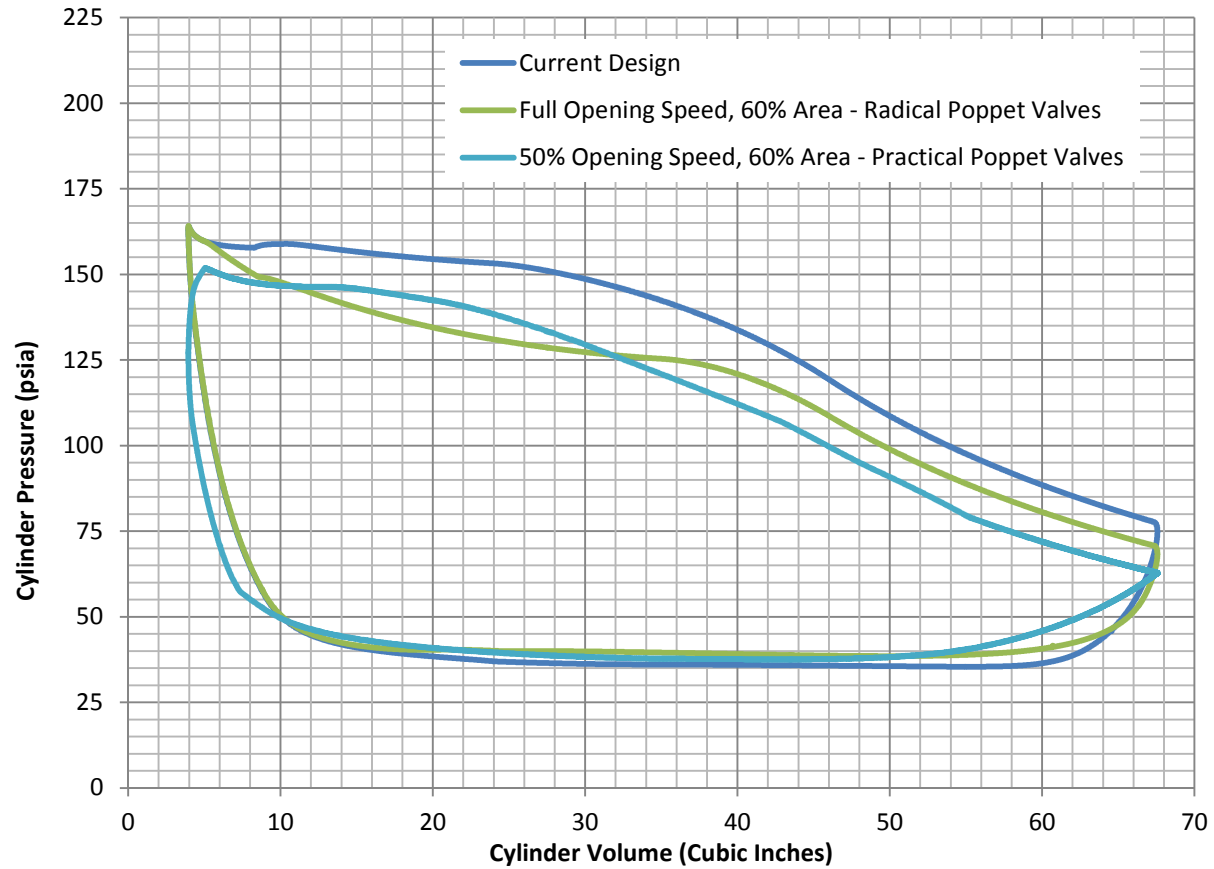


Figure 5-4. PV diagrams for varying valvetrain designs.

The PSE with the current design yielded 27% greater power than the engine with the practical poppet valve. It should be noted that the exhaust timing on the practical poppet valve configuration had to be retarded by 13° to prevent over compression.

6. MODELING OF EXPERIMENTAL APPLICATION

a. INTRODUCTION

A functional prototype has been manufactured and installed in the Western Ave Plant of Enwave Seattle (formerly Seattle Steam). This engine will be experimentally tested. Chapter 6 describes the experimentation of the engine. Chapter 5 provides a theoretical model of that engine such to be compared to experimental results. This model will project power, flow, efficiency, heat transfer losses and more at a variety of operating conditions.

The basic input conditions of the model are as described below.

- Inlet Steam Conditions
 - o Approximately Saturated Vapor (quality 99% +/- 1%)
 - o Pressure = 100-130 psig
 - o Temperature = 362.9 °F
- Outlet Steam Conditions
 - o Pressure = 18 psig.
 - o Temperature = 255.3 °F
- Heat loss inputs
 - o 75F ambient temperature
 - o No insulation on engine block

- Nominal conditions
 - o 1800 rpm operation
 - o Power output up to approximately 45 kWe (net at output of generator)
 - o Generator efficiency according to datasheet provided in Appendix A.

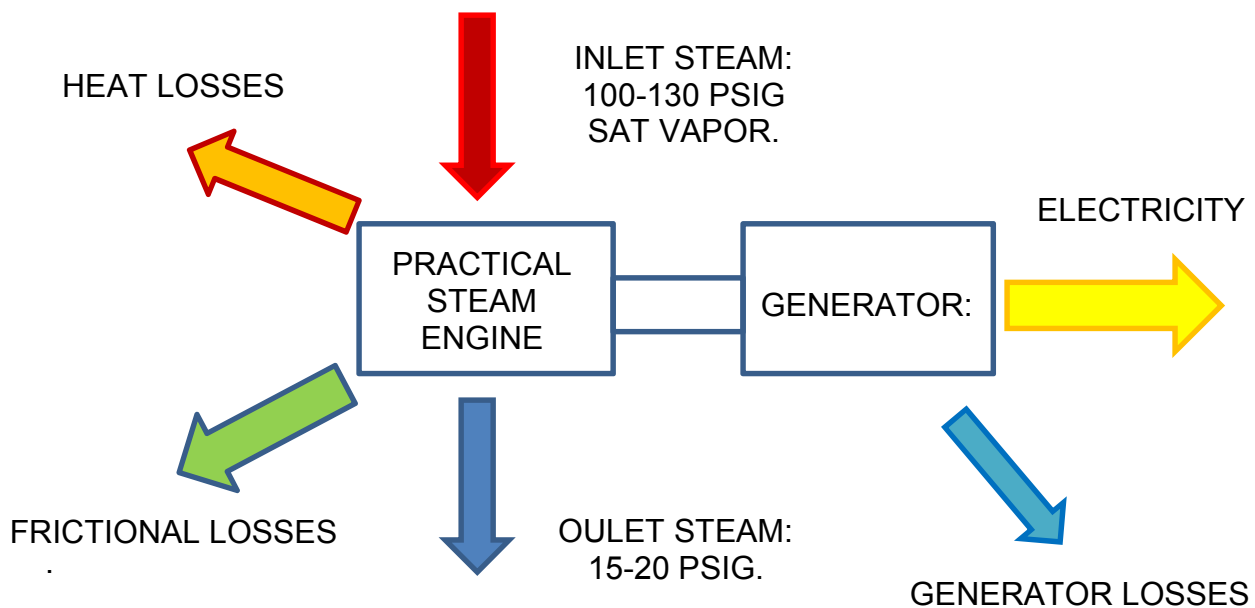


Figure 6-1. Layout of model for analysis

Figure 6-1 shows a basic layout of the system to be analyzed.

For this model, steam flow analysis was used instead of compressible flow. Heat transfer was considered. The intended application of this engine in this setting is to produce a constant 80 kWe and therefore this model focuses on operation with this load profile.

b. OPERATIONAL CHARACTERISTICS AT 80 kWe BASE LOAD

To achieve 80 kWe base load output at generator at a speed of 1800 rpm with the conditions given above, the analytical model suggests the Practical Steam Engine requires about 80 percent corresponding to 71 percent cutoff. At this condition, the friction power consumption is approximately 43.1 HP (32.2 kW). Additionally, to achieve 80 kWe net output of a generator with corresponding efficiency, 84.2 kW of shaft power must be supplied to the generator. After considering friction losses as described, the PSE must produce approximately 116.4 kW of indicated power. Figure 6-2 and Figure 6-3 illustrate the differences in these power levels for an example inlet pressure of 130 psig and outlet pressure of 18 psig. As shown, the indicated power and efficiency is the greatest. Brake power is found by subtracting power to overcome friction from the indicated power output. Net total power output at the generator is simply the efficiency of the generator multiplied by the brake power output. The efficiencies are in reference to the power specific steam consumption versus that of isentropic expansion from saturated vapor at inlet pressure to a saturated mixture at the outlet pressure. As seen from Figure 6-3, the indicated efficiency of the engine reaches an impressive 85%. However, the brake efficiency and net efficiency are around the 50% range in a typical operating range. For sake of comparison, single stage steam turbines typically operate in the 45-55% efficiency range. For the current application, requiring a net output of 80kWe, if provided 130 psig inlet pressure, the graph suggests a flow rate of around 4300 lbs/hr with a corresponding efficiency of around 57%.

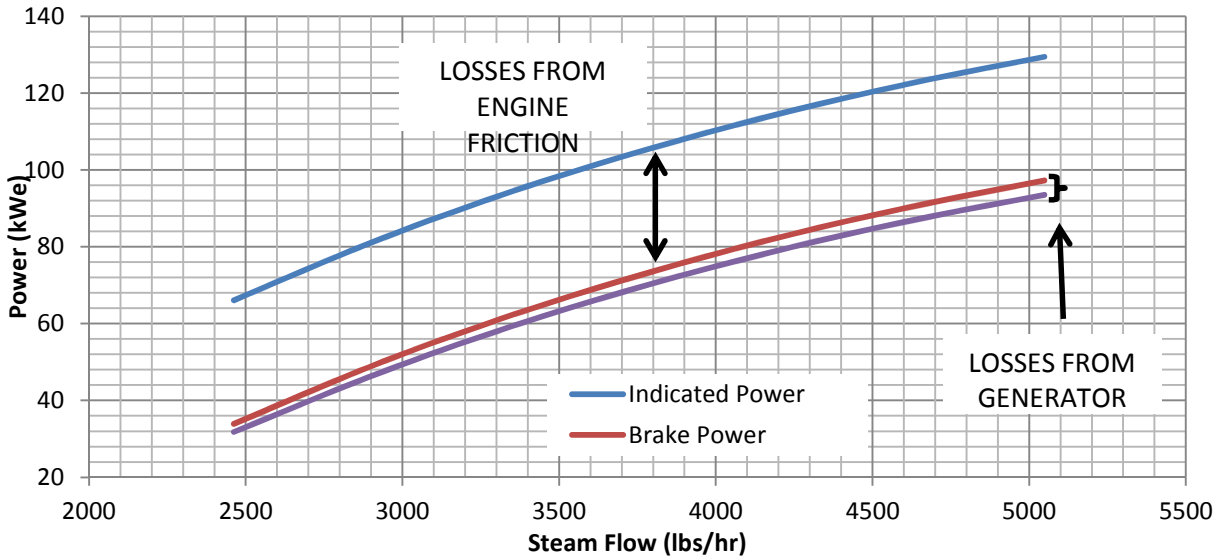


Figure 6-2. Indicated power, brake power and net power versus flow rate for 130 psig to 18 psig backpressure.

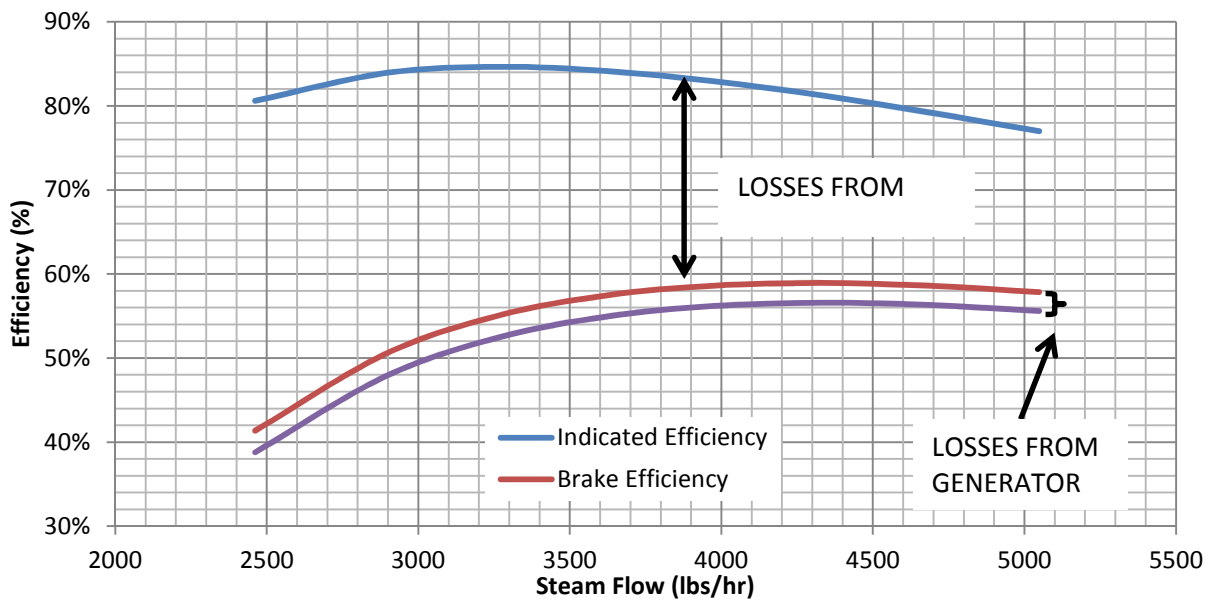


Figure 6-3. Indicated efficiency, brake efficiency and net efficiency versus flow rate for 130 psig to 18 psig backpressure.

A detailed model analysis of the engine in the current application yielding 45kW_e has been performed. Figure 6-4 illustrates the P-V diagram of a cycle of engine

operation. The valve events, intake opening, intake closing, exhaust opening and exhaust closing are noted on the figure. The cycle includes four primary events: admission, expansion, exhaust and compression. These events are noted on the P-V diagram. The area enclosed inside the P-V diagram represents the work output of the engine per cycle. This work per cycle is calculated to be approximately 4,388 in-lbs. At an operational speed of 1800 rpm, this equates to roughly 119 kW of indicated power. This equates to 86.4 kW brake power and around 82 kW of net power output.

In addition to gas exchange process and thermodynamic processes occurring on the steam, there is another significant process occurring during this cycle: heat transfer. A temperature differential between the steam and the cylinder wall surfaces produce a transfer of heat energy between the two.

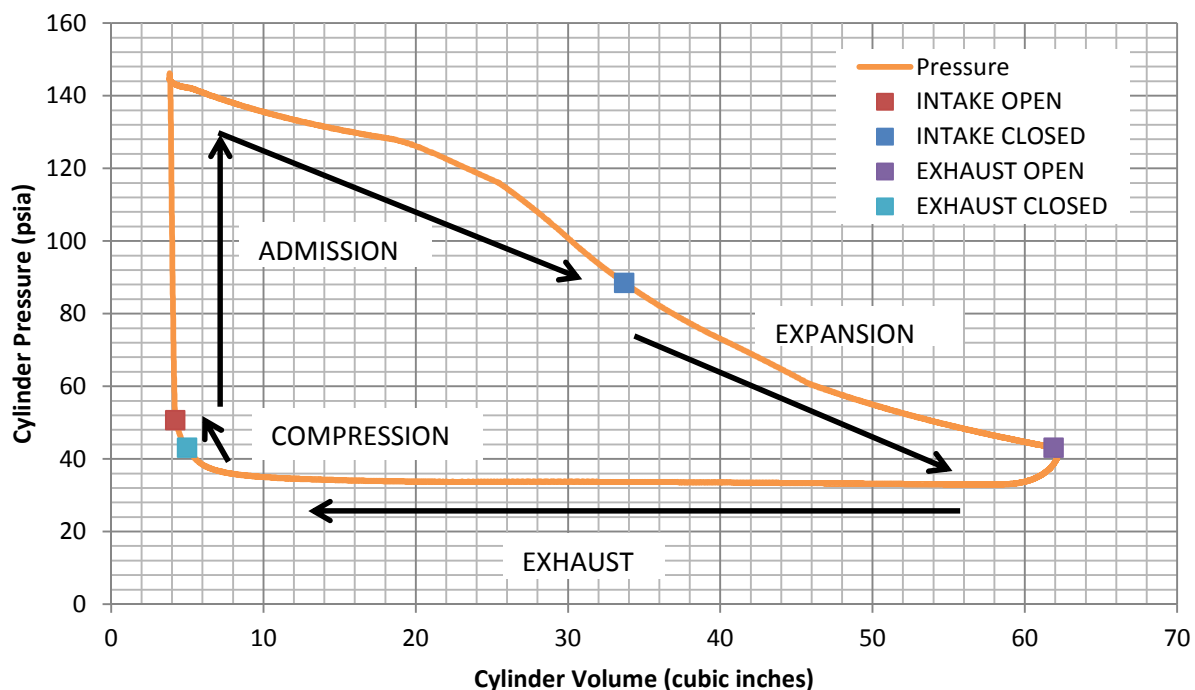


Figure 6-4. Modeled P-V diagram for a cycle of operation of the PSE.

The first step taken in order to model and analyze the heat transfer was to calculate the cylinder wall temperature. This was done by an iterative process. The process included simulating an adiabatic cycle of the engine to approximate the steam temperature profile. Then heat transfer characteristics were calculated as a function of the interaction between the calculated steam temperature profile and an initial guess of the wall temperature. The results of this heat transfer analysis were used to refine the wall temperature profile guess. This process was repeated until a proper wall temperature profile was calculated. The resulting wall temperature profile is given in Figure 6-5. This profile is not only a function of steam temperature but also of oil temperature, ambient temperature, insulation, geometries and other.

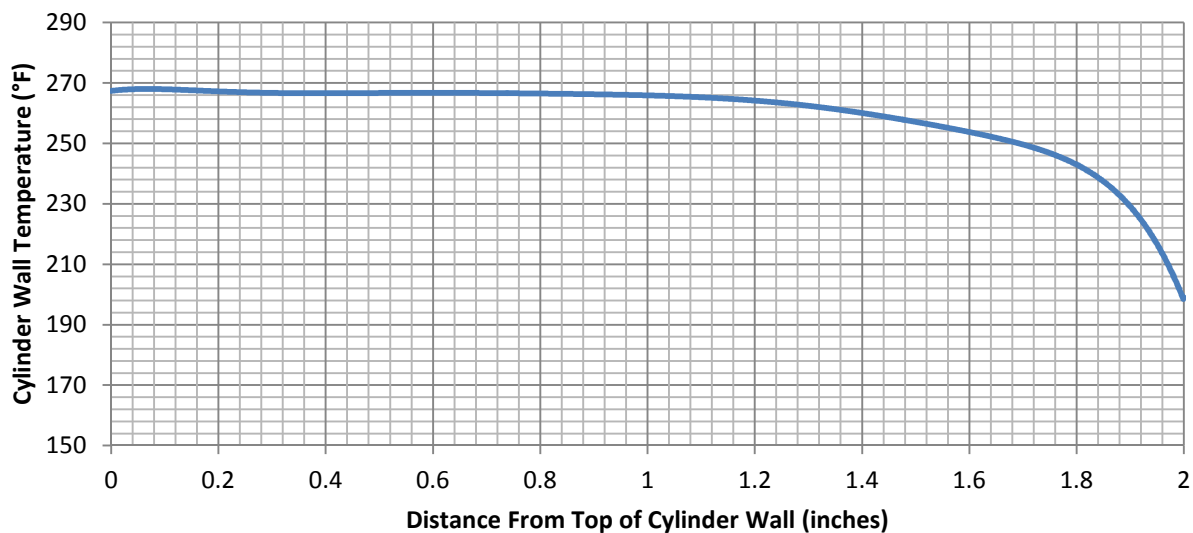


Figure 6-5. Calculated cylinder wall temperature profile.

The steam temperature in the cylinder varies throughout the cycle and at some points is greater than the wall temperature and some points less. Figure 6-6 shows the steam temperature throughout the cycle.

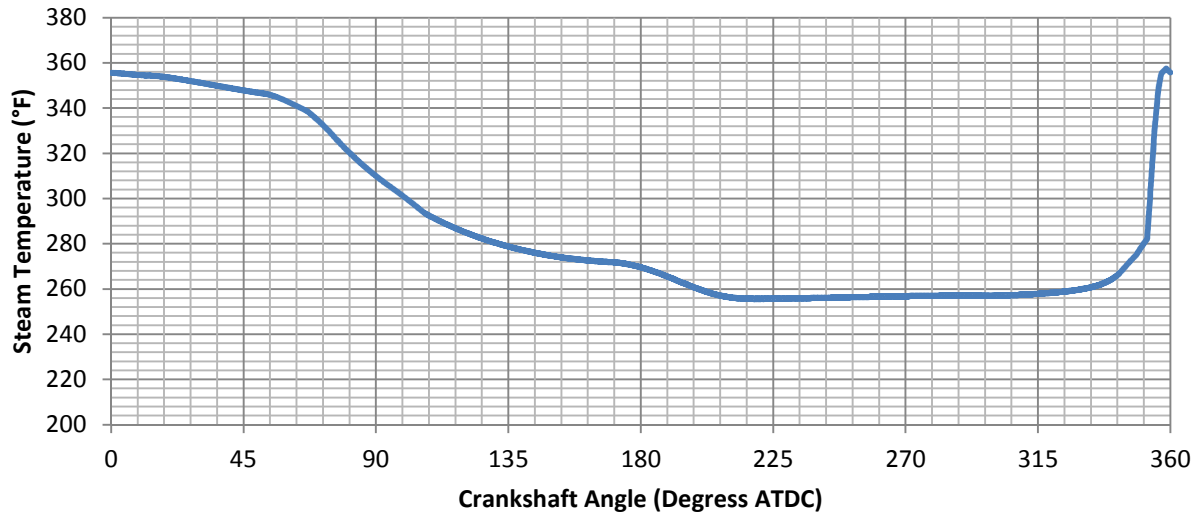


Figure 6-6. Calculated steam temperature throughout the cycle.

With these two profiles determined, the heat transfer throughout the cycle was modeled and calculated. Figure 6-7 illustrates the heat transfer calculated. Positive values represent heat being transferred from the steam to the cylinder wall while negative values represent heat being transferred from the cylinder wall to the steam.

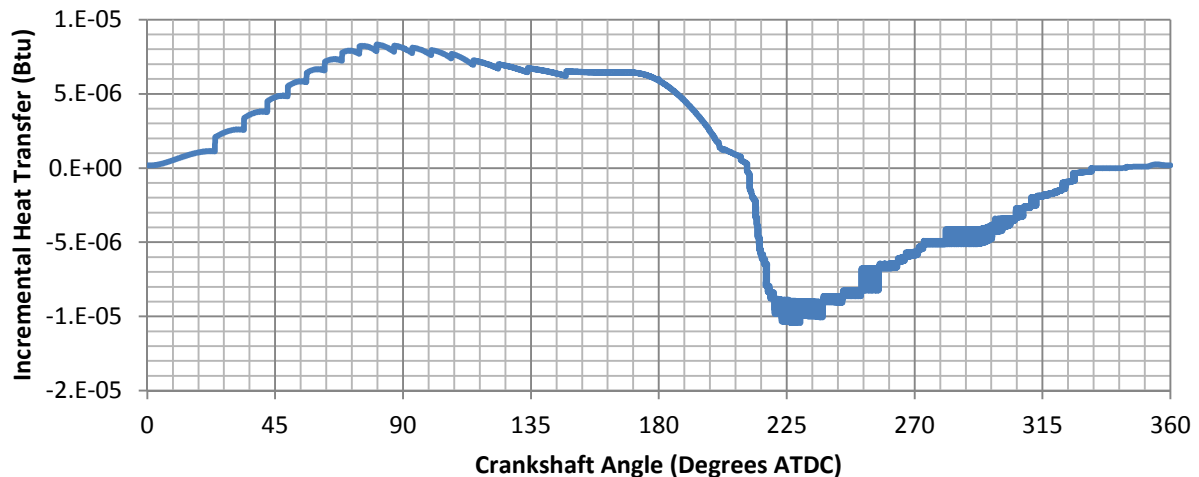


Figure 6-7. Graphical representation of the modeled and calculated heat transfer throughout a cycle.

At first glance, it may seem surprising to see similar or even greater max values of heat transfer into the steam than out of the steam due to the fact that there is a much larger temperature differential realized during peak modes of heat transfer from steam to the wall. However, this counter intuitive result is practically understood when studying the differing modes of heat transfer. The convective heat transfer coefficient is heavily influenced by the mode of heat transfer. For heat transfer from the steam to the wall, the mode of heat transfer is basic heat transfer (no phase change) and condensation. However, for heat transfer from the wall to the steam, the primary mode of heat transfer is evaporation or boiling. To describe to even more specific detail, the primary mode of boiling in this application is nucleate boiling. As recalled from the boiling curve as shown in Figure 4-3, nucleate boiling generates tremendous heat transfer rates and thus explains the significant heat transfer rates seen here with relatively small temperature differences. The heat transfer losses were found to be small in comparison to the power output of the engine. The total heat loss through the cycle is the sum of the heat transfer throughout the cycle. In this instance, it was calculated to be 0.0046 Btu/cycle. The cumulative sum of the heat transfer throughout the cycle is displayed in Figure 6-8. Considering the operational speed of 1800 rpm, this value may be converted to heat rate in terms of kW. This analysis projects a total heat loss rate of 0.15 kW.

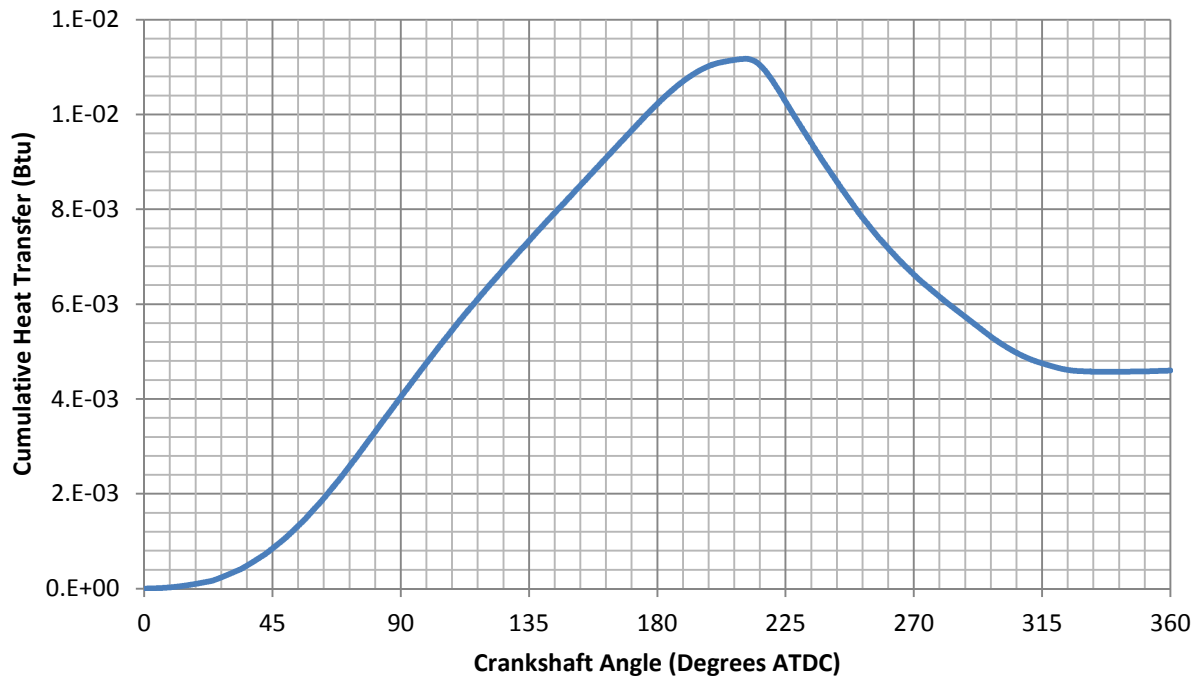


Figure 6-8. Calculated cumulative heat transfer throughout a cycle.

All of this data and calculations lead to more important information: what does the overall practical operation of the engine look like. This analysis is intended to analyze practical application of the PSE. Therefore, it is important to understand basic operational parameters of the engine such as outlet pressure, outlet quality, isentropic efficiency, total efficiency, etc. Figure 6-9 provides the projected pressure profile as well as the projected outlet pressure. As shown, the projected outlet pressure is 22.3 psig. This is calculated by performing a mass-averaged calculation of outlet pressures, including valve leakage. From a practical standpoint, this steam pressure will mix and expand in the exhaust stream, coming to an equilibrium pressure of 18 psig due to relatively large 18 psi reservoir.

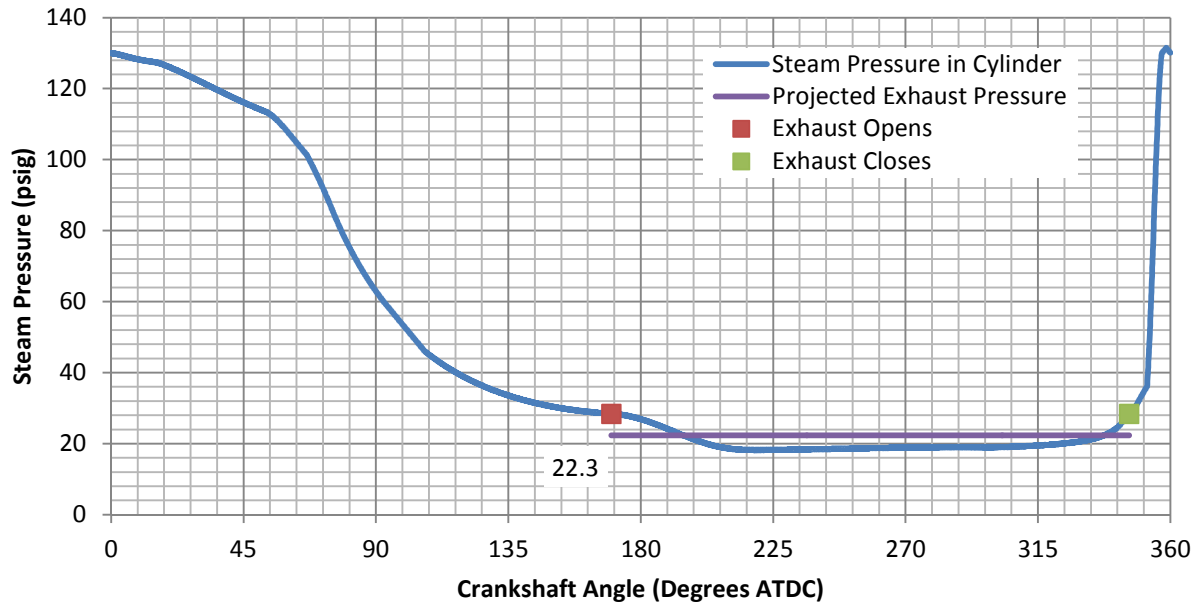


Figure 6-9. Calculated pressure throughout the cycle as well as the projected outlet pressure.

In addition to pressure, it is also very important to understand the quality of the steam exhausted from the engine. Figure 6-10 shows the quality profile in the cylinder as well as the mass-averaged outlet quality. This projected outlet quality is 91.4% as compared to 90.3% quality projected for isentropic expansion. Quality is defined as the percentage of the mass of the flow that is in vapor form. This can be written in the following configurations.

$$X = \frac{\dot{m}_{vapor}}{\dot{m}_{total}} \quad (6.1)$$

Or

$$X = 1 - \frac{\dot{m}_{liquid}}{\dot{m}_{total}} \quad (6.2)$$

Where

$\dot{m}_{vapor}$ is mass flow rate of vapor

$\dot{m}_{total}$ is total mass flow rate

$\dot{m}_{liquid}$ is mass flow rate of liquid

Figure 6-11 illustrates an overview of the of the pressure reduction through the engine in terms of thermodynamic properties. Figure 6-11 also shows an isentropic expansion process in comparison to the actual expansion process.

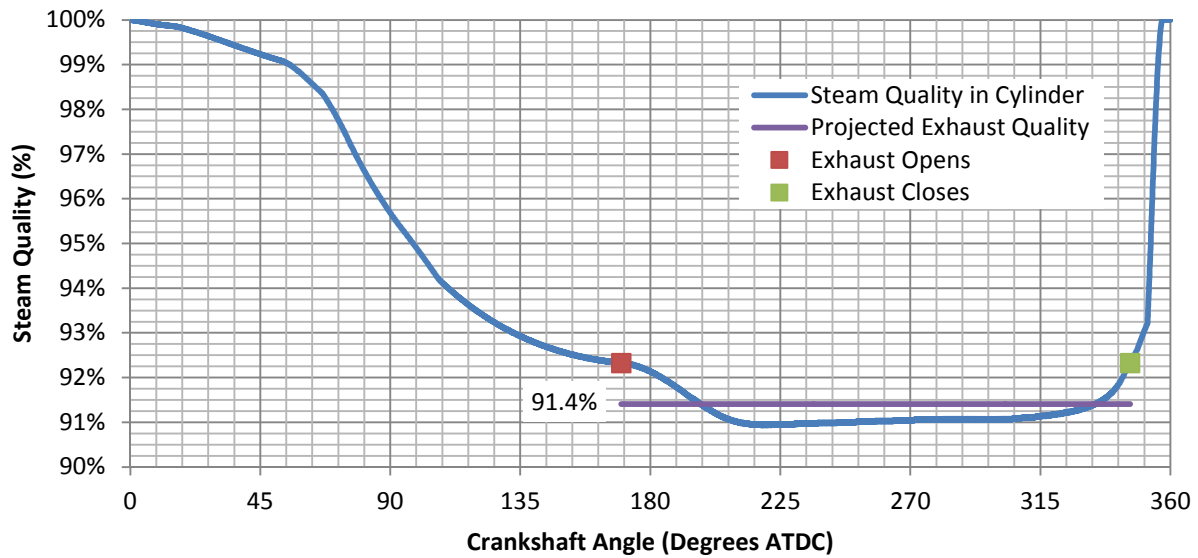


Figure 6-10. Calculated steam quality throughout the cycle as well as the projected outlet quality.

The isentropic efficiency may be ascertained from the data in the T-S diagram shown in Figure 6-11. The data points are tabulated in Table 6-1.

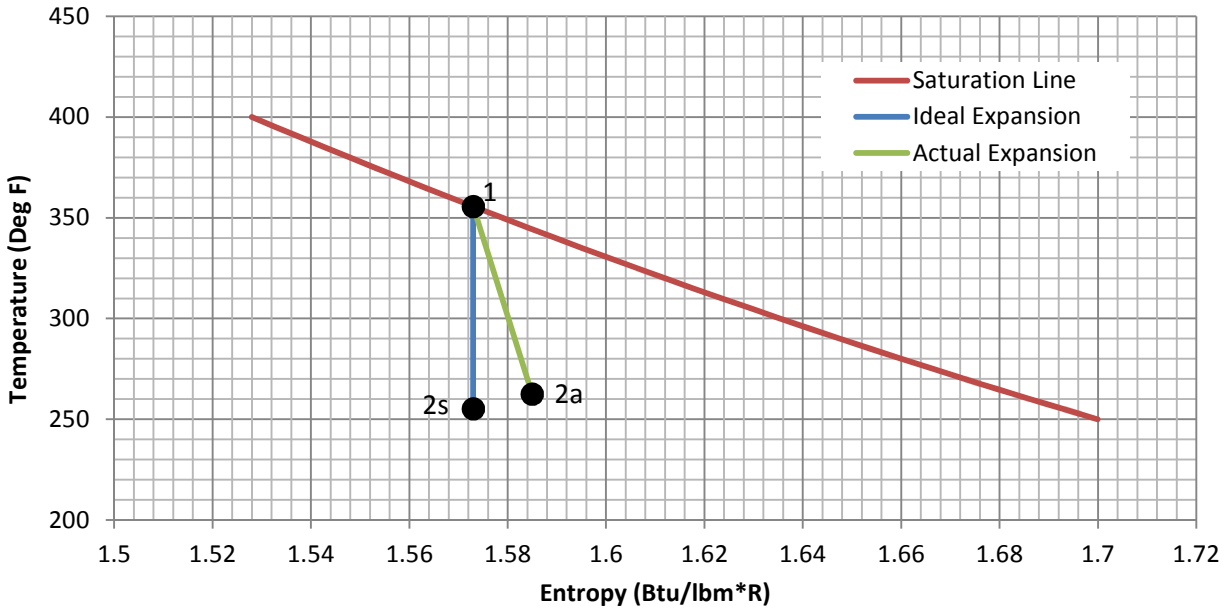


Figure 6-11. Expansion process of the PSE in comparison to isentropic expansion on T-S diagram.

The points 1, 2s and 2a in Figure 6-11 refer to initial condition, end of isentropic expansion and end of actual expansion, respectively. The conditions at the three points are given in Table 6-1.

Table 6-1. Thermodynamic properties for the points in Figure 6-11.

	Point 1	Point 2s	Point 2a
Pressure (psig)	130.0	18.0	22.3
Temperature (°F)	355.6	255.3	262.5
Enthalpy (Btu/lbm)	1193.9	1080.3	1097.8
Entropy (Btu/lbm*R)	1.573	1.573	1.585
Quality (%)	100%	90.9%	92.5%

From this data, the isentropic efficiency is calculated to be 84.6% based on the following equation:

$$\eta_{isn} = \frac{H_1 - H_{2a}}{H_1 - H_{2s}} \quad (6.3)$$

The isentropic efficiency contributes to the total efficiency of the engine. Other efficiencies that contribute are mechanical efficiency (losses due to friction) and generator inefficiency. For this application, the total net efficiency defined as the efficiency in which the engine converts a change in steam enthalpy into electricity, is 40.3%. At the anticipated operating condition, the model suggests 3,250 lbs/hr of steam flow and produces 43.6 kW. Chapter 6 describes a test procedure such to experimentally test the validity of the model.

c. SOURCES OF LOSSES IN THE PSE

There are numerous sources of inefficiencies and irreversibilities in the steam engine that cause it to deviate from an ideal steam expansion cycle. These primarily include throttling losses due to finite sized valves, engine friction, heat losses, valve leakage and entropy generation.

It is of interest to categorize the losses of the practical steam engine for purposes of improvement. Losses can be in the form of energy loss and exergy destruction as well as flow limitations. Each of these types of losses are illustrated in terms of percentage of contribution.

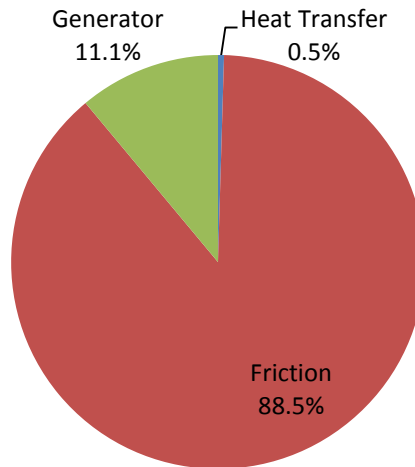


Figure 6-12. Pie chart of contribution of sources of energy loss.

Clearly mechanical inefficiency is the primary source of energy loss, or 1st law inefficiency, with nearly 90% of all losses being consumed by the friction. Generator losses make up most of the remaining losses, while heat transfer losses were very small.

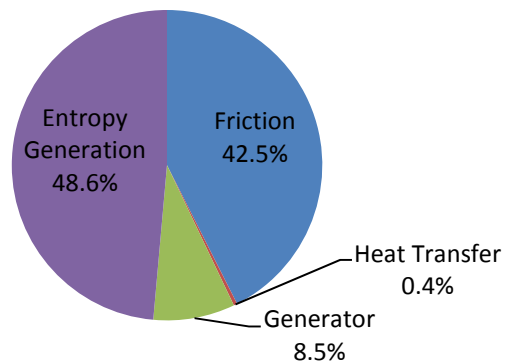


Figure 6-13. Pie chart of contribution of sources of exergy destruction.

The source of heat transfer does not include entropy generation due to heat transfer. All sources of entropy generation were lumped together. As seen, entropy generation, along with mechanical inefficiency, are the primary causes of exergy

destruction. Generator losses account for most of the remaining losses while exergy destruction due to heat transfer was minor.

Another area of interest is to understand what sources cause deviation from a perfectly efficient, ideal, isentropic steam engine. These sources are illustrated below.

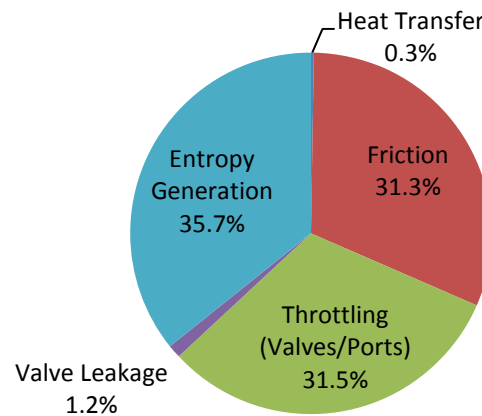


Figure 6-14. Pie chart of contributions of sources of deviation from ideal steam engine.

Sources of deviation from ideal steam engine were found to be fairly evenly distributed among three categories: entropy generation, throttling and friction. Each of these categories contributed to approximately one-third of the deviation from ideal, while valve leakage and heat transfer were other minor sources. Losses due to throttling was derived by comparing current valve performance to that achieved with infinitely large valves that open and close instantaneously.

All losses contributing to deviation from ideal steam engine should be examined in an effort to identify opportunities for improvement. Some of which may be very difficult to improve substantially (e.g., throttling through ports and valves, entropy

generation), while others may be improved with design enhancements (e.g., friction, generator losses, valve leakage).

7. TESTING/EXPERIMENTATION

Portions of this chapter were presented in June 2014 at the International District Energy Association Annual Conference in Seattle, Washington.

a. INTRODUCTION

A small-scale functional prototype has been designed, constructed and tested. The results of the tests show the performance of the engine to be extremely similar to that of theoretical calculations in terms of power and efficiency. After thorough testing of this unit, a full-scale unit was designed and manufactured. This unit was installed in the Western Avenue plan of Enwave Seattle (formerly Seattle Steam) in application of PRV parallel. This application of this PRV parallel is pressure reduction for low pressure steam supply to local customers for district heating. Currently, a PRV reduces 145 +/- psig steam to 18-22 +/- psig steam such to be delivered directly to customers. The process using a PRV, while theoretically isenthalpic, creates excess superheat, reduces the exergy of the steam and may be economically improved upon by use of a backpressure steam powered generator. This is the exact purpose of the PSE in this application. The PSE is designed to reduce the pressure as close to isentropic as possible and utilized the reduction in enthalpy to generate electricity.

The flow may vary from a low of roughly 5,000 lbs/hr in the summer to 15,000-20,000 lbs/hr in the winter. The flow capacity of the PSE in this application is around 5,000 lbs/hr. Therefore, the PSE is being installed in parallel to the PRV and designed

to allow 5,000 lbs/hr of steam through the PSE and the excess is routed through the PRV. This parallel installation allows redundancy and allows the PSE to be shut down for inspection or maintenance without compromising the operation of the system. A diagram of this installation is shown in Figure 7-1.

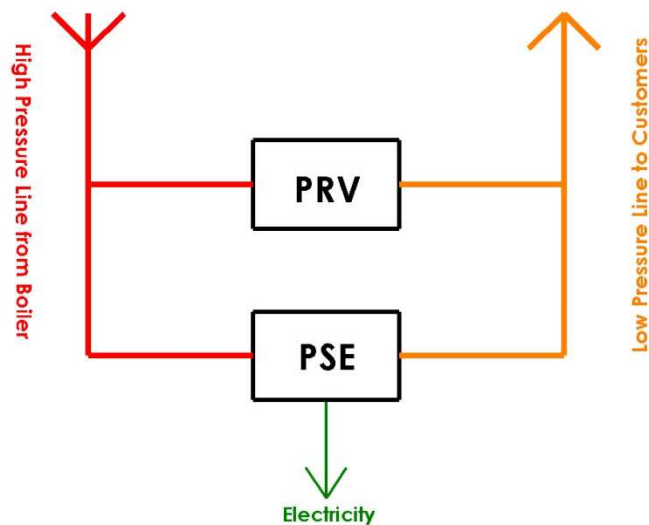


Figure 7-1. Diagram of how the PSE is installed in Enwave Seattle's system.

b. SETUP/EQUIPMENT

i. Installation

In order for the engine to be installed a concrete foundation must be poured, a high pressure steam line (insulated 2" schedule 80 pipe) and a low pressure line (insulation schedule 40 4" pipe) must be fabricated and electrical wiring must be supplied to the skid. After the concrete foundation was poured, the skid with engine was mounted. After the skid was installed on the foundation, the



Figure 7-2. Full-scale PSE being moved in on the concrete foundation during installation.

pipe was fabricated and wiring was supplied. The piping of the system can be seen in Figure 7-3. In this figure, the high pressure line and the low pressure line are shown.



Figure 7-3. Final installation of the PSE at Enwave Seattle.

ii. Equipment

A flow meter was installed in the high pressure line in order to monitor and record flow data. The flow meter selected is the Proline Prowirl F 200, manufactured by Endress + Hauser. The device uses a Vortex measuring principle. The unit is pressure and temperature compensated with an uncertainty of $\pm 1.7\%$ when measuring mass

flow rate of steam. A picture of the installation of the flow meter is shown in Figure 7-4. Specifications for the flow meter are given in Appendix A.



Figure 7-4. Installation of the flow meter.

The power output of the engine will be measured electrically. The PSE is driving a Marathon® MagnaPlus® 3 phase, 150 kW, 1800 rpm, 480 volt, generator. The unit model number is 363PSL1607 with a PMG excitation source and a DVR2000E+ voltage regulator. The efficiency of the generator varies from just under 93% up to almost 96%, depending on load. The generator datasheet including a graph of the generator efficiency is provided in Appendix A.

In order to test the engine performance, a simulated electrical load is required. For this purpose, a load bank is used. The load bank is an Avtron Freedom with a capacity of 100 kW. The load bank uses a Shark® 100 multifunction power and energy meter. This meter measures and calculates voltage, current, power and frequency. The

meter has various accuracy levels for various readings. A chart of these accuracy levels is provided in Appendix A. An image of the load bank with power meter is shown in Figure 7-5.



Figure 7-5. Avtron Freedom load bank and Shark® 100 power meter.

The principle of operation of the load bank is to send electrical current through various resistors in order to create an electrical load. The load may be changed by simply adjusting which resistors, and how many, are carrying current. This resistance through the resistors generates heat and the resistors are air-cooled by forced airflow from axial fans. The load bank has a load resolution of 5 kW, which may be manually

increased or decreased by the user. This load is applied to wires input into the load bank. Therefore, to apply load to the generator, and consequently engine, wires must be connected from the generator to the load bank. The Avtron Freedom Load Bank datasheet is given in Appendix A.

To connect the generator to the load bank, an electrical panel was installed with a 150 amp circuit breaker. A picture of this is shown in Figure 7-6.

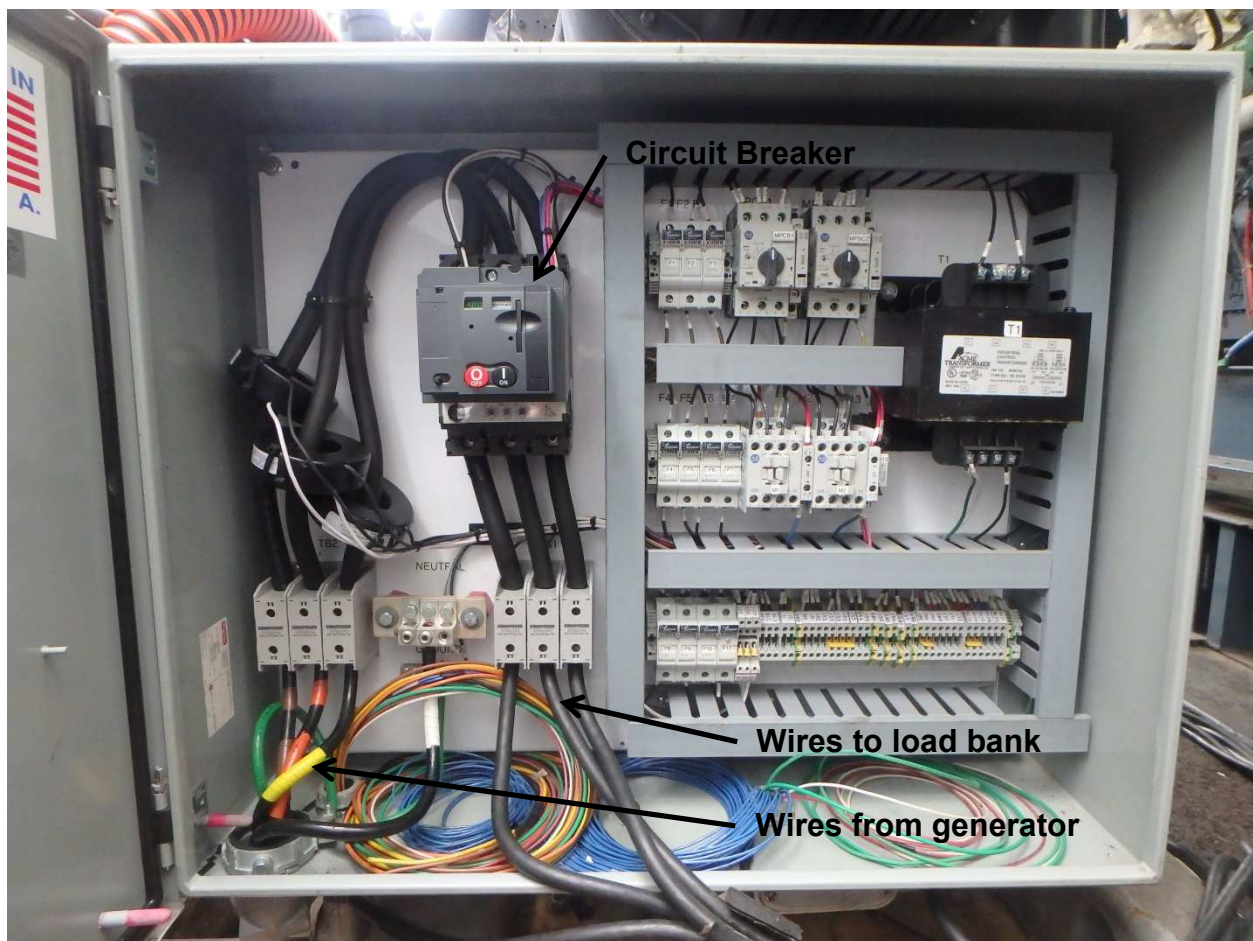


Figure 7-6. Electrical panel in which the generator is connected to the load bank via circuit breaker.

A calibrated torque wrench is used to measure friction torque. The torque wrench used was a calibrated Stanley Proto, model number J6014C with an accuracy of $\pm 3\%$ of reading. See Appendix A for copy of calibration certification.

Engine speed is measured using a Shimpo Instruments DT-725 stroboscope. The accuracy of the unit is $\pm 0.02\%$ of reading. This equates to ± 0.36 rpm in terms of engine speed. An image of the stroboscope is shown in Figure 7-7. The datasheet for the stroboscope is given in Appendix A. In addition to use of the stroboscope, frequency measurements from the Shark® 100 power meter were used to verify engine speed. The accuracy of the frequency measurement from the power meter is $\pm .01$ Hz. This equates to ± 0.6 rpm in terms of engine speed.



Figure 7-7. Image of the DT-725 stroboscope used to measure engine speed.

A Fluke® 62 Max IR Thermometer is used to measure surface temperatures. The accuracy of the meter is $\pm 3\text{ }^{\circ}\text{F}$ or $\pm 1.5\%$ of reading, whichever is greater. Therefore, for temperatures greater than $200\text{ }^{\circ}\text{F}$, the accuracy is $\pm 1.5\%$. For temperatures less than $200\text{ }^{\circ}\text{F}$, the accuracy is $\pm 3\text{ }^{\circ}\text{F}$. The specifications for this meter are given in Appendix A.

As documented, the steam power generation device being tested is the PSE. A picture of the PSE is given in Figure 7-8.



Figure 7-8. PSE.

c. PROCEDURE

i. Trial Operation And Monitoring Of Operational Parameters

Once the installation was completed the unit was operated for approximately 100 hours of run time in order to ensure all parameters are in order. After this initial trial period was completed, preparations were made for full testing. In this trial period, it was found that significant flow restrictions exist in the piping such as moisture separators and valves. These restrictions cause a pressure drop resulting in a less than expected supply pressure to the inlet of the engine. The supply pressure versus flow rate is provided in Figure 7-10. All other operating parameters seem to be in order. As a note, friction torque was noticed to be high.

ii. Friction Coefficient Determination

The first test procedure is to test the friction load on the engine in order to determine the coefficient, C_1 , in equation (4.49). To do so, a torque wrench was used to measure friction torque. The engine was warmed to a steady state temperature to best replicate operating conditions. Upon shutdown, the torque wrench was used to measure initial resistance as shown in Figure 7-9.

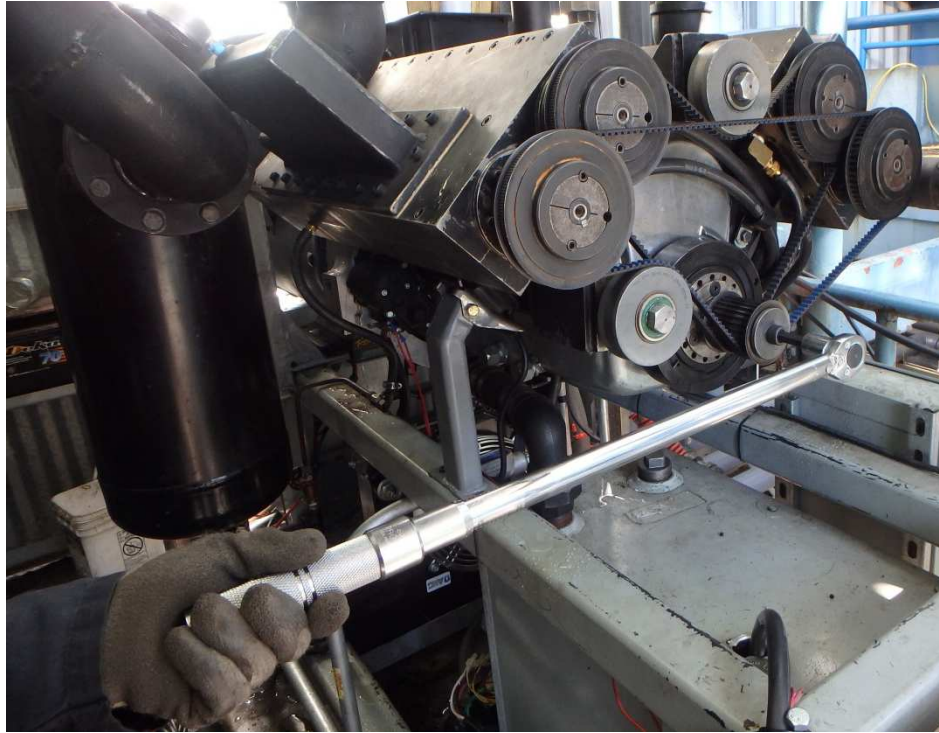


Figure 7-9. Process for measuring friction torque using a calibrated torque wrench.

This procedure was repeated multiple times to get a total of 5 measurements in order to establish a reasonable standard deviation of data points measured. Additionally, the friction torque was measured at initial break-away and post break-away as well as at different engine temperatures.

iii. Load Testing

Next, the load testing began. This procedure consisted of throttling the engine up to operating speed for a warm-up period. The surface temperature of the engine block and mass flow rate of steam were monitored until they reached a steady-state point. Once steady-state point was reaching, the engine was determined to be warmed up and

ready to test. At such point, with circuit breaker closed and generator connected to load bank, the load was increased on the load bank in 5 kWe increments from 0 kWe up to max output. At each increment, the engine throttle would be manipulated such to maintain operation at 1800 rpm. At each point, supply pressure, exhaust pressure, steam flow rate, engine speed, electrical frequency, power factor and electrical power were measured and recorded. The engine was operated at each point for no less than 2 minutes to ensure proper stabilized readings. Upon completion of all recorded data for each load point, the load was increased by 5 kWe and the recording process was repeated. After max load was reached and recorded, the load was reduced, the engine was throttled down and then the load increased back up to max output and the engine throttled up to again reach operational speed. At this point, a second recording was taken for max load. Upon completion of this recording, the load was reduced in 5 kWe increments all the way down to 0 kWe, resulting in two sets of data for each point. This entire process was repeated until the average standard deviation for each increment was less than 6%. To accomplish this, data was taken at each point five times.

d. RESULTS

i. Friction Torque Testing

The friction torque levels measured are given in Table 7-1.

Table 7-1. Recorded friction torque readings.

Reading	Breakaway Torque (ft-lbs)	Post Breakaway Torque (ft-lbs)
1	176	155
2	175	156
3	173	151
4	178	157
5	171	152
Mean	174.6	154.0
Standard Deviation	2.7	5.6
Variance 95% Confidence Interval	5.3	5.0

Furthermore, it was found that immediately after break-away friction was overcome, the torque resistance reduced to approximately 87-89% of the initial break-away torque on average. It is hypothesized that the reduction in torque is due to kinetic coefficient of friction as compared to static. Therefore, it is estimated that the post break-away torque best represents the actual friction torque although there is some doubt in this thought. Therefore, the initial break-away torque was used to determine friction torque. The mean value was found to be 154.0 ft-lbs with a standard deviation of 5.6 ft-lbs. Therefore, the resulting 95% confidence interval is 154.0 ft-lbs +/- 5.0 ft-lbs. From this, based on the mean value of friction torque, the coefficient in equation (4.47) was calculated to be 0.74. By plugging the determined coefficient into equations

(4.50). (4.48) and (4.49) the estimated friction power consumption is 49.2 kW at 1800 rpm.

ii. Supply Pressure Versus Flow

A significant correlation between mass flow rate and supply pressure has been found. Figure 7-10 shows how the supply pressure to the engine changes with flow rate. The raw data for this figure is supplied in Appendix B as well as a sample calculation for uncertainty as shown with error bars in the figure.

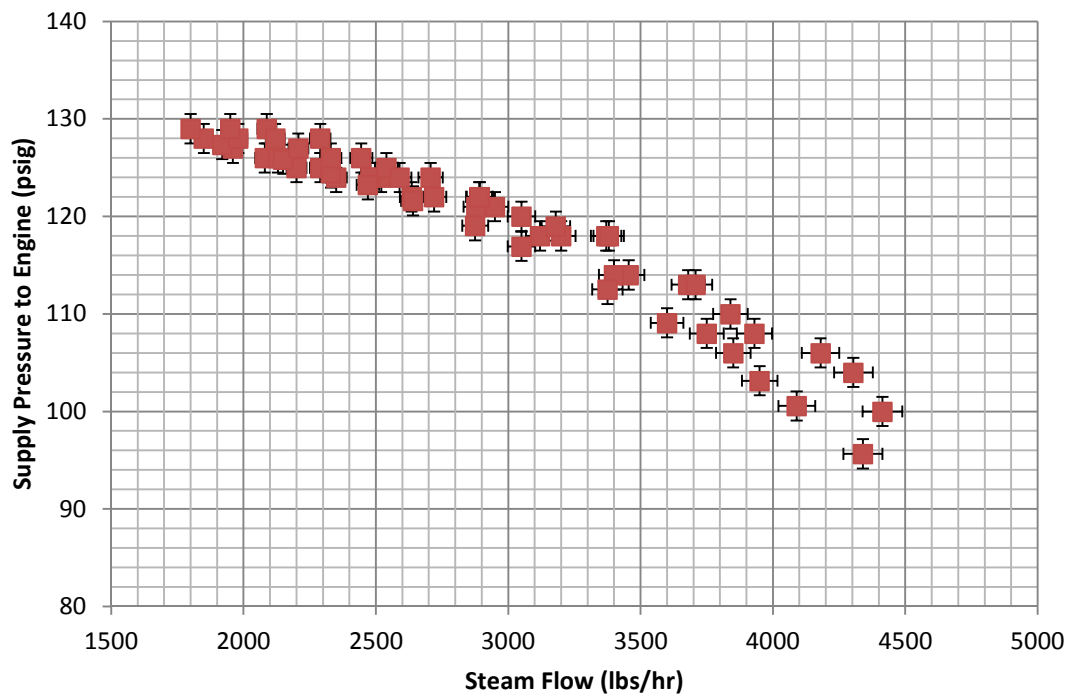


Figure 7-10. Supply pressure at the engine versus flow rate.

iii. Load Testing

Load testing has been completed and a clear trend for power output has been observed as illustrated in Figure 7-11. The corresponding efficiencies are given in

Figure 7-12. The figures include error bars according to calculated uncertainty as shown in Appendix B. The raw data for these figures is also given in Appendix B.

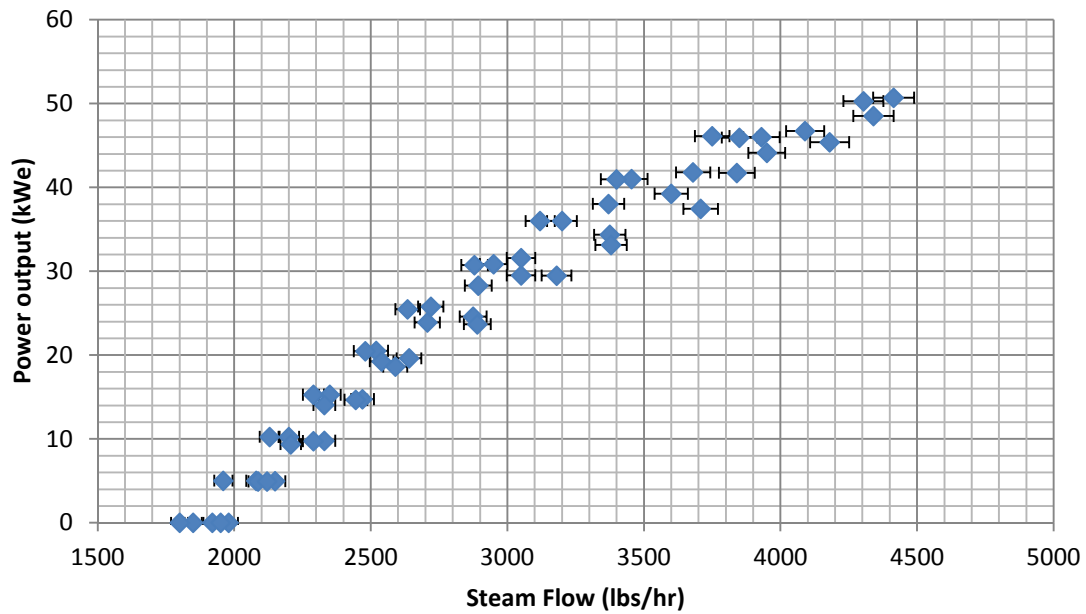


Figure 7-11. Electrical power output measured versus measured steam flow rate.

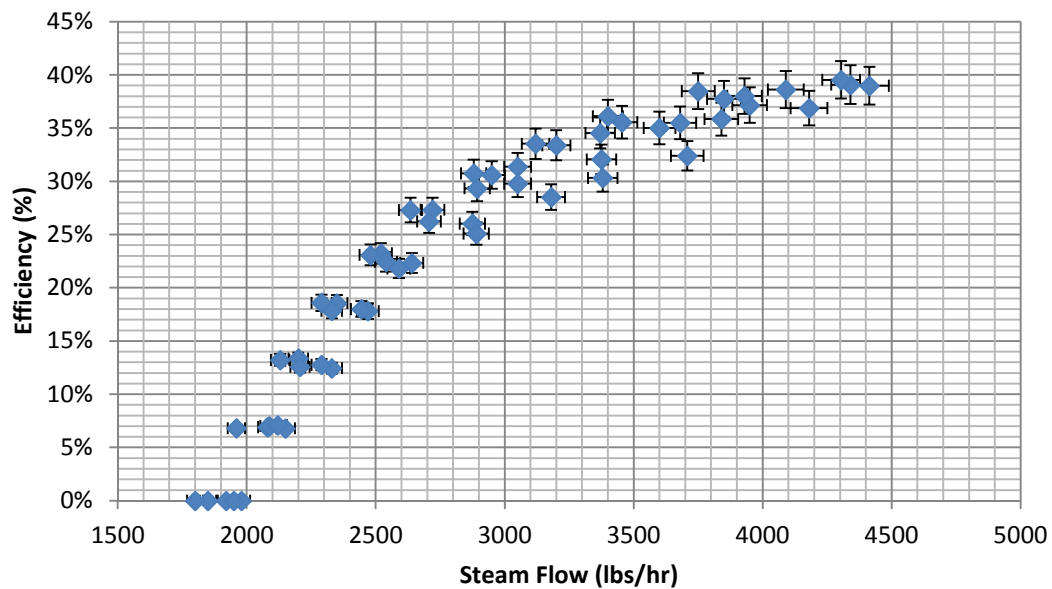


Figure 7-12. Corresponding efficiencies to the data points shown in Figure 7-11.

e. SUMMARY/COMMENTS

While the pressure drop in the engine supply piping resulted in unexpectedly reduced inlet pressures to the PSE at increase flow rates, all data showed clear trends. These trends were all supportive of conventional wisdom and conventional thinking. However, it should be noted that the calculated isentropic efficiency was extremely high, as predicted.

Furthermore, the actual measured efficiency was less than desired and it seems this was almost exclusively due to excess friction from pistons. The recorded friction was nearly 60% more than expected based on spark-ignited engine correlations. As a note, the friction of the engine when cold was found to be extremely low. This cold friction was unable to be measured as it was found to be less than the lower measuring limit of the torque wrench (min. 50 ft-lbs). This significant variation in friction with temperature change supports the suspicion that pistons are the primary source of friction.

With exception to excessive friction and pressure drop in piping to the engine, experimental results were very positive and favorable. The trend of power and efficiency was as expected.

8. EXPERIMENT CONCLUSIONS & MODEL VALIDATION

a. DISCUSSION

The experimental results found in Chapter 6 were compared and contrasted to the projections based on the analytical model as outlined in previous chapters. The first and most obvious comparison to observe is projected power output versus steam flow. The comparison versus initial projection provided in chapter 5 is given in Figure 8-1.

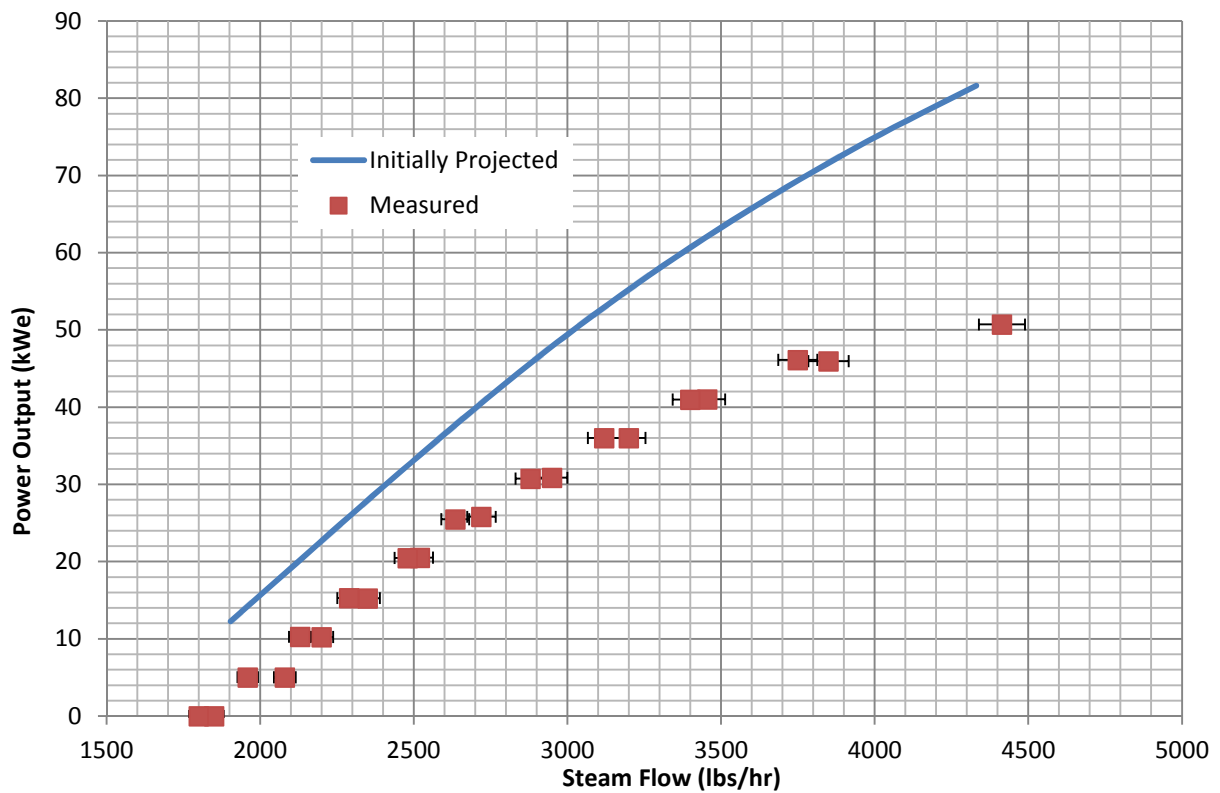


Figure 8-1. Comparison between measured power output and initial projections.

As can easily be seen from the figure, the projected power is significantly greater than actual measured power. This is for two distinct reasons. First, the projection was based on 130 psig inlet pressure. While that was the inlet pressure at 2000 lbs/hr of steam flow, the pressure drop in the pipe reduced that inlet pressure down below 105 psig at the higher end of the flow rates. Obviously, this would cause a significant error in power projections. The second reason is the underestimation of friction in the engine in the analytical model created high projections. After taking physical friction measurements, as shown in the experiment chapter, the friction model can be corrected. The analytical model was run again with inlet pressures corrected as well as the friction model modified. Figure 8-2 shows the analytical projection with corrected inlet pressures and with the friction model modified using both break-away friction and post break-away friction.

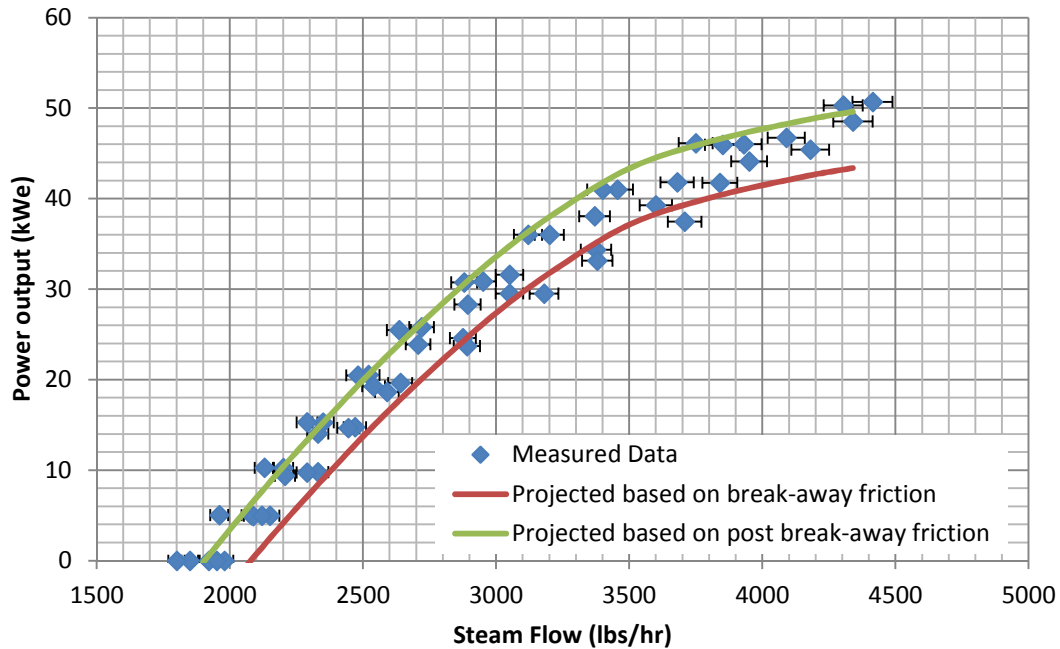


Figure 8-2 Graphical comparison of recorded data to the projected data on two different basis of friction.

As seen from the figure, the recorded data follows the general trend of the corrected projected data. Furthermore, it seems that the post break-away torque gives a more accurate representation of the friction. Additionally, it is worth pointing out the correlation found between a single set of data and that the projections. Figure 8-3 shows a tremendous correlation between the analytically projected performance based on post break-away friction torque and the performance in the last series of measurements.

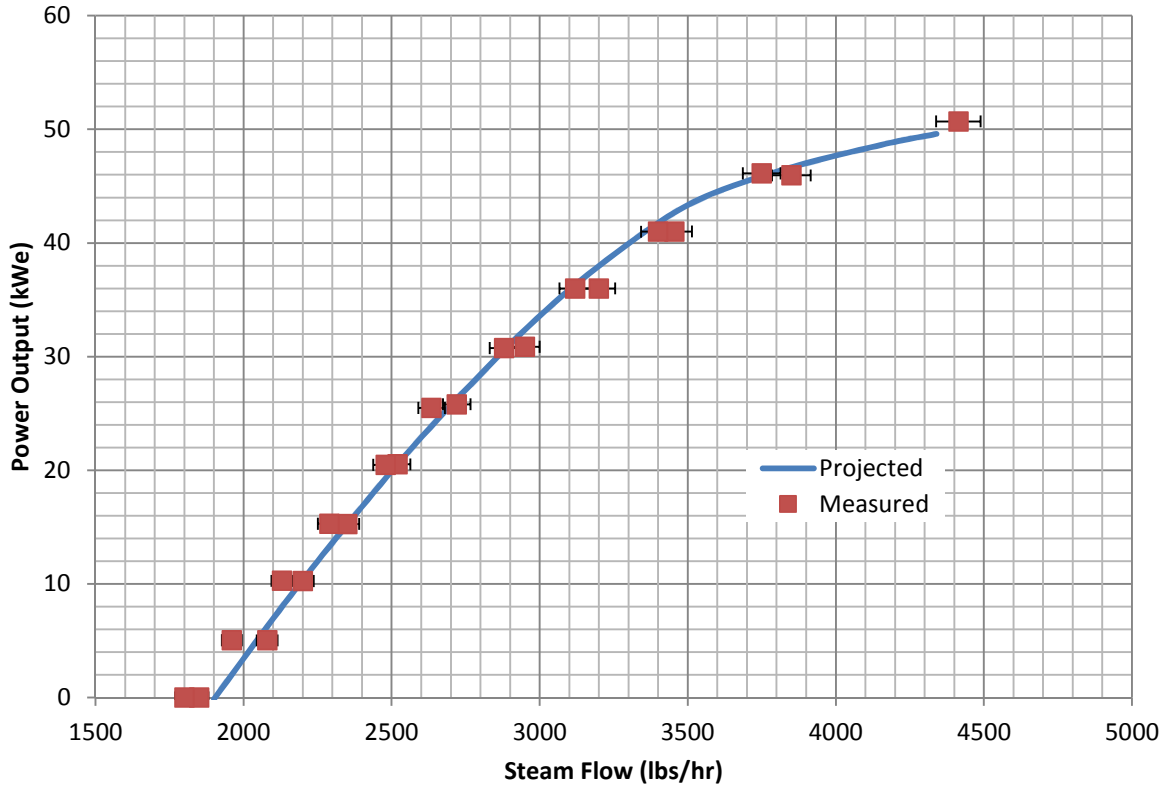


Figure 8-3. Comparison of the projected data based on post break-away friction and the last series of data points measured.

Figure 8-3 clearly illustrates that the analytical projections accurately represent the actual performance. The analytical projected data represents the experimental data well with average error of 2.35% for most flow ranges. Therefore, it can be deduced that the friction power consumption produced in the analytical model is substantially accurate. From this, the friction power consumption and the generator losses can be added to the measured power resulting in the indicated power output as shown in Figure 8-4. Error bars are shown that include uncertainty introduced from measuring equipment and from 95% confidence interval of friction torque measurement.

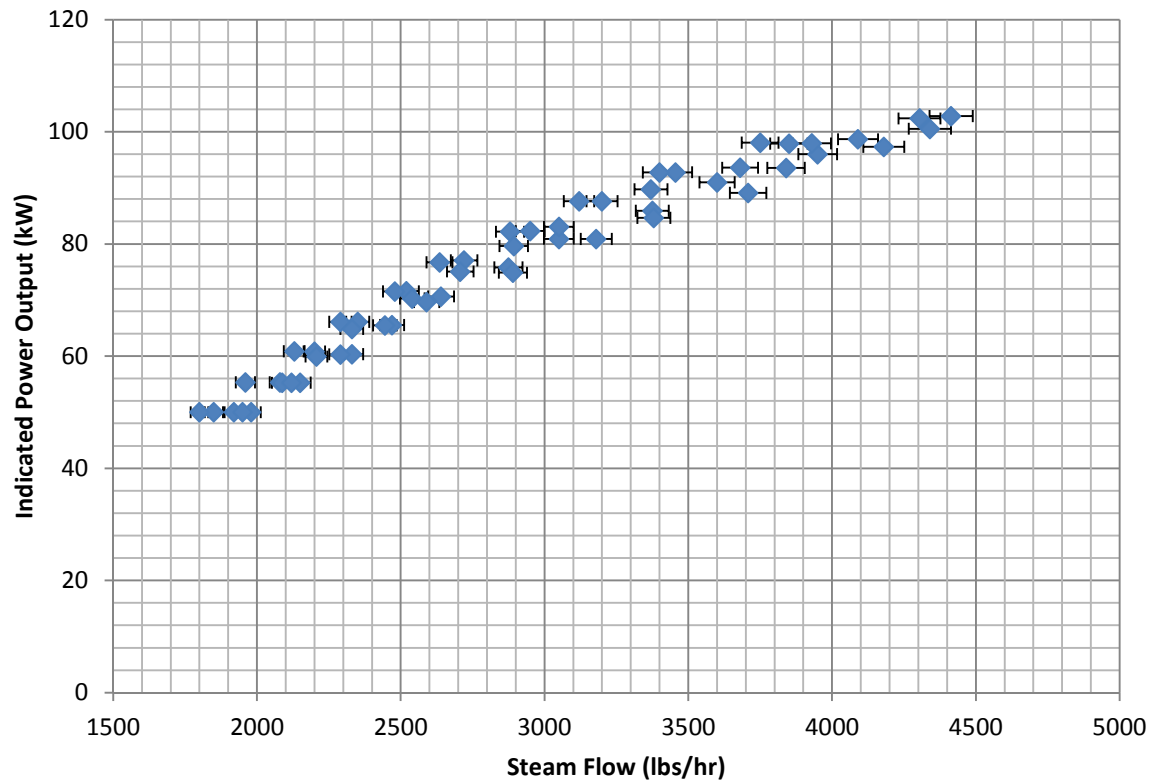


Figure 8-4. Calculated indicated power of the PSE.

These values represent the total power output of the engine without friction or generator losses. From this, the indicated specific steam consumption can be calculated as well as the isentropic efficiency. The resulting isentropic efficiency is given in Figure 8-5. This resulting isentropic efficiency is well above that for standard industrial single stage steam turbines.

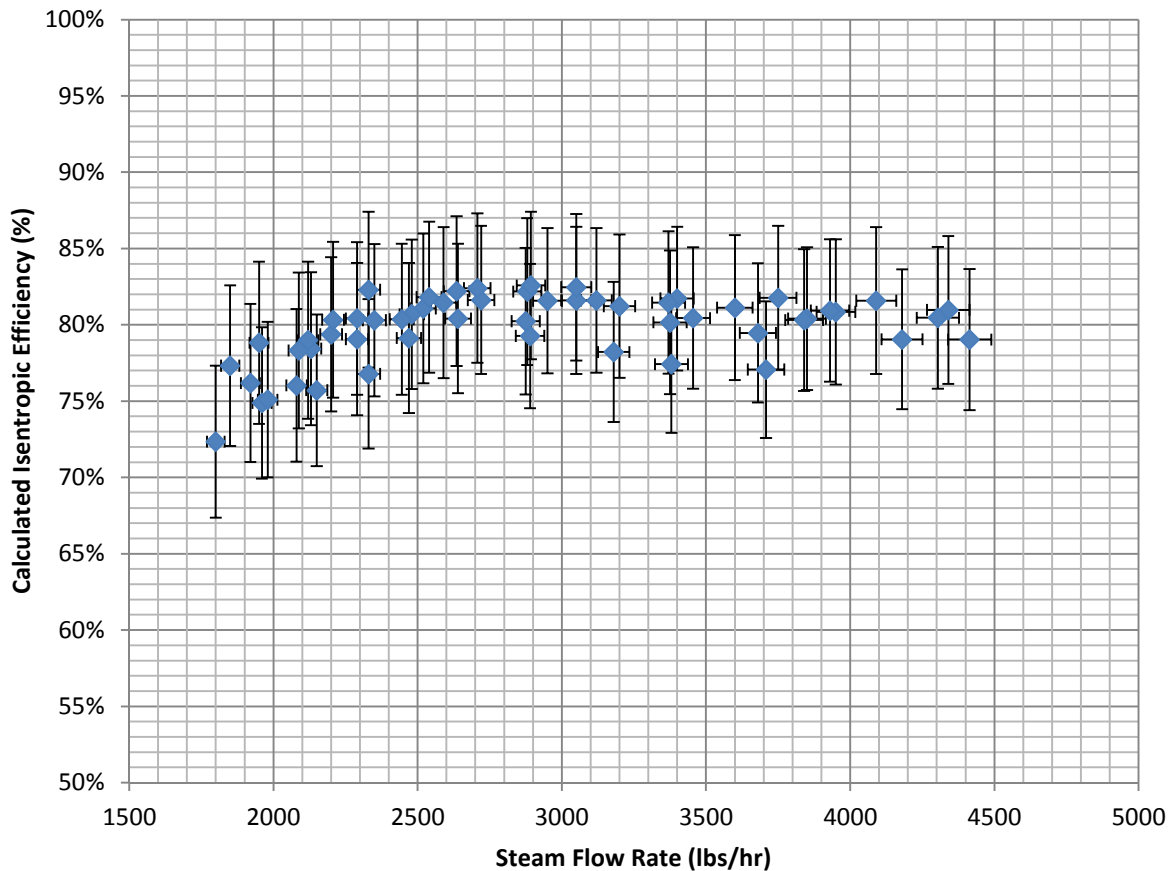


Figure 8-5. Calculated isentropic efficiency for the PSE.

From this, the power and efficiency of the PSE for applications in which the friction was closer to expected based on spark ignited engine correlations can be ascertained. This data has been calculated for the last set of data points taken and is given in

Figure 8-6.

The efficiency, assuming normal friction reaches 65%, a more than 25% increase from measured power. Additionally, the max power increases by around 70% using expected friction values. Furthermore, this increased power level does not include an increase in power due to higher inlet pressure which may be realized by revised piping.

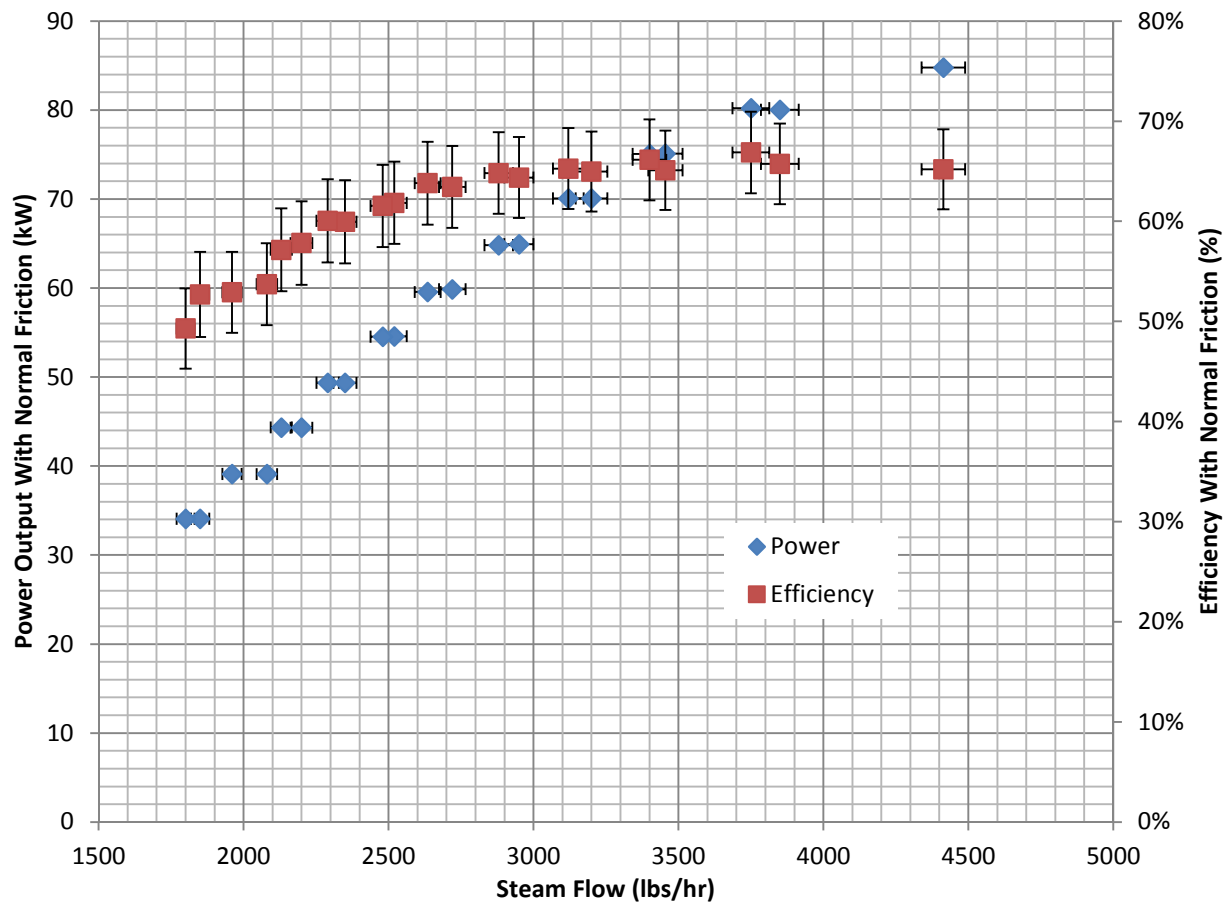


Figure 8-6. Power and efficiency of the PSE that can be expected when normal friction is present.

b. CONCLUSIONS

From the data shown, validity of the analytical model is clearly verified. Once the model was adjusted to incorporate the friction coefficient calculated from the post-breakaway torque instead of the break-away torque, recorded data closely followed projected data. Furthermore, the model allows projection of PSE performance once the friction issues are resolved. This resulting projected performance is superior to all

commercially marketed products designed to operate in applications such as the one tested.

9. FIELD APPLICATION CASE STUDY

Portions of this chapter will be presented at the International District Energy Association Annual Conference in Boston, MA in June 2015.

a. BACKGROUND

The application of backpressure steam power generation devices is common in district heating systems today. These devices are used to reduce the pressure of steam

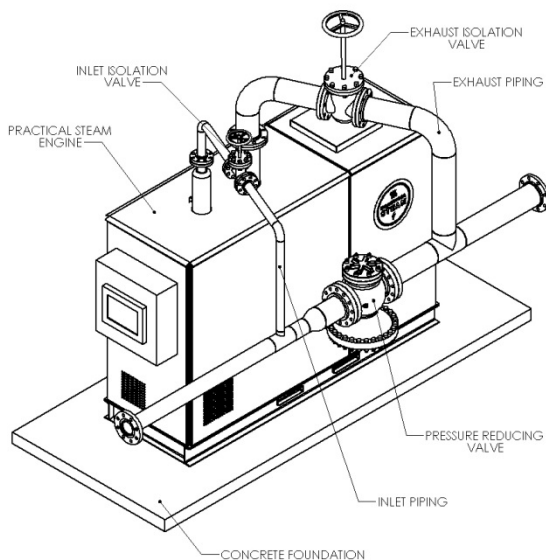


Figure 9-1. 3D representation of the system with the PSE installed in parallel to a PRV.

where needed while generating power, whether electrical or mechanical, for some useful purpose such to reduce operational costs. This pressure reduction is normally accomplished by use of a throttling valve called a pressure reducing valve (PRV). While steam power generation devices are not 100% efficient, PRVs are typically referred to as close to isenthalpic with minor inefficiencies. Therefore, it is of particular

interest to investigate the merits of use of a back pressure steam power generation device for pressure reduction as a method to reduce operational costs. The specific steam power generation device of interest for this study is the Practical Steam Engine

(PSE), a reciprocating steam engine. This paper analyzes the economic value of a system with the PSE installed in parallel to a PRV. Figure 9-1 shows a 3D representation of the system with the PSE installed in parallel to a PRV.

b. SYSTEM

The systems analyzed are illustrated in Figure 9-2. The system primarily consists of a natural gas fueled boiler producing 150 psig saturated steam. The pressure of this steam is reduced to 20 psig, either by PRV or backpressure steam power generation device. The 20 psig steam is delivered to a 1.5MW heating load from which 20 psig condensate is disposed.

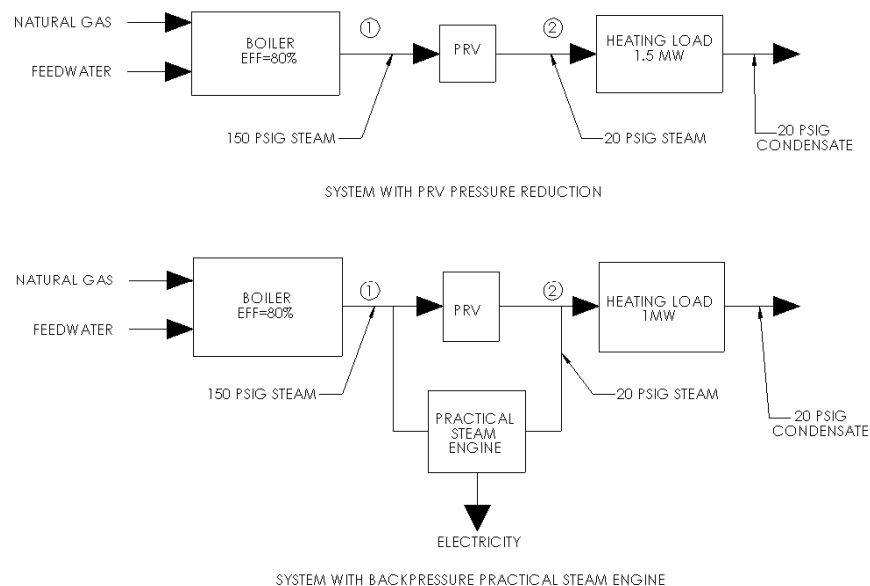


Figure 9-2. Two steam heating systems being analyzed.

c. METHOD

A thermodynamic analysis is performed such to quantify the conditions of the steam at supply to boiler, exit of the boiler, entrance to the load and exit of the load. From these conditions, mass flow rate, power generation, fuel costs and value of power generation may be calculated.

d. EQUIPMENT

- The PSE produces an electrical power output and has an average isentropic efficiency of 80%, with a mechanical efficiency of 80% and a generator efficiency of 95%¹.
- For this analysis, the characteristics of the PRV are taken as isenthalpic.
- A natural gas fuel boiler generating 150 psig saturated steam with an efficiency of 80% is considered.
- A 1.5 MW (5.118 MMBtu/hr) heating load is considered.

e. ASSUMPTIONS

- Natural gas: \$5/MMBtu²
- Electricity: \$0.075/kWh²
- Boiler feedwater: Saturated liquid at 150 psig³
- Condensate exiting load: Saturated liquid at 20 psig
- High pressure steam exiting boiler: Saturated vapor at 150 psig

¹ Efficiencies based on experimental data.

² Based on data for Seattle Steam Western Ave Plant Seattle, WA.

³ This analysis neglects heat addition required to preheat feedwater to this condition as that does not affect results of analysis.

- Operational hours: 8,500 per year
- Normalized maintenance costs of PSE: For 25 year life cycle, annual maintenance costs are approximately \$4,000 more than those for a PRV

f. ENERGY ANALYSIS

The first analysis to be performed is one considering a 1st law approach, which is an energy analysis. This analysis considers the transfer of energy throughout the system. To begin, feedwater enters the boiler as a saturated liquid at 150 psig with a corresponding enthalpy of 338.55 Btu/lbm. Steam exits the boiler as a saturated vapor at 150 psig with a corresponding enthalpy of 1,195.96 Btu/lbm. Therefore, based on a boiler efficiency of 80% the specific heat addition to the boiler is as follows:

$$Heat\ Addition = \frac{H_1 - H_{feedwater}}{\eta_{boiler}} = \frac{1,195.96 \left[\frac{Btu}{lbm} \right] - 338.55 \left[\frac{Btu}{lbm} \right]}{80\%} = 1,071.76 \left[\frac{Btu}{lbm} \right] \quad (9.1)$$

First considering the PRV system, the enthalpy of the low pressure steam is found based upon an isenthalpic pressure reduction through the PRV as follows:

$$h_{2_{PRV}} = H_1 * \eta_{PRV} = 1,195.96 \left[\frac{Btu}{lbm} \right] * 100\% = 1,195.96 \left[\frac{Btu}{lbm} \right] \quad (9.2)$$

From this result and the heat load condensate exit enthalpy of 227.51 Btu/lbm, found from the assumed condensate exit load condition, the mass flow rate of steam can be calculated as a function of heat load as shown below:

$$\dot{m}_{PRV} = \frac{Q}{H_{2PRV} - H_{condensate}} = \frac{5.118 [MMBtu]}{1,195.96 \left[\frac{Btu}{lbm} \right] - 227.51 \left[\frac{Btu}{lbm} \right]} = 5,284.9 \left[\frac{lbs}{hr} \right] \quad (9.3)$$

From this, total heat addition for PRV system can be calculated as shown:

$$\begin{aligned} Total\ Heat\ Addition_{PRV} &= Heat\ Additon * \dot{m}_{PRV} = \frac{1,071.76 \left[\frac{Btu}{lbm} \right] * 5,284.9 \left[\frac{lbs}{hr} \right]}{1,000 [MMBtu/Btu]} \\ &= 5.664 \left[\frac{MMBtu}{hr} \right] \end{aligned} \quad (9.4)$$

This heat input results in annual fuel cost as follows:

$$\begin{aligned} Annual\ Fuel\ Costs_{PRV} &= Total\ Heat\ Addition_{PRV} * Cost\ Per\ MMBtu * Hours\ of\ operation \\ &= 5.664 \left[\frac{MMBtu}{hr} \right] * 5 \left[\frac{\$}{MMBtu} \right] * 8,500 [hrs] = \$241,000 \end{aligned}$$

(9.5)

Now, considering the Practical Steam Engine, the enthalpy of the low pressure steam is found, based on the 80% isentropic efficiency of the engine, as follows:

$$\begin{aligned}
 h_{2PSE} &= H_1 - (H_1 - H_{2PSE_s}) * \eta_{isentropic} \\
 &= 1,195.96 \left[\frac{Btu}{lbm} \right] - \left(1,195.96 \left[\frac{Btu}{lbm} \right] - 1,076.72 \left[\frac{Btu}{lbm} \right] \right) * 80\% \\
 &= 1,100.57 \left[\frac{Btu}{lbm} \right]
 \end{aligned}$$

(9.6)

From this result and the heat load condensate exit enthalpy of 227.51 Btu/lbm, found from the assumed condensate exit load condition, the mass flow rate of steam can be calculated as a function of heat load as shown below:

$$\dot{m}_{PSE} = \frac{Q}{H_{2PSE} - H_{condensate}} = \frac{5.118 [MMBtu]}{1,100.57 \left[\frac{Btu}{lbm} \right] - 227.51 \left[\frac{Btu}{lbm} \right]} = 5,862.4 \left[\frac{lbs}{hr} \right]$$

(9.7)

From this, total heat addition for PRV system can be calculated as shown:

$$\begin{aligned}
Total\ Heat\ Addition_{PSE} &= Heat\ Additon * \dot{m}_{PSE} = \frac{1,071.76 \left[\frac{Btu}{lbm} \right] * 5,862.4 \left[\frac{lbs}{hr} \right]}{1,000 [MMBtu/Btu]} \\
&= 6.283 \left[\frac{MMBtu}{hr} \right]
\end{aligned}
\tag{9.8}$$

This heat input results in annual fuel cost as follows:

$$\begin{aligned}
Annual\ Fuel\ Costs_{PSE} \\
&= Total\ Heat\ Addition_{PSE} * Cost\ Per\ MMBtu * Hours\ of\ operation \\
&= 6.283 \left[\frac{MMBtu}{hr} \right] * 5 \left[\frac{\$}{MMBtu} \right] * 8,500 [hrs] = \$267,000
\end{aligned}
\tag{9.9}$$

Power generation is calculated using mass flow rate and known value of low pressure steam enthalpy when considering 80% mechanical efficiency of the Practical Steam Engine and the 95% generator efficiency. The calculation is as follows:

$$\begin{aligned}
\dot{W}_{PSE} &= (H_1 - H_{2PSE}) * \dot{m}_{PSE} * \eta_{mech} * \eta_{generator} \\
&= \frac{1,195.96 \left[\frac{Btu}{lbm} \right] - 1,100.57 \left[\frac{Btu}{lbm} \right]}{3,412.14 \left[\frac{kW}{Btu/hr} \right]} * 5,862.4 \left[\frac{lbs}{hr} \right] * 80\% * 95\% \\
&= 125.4[kW]
\end{aligned}
\tag{9.10}$$

The value of the power generation is easily calculated from power output if the system hours of operation and power cost are known:

$$\begin{aligned}
\text{Value of Annual Power Production} &= \dot{W}_{PSE} * \text{Hours of operation} * \text{Cost per kWh} \\
&= 125.4[kW] * 8,500[hrs] * 0.075 \left[\frac{\$}{kWh} \right] = \$80,000
\end{aligned}
\tag{9.11}$$

From this the annual savings by use of a backpressure Practical Steam Engine in lieu of a PRV are calculated as follows:

Annual Savings

$$\begin{aligned} &= \text{Annual Fuel Costs}_{PRV} \\ &- (\text{Annual Fuel Costs}_{PSE} - \text{Value of Annual Power Production}) \\ &- \text{Annual Maintenance Costs} \\ &= \$241,000 - (\$267,000 - \$80,000) - \$4,000 = \$50,000 \end{aligned}$$

(9.12)

g. EXERGY ANALYSIS

The energy analysis method above is the most commonly used method in today's engineering practice to determine the validity of energy efficiency products and applications. However, a 2nd law analysis, that is an analysis of the exergy of the system, may prove to provide more valuable insight into where work potential is being lost.

Exergy, the ability to do work, is often destroyed when energy is lost. Consider the PRV. The energy analysis above suggests that all energy is conserved across the valve. While this is technically true based upon an isenthalpic assumption, a 2nd law analysis would suggest some ability to do work was lost across the PRV. Exergy of a fluid, neglecting kinetic and potential energy, can be calculated as a function of enthalpy and entropy as given below:

$$\psi = (h - h_0) - T_0(s - s_0) \quad (8.13)$$

Where:

h is the current state enthalpy of the fluid

h_0 is dead state enthalpy of the fluid

T_0 is dead state temperature

s is the current state entropy of the fluid

s_0 is the dead state entropy of the fluid

Consider the PRV energy analysis above. The exergy coming into the PRV is 369.4 Btu/lbm while all exergy leaving the PRV in form of steam is 282.2 Btu/lbm. This is an exergy destruction of 23.6% and a resulting 2nd law efficiency of 76.4%. This means that the steam produced by the boiler has lost 23.6% of its ability to perform useful work.

Now consider the PSE. The exergy coming into the PRV is the same 369.4 Btu/lbm while all exergy leaving the PRV in form of steam is 256.4 Btu/lbm. However, an additional amount of work was produced at a rate of 73.0 Btu/lbm bringing the total outlet exergy to 329.4 Btu/lbm. This results in an exergy destruction of 10.8% and a resulting 2nd law efficiency of 89.2%. This means that only 10.8% of the steam's ability to do work was lost through the PSE compared to 23.6% through the PRV.

h. SUMMARY/REMARKS

The study clearly illustrates an example of the potential savings realized by use of a backpressure PSE in parallel to a PRV for district energy applications. While using the PSE resulted in a 10.9% increase in the overall heat input, and thus fuel cost, this was more than offset by electrical power generation. Using the PSE reduced annual operational costs for this system by 20.7%, a \$50,000 annual savings. Furthermore, the 2nd law analysis showed that the PSE better conserved the steam's ability to do work than did the PRV.

Special consideration should be given to the fact that this analysis considers electrical rates in Seattle, WA while many places have rates much greater. For example, consider New York City where electrical rates average \$0.19/kWh. In this setting, the exact same engine installation discussed here would yield annual savings of \$127,000 in NYC, as compared to \$50,000 in Seattle, and reduce the annual operational costs of the system by 52.6%. The resulting return on investment (ROI) in NYC would be less than 100%, yielding payback periods less than one year.

Additionally, this study did not consider value added from green incentives, tax credits, backup power capability, etc. With these factors considered, further savings may be realized. Therefore, the PSE proves to be a viable economic solution for a pressure reducing station for district energy.

10. CONCLUSIONS

a. SUMMARY/REMARKS

This study has provided valuable information regarding the history, development, modeling, testing, validation and case study of the PSE. A literature review was performed in order to best understand what research has been performed and what gaps exist in the research. A technical history of steam engine operation was given as well as specific discussion of design focus for the PSE. The basis for an analytical model to assist in design of the PSE, as well as predict performance, has been presented. Utilization of this model to predict performance in an experimental setting was shown and projected data was provided. A detailed experimental study of an operational PSE to measure performance and compare to analytically modeled data was outline and results presented. The results of the experimental study were compared to analytical data to validate the model as clearly demonstrated. Finally, using a combination of experimental and analytical data, a case study of a commercial application of the PSE was analyzed resulting in demonstration of economic benefit of the PSE in a district energy application.

The summation of this study suggests that (1) an analytical model can be created to accurately predict the performance of the PSE, (2) the PSE can be experimentally operated and successfully demonstrate the ability to perform, and (3) there is economic viability to commercial applications of the PSE.

However, the results of this study create several questions. A few specific questions include: what can be done to reduce friction in the PSE, how can we accurately model the friction, how can we improve the model in terms of fluid flow and heat transfer, and what other practical improvements can be made to the PSE? With these questions in mind, suggestions for future work are made.

b. SUGGESTIONS FOR FUTURE WORK

The research here has shown much promise for the applicability of a PSE in practical applications. However, this research has raised several questions and areas on which to be focused.

First, the PSE was recorded to have significantly more friction than a standard spark-ignited engine. While it is well understood that testing of a new form of abradable piston coating significantly to this excess friction, it is not well understood what other contributions exist. This leads to a future area of work to include proper derivation of friction correlations for the PSE.

Second, related to the first topic, is to determine methods to reduce engine friction while not compromising lubrication and functionality of the PSE. Use of the abradable piston coating was intended to reduce friction and improve lubricity between surfaces in which steam may have compromised standard oil lubrication. Research is needed to find a better solution to this problem. Improvement in this area would allow the PSE efficiency to approach the calculated isentropic efficiency.

Third, while not mentioned in this research, it was found that cold startup of the PSE can be problematic due to excess condensation formation in the cylinder. It has been found that the exhaust valve timing should be significantly retarded during startup

to prevent compression of condensate in the cylinder and any resulting damage and failure. However, finding an alternative solution, or a method of using automated processes to retard exhaust timing during warm-up and advance it back for standard operation is highly desirable.

Forth, while possibly related to the third topic, the use of steam jacketing should be researched. Use of this process prior to startup could potentially reduce or eliminate warm-up valve timing requirements. Additionally, this process could reduce the condensation formation in the cylinder and therefore cylinder wall cooling.

Fifth, while not specifically mentioned in this research, it has been determined that a significant area of difficulty is separation of condensate out of the lubrication oil and oil out of the exhaust steam. The condensate in the oil impairs the lubricating properties of the oil and therefore creates risk of failure to the engine. The oil in the exhaust steam presents environmental issues as well as additional operating expense and possible risk to the downstream steam system. These areas of research could require significant effort but come with a worthy goal.

Sixth, as an expansion on Chapter 4, fluid flow modeling to understand how the flow deviates from isentropic flow through the valves could be quite valuable research. Not only could this help better describe the basic gas exchange process, but it could also better define valve leakage.

Seventh, as a specific alteration of Chapter 4, consideration of a time dependent wall temperature profile could be considered. The analysis presented here assumes a constant wall temperature profile throughout the cycle. While it is expected that this

assumption will yield similar results to one with a time dependent profile, this is an area to be considered.

Eighth, as another expansion on Chapter 4, research to determine effects from non-homogenous cylinder conditions may prove to be valuable. The research presented here assumes the steam conditions in the cylinder are homogenous. However, it is suspected that the conditions of the steam close to the cylinder wall may differ from those near the center of volume of the cylinder. Further, velocity fields inside the cylinder may significantly affect heat transfer rates.

Lastly, in general, the research here could be expanded to include other types of applications of the PSE namely condensing applications (e.g. biomass, biogas, waste heat recovery, waste steam recovery, etc.) and cogeneration/trigeneration applications. Furthermore, the analysis could be broadened to consider the additional equipment of the system (e.g. gasifiers, boilers, heat exchangers, boiler feed systems, etc.) in order to understand the practicality, efficiency and impact of an entire system implementing the PSE.

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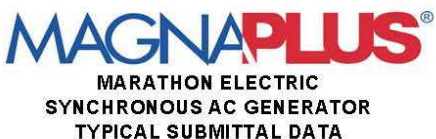
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APPENDIX A – EQUIPMENT DATASHEETS

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a. MARATHON® MAGNAPLUS® 363CSL1607 GENERATOR DATASHEET



Basic Model: 363CSL1607/363PSL1607

Date: 3/28/13

Kilowatt ratings at		1800 RPM		60 Hertz		12 Leads			
kW (kVA)		3 Phase		0.8 Power Factor			Dripproof or Open Enclosure		
	Class B	Class F					Class H		
Voltage*	80° C ①	90° C ①	95° C ①	105° C †	105° C ①	130° C ①	125° C †	125° C ①	150° C ①
	Continuous	Lloyds	ABS	British Standard	Continuous	Standby	British Standard	Continuous	Standby
240/480	110 (138)	115 (144)	115 (144)	125 (156)	125 (156)	135 (169)	135 (169)	135 (169)	150 (188)
230/460	110 (138)	115 (144)	115 (144)	125 (156)	125 (156)	135 (169)	135 (169)	135 (169)	150 (188)
220/440	105 (131)	110 (138)	110 (138)	115 (144)	115 (144)	130 (163)	130 (163)	130 (163)	140 (175)
208/416	100 (125)	105 (131)	105 (131)	110 (138)	110 (138)	125 (156)	125 (156)	125 (156)	135 (169)
190/380	90 (113)	95 (118)	95 (118)	100 (125)	100 (125)	115 (144)	115 (144)	115 (144)	125 (156)

☉ Rise by resistance method, Mil-Std-705, Method 680.1b.

† Rating per BS 5000.

Submittal Data: 480 Volts, 169 kVA, 1800 RPM, 60 Hz, 3 Phase					
Mil-Std-705B			Mil-Std-705B		
Method	Description	Value	Method	Description	Value
301.1b	Insulation Resistance	> 1.5 Meg	505.3b	Overspeed	2250 RPM
302.1a	High Potential Test		507.1c	Phase Sequence CCW-ODE	ABC
	Main Stator	2000 Volts	601.4a	L-L Harmonic Maximum - Total (Distortion Factor)	3.5%
	Main Rotor	1500 Volts	601.4a	L-L Harmonic Maximum - Single	2.5%
	Exciter Stator	1500 Volts	601.1c	Deviation Factor	7.0%
	Exciter Rotor	1500 Volts	---	TIF (1960 Weightings)	<50
401.1a	Stator Resistance, Line to Line		Additional Prototype Mil-Std Methods are Available on Request.		
	High Wye Connection	0.0692 Ohms			
	Rotor Resistance	1.365 Ohms			
	Exciter Stator	23.5 Ohms			
	Exciter Rotor	0.12 Ohms			
410.1a	No Load Exciter Field Amps at 480 Volts Line to Line	0.52 A DC	--	Generator Frame	360
420.1a	Short Circuit Ratio	0.485	--	Type	Ext. Voltage Regulated, Brushless
421.1a	Xd Synchronous Reactance	2.41 pu	--	Insulation	Class H
422.1a	X2 Negative Sequence Reactance	0.108 pu	--	Coupling - Single Bearing	Flexible
423.1a	X0 Zero Sequence Reactance	0.034 pu	--	Amortisseur Windings	Full
425.1a	X'd Transient Reactance	0.152 pu	--	Cooling Air Volume	700 CFM
426.1a	X''d Subtransient Reactance	0.108 pu	--	Exciter	Rotating
427.1a	T'd Transient Short Circuit Time Constant	0.06 sec.	--	Voltage Regulator	SE350
428.1a	T''d Subtransient Short Circuit Time Constant	0.008 sec.	--	Voltage Regulation	1%
430.1a	T'do Transient Open Circuit Time Constant	0.8 sec.			
432.1a	Ta Short Circuit Time Constant of Armature Winding	0.02 sec.			

* Voltage refers to wye (star) connection, unless otherwise specified.

www.marathonelectric.com

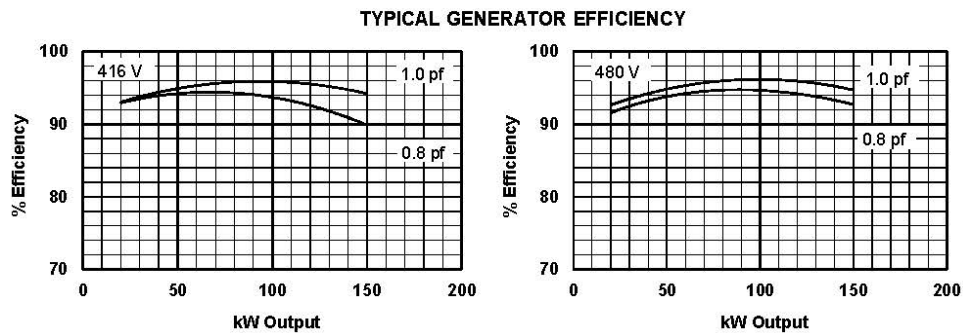
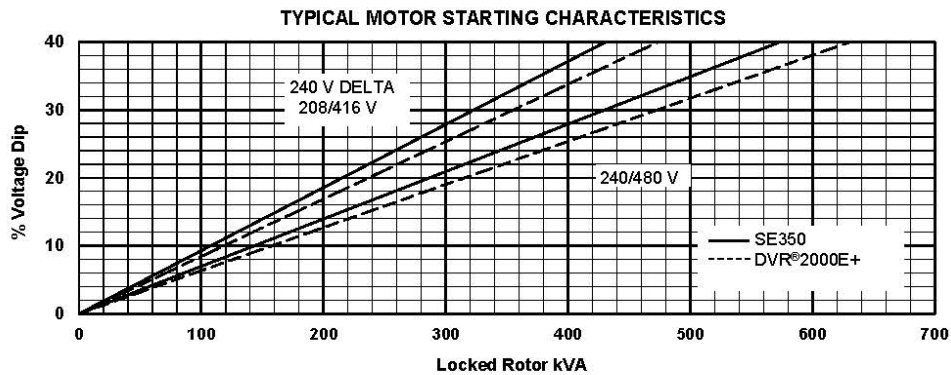
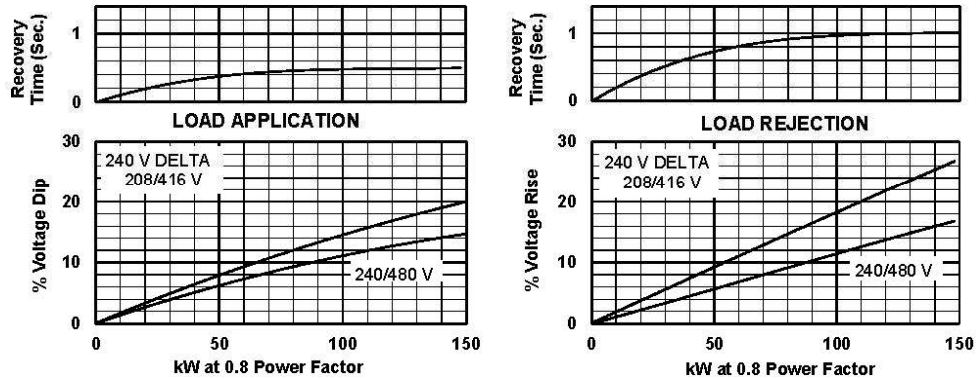


**MARATHON ELECTRIC
SYNCHRONOUS AC GENERATOR
TYPICAL DYNAMIC CHARACTERISTICS**

Basic Model: 363CSL1607/363PSL1607

Date: 3/28/13

60 HERTZ



Voltage refers to wye (star) connection, unless otherwise specified.

www.marathonelectric.com

b. STANLEY PROTO J6104C TORQUE WRENCH CALIBRATION CERTIFICATION

Stanley Proto Industrial Tools™					
Torque Wrench Calibration Form					
This Torque Wrench, Model # <u>J6014C</u> Serial # <u>DPK 57397</u>					
has been calibrated with test equipment with a certified accuracy of $\pm 1/2\%$, traceable to the United States National Institute of Standards and Technology. This torque wrench has been found to conform to the accuracy requirements of the American National Standard, ASME B107.14M-1994.					
Accuracy $\pm 3\%$ Clockwise			Accuracy $\pm 6\%$ Counter Clockwise		
Capacity of Wrench	Wrench Setting	Average Reading	Capacity of Wrench	Wrench Setting	Average Reading
20%	<u>50</u>	<u>50.51</u>	20%	<u>50</u>	<u>50.43</u>
60%	<u>150</u>	<u>152.7</u>	60%	<u>150</u>	<u>150.7</u>
100%	<u>250</u>	<u>253.6</u>	100%	<u>250</u>	<u>251.2</u>
Authorized Repair Center			Inspector: <u>LLE</u>		
Stanley Proto Industrial Tools			Date: <u>12/19/14</u> Station No. <u>2076</u>		
2195 East View Parkway, Suite 103			Customer "In Service Date": _____		
Conyers, GA 30013			Calibration Due Date: _____		
FORM X6016CAL REV C					

c. SHARK® 100 POWER METER ACCURACY

2: Meter Overview and Specifications

2.4: Accuracy

For 23° C, 3 Phase balanced Wye or Delta load, at 50 or 60 Hz (as per order), 5A (Class 10) nominal unit (see next page for Shark® 50 accuracy chart):

Parameter	Accuracy	Accuracy Input Range
Voltage L-N [V]	0.1% of reading ²	(69 to 480)V
Voltage L-L [V]	0.1% of reading	(120 to 600)V
Current Phase [A]	0.1% of reading ¹	(0.15 to 5)A
Current Neutral (calculated) [A]	2.0% of Full Scale ¹	(0.15 to 5)A @ (45 to 65)Hz
Active Power Total [W]	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0.5 to 1) lag/lead PF
Active Energy Total [Wh]	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0.5 to 1) lag/lead PF
Reactive Power Total [VAR]	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0 to 0.8) lag/lead PF
Reactive Energy Total [VARh]	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0 to 0.8) lag/lead PF
Apparent Power Total [VA]	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0.5 to 1) lag/lead PF
Apparent Energy Total [VAh]	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0.5 to 1) lag/lead PF
Power Factor	0.2% of reading ^{1,2}	(0.15 to 5)A @ (69 to 480)V @ +/- (0.5 to 1) lag/lead PF
Frequency	+/- 0.01Hz	(45 to 65)Hz
Total Harmonic Distortion (%)	5.0% ¹	(0.5 to 10)A or (69 to 480)V, measurement range (1 to 99.99)%
Load Bar	+/- 1 segment ¹	(0.005 to 6)A

¹ For 2.5 element programmed units, degrade accuracy by an additional 0.5% of reading.

- For 1A (Class 2) Nominal, degrade accuracy by an additional 0.5% of reading.
- For 1A (Class 2) Nominal, the input current range for Accuracy specification is 20% of the values listed in the table.

d. SHIMPO INSTRUMENTS DT-725 STROBOSCOPE DATASHEET

Bulletin TK009-721

CHECK·LINE®
BY ELECTROMATIC

Low Cost STROBOSCOPE

DT-721 (AC-Powered)

DT-725 (Battery-Powered)

40 – 12,500 RPM/FPM

and higher using harmonics

The CHECK·LINE® DT-721 and DT-725 Digital Stroboscopes are highly stable, industrial-quality instruments for non-contact RPM measurement as well as for inspection and observation of moving parts by “freezing” or “slowing down” the action.

A “Kit” option is offered for both models which also includes a carrying case and spare flash tube at minimal additional cost.



Optional Complete Kit



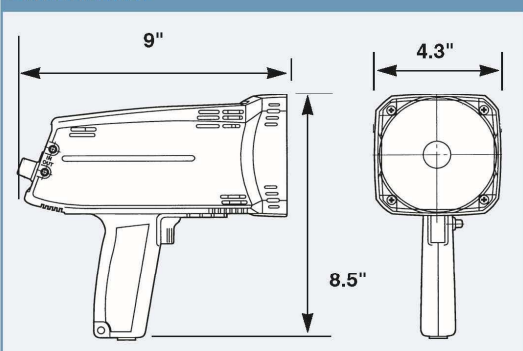
FEATURES

- AC or battery-powered models
- Highly accurate speed control with drift-free operation
- Bright flash intensity with life of 100 million flashes
- Easy-to-use controls with bright LED display
- Instant *double* (2X) and *half-rate* ($\div 2$) flash selection
- Accepts input from sensor for external flash triggering
- Phase shift adjustment permits movement of objects to desired position for convenient observation
- Extremely lightweight and rugged

over ➡

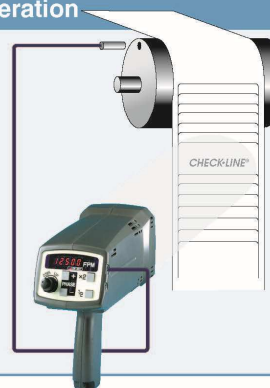
DT-721/725

Dimensions



External Trigger Operation

Suitable for applications requiring external control of the flash rate using a sensor or other input device. An external power source may be required.



Specifications

Flash Range	40 – 12,500 FPM
Accuracy	± 0.02% of reading
Resolution	(0.1) 40.0 – 4,999.9 (0.2) 5,000.0 – 7,999.8 (0.5) 8,000.0 – 9,999.5 (1.0) 10,000.0 – 12,500.0
Display	5-Digit LED, 8 mm high
Flash Tube	Xenon, 10 watt
Tube Life	100 million flashes
Flash Duration	10 – 15 µs
Signal Output	NPN Open Collector, 24 VDC max. at 50 mA (150 µs typical)
Phase Shift	0 – 359° (Internal Mode)

Duty Cycle	Continuous
Battery Life	1 hour (Model 725 only)
Re-charging Time	12 hours (4 hours with optional Fast Charger)
Low Battery Indicator	Model 725 only
Operating Temperature	32 to 105 °F (0 to 40 °C)
Weight	Model 721 1.50 lbs. (680 g) Model 725 2.75 lbs. (1.25 Kg)
Power Requirements	Model 721 115 VAC ± 10%, 50/60 Hz Model 725 Internal Battery (Lead Acid Rechargeable)

Specifications subject to change without notice.

External Trigger Input Signal

- a) 12 – 24 VDC thru a 3-wire NPN sensor (requires external power supply)
- b) 12 VDC thru a 2-wire proximity sensor (Leakage Current 1 mA max. Load current 8 mA min.)
- c) Switch or Relay Contact

Ordering Guide

For use in countries with 115 VAC	DT-721	For use in countries with 230 VAC	DT-721-2
	DT-725		DT-725-2
	DT-721-KIT		DT-721-2-KIT
	DT-725-KIT		DT-725-2-KIT

CHECK·LINE® – PRECISION QUALITY CONTROL INSTRUMENTS

Electromatic Equipment Co., Inc.
600 Oakland Ave.
Cedarhurst, N Y 11516 —USA

Tel: (800) 645-4330 (USA & Canada)
Tel: (516) 295-4300
Fax: (516) 295-4399

Email: info@checkline.com
Website: www.checkline.com

FOR ADDITIONAL INFORMATION OR TO PLACE AN ORDER CALL TOLL FREE 1-800-645-4330

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Printed in U.S.A.

e. FLUKE 62 MAX IR THERMOMETER SPECIFICATIONS

Infrared Thermometer Specifications

Specifications

	62 MAX	62 MAX +
Temperature Range	-30 °C to 500 °C (-22 °F to 932 °F)	-30 °C to 650 °C (-22 °F to 1202 °F)
Accuracy (Calibration geometry with ambient temperature 23 °C ±2 °C)	≥0 °C: ±1.5 °C or ±1.5 % of reading, whichever is greater (≥32 °F: ±3 °F or ±1.5 % of reading, whichever is greater) ≥ -10 °C to <0 °C: ±2 °C (≥14 °F to <32 °F: ±4 °F) < -10 °C: ±3 °C (<14 °F: ±6 °F)	≥0 °C: ±1 °C or ±1 % of reading, whichever is greater (≥32 °F: ±2 °F or ±1 % of reading, whichever is greater) ≥ -10 °C to <0 °C: ±2 °C (≥14 °F to <32 °F: ±4 °F) < -10 °C: ±3 °C (<14 °F: ±6 °F)
Response Time (95 %)	<500 ms (95 % of reading)	<300 ms (95 % of reading)
Spectral Response	8 to 14 microns	
Emissivity	0.10 to 1.00	
Temperature Coefficient	±0.1 °C/°C or ±0.1 %/°C of reading (whichever is greater)	

7

62 MAX/62 MAX + Users Manual

Optical Resolution	10:1 (calculated at 90 % energy)	12:1 (calculated at 90 % energy)
Display Resolution	0.1 °C (0.2 °F)	
Repeatability (% of reading)	±0.8 % of reading or ±1.0 °C (2 °F), whichever is greater	±0.5 % of reading or ±0.5 °C (1 °F), whichever is greater
Power	1 AA IEC LR06 Battery	
Battery Life	10 hours with laser and backlight on	8 hours with laser and backlight on
Weight	255 g (8.99 oz)	
Size	(175 x 85 x 75) mm (6.88 x 3.34 x 2.95) inches	
Operating Temperature	0 °C to 50 °C (32 °F to 122 °F)	
Storage Temperature	-20 °C to 60 °C (-4 °F to 140 °F), (without battery)	
Operating Humidity	10 % to 90 % RH non-condensing @ 30 °C (86 °F)	
Operating Altitude	2000 meters above mean sea level	
Storage Altitude	12,000 meters above mean sea level	

8

f. AVTRON FREEDOM LOAD BANK DATASHEET

Avtron LPH100

Compact, Portable Load Bank - 100 KW

Testing Your Emergency Standby Generator with a Load Bank Means Reliability When it Matters Most.

Key Features

- Rated for Continuous Operation
- Compact, Portable, Easy to Transport and Operate
- Large Load Capacity - up to 100 KW
- Lightweight Aluminum and Aluminized Steel Enclosure with Casters
- Convenient Operator Controls with Digital Metering and Data Logging Software included

Superior Avtron quality combined with simple operation and maintenance will provide years of trouble free service.

Avtron's extensive line of Load Bank and Industrial Resistor Products are solid performers used throughout the world.

For total technical support or additional information, please contact Avtron at (216) 573-7600 or LBsales@Emerson.com.

Quality System Certified to ISO 9001

The Avtron Model LPH100 Load Bank is a portable, self-contained AC resistive model designed for indoor use.

The Avtron LPH100 Load Bank gives users the freedom of a high capacity load in an ultra-compact package. The LPH100 is ideal for routine testing and maintenance of AC power sources such as stand-by-emergency generators and UPS systems. The load bank is rated at 100 KW at 480 VAC and 93.75 KW at 240 VAC at 3 phase, 60 Hz.

This versatile load bank weighs only 105 lbs. [48 kg] and is less than half the weight and size of traditional load banks of similar capacity. The load bank features horizontal air discharge to provide a low profile design. Quick-connect load terminals and simple operation allow the load bank to be set up and operated by a single technician, saving valuable time in the field.

The load bank controls and cooling fans are powered by any standard 120 VAC, single phase wall outlet with detachable line cord provided. An integral control panel is provided for load bank operation and includes the necessary controls and status indicators.

The LPH100 Load Bank features the standard in Digital Metering which includes the Avtron Advanced Digital Monitoring System (ADMS™). The ADMS meter provides faceplate digital display of Volts, Amps, Frequency, and KW. The Communicator EXT™ Data Logging Software allows Real Time Data Acquisition and trending of all meter parameters from your Laptop PC or PDA. An Infrared/USB interface cable is included for "Plug and Play" convenience.



LPH100 100 KW
Compact, Portable
Avtron Load Bank.

AVTRON


EMERSON
Network Power

Avtron LPH100

Compact, Portable Load Bank - 100 KW

Specifications subject to change without notice.

Construction

A compact aluminized steel enclosure with screened inlet and exhaust openings. Four swivel casters provide mobility. Heavy duty lifting handles are also provided.

Control Power

The load bank requires control power of 120 VAC, single phase, 60 Hz (line cord provided).

Cooling

The resistive elements are air cooled by four high velocity muffin fans for quiet operation.

Controls

The operator control panel includes a MASTER LOAD ON-OFF switch, load step ON-OFF industrial toggle switches, a BLOWER ON-OFF switch, and a LOAD VOLTAGE selector switch (Dual Voltage model only).

Digital Metering

Avtron Advanced Digital Monitoring System (ADMS™) is provided for display of Volts, Amps, Frequency, and KW. ADMS™ also includes Communicator EXT™ Data Logging Software.

Safety Circuits

The LPH100 Load Bank features main line circuit fusing, wrong voltage protection (Dual Voltage model only), and overtemperature protection.

Load Capacity Ratings: (3 phase, 60 Hz)

Available Load Step Resolution: 5 KW

Rated KW	Rated Voltage	Load Step Resolution
100	480	5, 10, 10, 25, 50
93.75	240	3.75, 7.5, 7.5, 25, 50

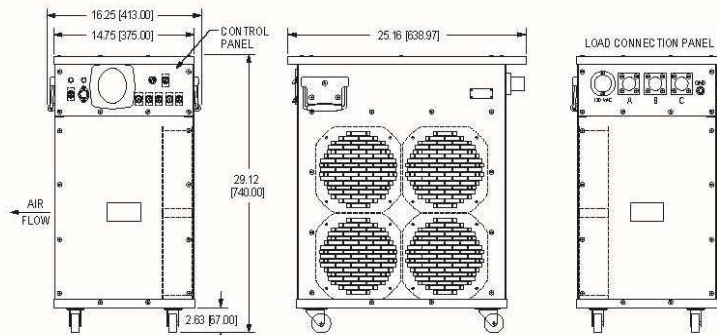
NOTE Designs for international 50/60 Hz voltages or custom DC designs are also available. Consult factory.

Load Bank Weight (approx.)

105 lbs. [48 kg]

Options

- 20' cable set with connectors
- Heavy duty shipping case



All dimensions are in inches [millimeters].



Emerson Network Power

Avtron • 6255 Halle Drive, Cleveland, Ohio 44125, U.S.A.
216-573-7600 • FAX: 216-573-5953 • l3sales@Emerson.com

EmersonNetworkPower.com/loadbank

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Communicator EXT™ is a trademark of Electro Industries/GaugeTech. ADMS™ is a trademark of Avtron. Specifications subject to change without notice.

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Bulletin 428 • Rev. G

EMERSON. CONSIDER IT SOLVED.®

g. PROLINE PROWIRL F 200 FLOW VORTEX FLOW METER SPECIFICATIONS

Proline Prowirl F 200

Nominal discharge current (8/20 µs)	10 kA
Temperature range	-40 to +85 °C (-40 to +185 °F)

1) The voltage is reduced by the amount of the internal resistance $I_{min} \cdot R_i$

i Depending on the temperature class, restrictions apply to the ambient temperature for device versions with overvoltage protection (→ 46)

Performance characteristics

Reference operating conditions

- Error limits following ISO/DIN 11631
- +20 to +30 °C (+68 to +86 °F)
- 2 to 4 bar (29 to 58 psi)
- Calibration system traceable to national standards
- Calibration with the process connection corresponding to the particular standard

i To obtain measured errors, use the *Applicator* sizing tool (→ 92)

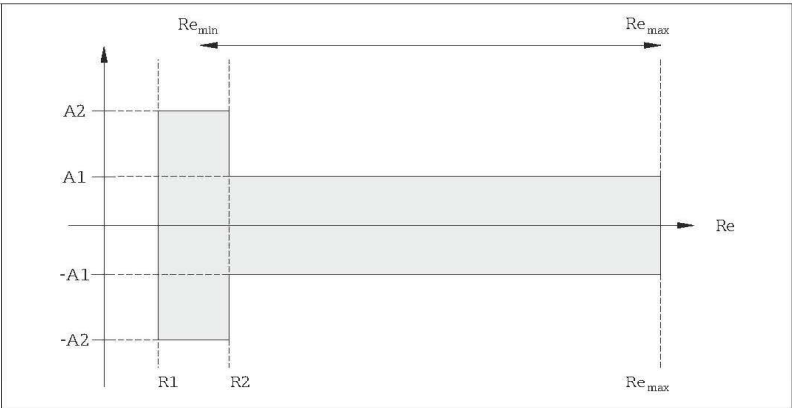
Maximum measured error

Base accuracy

o.r. = of reading, Re = Reynolds number

Volume flow

The measured error for the volume flow is as follows depending on the Reynolds number and the compressibility of the medium under measurement:



Deviation of volume flow value (absolute) from the reading			
Medium type		Incompressible	Compressible ¹⁾
Re range	Measured value deviation	Standard	Standard
R1 to R2	A2	< 10 %	< 10 %
R2 to Re_{max}	A1	< 0.75 %	< 1.0 %

1) Accuracy specifications valid up to 75 m/s (246 ft/s)

Reynolds numbers	Incompressible	Compressible
	Standard	Standard
R1	5 000	
R2	20 000	

Temperature

- Saturated steam and liquids at room temperature if $T > 100\text{ °C}$ (212 °F) applies: $< 1\text{ °C}$ (1.8 °F)
- Gas: $< 1\text{ % o.r. [K]}$

Rise time 50 % (stirred under water, following IEC 60751): 8 s

Mass flow (saturated steam)

- Flow velocities 20 to 50 m/s (66 to 164 ft/s), $T > 150\text{ °C}$ (302 °F) or (423 K)
 - Re $> 20\,000$: $< 1.7\text{ % o.r.}$
 - Re between 5 000 to 20 000: $< 10\text{ % o.r.}$
- Flow velocities 10 to 70 m/s (33 to 210 ft/s), $T > 140\text{ °C}$ (284 °F) or (413 K)
 - Re $> 20\,000$: $< 2\text{ % o.r.}$
 - Re between 5 000 to 20 000: $< 10\text{ % o.r.}$



The use of a Cerabar S is required for the measured errors listed in the following section. The measured error used to calculate the error in the measured pressure is 0.15%.

Mass flow of superheated steam and gas (single gas, gas mixture, air: NEL40; natural gas: ISO 12213-2 contains AGA8-DC92, AGA NX-19, ISO 12213-3 contains SGERG-88 and AGA8 Gross Method 1)

- Re $> 20\,000$ and process pressure $< 40\text{ bar abs.}$ (580 psi abs.): 1.7 % o.r.
- Re between 5 000 to 20 000 and process pressure $< 40\text{ bar abs.}$ (580 psi abs.): 10 % o.r.
- Re $> 20\,000$ and process pressure $< 120\text{ bar abs.}$ (1740 psi abs.): 2.6 % o.r.
- Re between 5 000 to 20 000 and process pressure $< 120\text{ bar abs.}$ (1740 psi abs.): 10 % o.r.

abs. = absolute

Mass flow (water)

- Re 20 000: $< 0.85\text{ % o.r.}$
- Re between 5 000 to 20 000: $< 10\text{ % o.r.}$

Mass flow (user-defined liquids)

To specify the system accuracy, Endress+Hauser requires information about the type of liquid and its operating temperature or information in table form about the dependency between the liquid density and the temperature.

Example

- Acetone is to be measured at fluid temperatures between $+70$ to $+90\text{ °C}$ ($+158$ to $+194\text{ °F}$).
- For this purpose the **Reference temperature** parameter (7703) (here 80 °C (176 °F)), **Reference density** parameter (7700) (here 720.00 kg/m^3) and **Linear expansion coefficient** parameter (7621) (here $18.0298 \times 10^{-4}\text{ 1/°C}$) must be entered in the transmitter.
- The overall system uncertainty, which is smaller than 0.9 % for the example above, is comprised of the following uncertainties of measurement: uncertainty of volume flow measurement, uncertainty of temperature measurement, uncertainty of the density-temperature correlation used (incl. the resulting uncertainty of density).

Mass flow (other media)

Depends on the selected fluid and the pressure value, which is specified in the parameters. Individual error analysis must be performed.

Diameter mismatch correction

Prowirl 200 can correct shifts in the calibration factor which are caused, for example, by diameter mismatch between the device flange (e.g. ASME B16.5/Sch. 80, DN 50 (2")) and the mating pipe (e.g. ASME B16.5/Sch. 40, DN 50 (2")). Only apply diameter mismatch correction within the following limit values (listed below) for which test measurements have also been performed.

Flange connection:

- DN 15 ($\frac{1}{2}$ ") : $\pm 20\text{ %}$ of the internal diameter
- DN 25 (1") : $\pm 15\text{ %}$ of the internal diameter
- DN 40 ($1\frac{1}{2}$ ") : $\pm 12\text{ %}$ of the internal diameter
- DN ≥ 50 (2") : $\pm 10\text{ %}$ of the internal diameter

If the standard internal diameter of the ordered process connection differs from the internal diameter of the mating pipe, an additional measuring uncertainty of approx. 2 % o.r. must be expected.

Example

Influence of the diameter mismatch without using the correction function:

- Mating pipe DN 100 (4"), schedule 80
- Device flange DN 100 (4"), schedule 40
- This installation position results in a diameter mismatch of 5 mm (0.2 in). If the correction function is not used, an additional measuring uncertainty of approx. 2 % o.r. must be expected.



For detailed information about diameter mismatch correction, refer to the Operating Instructions (→ 93)

Accuracy of outputs

o.r. = of reading

Current output

Accuracy	±10 µA
-----------------	--------

Pulse/frequency output

Accuracy	Max. ±100 ppm o.r.
-----------------	--------------------

Repeatability

o.r. = of reading

±0.2 % o.r.

Response time

If all the configurable functions for filter times (flow damping, display damping, current output time constant, frequency output time constant, status output time constant) are set to 0, in the event of vortex frequencies of 10 Hz and higher a response time of max(T_v , 100 ms) can be expected.

In the event of measuring frequencies < 10 Hz, the response time is > 100 ms and can be up to 10 s. T_v is the average vortex period duration of the flowing fluid.

Influence of ambient temperature

o.r. = of reading

Current output

Additional error, in relation to the span of 16 mA:

Temperature coefficient at zero point (4 mA)	0.02 %/10 K
Temperature coefficient with span (20 mA)	0.05 %/10 K

Pulse/frequency output

Temperature coefficient	Max. ±100 ppm o.r.
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APPENDIX B – RAW DATA FROM EXPERIMENTATION AND UNCERTAINTY

a. SUPPLY PRESSURE VERSUS FLOW

i. Raw Data

Steam Flow (lbs/hr)	Steam Flow Uncertainty (lbs/hr)	Supply Pressure at Engine (psig)	Supply Pressure at Engine Uncertainty (psig)
1850	31.45	128	1
2080	35.36	126	1
2200	37.4	125	1
2350	39.95	124	1
2520	42.84	124	1
2720	46.24	122	1
2950	50.15	121	1
3200	54.4	118	1
3455	58.735	114	1
3850	65.45	106	1
4414	75.038	100	1
3750	63.75	108	1
3400	57.8	114	1
3120	53.04	118	1
2880	48.96	121	1
2635	44.795	122	1
2480	42.16	124	1
2290	38.93	125	1
2130	36.21	126	1
1960	33.32	127	1
1800	30.6	129	1
1980	33.66	128	1
2088	35.496	129	1
2207	37.519	127	1
2330	39.61	126	1
2540	43.18	125	1
2707	46.019	124	1
2893	49.181	122	1

3050	51.85	120	1
3370	57.29	118	1
3680	62.56	113	1
3930	66.81	108	1
4304	73.168	104	1
1950	33.15	129	1
2120	36.04	128	1
2290	38.93	128	1
2445	41.565	126	1
2590	44.03	124	1
2890	49.13	122	1
3180	54.06	119	1
3380	57.46	118	1
3708	63.036	113	1
3840	65.28	110	1
4180	71.06	106	1

ii. Uncertainty Sources

Pressure uncertainty:

Only source of uncertainty is pressure gauge which is +/- 1 psi.

Flow uncertainty:

Only source of uncertainty is flow meter which is +/- 1.7%

b. LOAD TESTING

i. Raw Data

Speed (rpm)	Frequency (Hz)	Exhaust Pressure (psig)	Inlet Pressure (psig)	Power (kW)	Steam Flow (lbs/hr)
1795	59.63	15.0	128.0	0.0	1850
1803	60.14	14.5	126.0	5.0	2080
1800	60.11	14.5	125.0	10.2	2200
1799	59.89	14.0	124.0	15.3	2350
1803	60.10	14.0	124.0	20.5	2520
1800	60.03	14.0	122.0	25.8	2720
1799	60.00	14.5	121.0	30.9	2950
1796	59.95	14.5	118.0	36.0	3200
1791	59.67	14.0	114.0	41.0	3455

1812	60.17	14.5	106.0	46.0	3850
1765	58.84	16.0	100.0	50.7	4414
1804	60.08	14.5	108.0	46.1	3750
1797	59.91	14.0	114.0	41.0	3400
1805	60.16	13.5	118.0	36.0	3120
1802	60.03	13.8	121.0	30.8	2880
1802	60.10	13.0	122.0	25.5	2635
1799	59.97	13.0	124.0	20.5	2480
1797	59.93	13.0	125.0	15.3	2290
1800	59.99	12.5	126.0	10.3	2130
1798	59.95	11.0	127.0	5.0	1960
1798	59.97	10.5	129.0	0.0	1800
1770	58.99	16.0	127.4	0.0	1920
1780	59.63	16.0	125.9	5.0	2150
1795	59.72	16.0	124.5	9.8	2330
1786	59.69	16.0	123.2	14.8	2470
1801	59.91	16.0	121.6	19.6	2640
1797	59.87	16.0	119.1	24.6	2875
1803	60.06	16.0	116.9	29.5	3050
1816	60.70	16.0	112.5	34.4	3375
1801	60.09	16.0	109.1	39.3	3600
1811	60.56	16.0	103.1	44.1	3950
1748	58.22	16.0	100.6	46.7	4090
1763	58.81	16.0	95.7	48.5	4340

Speed Uncertainty (rpm)	Frequency Uncertainty (Hz)	Exhaust Pressure Uncertainty (psig)	Inlet Pressure Uncertainty (psig)	Power Uncertainty (kW)	Steam Flow Uncertainty (lbs/hr)
0.359	0.01	1	1	0.0	31.45
0.361	0.01	1	1	1.0	35.36
0.360	0.01	1	1	2.0	37.40
0.360	0.01	1	1	3.1	39.95
0.361	0.01	1	1	4.1	42.84
0.360	0.01	1	1	5.2	46.24
0.360	0.01	1	1	6.2	50.15
0.359	0.01	1	1	7.2	54.40
0.358	0.01	1	1	8.2	58.74
0.362	0.01	1	1	9.2	65.45
0.353	0.01	1	1	10.1	75.04
0.361	0.01	1	1	9.2	63.75
0.359	0.01	1	1	8.2	57.80
0.361	0.01	1	1	7.2	53.04
0.360	0.01	1	1	6.2	48.96

0.360	0.01	1	1	5.1	44.80
0.360	0.01	1	1	4.1	42.16
0.359	0.01	1	1	3.1	38.93
0.360	0.01	1	1	2.1	36.21
0.360	0.01	1	1	1.0	33.32
0.360	0.01	1	1	0.0	30.60
0.354	0.01	1	1	0.0	32.64
0.356	0.01	1	1	1.0	36.55
0.359	0.01	1	1	2.0	39.61
0.357	0.01	1	1	3.0	41.99
0.360	0.01	1	1	3.9	44.88
0.359	0.01	1	1	4.9	48.88
0.361	0.01	1	1	5.9	51.85
0.363	0.01	1	1	6.9	57.38
0.360	0.01	1	1	7.9	61.20
0.362	0.01	1	1	8.8	67.15
0.350	0.01	1	1	9.3	69.53
0.353	0.01	1	1	9.7	73.78

ii. Uncertainty Sources

Power uncertainty:

Only source of uncertainty is power meter which is +/- 0.2% of reading.

Flow uncertainty:

Only source of uncertainty is flow meter which is +/- 1.7% of reading.

Speed uncertainty:

Only source of uncertainty is stroboscope which is +/- 0.02% of reading

Pressure uncertainty:

Only source of uncertainty is pressure gauge which is +/- 1 psi.

Electrical frequency uncertainty:

Only source of uncertainty is power meter which is +/- 0.01 Hz.

Specific consumption uncertainty:

Sources of uncertainty are combined uncertainty of power and flow.

$$Specific\ Consumption_{Actual} = \frac{Flow}{Power} \quad (0.1)$$

$$\begin{aligned} U_{Specific\ Consumption_{Actual}} &= \frac{\partial Specific\ Consumption_{Actual}}{\partial Flow} U_{Flow} \\ &+ \frac{\partial Specific\ Consumption_{Actual}}{\partial Power} U_{Power} \\ &= \frac{1}{Power} * 1.7\% * Flow + \frac{Flow}{Power^2} * 0.02\% * Power \end{aligned} \quad (0.2)$$

Efficiency uncertainty:

The sources of efficiency uncertainty are uncertainty of specific consumption and uncertainty in theoretical steam rate due to uncertainty of pressure readings and inlet quality.

$$Eff = \frac{Specific\ Consumption_{Ideal}}{Specific\ Consumption_{Actual}} \quad (0.3)$$

$$\begin{aligned}
U_{eff} &= \frac{\partial Eff}{\partial Specific\ Consumption_{Ideal}} U_{Specific\ Consumption_{Ideal}} \\
&+ \frac{\partial Eff}{\partial Specific\ Consumption_{Actual}} U_{Specific\ Consumption_{Actual}} \\
&= \frac{1}{Specific\ Consumption_{Actual}} U_{Specific\ Consumption_{Ideal}} \\
&+ \frac{Specific\ Consumption_{Ideal}}{Specific\ Consumption_{Actual}^2} U_{Specific\ Consumption_{Actual}}
\end{aligned} \tag{0.4}$$

$$Specific\ Consumption_{Theo} = \frac{3412.14}{Inlet\ Enthalpy - Exhaust\ Enthalpy} \tag{0.5}$$

$$\begin{aligned}
&U_{Specific\ Consumption_{Ideal}} \\
&= \frac{\partial Specific\ Consumption_{Ideal}}{\partial Inlet\ Enthalpy} U_{Inlet\ Enthalpy} \\
&+ \frac{\partial Specific\ Consumption_{Ideal}}{\partial Outlet\ Enthalpy} U_{Outlet\ Enthalpy}
\end{aligned} \tag{0.6}$$

$$\begin{aligned}
U_{eff} &= \frac{1}{Specific\ Consumption_{Actual}} \left(\frac{\partial Specific\ Consumption_{Ideal}}{\partial Inlet\ Enthalpy} U_{Inlet\ Enthalpy} \right. \\
&+ \left. \frac{\partial Specific\ Consumption_{Ideal}}{\partial Outlet\ Enthalpy} U_{Outlet\ Enthalpy} \right) \\
&+ \frac{Specific\ Consumption_{Ideal}}{Specific\ Consumption_{Actual}^2} \left(\frac{1}{Power} * 1.7\% * Flow + Flow * 0.02\% \right. \\
&\left. * Power \right)
\end{aligned}$$

(0.7)¹

¹ Equations for the inlet and outlet enthalpies are easily determined, especially for multiple independent variables (e.g. Inlet pressure, outlet pressure and inlet quality). Consequently, these uncertainties are difficult to compute algebraically. Therefore, these uncertainties are calculated numerically and included in total uncertainty shown in the raw data.