Repurposing mass-produced internal combustion engines
Quantifying the value and use of low-cost internal combustion piston engines for modular applications in energy and chemical engineering industries

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ABSTRACT

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This thesis proposes that internal combustion piston engines can help clear the way for a transformation in the energy, chemical, and refining industries that is akin to the transition computer technology experienced with the shift from large mainframes to small personal computers and large farms of individually small, modular processing units. This thesis provides a mathematical foundation, multi-dimensional optimizations, experimental results, an engine model, and a techno-economic assessment, all working towards quantifying the value of repurposing internal combustion piston engines for new applications in modular, small-scale technologies, particularly for energy and chemical engineering systems.

Many chemical engineering and power generation industries have focused on increasing individual unit sizes and centralizing production. This “bigger is better” concept makes it difficult to evolve and incorporate change. Large systems are often designed with long lifetimes, incorporate innovation slowly, and necessitate high upfront investment costs. Breaking away from this cycle is essential for promoting change, especially change happening quickly in the energy and chemical engineering industries. The ability to evolve during a system’s lifetime provides a competitive advantage in a field dominated by large and often very old equipment that cannot respond to technology change.

This thesis specifically highlights the value of small, mass-manufactured internal combustion piston engines retrofitted to participate in non-automotive system designs. The applications are unconventional and stem first from the observation that, when normalized by power output, internal combustion engines are one hundred times
less expensive than conventional, large power plants. This cost disparity motivated a look at scaling laws to determine if scaling across both individual unit size and number of units produced would predict the two order of magnitude difference seen here. For the first time, this thesis provides a mathematical analysis of scaling with a combination of both changing individual unit size and varying the total number of units produced. Different paths to meet a particular cumulative capacity are analyzed and show that total costs are path dependent and vary as a function of the unit size and number of units produced. The path dependence identified is fairly weak, however, and for all practical applications, the underlying scaling laws seem unaffected. This analysis continues to support the interest in pursuing designs built around small, modular infrastructure.

Building on the observation that internal combustion engines are an inexpensive power-producing unit, the first optimization in this thesis focuses on quantifying the value of engine capacity committing to deliver power in the day-ahead electricity and reserve markets, specifically based on pricing from the New York Independent System Operator (NYISO). An optimization was written in Python to determine, based on engine cost, fuel cost, engine wear, engine lifetime, and electricity prices, when and how much of an engine’s power should be committed to a particular energy market. The optimization aimed to maximize profit for the engine and generator (engine genset) system acting as a price-taker. The result is an annual profit on the order of $30 per kilowatt. The most value in the engine genset is in its commitments to the spinning reserve market, where power is often committed but not always called on to deliver. This analysis highlights the benefits of modularity in energy generation and provides one example where the system is so inexpensive and short-lived, that the optimization views the engine replacement cost as a consumable operating expense rather than a capital cost.

Having the opportunity to incorporate incremental technological improvements in
a system’s infrastructure throughout its lifetime allows introduction of new technology with higher efficiencies and better designs. An alternative to traditionally large infrastructure that locks in a design and today’s state-of-the-art technology for the next 50 - 70 years, is a system designed to incorporate new technology in a modular fashion. The modular engine genset system used for power generation is one example of how this works in practice.

The largest single component of this thesis is modeling, designing, retrofitting, and testing a reciprocating piston engine used as a compressor. Motivated again by the low cost of an internal combustion engine, this work looks at how an engine (which is, in its conventional form, essentially a reciprocating compressor) can be cost-effectively retrofitted to perform as a small-scale gas compressor. In the laboratory, an engine compressor was built by retrofitting a one-cylinder, 79 cc engine. Various retrofitting techniques were incorporated into the system design, and the engine compressor performance was quantified in each iteration. Because the retrofitted engine is now a power consumer rather than a power-producing unit, the engine compressor is driven in the laboratory with an electric motor. Experimentally, compressed air engine exhaust (starting at elevated inlet pressures) surpassed 650 psia (about 45 bar), which makes this system very attractive for many applications in chemical engineering and refining industries. A model of the engine compressor system was written in Python and incorporates experimentally-derived parameters to quantify gas leakage, engine friction, and flow (including backflow) through valves. The model as a whole was calibrated and verified with experimental data and is used to explore engine retrofits beyond what was tested in the laboratory. Along with the experimental and modeling work, a techno-economic assessment is included to compare the engine compressor system with state-of-the-art, commercially-available compressors. Included in the financial analysis is a case study where an engine compressor system is modeled to achieve specific compression needs. The result of the assessment is
that, indeed, the low engine cost, even with the necessary retrofits, provides a cost advantage over incumbent compression technologies.

Lastly, this thesis provides an algorithm and case study for another application of small-scale units in energy infrastructure, specifically in energy storage. This study focuses on quantifying the value of small-scale, onsite energy storage in shaving peak power demands. This case study focuses on university-level power demands. The analysis finds that, because peak power is so costly, even small amounts of energy storage, when dispatched optimally, can provide significant cost reductions. This provides another example of the value of small-scale implementations, particularly in energy infrastructure. While the study focuses on flywheels and batteries as the energy storage medium, engine gensets could also be used to deliver power and shave peak power demands.

The overarching goal of this thesis is to introduce small-scale, modular infrastructure, with a particular focus on the opportunity to retrofit and repurpose inexpensive, mass-manufactured internal combustion engines in new and unconventional applications. The modeling and experimental work presented in this dissertation show very compelling results for engines incorporated into both energy generation infrastructure and chemical engineering industries via compression technologies. The low engine cost provides an opportunity to add retrofits whilst remaining cost competitive with the incumbent technology. This work supports the claim that modular infrastructure, built on the indivisible unit of an internal combustion engine, can revolutionize many industries by providing a low-cost mechanism for rapid change and promoting small-scale designs.
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———

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Chapter 1

Introduction and Motivation

1.1 Designing for flexibility

Over time, technology has gradually made changes. In some fields, it is individual unit size that is impressive; in other fields, it is the number of units produced. This thesis looks at ways of pushing the boundary of large versus small in technology designs. Previous research (including \cite{13}, \cite{2}, \cite{6}, \cite{15}, \cite{7}, \cite{12}, \cite{9}, \cite{50}) has shown that mass manufacturing and automation open the possibility for significant cost reductions. This flies in the face of conventional wisdom because technologies are typically scaled up in individual unit size to reduce costs. Rather than scaling to larger individual units, downscaling technologies provides another avenue to lowering costs by taking advantage of the benefits of mass-manufacturing.

The learning curve is built on the observation that with every doubling in cumulative capacity, costs decline by a factor, typically around 10-20\%. This provides a way for mass-manufacturing to achieve cost reductions per unit of total capacity on par with the reductions from scaling up individual unit capacity \cite{13}. Pursuing systems built on large numbers of smaller units rather than few, but large, individual units, has historically been unfavorable due to constraints posed by personnel costs. If each unit, for example, requires human supervision, labor costs are significantly reduced with fewer, and therefore larger, units. Advances in computing power and automation break this constraint. Small-scale technologies have the opportunity now to compete with larger systems by taking advantage of automation and mass-manufacturing.

This thesis focuses on one particular technology, the internal combustion engine,
that has been mass-manufactured for more than a century and is low-cost. This work explores specific ways to take advantage of a small engine in non-automotive applications. Internal combustion engines have been extensively produced due to automobile mass-manufacturing. Mass-production has enabled cars, and therefore engines, to come down the learning curve. An example of the learning curve, specifically applied to automobiles, is provided in Fig. 1.1 which shows the price of the Ford Model T versus the cumulative production. From 1909 to 1923, the learning rate was about 15%, meaning that every time cumulative production doubled, the price of the car decreased by 15%. The cost reductions captured in this Model T example are representative of the trend in cost reductions across the automotive industry over the last century. Although internal combustion engines will have their own learning curve, the more automobiles are produced, the more engines are produced, which will lower engine costs.

Figure 1.1: Model T learning curve from 1909 to 1923 (log-log plot of price vs. cumulative production); reproduced from [2]

Mass producing internal combustion engines has brought their cost down to about $10/kW, a figure two orders of magnitude lower than the cost of power plant technology [29]. There are three statements that are likely true and contribute to this
100 fold difference between the cost of an engine and a power plant: (1) at the start, internal combustion engines and power plants were competitive on a $/kW basis; (2) engines have exhibited a higher learning rate (steeper learning curve); and (3) engines have come further down their learning curve. The first point comes from data on costs (and capacity) of early internal combustion engines and Edison’s first power plant. Estimates and data on these costs suggests that they were similar at the beginning on a per power output basis (more on this in Chapter 2). Note that Edison’s Pearl Street Station power plant was built in 1882 during the time two-stroke engines were just being developed. It wasn’t until 1892 that Rudolf Diesel patented the compression-ignition engine [22]. The second point comes from the observation that small technologies exhibit higher learning rates compared with larger systems [13]. Dahlgren et al. explain two phenomena: (1) “continuous improvement can lead to significant cost reductions as production volumes increase” and that (2) “cost reductions attributed to learning can reverse themselves given extended breaks in production” [13]. The first phenomena describes a small, internal combustion engine where new ideas, designs, or manufacturing skills can quickly, easily, and continuously be introduced, whereas the second describes a power plant with its long-lived infrastructure. Due to the large gap in time between building plants, there can be “forgetting,” or reversal on the learning curve [13]. The third point is simply based on numbers: more engines have been manufactured than power plants built and costs decline as a function of cumulative output.

Because engines are now so inexpensive, this raises the question: could they be used for other (non-automotive) applications? This thesis is focused on exploring, in detail, specific examples where the answer to this question is indeed, yes. The claim made here is that systems gain a real advantage by capitalizing on engines’ low costs, even in situations where the engine is not a perfect design.

This thesis proposes that internal combustion piston engines can help clear the
way for a transformation in the energy, chemical, and refining industries that is akin to the transition computer technology experienced with the shift from large mainframes to small personal computers and large farms of individually small, modular processing units. This thesis provides a mathematical foundation, multi-dimensional optimizations, experimental results, an engine model, and a techno-economic assessment, to quantify the value of repurposing internal combustion piston engines for new applications in small-scale technologies, particularly in energy and chemical engineering systems.

1.2 Goals and outline

Broadly speaking, one of the items this thesis focuses on is a systems-level view of energy systems by specifically engaging small-scale, modular designs. This work builds on the mathematical foundation others have established, quantifying the value of mass-manufacturing. This research begins with the knowledge that producing large numbers of units brings costs down. This thesis uses this knowledge and explores specific case studies to quantify the value of bringing mass-produced internal combustion engines into new fields, in particular power generation and chemical engineering. This thesis begins with a mathematical analysis to show how a technology’s total costs in achieving a particular cumulative capacity depend on both the unit size and number of units produced (Chapter 2). Following this, a study is presented which quantifies the potential for small, internal combustion engines participating in the energy market (Chapter 3 and Appendix A). The next three chapters investigate how an internal combustion piston engine can be retrofitted to function as a small-scale gas compressor. Compression technologies, particularly on the small scale, are often uneconomical due to high costs. High compression costs prohibit new, often small-scale systems from entering the market. Creating a cost-efficient way to compress on the small-scale opens the door to disruptive technologies in energy and chemical engineering infras-
tructure. Chapter 4 presents the experimental results of retrofitting and testing an engine designed to run as a small air compressor in the laboratory. Chapter 5 presents a model designed and executed in Python to simulation an engine running as a gas compressor. Chapter 6 brings the experimental work and the modeling together to validate the model and to explore the opportunities to push the system beyond what was achieved in the laboratory. Following this analysis, Chapter 7 presents a techno-economic assessment for this technology. Included in Appendix B is a study focused on small-scale energy storage technologies specifically optimized for deploying power to shave peak power demands. As noted in Appendix C, the Python code developed for these studies is available online.

1.3 Scaling

The need for energy is ubiquitous and drives decision-making across many disciplines, including politics, economics, and engineering. Over the last 135 years, power plants have evolved and grown tremendously in terms of both physical scale and technical sophistication. In 1882, Thomas Edison’s Pearl Street Station in New York became the first coal-fired power plant in the world, providing a total capacity of 600 kW, which comprised six-100 kW generators [44]. In 2017, Plant Scherer in Georgia, U.S.A. has a capacity of 3,600 MW [35], or 6,000 times the total capacity of the Pearl Street Station. The individual generator unit size increased 9,000 times from 100 kW in 1882 [44] to 900 MW in 2017 [35]. This power plant comparison is representative of a trend across both energy systems and chemical engineering plants: technologies have gotten bigger. Both total plant capacity and individual unit size have increased dramatically over the last century.

This thesis is built on the foundation others have established, questioning the trend in energy and chemical engineering industries towards larger scales and showing the attractiveness of small-scale technologies ([13], [12], [9], [50]). The work here
proposes options to create disruptive change to archaic and monolithic infrastructure by using small, inexpensive, mass-manufactured internal combustion piston engines to pursue flexible and resilient designs in energy infrastructure. Now more than ever, the ability to accommodate rapid change in energy infrastructure with small-scale, modular systems creates a competitive advantage in a field dominated by large, long-lived plants. Traditional systems require high upfront costs and must be kept in use for decades to gain returns on investments. This approach is inflexible to change and prevents introduction of new technologies. Energy systems today must solve constantly evolving optimizations driven by everything from advances in power grids to environmental constraints driven by pollution and climate change. This thesis presents a case for pursuing small-scale technologies with components built from repurposing already mass-manufactured, inexpensive internal combustion engines.

In the earlier comparison between Pearl Street Station and Plant Scherer, notice the difference between plant size (total capacity) and individual unit (generator) size. This distinction builds on work by Eric Dahlgren who, in his Ph.D. thesis ([12]), showed that empirically, cost reductions from scaling up in unit size and scaling up in number of units are on par with each other when measured against total capacity built. This finding suggests that some systems may perform better when scaled up in numbers rather than in unit size, and/or that there exists an optimal combination of scaling in unit size and scaling in number of units to achieve a particular total capacity size. The path dependency of scaling is explored in Chapter 2 and total cost is found to be path dependent. This suggests that total costs may be minimized by pursuing a particular scaling path. The path dependency identified is weak, however, and for all practical applications, the underlying scaling laws hold.
1.4 Engines as modular power plants

Common amongst conventional energy systems is the emphasis on centralization and large individual unit sizes. Chapter 3 provides the background and supplementary material to accompany the paper draft included in Appendix A which quantifies the value of internal combustion engine generators (engine gensets) in the energy and reserve markets.

The work included in Appendix A is a copy of a draft of a paper currently under review for publication. The paper presents the results of an optimization to determine how an engine genset should commit power in energy and reserve markets to maximize profit. The model was written in Python and incorporates engine cost, engine wear, engine lifetime, fuel costs, and energy prices. Prior to optimizing the engine power commitments, an optimization was executed to determine the engine operating points. Based on an engine model created in GT-POWER, the trade-off between power output and fuel consumption, as functions of speed and air-to-fuel ratio, was mapped. Based on costs of engine wear (which translate to engine replacement costs) and the cost of fuel, engine speed and air-to-fuel ratio were optimally selected to achieve any power output (within the engine’s range) based on minimizing total system costs. Based on these costs, the optimization evaluates the price of electricity in the day-ahead energy and reserve markets and determines a commitment level to one, both, or neither of the markets. An important distinction between committing in the two markets is that power commitments in the reserve markets are not always called upon to deliver. A generator is paid for the capacity committed, regardless of if it is needed. An additional payment is provided when power is delivered. For a large portion of the time, therefore, capacity is committed but not called upon to deliver. This means that the price threshold to enter the spinning reserve market is lower than the price threshold to enter the energy market as a function of the probability of being called upon to deliver spinning reserves. To commit engine capacity to a
particular market, the price of electricity must be high enough to overcome engine wear and fuel costs. In the energy market, engine wear and fuel costs are always incurred; in the spinning reserve market, engine wear and fuel costs are incurred as a function of the probability of being called upon to deliver the committed capacity.

The optimal engine operating point is not at peak power as was determined by evaluating engine wear and specific fuel consumption. The analysis was performed for a single engine, but as long as a fleet of engines is small enough to not substantively change total supply, this optimization can be extended to a fleet of engines. The optimal strategy for one engine genset will hold for a group of engine gensets. The analysis provided fully in Chapter 3 and Appendix A finds that annual profits are on the order of $30/kW. This result is encouraging and highlights the value of small-scale technology in the energy realm. In particular, the profit is largely a result of commitments in the reserve market, which also suggests that if reserve prices increase due to higher variability as renewable energy penetration on the grid increases, the engine genset will increase in value.

The engine genset in this model is a price-taker, which means that the additional capacity added by this system does not affect electricity prices. If, however, a fleet of engine gensets are made available, the increase in available capacity may begin to lower prices. A sensitivity analysis incorporating lower spinning reserve prices is evaluated, but specifically quantifying by how much engine genset capacity will change market prices is not considered in this analysis. This study always considers the engine genset a price-taker as opposed to a price-setter. Future work focused on quantifying the potential of engine gensets in energy and reserve markets should consider this scenario.
1.5 Engines as small-scale gas compressors

State-of-the-art industrial gas compression technology ranges from about $1,000 to $5,000 per horsepower of effective compressor power [41]. The work in this thesis aims to cut these costs by one to two orders of magnitude. Competing with well-honed large systems is difficult, and starting at the top of the learning curve for new systems would be expensive. Instead, the research here proposes a design based on modifying existing mass-manufactured internal combustion engines into externally driven, small-scale gas compressors (i.e., the engine is consuming rather than producing power). The low cost of mass-produced engines provides a starting point for creating single- and multi-stage compressors that are both small and inexpensive. By striking a balance between custom modifications and efficiency penalties compared to expensive purpose-built equipment, this system design aims to take advantage of the low cost of mass-produced machinery.

This thesis proposes a compressor design that begins with an internal combustion engine. Recall engines are on the order of $10/kW [29], or $7.5/HP, which is much less than the compressor cost, even if one accounts for the fact that the nominal engine power may be significantly larger than the effective compressor power (as a note, the effective compressor power may also be larger than the nominal engine power, as is shown in an example in Chapter 7). If an engine can cost-efficiently be retrofitted and repurposed to perform as a small-scale gas compressor, the cost differential between this system and industrial compressors may be significant. Answering this question is the bulk of this thesis and is covered in Chapter 4 with laboratory experiments, in Chapter 5 with an engine compressor model written in Python, in Chapter 6 with a comparison of experimental and simulated data to verify the model and propose additional retrofits, and in Chapter 7 with a techno-economic assessment.

This thesis finds that an internal combustion engine can be cost-efficiently repurposed to function as a gas compressor, with positive results both in simulations and in
experimentation. The experimental analysis focused on compressing air. The engine compressor built in the laboratory shows engine exhaust can be stored as compressed air. Stored exhaust air surpassed 650 psia with air entering the engine at elevated inlet pressures. The techno-economic assessment in Chapter 7 provides the costs of the engine compressor system built in the laboratory as part of this work and includes a projection of the reduction in cost for future systems. Conservatively, this financial assessment finds that an order of magnitude reduction in costs per unit of horsepower is achievable for the engine compressor system compared with commercially-available options. The expected engine compressor lifetime was not determined experimentally but is approximated. A discussion is included in Chapter 7 regarding capital versus operational costs, as system efficiency and lifetime affect both. With regards to short-lived infrastructure, the analysis in Chapter 3 and Appendix A shows that short lifetimes are not, by nature, disadvantageous. This applies also to the short engine compressor lifetime compared with industrial compressor lifetimes.

The experimental focus in this work is on exploring a series of engine retrofits to first demonstrate air at elevated pressure can be stored as exhaust from the engine compressor, and then improving performance by decreasing engine cylinder clearance volume to increase system efficiency and by facilitating heat exchange during compression. The first retrofit focused on redesigning intake and exhaust valves while using the engine’s original cylinder head. The pushrods were removed to keep the engine’s intake and exhaust valves closed. Flow in and out of the engine cylinder are now through two check valves, which access the cylinder via the spark plug hole. The spark plug was removed and in its place, a retrofitted “valvetrain” was added. The valvetrain consists of piping that holds the check valves, instrumentation for measuring temperatures and pressures, and a safety pressure relief valve.

The next two major engine retrofits focused on improving system performance by decreasing cylinder clearance volume. This required designing and fabricating new
engine cylinder heads. The first design used a 1/2” thick acrylic plate. Instead of a
domed cylinder head, the flat plate reduces dead volume, and therefore increases the
amount of expelled air on every cycle. This design was further improved by fabricating
the next cylinder head out of a 1” thick aluminum plate with built-in channels to
provide water cooling to the cylinder head. The details and results of each of these
retrofits are tested in the laboratory as per the details given in Chapter 4 and modeled
in Python as described in Chapter 5. The custom-designed aluminum cylinder head
shows the most promise for future designs. Efficiencies vary as a function of cylinder
head, inlet pressure, cylinder head cooling, and engine speed.

1.6 Peak demand shaving
The final contribution of this thesis is in providing a tangential study of the value
of small-scale technologies, this time focused on energy storage 27. Many utility
companies offer varying electricity prices, or tariffs, that charge a high price for the
peak power demand in a given interval throughout the billing period. The charge
can be above $15 - 20/kW and can account for a large portion of the electric bill.
Because this peak power demand is so costly, decreasing peak power demand by even
small amounts can lead to significant cost reductions. Included in Appendix B is a
paper that quantifies the value of small-scale, onsite energy storage that is deployed
during peak demand periods. The study finds that for every technology considered,
positive net present values are realized over a technology’s lifetime (on average about
$400/kWh). To decrease costs, the energy storage must be deployed precisely during
the interval of peak power during the billing period. An algorithm was written in
Python to predict when this peak power demand would occur. The algorithm was not
consistently reliable in predicting the peak during every billing cycle, which meant
that, in some months, there was no storage defense available precisely when it was
needed most during the interval of peak power demand. Even with this handicap, the
study’s results are very positive and point to even more profitable net present values with a smarter dispatch algorithm that is built on principles of machine learning. This would allow more peaks to be shaved and higher returns on investments. The paper is included in Appendix B because it further highlights the value of small technologies, in particular in the energy field.

A point to add about the study in Appendix B is that, although batteries and flywheels are considered as the energy storage options, an additional scenario is plausible, where engine gensets (in a manner posited in Chapter 3 and Appendix A) could become the peak shaving mechanism. If there is fuel available, engine gensets can quickly provide the power needed to reduce grid purchases. Recall that in the case with batteries, if the algorithm incorrectly predicts the threshold above which storage should be discharged to reduce a peak power demand, the capacity may be fully depleted when it is needed most. In the case of engine gensets, fuel may be plentiful and provide a buffer in the required accuracy of the algorithm predictions.

1.7 Summary

This thesis brings together economics of scaling, modeling and optimizations in Python, experimental work, and a techno-economic assessment to provide a robust view of small-scale technologies and their potential in bringing about disruptive change in many industries. The research included in this dissertation is in many ways still at the beginning. The value of small-scale technologies, in particular ones based on repurposing internal combustion engines, is quantified through experiments, modeling, and financial analyses. The aim of this work has been to provide an understanding of small-scale systems from both engineering and economic perspectives. The task going forward is to demonstrate small-scale technologies can hold their own in commercial applications.

While the results of the simulations and laboratory-scale experimental work in-
cluded in this dissertation provide compelling reasons to continue pursuing small-scale infrastructure, there are two main hurdles. The first challenge is in moving from laboratory to pilot and commercial scales, which requires testing the resilience of the small-scale infrastructure designed in this thesis as well as incorporating additional balance of system constraints not addressed at the laboratory scale. The second difficult task is in being able to interrupt the inertia in the incumbent technology. Introducing change in many power generation and chemical engineering industries is unwelcome. Overcoming these challenges is required for the specific applications presented in this work and also for the broader community of small-scale infrastructure.
Chapter 2

Scaling, modularity, and internal combustion piston engines as building blocks

2.1 Scaling background

Historically, technologies in energy and chemical engineering industries have grown to increasingly large scales [13]. Reasons for scaling up include spreading fixed costs over a larger capacity. In 2013, Dahlgren et al. pointed out, however, that the cost reductions from scaling up in individual unit size can be matched by cost reductions from scaling up in numbers by learning and mass manufacturing, largely due to advances in automation [13].

Dahlgren et al. point out that one reason technologies have tended to scale up is that material costs of construction are thought to scale with the ratio of surface area to volume, which, for a cylinder, is $2\pi rh$ to $\pi r^2h$. If costs did scale like this, it would, indeed, be less expensive to build bigger. Nature shows that this isn’t the case, however, because wall thickness increases as volume increases, so there is a disproportionate increase in materials, which reduces the benefits of scaling up from the non-linear surface area to volume ratio. An added level of design and construction complexity also accompanies building larger systems, which is costly. Dahlgren et al. point out that another reason often cited for increasing individual unit size is labor costs [13]. One hundred or even fifty years ago, computing technology was minimal compared with today’s capabilities. This meant that it was more economical to have large, centralized units to decrease personnel costs. This restriction is loosened today
due to advances in automation. Computer programs can control one, ten, hundreds, thousands of units, centralized or distributed, in parallel. Labor costs are no longer prohibitive in many industries.

Contrary (or in addition) to scaling up in unit size, technologies can be scaled in number of units. Modular and small-scale units allow for distributed operation while retaining the option for centralization. They provide increased flexibility to engage markets due to shorter lead times. Small and short-lived units allow firms to more easily disengage from unprofitable ventures without stranding a significant investment meant to last for decades. Finally, small unit size provides redundancy whereby failure of a single component translates to partial (as opposed to total) outage. Additional benefits of modularity are described in a 1994 paper by Baldwin and Clark [7]. They show how incorporating modularity in system designs increases the ability for small-scale custom designs, lowers maintenance costs, and allows for upgrades over the technology lifetime [7].

The mathematical derivations of scaling in unit size or scaling in number of units have already been worked out (see [13] and [12]). Dahlgren et al. found that, empirically, the cost reductions from scaling up in individual unit size are on par with the cost reductions from scaling in number of units produced. The mathematical foundation on scaling laws is summarized below from [13] and [12]. Building on these fundamentals, a new analysis is included here that looks at the result of scaling across both dimensions: size and number. This developed from the observation that if scaling up in unit size and scaling out in number of units are comparable, do all development paths to a particular cumulative capacity as a function of unit size and number of units produced lead to the same total costs? The aim of this analysis is to determine the path dependence (or independence) of total costs as a function of unit size and number of units produced.

“Economies of unit scale” are the cost reductions associated with scaling the
capacity of an individual unit; (2.1) shows the cost, \( k \), of the \( n^{th} \) unit. Capacity size is designated with \( c \), unit number is designated with \( n \), and \( \text{ref} \) refers to a reference capacity/cost.

\[
k_n = k_{\text{ref}} \left( \frac{c_n}{c_{\text{ref}}} \right)^\alpha
\]  

(2.1)

In (2.1), \( \alpha \) is typically between 0.6 and 0.8, which is why this expression is often called the “two-thirds law.” It captures the non-linearity between costs and size and shows that capacity increases faster than costs rise (i.e., bigger units are less expensive per unit of capacity). This is also analogous to the volume to surface area relationship described at the beginning of this chapter.

Scaling, instead, in numbers (economies of mass-manufacturing) takes advantage of learning, which is the finding that with every doubling in cumulative output, costs decrease by a factor, \( \epsilon \), called the progress ratio \([38]\). \( \epsilon \) is less than one and often around 0.81 \([13]\). The term “learning rate” refers to one minus the progress ratio. The expression for learning is given in (2.2), where \( k \) is again cost and \( n \) is the number of units produced. Recall the earlier example of the Ford Model T learning curve shown in Fig. 1.1; this is the equation that describes the cost reduction behavior seen in that example (for that example, \( \epsilon \) is about 0.85).

\[
\frac{k_{2n}}{k_n} = \epsilon
\]  

(2.2)

Using (2.2), the expression for the cost, \( k \), of the \( n^{th} \) unit from learning is derived as shown in (2.3).

\[
k_n = k_{\text{ref}} \epsilon^{\log_2 n} = k_{\text{ref}} n^{\log_2 \epsilon}
\]  

(2.3)

The cost of the \( n^{th} \) unit that results from either scaling up in individual unit size or scaling up in number of units produced is given by (2.1) and (2.3), respectively. To compare the cost reductions from each scaling force, the total cost of producing
a total capacity $N$ times larger than the initial capacity, $c_{\text{ref}}$, is found. The result of scaling in unit size is to build one unit at a capacity of $Nc_{\text{ref}}$, which results in the expression shown in (2.4).

$$k(Nc_{\text{ref}}) = k_{\text{ref}}N^\alpha$$ (2.4)

The alternative is to keep unit size unchanged at $c_{\text{ref}}$ and scale to $N$ units. To do this, one must integrate the expression in (2.3) over the total number of units, $N$. The result is the total cost, $K$, and is shown in (2.5).

$$K(N) = \frac{k_{\text{ref}}}{1 + \log_2 \epsilon} N^{1+\log_2 \epsilon}$$ (2.5)

Since $\alpha \approx 1 + \log_2 \epsilon$ (where $\epsilon$ is about 81% and therefore $\alpha$ about 0.7), the total cost after scaling in numbers (via mass manufacturing) is on par with the total cost after scaling in individual unit size.

What Dahlgren et al.’s comparison does not include, however, is a case where a technology varies unit size while more units are produced. This combination of scaling is evaluated here to determine which method (or combination of methods) of scaling is (are) optimal for minimizing cumulative costs. The next section covers the mathematics describing the case of scaling in both unit size and number of units produced.

### 2.2 Path dependence in total cost

Dahlgren et al. evaluate scaling in unit size and scaling in number of units produced independently. In this section, the results of combining these two scaling laws are derived to determine the relationship between unit capacity, number of units, and total cost of installing a total capacity, $C$.

At the level of analysis included in this model, the type of unit is not explicitly chosen. This analysis could be confined to certain subsystems where the rest of
the balance of system is not included, which could be important if the subsystem chosen is modular, whereas the surrounding support infrastructure is not, or where other parts of the system may have much larger production numbers due to other applications. In this latter case, one would not expect learning in the context of this use to significantly reduce costs. At the resolution of the analysis included here, the model is not committed to what type of unit it is. Here the analysis looks only for qualitative results and explores trends.

In the simple cost comparison of scaling up from a single test unit to a single, much larger unit versus building \(N\) units equal in size to the first one, this analysis found that the power law describing the scaling is the same, but the coefficient in front is slightly changed. So in terms of the shape of the curve, the two paths result in the same answer; in terms of the far less certain coefficient in front of the power law the numbers are not identical, but similar. Here the question is whether this near path dependence is nearly universal, or is limited to the two end members in the path, where scaling up is accomplished either by solely increasing the unit size, or solely scaling up in numbers.

Describing path dependence begins with determining the cost of the \(n^{th}\) unit produced when a technology is scaled in both unit size and in numbers; (2.6) shows this expression. Recall that (2.1) and (2.3) show the costs of the \(n^{th}\) unit for these two scaling metrics acting in isolation; (2.6) combines the two.

\[
k_n = k_{\text{ref}} n^{\log_2 \epsilon} \left( \frac{c}{c_{\text{ref}}} \right)^\alpha
\]

(2.6)

If \(k_{\text{ref}}\) in (2.6) is in terms of dollars per unit output, the change in capacity must be addressed by multiplying by the ratio as shown in (2.7).

\[
k_n = k_{\text{ref}} n^{\log_2 \epsilon} \left( \frac{c}{c_{\text{ref}}} \right)^{\alpha-1}
\]

(2.7)
To determine total costs, $K$, as a function of total number of units, $N$, and total capacity, $C$, (2.6) is integrated as shown in (2.8).

\[ K(N, C) = \int_0^N k_n dn = \int_0^N k_{ref} n^{\log_2\epsilon} \left( \frac{c}{c_{ref}} \right)^\alpha dn \]  

(2.8)

Note that the total capacity, $C$, is a function of units produced and of their (average) size. It depends on $n$ and $c(n)$. To simplify the discussion, for a particular trajectory, $C$ is thought of as a function of $n$, $C = C(n)$. Different choices for unit sizes result in different total capacity functions. The derivative of the total capacity gives the unit capacity of the $n^{th}$ unit; see (2.9).

\[ c_n = \left( \frac{dC(n)}{dn} \right) \]  

(2.9)

Using the substitution in (2.9), (2.8) can be rewritten as shown in (2.10).

\[ K(N, C) = \int_0^N k_{ref} n^{\log_2\epsilon} \left( \frac{dC}{ dn c_{ref}} \right)^\alpha dn = k_{ref} c_{ref}^{-\alpha} \int_0^N n^{\log_2\epsilon} \left( \frac{dC}{dn} \right)^\alpha dn \]  

(2.10)

Recall that Dahlgren et al. found that scaling in unit size is on par with scaling in number of units. Specifically, they related $\alpha$ and $\epsilon$ as shown in (2.11).

\[ \alpha \approx 1 + \log_2 \epsilon \]  

(2.11)

For simplicity and substitution, part of the expression in (2.11) is rewritten as shown in (2.12).

\[ \log_2 \epsilon = -\gamma \]  

(2.12)

For now, the focus is on the integrand in (2.10) only (put another way, assume for now that $k_{ref} c_{ref}^{-\alpha} = 1$). The integrand is rewritten as shown in (2.13), where the substitution from (2.12) is incorporated.
\[ K(N, C) = \int_0^N n^{-\gamma} \left( \frac{dC}{dn} \right)^{1-\gamma} dn = \int_0^N n^{-\gamma} \left( \frac{dC}{dn} \right)^{-\gamma} \left( \frac{dC}{dn} \right) dn \] (2.13)

From (2.13), the bounds of integration are changed by moving from \( n \) as the independent variable to \( C \) as the independent variable. The integration is now over capacity instead of number of units, as shown in (2.14).

\[ K(N, C) = \int_0^C \left( \frac{n}{dC} \right)^{-\gamma} dC \] (2.14)

Recall that both \( n \) and \( \frac{dC}{dn} \) are functions of \( C \), and total capacity is a function of \( n \), so \( n(C(n)) = n \) (as written in (2.14)). Because (2.14) has no straightforward solution, it is further manipulated using the observation given in (2.15).

\[ \frac{d}{dn} \log n = \frac{1}{n} \] (2.15)

Using (2.15), (2.14) is rewritten as shown in (2.16).

\[ K(N, C) = \int_0^C \left( \frac{1}{n} \frac{dn}{dC} \right)^{\gamma} dC = \int_0^C \left( \frac{d}{dn} \log n \frac{dn}{dC} \right)^{\gamma} dC = \int_0^C \left( \frac{d}{dC} \log n \right)^{\gamma} dC \] (2.16)

A new function \( g(C) \) is defined as shown in (2.17).

\[ g(C) = \left( \frac{d}{dC} \log n(C) \right)^{\gamma} \] (2.17)

Using this new function, the expression for total cost resulting from both scaling in unit size and scaling in number of units produced is simplified and shown in (2.18).

\[ K(N, C) = \int_0^C g(C)dC \] (2.18)

In (2.18), the possibility that \( g(C) \) may become singular at zero is ignored. A better representation of (2.18) is shown in (2.19) where \( C_0 \) is the capacity of the first
unit and \( K_0 \) is the cost of the first unit. This caveat is noted, but for simplicity, this possibility is ignored and the analysis continues with the expression from (2.18).

\[
K(N, C) = \int_{C_0}^{C} g(C) dC + K_0
\]  

(2.19)

The value of the integral in (2.18) does depend on the particular path taken; it depends on the choice of \( n(C) \) or \( g(C) \). Because this is a simple dimensional analysis, a few capacity paths are evaluated. The path dependence can be shown by explicitly looking at different path examples. Table 2.1 provides eight different capacity paths, \( C(n) \). In cases 1 and 2, \( C \) and \( n \) are linearly related. Case 3 introduces a quadratic where \( C \) is equal to the square of \( n \). Case 4 gives the more general power law where \( C \) is equal to \( n \) raised to a power, \( \beta \). In cases 5 and 6, \( C \) is related to the logarithm of \( n \). Case 7 gives the exponential form, and case 8 is the logarithmic raised to a power. In case 8, \( N_\infty \) is the maximum number of units at infinite capacity. Table 2.1 shows the steps for solving for \( K \) as per (2.18), given the substitution defined in (2.17). Recall that the total costs in Table 2.1 neglect the factor outside of the integrand, \( k_{\text{ref}} c_{\text{ref}}^{-\alpha} \), which is constant for all cases.
Table 2.1: Total cost with different total capacity paths

<table>
<thead>
<tr>
<th>Case</th>
<th>( C(n) )</th>
<th>( n(C) )</th>
<th>( \log n )</th>
<th>( g(C) )</th>
<th>( K )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>( n )</td>
<td>( C )</td>
<td>( \log C )</td>
<td>( \left( \frac{1}{C} \right) )</td>
<td>( \frac{1}{1 - \gamma} C^{1-\gamma} )</td>
</tr>
<tr>
<td>2</td>
<td>( An )</td>
<td>( \frac{C}{A} )</td>
<td>( \log C - \log A )</td>
<td>( \left( \frac{1}{C} \right) )</td>
<td>( \frac{1}{1 - \gamma} C^{1-\gamma} )</td>
</tr>
<tr>
<td>3</td>
<td>( n^2 )</td>
<td>( C^{3/2} )</td>
<td>( \frac{1}{2} \log C )</td>
<td>( \left( \frac{1}{2C} \right) )</td>
<td>( \frac{2^{-\gamma}}{1 - \gamma} C^{1-\gamma} )</td>
</tr>
<tr>
<td>4</td>
<td>( n^\beta )</td>
<td>( C^{1/\beta} )</td>
<td>( \frac{1}{\beta} \log C )</td>
<td>( \left( \frac{1}{\beta C} \right) )</td>
<td>( \frac{\beta^{-\gamma}}{1 - \gamma} C^{1-\gamma} )</td>
</tr>
<tr>
<td>5</td>
<td>( \log n )</td>
<td>( e^C )</td>
<td>( C )</td>
<td>1</td>
<td>( C )</td>
</tr>
<tr>
<td>6</td>
<td>( 1 + \log n )</td>
<td>( e^{C-1} )</td>
<td>( C - 1 )</td>
<td>1</td>
<td>( C )</td>
</tr>
<tr>
<td>7</td>
<td>( e^n )</td>
<td>( \log C )</td>
<td>( \log(\log C) )</td>
<td>( \left( \frac{1}{C \log C} \right)^\gamma )</td>
<td>( \int_0^C \left( \frac{1}{C \log C} \right)^\gamma dC )</td>
</tr>
<tr>
<td>8</td>
<td>( \left( \frac{\gamma - 1}{\gamma} \log \frac{n}{N_\infty} \right)^{\gamma/(\gamma-1)} )</td>
<td>( N_\infty \exp \left( \frac{\gamma}{\gamma - 1} C^{(\gamma - 1)/\gamma} \right) )</td>
<td>( \log N_\infty + \frac{\gamma}{\gamma - 1} C^{(\gamma - 1)/\gamma} )</td>
<td>( \frac{1}{C} )</td>
<td>( \log C )</td>
</tr>
</tbody>
</table>
As shown in Table 2.1, the total cost is path dependent. As cumulative capacity paths are varied, total costs, $K$, come out differently. Even the shape of the scaling curve, its power or functional form, is affected by the choice of the path. Mathematically then, the way technology is scaled as a function of unit capacity and number of units produced matters.

To get a sense of the physical scaling in some of the capacity paths in Table 2.1 plots are provided in Fig. 2.1 and Fig. 2.2 with the goal of visually capturing how plant size varies in these scenarios. In cases 1 and 2, each unit size is the same because cumulative capacity, $C$, is linear in $n$. In case 3, each unit is larger than the next by a square. The choice of $\beta$ in case 4 will determine the relationship between each unit’s size. For $0 < \beta < 1$, each unit is smaller than the previous. For $\beta = 1$, each unit is the same size, and for $\beta > 1$, each unit is larger than the previous. In cases 5 and 6, unit size is exponentially decreasing. In case 7, unit size is increasing exponentially.

To demonstrate the results from Table 2.1 visually, total costs are first calculated for cases 1, 2, and 3. From the derivations in Table 2.1 the expectation is that the total cost in case 1 and 2 should be the same, while the total cost for case 3 should be $2^{-\gamma}$ times the cost of case 1, where $2^{-\gamma}$ is $\epsilon$. In case 2, $A$ is set to 3. For this example, total capacity, $C(N)$, is set to 100; $\epsilon$ is set to 0.81. Fig. 2.1 includes three plots; the top plot shows the capacity of each unit produced; the middle plot shows the cumulative capacity; the bottom plot shows the total cost as a function of units produced. The bottom plot in Fig. 2.1 shows that the total costs vary, which is expected from the results in Table 2.1. The costs for case 1 and case 2 are equivalent and just over 35, while the total cost for the path in case 3 is below 30 (costs are in arbitrary units for the arbitrary capacity of 100).

Notice in case 1, every unit has capacity equal to one (see Fig. 2.1 top plot). In case 2, each unit has capacity equal to three except for the last unit, which has capacity equal to one, which is necessary to reach a total capacity of 100. In case
Figure 2.1: Unit capacity (top), cumulative capacity (middle), and total cost (bottom) as a function of number of units produced when total capacity is set to 100

3, each unit is bigger than the previous because the cumulative capacity, $C(N)$, is a power law.

Case 1 and case 2 have equivalent expressions for total cost, which suggests that keeping unit size fixed (regardless of the unit size) results in the same total cost.

The total costs for cases 4 and 6 were also calculated, here with a total capacity, $C(N)$, of five. Case 5 is neglected since it gives the same results as in case 6. $\epsilon$ is still
Two simulations are included for case 4, one with $\beta$ set to $1/2$ and the second with $\beta$ set to 3. As Fig. 2.2 shows, the total costs vary (as given in Table 2.1). The cost for case 4 ($\beta = 1/2$) was almost 5.5, for case 4 ($\beta = 3$) was just over 3, and for case 6, total cost was 5.

Figure 2.2: Unit capacity (top), cumulative capacity (middle), and total cost (bottom) as a function of number of units produced when total capacity is set to 5

Notice in cases 4 ($\beta = 1/2$) and 6 as shown in Fig. 2.2 each unit is smaller than the previous unit. Decreasing unit size results in increasing unit cost. However, smaller
unit size results in more units being built, which lowers costs.

As these examples have shown in Table 2.1, Fig. 2.1, and Fig. 2.2, total cost is path dependent and a function of number of units produced and size of each unit. In Table 2.1, cases 1 - 4 have a similar factor $C^{1-\gamma}$, but this falls apart in cases 5 and 6 when cumulative capacity, $C(n)$ is logarithmic, and in cases 7 and 8. In cases 5 and 6, the total cost is equivalent to the cumulative capacity, $C$. In case 4, $\beta$ must be greater than 0 in order to satisfy the constraint that $C(n)$ is a monotonically rising function. Total costs decrease as $\beta$ increases (see Fig. 2.3 for a plot of total cost, $K$, as a function of the choice of $\beta$, for a total capacity set to 100).

![Figure 2.3: Total cost vs. $\beta$ (case 4 cost curves) when total capacity is set to 100](image)

This analysis shows that the total cost for achieving a particular cumulative capacity depends on the path taken to get there. Total cost will vary depending on whether all units have the same size, or whether the size of the unit is itself a function of the capacity. The path dependence identified here is fairly weak, and in practical applications may be overwhelmed by details that have been neglected in this simple model. For example, in practice, the physics of a particular process introduces natural scales into the system which could favor a particular size or sizes. Therefore,
there could be a preferred size, which is not captured in this model. Also in practice, machines are built from prefabricated parts, and it may not be practical to assume that subunits like pipes can be obtained in arbitrary combinations of wall thicknesses and diameters. Certain sizes may be more readily available and thus define a cost minimum.

Moreover the cost integral involves the function $g(C)$ as defined in (2.17), which only depends on the logarithm of $n$. As a result, changes in the cost due to changing unit size are rather small, and the scaling is very weak. For example, for a wide range of pathways one may want to chose, it would be reasonable to assume that the number of units that add up to a certain capacity scale with a simple power of the capacity, $n(C) = n_0 C^{1/\beta}$ (case 4). Because the total number of units has to scale monotonically upward with the capacity, $\beta$ has to be larger than zero, but is otherwise unspecified. Recall, if $\beta$ is equal to one, all units have the same size, if $\beta$ is larger than one, the unit size increases with capacity, if $\beta$ is between zero and one, the unit size shrinks as the capacity increases. In all these cases the cost can be written as $K = k_0 C^p$, where $k_0$ is the cost of the unit with capacity equal to 1 and $p$ is a power, which is a function of the learning coefficient, $\gamma$.

Unsurprisingly, the analysis shows that focusing learning at the beginning of a technology’s production is helpful, and that starting small and going to larger units with increasing capacity reduces costs. However, this is only reflected in the multiplier $k_0$ that scales the overall cost and varies with $\beta$. The power law that relates cost and capacity is not changed; $p$ does not depend on $\beta$. This is important because, for such simple first order cost estimates, the most important lesson is taken from the underlying scaling laws, which in this case seem unaffected as long as capacity and number are related by a power law.

Since the result is path dependent, it is possible to force a different trajectory that can break the simple power law, but this requires extreme changes in the relationship
between \( n \) and \( C \). If the relationship takes on logarithmic or exponential forms (cases 5 - 8), it becomes possible to change the relationship between total cost and total capacity installed. For example if \( C = \log n \) (case 5), which suggests that units become exponentially smaller, the power relationship moves from a two-thirds power to a linear relationship. If \( C \) scales like a power of \( n \) inside an exponential growth (case 8), then it is possible to drive the power law to zero and limit the cost growth to a logarithmic behavior. This is not likely to be a practical strategy, because individual units would have to scale up so rapidly, that after a few generations, the technology is likely to hit very serious physical constraints. The basic assumption, that units are similar to each other is likely to be violated. Indeed, if one wants a logarithmic relationship between capacity and cost, the total number of units remains finite even if one approaches infinite capacity (as in case 8).

To conclude, it may be possible to improve learning behavior slightly, by starting a process with small units and gradually moving to larger units. However, the improvements one can achieve seem small compared to other variables which are not considered in this simple model. The factor \((k_0)\) in front of the scaling law can fairly easily be modified, but the exponent \((p)\) characterizing the relationship between cost and total capacity is only affected in extreme situations, which are not likely to be realized in practice.

2.3 Scaling of engines and power plants

The focus described in the previous section on quantifying the results of scaling in both unit size and in number of units produced was driven by a comparison of the costs of small-scale internal combustion engine technology and large power plant infrastructure. Engines are about \$10/kW, whereas power plants are about \$1,000/kW. This cost disparity motivated an exploration of scaling to see if production numbers and capacity over the technologies’ lifetimes would predict this result.
Included in this section are two methods that were explored to make sense of the two order of magnitude difference in costs between an engine and a power plant. The first analysis looks at the difference in today’s engine and power plant capacity and relates them by adjusting for system lifetime, average size, and rate of units built. The second analysis dives deeper and notes that the story is complicated by the fact that average unit sizes have changed over the technologies’ lifetimes. This analysis takes changing unit size into account.

2.3.1 Neglect changing unit sizes

The first, high-level analysis to compare the costs per unit power output of power plants and car engines assumes a uniformity in average unit size across the history of each technology. The total aggregate capacity of power plant generators is a combination of the rate of units built, the average size of a unit, and the normalized lifetime, as shown in (2.20) where \( P_T \) is today’s total capacity of power plant generators (in GW), \( P_n \) is the rate at which power plant generators are built, \( P_s \) is the average power plant generator size, and \( P_l \) is the normalized power plant generator lifetime, where \( P_l \) is set to a value of 1.

\[
P_T = P_n P_s P_l \tag{2.20}
\]

Today’s total capacity of engines is represented the same way as given in (2.21), where \( E_T \) is today’s total engine capacity, \( E_n \) is the rate at which engines are built, \( E_s \) is the average engine size, and \( E_l \) is the engine lifetime in terms of the power plant lifetime.

\[
E_T = E_n E_s E_l \tag{2.21}
\]

Today’s total capacity of power in engines is about 25 times that for power plants \cite{25}. Therefore, \( 25 P_T = E_T \). The lifetime of an internal combustion engine is about
15 years, whereas a power plant is about 60. Therefore, \( P_t = 4E_t \). Engines have an average size of 111 kW \[25\], whereas U.S. coal-fired power plant generators today have an average size of about 600 MW \[3\]. Therefore, \( P_s = 5400E_s \). Putting it all together yields (2.22).

\[
25P_T = E_T = E_n \left( \frac{1}{5400} P_s \right) \left( \frac{1}{4} P_t \right)
\]  

(2.22)

The expression in (2.22) is rearranged to give the ratio of the rate of engines built to the rate of power plants built, as shown in (2.23). This ratio is assumed to be true throughout the history of both technologies.

\[
\frac{E_n}{P_n} = 25 \times 5400 \times 4 = 540,000 = 2^{19}
\]  

(2.23)

To see if this figure of 540,000 is reasonable, the number of power plants and engines produced are needed. Based on data from the U.S. Energy Information Administration, 1,453 coal-fired generators have been built in the U.S. since 1921 \[3\]. According to the U.S. Department of Transportation, the number of vehicles produced in the U.S. since 1960 is about 554 million as shown in Fig. 2.4 \[48\]. Extrapolating back to the early 1900s, an additional vehicle production needs to be quantified. It is estimated here as about 150 million from the early 1900s to 1960. Therefore, a total of about 700 million vehicles has been produced. Based on these figures, the ratio in (2.23) is \((700 \times 10^6)/1,453\), which is about 500,000. The figure in (2.23), therefore, is a remarkably good estimate. While this simple analysis does not rule out additional metrics in this cost/capacity comparison of engines versus power plants, it is noteworthy how well it holds up.

Based on the calculations that achieved the ratio shown in (2.23), when size and lifetime are taken into account, there have been \(2^{19}\) more engines than power plants built to get to today’s total capacity. Recall that, based on the learning rate and economies of numbers, \( k_n = k_{ref} \epsilon^{\log_2 n} \). If engines and power plant generators started
at approximately the same cost per unit output, then the learning rate should predict the difference between the two costs per unit output due to the fact that many more engines have been produced than power plant generators built. The ratio of $k_n$ to $k_{ref}$, therefore, can be thought of as the cost per unit output of an engine to the cost per unit output of a power plant generator. By selecting a learning rate of 15%, $\epsilon$ is equal to 0.85. From (2.23), $\log_2 n$ is found to be 19. As shown in (2.24), the fact that more engines have been produced predicts that an engine today should be about 4.6% the cost per unit output of a power plant generator.

$$0.85^{19} = 0.046$$  \hspace{1cm} (2.24)

Since engines (on a per capacity basis) are 1% of the power plant generator cost ($10/kW$ versus $1,000/kW$), it is clear that one or more of the following are true: 1) engines and power plant generators did not start at the same cost per kW; 2) engines and power plants did not follow the same learning rate; or 3) engines and power plant generators have changed average unit size over their histories. Point 3 is known to be
true from historical data. Point 2 is likely true based on data showing higher learning rates for small rather than large infrastructure (see [13]). Point 1 is likely false based on data and estimates of these early costs, which show the costs were competitive.

Incorporating changing unit size (point 3) is addressed next.

2.3.2 Account for changing unit sizes

In Section 2.2, total costs as a function of a technology’s unit size and number of units produced (cumulative capacity path) were derived. To understand today’s unit cost difference (per unit output) in engines and power plants, the capacity path is mapped for each technology.

2.3.2.1 Internal combustion engines

Fig. 2.4 shows cumulative U.S. vehicle production at about 554 million since 1960. For production numbers before 1960, Ford data is used as a proxy. Based on [2], about 40 million Ford vehicles were produced by 1960. Ford was not the only company producing vehicles in the first half of the twentieth century in the U.S. Therefore, the total number of cars produced in the U.S. since the early twentieth century is estimated at about 700 million.

In 1910, the price of a Model T car was $3,000 (in 1958 dollars) [2], which is about $25,000 in 2016 dollars. Manufacturing costs account for about 50% of the MSRP and engine costs account for 18.5% of total manufacturing costs [10], a relationship assumed to be always true throughout history for this discussion. Therefore, engine costs are 9.25% MSRP.

For the Model T, this suggests that a 1910 engine cost about $2,300. The Model T had about 20 HP, or about 15 kW. Therefore, in 1910, an engine cost about $150/kW.

Using the Ford Model T data from [2], the cost of the first engine can be extracted. In 1910, already 50,000 Model Ts had been manufactured (see Fig. 1.1). Model Ts
were not the only vehicle being produced; so an estimated seven or eight times that figure may be reasonable, which means a total of 300,000 vehicles were manufactured in the U.S. by 1910. If a learning rate of 20.5% (the average learning rate for “small technologies” ([12]) is used, the first engine would cost about $9,800/kW (this is found by manipulating the equation for the learning curve as shown in (2.25)).

$$k(1) = \frac{k(n)}{n^{\log_2 \epsilon}} = \frac{150}{300,000^{\log_2 0.795}}$$  \hspace{1cm} (2.25)

Using this figure, the cost of an engine today (after 700 million units produced) can also be extracted by solving: \(k = $9,800/kW(700 \times 10^6)^{\log_2 0.795}\), which is about $12/kW. Again, this is a remarkable prediction of today’s engine costs per unit output, based on mass-manufacturing alone.

A 2017 Honda Accord starting MSRP is $22,355 and has 212 HP (158 kW). 9.25% of the MSRP is $2,068, which suggests the engine costs $13/kW. Therefore, the total production figures, learning rate, and engine cost data at the resolution of this analysis are well aligned. Mass manufacturing (producing millions of vehicle internal combustion engines) has led to these cost reductions.

As per (2.7), if a combination of both scaling in numbers and scaling in unit size is evaluated, as shown in (2.26), the cost per unit output for today’s engine is about half the value it should be. This shows that in the case of engines, it was mass-manufacturing that got them to their current cost per unit output. For all practical scaling applications, it is safe to assume an engine has maintained about the same size over its lifetime.

$$k_n = (\$9,800/kW)(0.795)^{29.3} \left( \frac{111kW}{15kW} \right)^{0.66-1}$$  \hspace{1cm} (2.26)
2.3.2.2 Power plant generators

Over time, power plant generator size has changed. Plotted in Fig. 2.5 are U.S. conventional steam coal power plant generator sizes as a function of year operation began (data from U.S. Energy Information Administration (EIA) [3]). This figure captures both an upward trend in generator size as well as the longevity of coal power plants. Plants from the 1950s and earlier have yet to be retired as of 2015 [3].

EIA data on coal-fired power plants begins in 1921. According to the EIA, 427 unique coal plants were operational in the U.S. in 2015 (of those, 425 are “conventional steam coal,” 2 are “coal integrated gasification combined cycle”). By 2015, 205 coal plants had been retired (of those, 204 were “conventional steam coal,” 1 was “coal integrated gasification combined cycle”). Therefore, a total of 632 coal-fired power plants have been built in the U.S.

In terms of generators, 968 coal generators were operational in the U.S. in 2015 (of those, 963 are “conventional steam coal,” 5 are “coal integrated gasification combined cycle”).
cycle”). By 2015, 485 coal generators had been retired (of those, 483 were “conven-
tional steam coal,” 2 were “coal integrated gasification combined cycle”). Therefore,
a total of 1,453 coal-fired generators have been built in the U.S.

The cost of Pearl Street Station (the first power plant) in 1882 was about $300,000,
which translates to about $6.7 million in today’s dollars. The total plant capacity
was 600 kW (made up of six-100 kW generators). The cost, therefore, was about
$11,000/kW. Today’s power plant costs somewhere between $1,000 and $2,000/kW.

For power plants, cost per power output ($/kW) may have decreased because 1) individual
units increased in size (two-thirds law), 2) more power plants were built (learning
curve), or 3) a combination of 1) and 2).

Based on the cost and size of Pearl Street Station, the cost of today’s plant can
be predicted based on each scaling law individually: first on the two-thirds law and
second on the learning rate. Referring back to the equation for scaling capacity in
unit size as per (2.1), and addressing for the change in generator size, today’s cost
is solved for in (2.27) by using the Pearl Street Station cost per unit output and
generator capacity size and an $\alpha$ equal to 0.7 (as per an estimate in [13]). The
capacity of today’s generator is calculated as the average of coal generators built in
the last decade, as shown in Fig. 2.5. The average size from 2005 to 2015 from this
data is 606 MW [3]. (For comparison, the average size of generators in operation in
2015 (regardless of year built) is 315 MW.)

$$k(c) = $11,000/kW \left( \frac{606 \text{ MW}}{0.1 \text{ MW}} \right)^{0.7-1} \tag{2.27}$$

Based on (2.27), today’s power plant should cost about about $800/kW. This
figure is lower than today’s costs likely due to the changes made in power plants over
the last decade. A modern power plant generator has more components (for example
those to address environmental concerns), which were not part of Edison’s power
plant.
If instead the cost reductions are based on the fact that more than one power plant has been built, economies of numbers predicts another value for today’s costs. The learning rate is set to 10.8% (\( \epsilon = 0.892 \)) based on an estimate from [13]. Therefore, if the first plant was $11,000/kW and about 1,453, or \( 2^{10.5} \), generators have been built, the estimate for today’s costs is: \( 11,000 \times (0.892)^{10.5} = $3,300/kW \). Recall that this figure assumes every unit has stayed about the same size.

Because power plants have scaled in size and numbers, it should be a combination of the above two laws that yields the cost reductions found today, as per the equation given in (2.7). Combining the above figures yields the calculation in (2.28).

\[
k_n = (\$11,000/kW)(0.892)^{10.5} \left( \frac{600}{0.1} \right)^{0.7-1} \tag{2.28}
\]

As (2.28) shows, the estimate for today’s power plant generator cost based on the number of units built as well as the change in capacity size results in a value of $250/kW, which is lower than today’s actual costs. There are two reasons this is reasonable. The first reason is that power plant generators likely learned at a lower rate, which results in a higher estimate for \( \epsilon \). At an \( \epsilon \) set at 1, which is essentially suggesting there was no learning, the estimate for today’s costs, however, is still just $800/kW. This figure is still too low, which raises the second reason for these low estimates, which was mentioned above: today’s power plant generators are more complicated than the first generators built.

What this analysis shows is that unlike engines where mass-manufacturing, and therefore the learning rate, brought their costs down, power plant generators are the opposite. Their cost reductions are largely due to the fact that they have gotten bigger in individual unit size and less so that more than one generator has been built.
2.3.3 Power plant generator cumulative capacity path

Earlier in this chapter total costs were explored by evaluating cumulative capacity paths. Because power plant generators and internal combustion engines have different cumulative capacities, it is not meaningful to compare their total costs. An analysis is included in this section, however, to show how a cumulative capacity path is constructed, here using power plant generators as an example.

For coal-fired power plant generators, the cumulative capacity is overlaid on the coal generator data from Fig. 2.5 as shown in Fig. 2.6.

Two fits were tested of the forms given in (2.29) and (2.30).

\[
C(n) = \frac{L}{1 + \exp(-k(n - n_0))} \quad (2.29)
\]

\[
C(n) = C_3 n^3 + C_2 n^2 + C_1 n \quad (2.30)
\]
The results of both fits are shown in Fig. 2.7 along with the actual calculated dataset. From (2.29), \( L \) is equal to 363.7 (the cumulative capacity, in GW), \( k \) is equal to 0.0065, and \( n_0 \) is equal to 950. In (2.30), \( C_3 \) is \(-5 \times 10^{-8}\), \( C_2 \) is 0.0003, and \( C_1 \) is \(-0.0584\). Unlike the previous plots, this plot shows the cumulative generator output, \( C(n) \), versus units produced, \( n \).

Figure 2.7: Cumulative capacity path, \( C(n) \) versus number of units produced, \( n \); includes actual path, and two calculated paths: an exponential and a power law

Putting it all together, Fig. 2.8 shows the original dataset with the cumulative capacity paths (actual and the two calculated) overlaid on the right y-axis.
2.4 Internal combustion engines as building blocks

The mathematical foundation behind scaling laws provides a justification for focusing on small-scale designs. When advanced automation and the capacity to mass-manufacture are available, the environment is ready for small-scale, modular technologies. The opportunity for low-cost, small-scale infrastructure enables one to push the boundary between large versus small scales. The following chapters focus specifically on exploring the value of internal combustion engines as building blocks, first specifically for power generation and then for a small-scale compressor for chemical engineering industries.

As was discussed in Section 2.1, modular designs, particularly in energy systems,
provide increased flexibility. To enable small-scale designs to start at a low-cost base, the focus on this thesis is on utilizing internal combustion engines and adapting them for new applications. Internal combustion engines are ubiquitous; they are manufactured as a component of automobile production across six continents and in over fifty countries [32]. According to the U.S. Department of Transportation, about 2.5 billion vehicles have been produced globally between 1961 and 2015 [49]. The annual global production of cars and vehicles is likely to pass 100 million before 2020 as per the rates shown in Fig. 2.9.

![Figure 2.9: Global number of vehicles produced annually from 1999 to 2016](image)

For over a century, automobiles have been produced, which (as was detailed earlier in this chapter) has brought the cost of engines down to about $10/kW [29]. German engineer, Nikolaus Otto first developed internal combustion engines over 140 years ago, and Rudolf Diesel followed in 1892 with the development of the compression-ignition (diesel) engine [22].

In this thesis, internal combustion engines are used for two main applications. In Chapter 3 and Appendix A the value of small, modular engine gensets committing
to power delivery in energy and reserve markets is quantified. In this study, engines are used as a means of producing power in the electricity sector (think broadly of a power plant comprising many small engine genests). The analysis shows that the greatest potential is in reserve markets. Because power is committed as standby and not always called upon to be delivered in the reserve markets, there is the potential for profits even when prices are low. This application truly takes advantage of the low capital cost of an internal combustion engine. With increased penetration of renewables, engine gensets as reserve power generators become even more interesting because engines are inexpensive and can quickly ramp up to full power output.

In Chapter 4, Chapter 5, and Chapter 6, a piston engine is rebuilt and modeled to function as a small-scale gas compressor. Because gasoline engines are designed for combustion, the engine head and cylinder can withstand high pressures. The engine durability combined with its low cost makes it an attractive component for repurposing as a small compressor. Included in Chapter 7 is a look at how the system economics play out.

The scaling laws explored at the beginning of this chapter provide a foundation for pursuing small scales and begin to show one of the benefits that arise from taking advantage of mass-manufactured components which have come down their learning curves. The specific applications for engine technology explored in the next few chapters build off this foundation and show favorable results for using engines in and adapting engines for unconventional applications.
Chapter 3

The value of small-scale internal combustion engine gensets as energy infrastructure

This chapter provides additional background details and supplementary information to accompany the paper provided in Appendix A. The paper in Appendix A is a draft of a current paper under review for publication. This research quantifies the value of engine generator systems (engine gensets) participating in energy and reserve markets. An optimization is written to maximize profit by determining the optimal engine power commitments to the energy market, reserve market, both markets, or neither market. The optimization is a function of engine cost, engine wear, engine lifetime, and fuel cost. This analysis finds that the annual profit for an engine genset acting as a price-taker is on the order of $30/kW. The full details, including sensitivity analyses are included in Appendix A. This chapter provides a more detailed background and motivation for the paper and includes additional information on mapping engine performance.

3.1 Engines as energy infrastructure

One of the major contributions of this thesis is looking at ways to incorporate internal combustion engine technology in energy infrastructure. The first analysis focuses on using internal combustion engine gensets for power production. Rather than traditional, large generators, this design focuses on viewing power plants as many small, modular, flexible internal combustion engine gensets. More specifically, this research
evaluates the opportunity to use internal combustion engine gensets for both energy and reserve market commitments. Because engines can reach 100% load delivery in a fraction of the time of conventional gas and steam turbines, there is a compelling argument for using engines for spinning reserves and grid balancing. The trade-off between committing to and delivering power in the energy market, committing to and delivering when called upon in the reserve market, and committing to deliver in neither market is evaluated. This optimization is a function of the electricity and spinning reserve prices provided by the New York Independent System Operator (NY-ISO), the likelihood of delivering spinning reserve capacity, the price of fuel (natural gas), and engine fuel consumption, wear, cost, and lifetime.

The full analysis is provided in Appendix A. The remaining sections of this chapter provide a background and additional details to supplement the paper.

3.1.1 Findings

The paper’s abstract is reproduced here:

When normalized by power output, internal combustion engines are one hundred times less expensive than conventional, large power plants. Engines are mass-manufactured, low-cost, short-lived power generators that can rapidly change output from zero to full power. In this analysis we treat them as consumables, as an operational rather than capital cost. This paper briefly reviews the economics of scaling, modularity, and centralization vs. distribution in the context of power generation and then presents a model that optimizes how an internal combustion engine and electric generator (‘engine genset”) could participate simultaneously in the day-ahead energy and ancillary services (spinning reserve) markets. The paper quantifies the value of small-scale, modular, mass-manufactured components in energy markets, with an example focused on internal combustion engine gensets for electricity generation. The optimization shows that net annual profits from committing power output from an engine genset in the day-ahead energy and spinning reserve markets are positive in all cases analyzed.

The optimization in the paper is run at every hour for a year. At each time step, the model optimizes power commitments based on the engine costs (wear and fuel)
and the electricity prices. The optimization finds that the expected annual profit of
an engine participating in energy and reserve markets is on the order of $30/kW. Over
80% of the profit comes from committing power to the spinning reserve market. In all
sensitivity analyses (where fuel price, electricity prices, engine cost, and probability of
being called on to deliver in the reserve market are varied), the majority of the hours
of the year are spent making power commitments in the spinning reserve market only.
This shows a strong incentive to focus efforts on the reserve market and provides a
motivation to pursue this technology further, given that reserve prices may increase
as a function of higher penetration of renewables on the grid.

A unique feature of the model is how the cost of the energy generation technology
(the engine genset) is captured. The first engine purchased is a capital cost. Every
engine purchased thereafter (the replacements) is an operational cost. The engine’s
short lifetime has little impact on the overall system cost because the engine costs are
so low. In fact, the cost of fuel overshadows the engine replacement cost. This makes
the system very unique because it has the ability to trade engine wear against the cost
of power, which is very different from what a large power plant can do because the
plant is viewed as a capital investment. The generator is longer-lived and is viewed
as a capital cost in this analysis, though this may not need to be the case in practice
if less expensive generators can be used.

3.1.2 Background
In his 2013 Master’s thesis, Malco Parola leverages the idea of modularity and eval-
uates how a power plant comprising internal combustion engines would compare to
a conventional power plant [34]. He found that, because engines can ramp up and
down almost instantaneously, the power plant made of engines can essentially turn on
and off at no cost. Parola finds that this feature makes them flexible in capitalizing
on high electricity prices. Parola focused on evaluating how a power plant with total
capacity of 500 MW, constructed from 10,000 small, 50 kW engines compares to a
conventional plant. To do so, he simplifies his approach by staying at a high level in
terms of mapping engine performance and describing the transition from mechanical
engine output to electrical power commitments. Parola performs sensitivity analy-
ses and finds that net present values are positive when natural gas prices are below
$10/MMBTU, engine (power plant) efficiency is above 6 or 7%, and operational and
maintenance costs are below about $90,000/hr [34]. Parola finds that his analysis
results in a very low capital cost, just one tenth of the overnight cost of $977/kW for
a 540 MW gas or oil combined cycle power plant as per the U.S. Energy Information
Administration. Parola’s work provides a data point for a high, systems-level view of
the potential role of engines in energy systems. Perhaps the most significant part of
Parola’s work is how he treats the actual engines in his financial analysis; the engines
are viewed as consumables, as operational rather than capital expenses.

3.1.3 Motivation

With Parola’s work as a foundation, a more sophisticated model and optimization has
been performed. A paper written on this topic is included in Appendix [A]. Leveraging
the SciPy package, an optimization was implemented in Python to determine how
an engine genset should commit to delivering power in the New York Independent
System Operator (NYISO) energy and reserve markets. Engine performance was
modeled in GT-POWER. Day-ahead electricity prices from NYISO were put into the
model at every hour for a year.

A few major changes from Parola’s work are in the way this study characterizes
engine performance, in the inclusion of generators, in the ability to commit to deliv-
ering in both energy and reserve markets, and in the optimization over one-20 kW
engine rather than an entire power plant. With regards to optimizing over one en-
gine only, note that this optimization finds that optimal power output is not at peak
power output, so optimizing across one engine is in fact the same as optimizing over hundreds or thousands of engines, since it will be optimal for all engines to perform as the one engine optimized in this study performs. Implicit in this conclusion is the assumption that this plant is a price-taker and not a price-setter. If the plant were large enough that its operation would significantly change supply, then the price offered would depend on the number of engines running. However, the assumption here is that each engine sees the same market and therefore will perform the same optimization. In this analysis, the initial engine purchased is a capital cost, but every engine after the initial purchase is treated as a consumable. Its replacement, therefore, becomes an operational cost. Structuring the analysis this way is based on Parola’s work, as described earlier.

3.2 Mapping engine performance

One important contribution of this analysis is characterizing engine performance under different operating conditions. This work quantifies the trade-off between fuel consumption and power output and uses this information to strategically determine how an engine should optimize its power commitments.

To complement the information provided in Appendix A, the following sections include the fundamental equations governing engine performance and expand on how the engine simulation was designed in GT-POWER to create the engine performance map used in this optimization.

3.2.1 Engine map fundamentals

Engine performance is described by power output and specific fuel consumption; to understand the relationship between the two, engine performance maps are used. Performance maps plot contours of constant power and specific fuel consumption on a plot of brake mean effective pressure versus piston speed.
Contours of brake mean effective pressure are determined from power per area and piston speed as shown in (3.1), where $P_b$ is brake power, $A_p$ is the piston area, $n$ is the number of cylinders, $X$ is 2 for 4-stroke engines and 1 for 2-stroke engines, and $\bar{V}_p$ is the average speed.

\[
\text{bmep} = \left( \frac{P_b}{A_p n} \right) \frac{2X}{\bar{V}_p}
\]  

(3.1)

Fig. 3.1 shows a performance map for a carbureted engine (reproduced from [30]), which illustrates the relationship between the parameters discussed above. Fig. 3.1 shows bmep (brake mean effective pressure) versus piston speed. Plotted in this space are bsfc (brake specific fuel consumption) contours and contours of power per unit of piston area. The optimal operating point is where specific fuel consumption is lowest and power is highest. The area of lowest specific fuel consumption is inside the oval in the upper left of the plot, while constant power contours increase from left to right across the plot. There is a trade-off between minimizing fuel consumption and maximizing power output.

While constant specific power can be calculated from knowing engine specifications alone (piston size, number of cylinders, 4 versus 2-stroke), specific fuel consumption cannot. Specific fuel consumption is determined experimentally by running an engine over full range of load and speed and measuring torque and fuel flow-rate data from a dynamometer.

An alternate method for characterizing engine performance data without running an engine on a dynamometer is to model the engine. In this research, GT-POWER Engine Simulation Software, an industry standard in engine simulations, was used to model a generic, one-cylinder engine. The engine model provided an environment to run simulations with varying engine speed and air-to-fuel ratio in order to construct an engine performance map that could be used as a tool in the engine model optimizations. The performance map provides a method for quantifying the trade-off
3.2.2 Engine simulation

GT-POWER is a robust engine simulation software and the industry standard. GT-SUITE is used by numerous companies, including GM, Ford, BMW, Volvo, Mercedes-Benz, Cummins, and Chrysler \[45\]. Modeling the engine in this software provided the flexibility to look at different engine types and to cover a much wider parameter range more quickly than could be captured on a test stand. GT-POWER provides users with an environment for designing all aspects of the engine, cranktrain, valvetrain, and fuel system.
cooling, and many more features.

The simulation environment is put together by defining individual objects, which are then connected to produce the full model. The design for the one-cylinder engine model built for this application was largely based on one of the GT-SUITE Engine Performance Tutorials published by Gamma Technologies (Version 2016) [19]. The tutorial provides a guideline for creating objects, using built-in models (for heat transfer, friction, etc.), and running simulations. The tutorial steps are summarized below. For full details of the process, see [19].

For this application, a number of parameters/objects are defined:

- the environment inlet and outlet conditions, including temperature, pressure, and composition,
- the runner pipes connecting ambient air to the engine intake ports, which are characterized by their pipe geometry, roughness, and wall temperature properties,
- the intake and exhaust valves,
- the valve timing,
- the engine cylinder, which consists of defining the cylinder heat transfer (this simulation uses the built-in Woschni heat transfer model as per the suggestion of [19] because it is the industry standard for in-cylinder heat transfer),
- the combustion object, which uses the Wiebe curve to model the combustion burn rate as per [19],
- the fuel injector, which includes setting the fuel delivery rate, the air-to-fuel ratio, injection timing angle, and injected fluid temperature, and
the engine cranktrain object, which sets the number of strokes (this system uses a four-stroke engine), the engine speed, engine friction (defined by the Chen-Flynn model\cite{19}), and engine cylinder geometry.

Because this model is used to map engine performance, the engine is run over a range of speeds (from \(<1,000\) to \(8,000\) rpm) and air-to-fuel ratios (from 12 to 21). This provides the data to construct a plot of constant air-to-fuel ratio contours in two coordinates: first on a map of power output (kW) as a function of speed (rpm) and second on a map of brake specific fuel consumption (g/kWh) as a function of speed. This data is used to construct the engine performance map needed to optimize how an engine should be run given various boundary conditions. For the resulting engine performance map, see Fig. A.4.

3.3 Other notes

A few additional comments on electricity markets and prices, the electric generator, the choice to be asynchronous rather than synchronous, and the choice to optimize over one engine are provided in this section.

3.3.1 Electricity markets

Two electricity markets (energy and reserve) are included in this optimization. In the energy market, a power commitment requires that the energy be delivered to the grid at the agreed upon time, and a price is paid for that energy. In the reserve market, capacity is committed, but not always called upon to be delivered to the grid. In this market, a price is paid for the capacity committed and an additional price is paid for the energy delivered if/when called upon.

For this optimization, two specific markets were selected from these two groups. In both cases, day-ahead prices are used, which takes away speculation. The Locational Based Marginal Price (LBMP) is used for the energy market; the 10-minute Spinning
Reserve is used for the reserve market. These markets were selected because they are well established. Future work should focus on optimizing across various other electricity markets. Because the engine is so fast in ramping up, it may be useful for other services that require a very quick response, like grid stabilization.

### 3.3.2 Electricity prices

In the paper, day-ahead New York Independent System Operator (NYISO) electricity prices are used in the optimization. Day-ahead prices were selected because in this model, commitments are set without the option to change in real-time as function of being called upon to deliver spinning reserve capacity. In real-time, one can imagine committing to spinning reserves, but if not called upon, changing the commitment to the energy market. This option was not considered in this optimization. The regulatory framework needs to be investigated in more detail for future work, because the option to jump from one market to another may create a regulatory issue. For example, if the generator is committing to provide real-time power, it does not make sense then to jump out of this market in order to deliver reserves. Therefore, what commitments are permitted and in what time frame should be investigated further.

Day-ahead energy and reserve prices are set one day in advance, with the aim of reducing real-time volatility. For comparison of day-ahead and real-time prices, monthly averages are found. Across the month of June 2015, the NYISO average day-ahead electricity price (in the zone/market used in this paper) was $0.0246/kWh, and the average real-time electricity price was $0.0262/kWh, or, on average, 6.6% higher. In a particular day, day-ahead prices may dominate (see Fig. 3.2), but real-time prices can have very large spikes (see Fig. 3.3). The zoomed in view of one day shown in Fig. 3.2 can be identified as the first day in Fig. 3.3. The week-long view in Fig. 3.3 shows the infrequency of day-ahead prices being significantly higher than real-time prices. Instead, there are scattered times of very high real-time prices (notice the
spikes over $1/kWh). Even though, on average, real-time prices are just 6.6% higher than day-ahead prices (in the month evaluated; this figure is closer to 12% across the year), the very high volatility in real-time prices (sometimes over forty times as high as the average electricity prices) means that there can be very rewarding times to commit power in this market. The optimization in this paper misses the opportunity to capitalize on these very high electricity prices.

For the reserve markets, real-time prices are often significantly lower than day-ahead prices. In June 2015, average day-ahead spinning reserve prices were $0.00283/kWh, whereas average real-time reserves were $0.000378/kWh. For reserves, real-time prices are, on average, just 13% those of day-ahead (in the month evaluated; this figure is below 10% across the entire year). For a significant portion of time, real-time reserves are priced at $0. This suggests that it may be advantageous to continue to commit in the day-ahead reserve market. If, when not called upon to deliver reserve capacity, the engine can deliver in the real-time energy market, there may be significant financial incentives for this system design, beyond what is captured.

Figure 3.2: NYISO real-time (5 min. resolution) and day-ahead (60 min. resolution) electricity prices for June 1, 2015

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in the paper in Appendix A. Fig. 3.4 shows the full month of reserve prices in June 2015. Notice that spikes in reserve prices in real-time are much less frequent than (and not as high as) spikes in electricity prices.

Figure 3.3: NYISO real-time (5 min. resolution) and day-ahead (60 min. resolution) electricity prices for June 1 - 7, 2015

Figure 3.4: NYISO real-time (5 min. resolution) and day-ahead (60 min. resolution) reserve prices for June 1 - 30, 2015
A future model should optimize with real-time prices and investigate energy and reserve commitments more to determine if it is possible to shift to the energy market if capacity is not called upon to deliver in the reserve market. It may also be possible to optimize across commitments both in the day-ahead and real-time markets.

Additionally, the relationship between real-time and day-ahead markets will likely vary in different ISOs. The NYISO has a Virtual Trading feature, which, according to [31], is used to:

- effectively improve the efficiency and accuracy of the day-ahead energy markets by allowing market participants to schedule financial transactions that arbitrage the price difference between the day-ahead market and the real-time market. Virtual trading reduces the risk premiums in the day-ahead markets by allowing market participants to account for uncertainties such as wind resource output, weather changes or unexpected transmission outages. Virtual trading results in a more efficient and competitive selection of resources to meet real-time conditions and encourages participation in the day-ahead market. [31]

NYISO’s Virtual Trading closes the gap between real-time and day-ahead electricity prices, a feature that may not be as strong in other ISOs.

### 3.3.3 Generator
In the paper, an asynchronous generator is used. In this analysis, this means that the generator can see varying speed from the engine. This design decision allows the analysis to optimize how the engine should be run based on varying both speed and air-to-fuel ratio. The optimization was based on choosing the lowest cost option for delivering a particular power output, which took into account engine wear and fuel costs.

If instead, the generator is synchronous, it would see one constant engine speed. If this was the case, the optimization becomes simplified because now, varying the air-to-fuel ratio is the only option for changing the power output at a constant engine speed. A comparison between the profit optimizations with either an asynchronous
or a synchronous generator show that, depending on the application, the additional infrastructure needed for the asynchronous case is outweighed by the benefit of having the flexibility to optimize and operate over the full range of engine speeds and so change power output by varying both engine speed and air-to-fuel ratio. Furthermore, for a large fleet of engines, there also would have to be a significant investment into assuring synchronization between all units. This can be avoided if all units feed into a DC power supply, which then connects to the grid with a single DC-AC conversion. Whether or not this advantage compensates for the overall cost depends on many external factors not considered here. For example, if the desired product is DC power, then being asynchronous would not add much of a disadvantage. Even a generator operating synchronously needs a control system/regulator to give it control over how much power it delivers to the grid. The power output controls fuel efficiency and wear on the engine, which both must be managed by the operator.

The difference in profit as a function of being asynchronous or synchronous is quantified next.

### 3.3.4 Quantifying the benefit of being asynchronous

To quantify the added value of an asynchronous generator (the choice for the analysis in the paper in Appendix A), the same optimization was run, again over a full year’s worth of data, for cases where the generator is synchronous with the grid. With this boundary condition, the engine runs at one constant speed. The range of power output is reduced with this restriction because now the only avenue for varying power output is by changing air-to-fuel ratio (since speed is set). Two speeds were selected for the analysis. The first speed of 1987 rpm was selected based on the minimum value on the original total cost per unit of power output curve. At the minimum cost, the power output is 5.6 kW (see Fig. 3.5). At this power output, the optimal speed is 1987 rpm. For the synchronous generator case, this speed is now set. Holding the
speed constant, the range of power output is now 5.4 to 7.2 kW (see Fig. 3.5), which is achieved by varying the air-to-fuel ratio (from a minimum of 17 (stoichiometric) to a maximum of 21 (a leaner mixture)).

![Figure 3.5: Total cost ($/kWh) of the engine as function of power output (kW) for the original (asynchronous generator) optimization and for the synchronous generator optimization with the engine running at 1987 rpm](image)

The results of this optimization over one same sample week, June 1 - 7, 2015, (the same sample week as shown for the asynchronous case in Fig. A.7) are included in Fig. 3.6. Notice that, in general, the trends are similar to those in the asynchronous case, but now the window of available power commitments is narrowed. While just one week’s of data is shown here, the optimization was again run over the full year so that annual profits can be compared for the asynchronous and synchronous cases. In the original analysis, the total annual profit is $27.55 (see Table A.2 in Appendix A). With the restriction on speed placed in the optimization here, the total annual profit is about $11.45/kW, or about 40% what is was in the original, asynchronous case. Note the engine capacity here is still 20 kW. If instead the annual profit is divided
by the peak power output at this engine speed (7.2 kW), then the total annual profit is over $31/kW. However, the return on the investment is greatly reduced.

![Graph showing power output and market prices](image)

**Figure 3.6**: Sample week of June 1-7, 2015 showing model output for power to the electricity market (P1) and the sum of power to both the electricity and the spinning reserve markets (P2) for engine running at 1987 rpm (top) and energy and spinning market prices (bottom)

To show how these results vary based on the speed selected, a higher speed of 4200 rpm was also tested. At this speed, the power output ranges from 11.7 to 16.3 kW by varying air-to-fuel ratio as shown in Fig. 3.7. The range of possible power output is larger than when speed was held constant at 1987 rpm because, as speed increases, the spread in power output due to changing air-to-fuel ratio increases (see Fig. A.4). In practice, however, the range of power output is quite small because the engine optimization will never select a power output between 11.7 and 14.3 (the beginning of the total cost curve as shown in blue in Fig. 3.7) because costs are decreasing in this range with increased power. It is never advantageous to select a power output where the slope of the total cost curve is negative because it is less expensive to commit a higher power output. Therefore, the actual range of power output is between 14.3
and 16.3 kW.

![Graph showing the total cost ($/kWh) of the engine as function of power output (kW) for the original (asynchronous generator) optimization and for the synchronous generator optimization with the engine running at 4200 rpm.]

Figure 3.7: Total cost ($/kWh) of the engine as function of power output (kW) for the original (asynchronous generator) optimization and for the synchronous generator optimization with the engine running at 4200 rpm

With the engine speed locked in at 4200 rpm, the total annual profit is $24.90/kW, which is over 90% the profit of the asynchronous generator case. The results of the same first week of June 2015 are shown for this optimization in Fig. 3.8. Notice how similar the result is in Fig. 3.8 with the original optimization results shown in Fig. A.7.

Notice that during the week shown in both Fig. 3.6 and Fig. 3.8 most of the profit comes from committing in the reserve market. The reduction in engine capacity, therefore, is largely what affects the profit. This is not always the case across the year, however. When energy prices are higher, the optimization will look a lot different. For this very reason, the optimization was run across an entire year.

This analysis shows that the constraint in engine speed for a synchronous generator design does not, by necessity, eliminate the ability to achieve similar profits to
Figure 3.8: Sample week of June 1-7, 2015 showing model output for power to the electricity market (P1) and the sum of power to both the electricity and the spinning reserve markets (P2) for engine running at 4200 rpm (top) and energy and spinning market prices (bottom)

those in the asynchronous case. On the other hand, the profit is reduced, and this reduction allows for a significant investment in conversion technology to connect to the grid. The engine speed selected for the synchronous generator will dictate the possible power output range and thereby the achievable system profit.

### 3.3.5 One-engine optimization

In the optimization of one-20 kW engine, the analysis finds that optimal engine performance is not at peak power. Optimizing how one engine would perform is in fact optimizing how hundreds of engines would run in a pack. A key ingredient here is that this analysis assumes the capacity of the market to accept the engine power output is infinite and the engine is a price-taker. If, on the other hand, the boundary condition was such that the engine needed to match demand, the optimization would
be different, and it would be drastically shaped by the assumptions of how demand is shaped by price. This was not the setup for this analysis, however. Here the assumption is that the engine farm is a price-taker.

3.4 Conclusions

The analysis provided in this chapter and in Appendix A shows a case for using small engine gensets in the energy and reserve markets. The results are compelling. Engine gensets can be profitable in energy markets, particularly in situations requiring a quick ramp up. For additional discussion, results, and sensitivity analyses, see Appendix A.
Designing and testing a retrofitted engine to compressor

4.1 Engines as compressors

Across many industrial processes, gas compression is cited as a critical component that resists cost-efficient downscaling. For some projects, compression technologies alone dominate system costs \([40], [33], [9], [1]\) (more on this in Chapter 7). Affordable compressors, particularly on small scales can shift many projects from non-viable to practical by significantly decreasing system costs. The challenging engineering task is to design a compressor that is low cost whilst maintaining similar efficiencies to the current state of the art. Beginning with a new, first of its kind compressor design is costly and will require much iteration to come down the learning curve to reduce costs, which is necessary to compete with incumbent technologies. Rather than design a small-scale compressor from scratch, this project focuses on a compressor design based on modifying an internal combustion engine, an already well-developed and low-cost component for the new purpose of gas compression. This new design begins with a low-cost foundation with its learning curve starting on the tail of the already mature learning curve of the internal combustion engine, which has been driven down by the large number of automobiles that has been produced.

This idea for redesigning an already mature technology for a new purpose can be extended beyond gas compressors. In fact, many new technologies do build off of other designs. They capture salient features of an older technology and reduce design risks by modifying an operational design. For this application, some of the key attributes that were looked for in selecting another technology to modify were
durability, affordability, and the capacity to withstand high pressures. The technology that encompasses all the desired attributes is an internal combustion engine (ICE). What is novel in this approach is not that a new system arises from modifying an existing design, but that the existing design was selected based on its high level of maturation and on its extremely low unit cost, rather than with a focus of having a closely related functionality.

Though designed for combustion processes, at their core, internal combustion engines are reciprocating piston compressors. Engine blocks are designed with one or more cylinders each containing a piston that moves from top to bottom dead center over a crankshaft revolution. Because the piston/cylinder is designed to handle combustion, cylinder pressure can be over 1,000 psi [20] (also see [22], [30], [39], and [5]). In normal use, an ICE uses the chemical energy from fuel combustion to drive the crankshaft and produce mechanical work output. In this modified design, there is no combustion, and therefore no work output. Instead, there is a work requirement to drive the engine as a gas compressor. In the laboratory, an electric motor drives the modified engine compressor. The shaft of the electric motor is coupled with the engine shaft, and the electric power input to the motor turns the engine. The choice of an electric motor was made for practical reasons of working in a laboratory. The work input could also easily be another, unmodified, internal combustion engine producing mechanical work output that drives the engine compressor. The power rating of the unmodified engine driving the engine compressor will depend on the effective compressor power rating, which is a function of the compression ratio and pressure regime in which the compressor operates.

Internal combustion engines are ubiquitous due to mass-manufacturing in the automobile industry over the past one hundred years, which has led to significant declines in costs. Engines cost about $10/kW [29] (about $7.5/HP). Current compression technologies are on the order of $1,000 - $5,000/HP [41]. This was pointed
out in Chapter 1, but important to note again: when comparing the engine and compressor power output figures, note that these are not the same power scales. The compressor output depends on the compression ratio, system efficiency, etc. Because mass-manufactured internal combustion engines are two orders of magnitude less expensive than state of the art compressors, there is a strong financial incentive to pursue a gas compression technology based on modifying low-cost engines. The caveat is, of course, that engine modifications must not be so costly that they render the engine financially uncompetitive.

Two intertwined efforts were pursued to explore how an engine would and could be redesigned to function as a small-scale gas compressor: an engine was retrofitted in the laboratory and an engine compressor model was designed and written in Python. These two tools interacted in an iterative fashion. Experimental analysis was used to both verify and calibrate the engine model and the model was used to inform the experiments. This chapter focuses on the experimental results; Chapter 5 focuses on the engine model: how it was designed, how calculations are performed, how parameters were set, and what output can be expected. Chapter 6 brings the experimental work and the modeling together. The model is calibrated and validated with experimental data. The agreement between experimental results and modeling is strong.

4.2 Compressor background

Compressors fall into two main categories: positive displacement or continuous-flow. Positive displacement compressors include reciprocating (single and double acting) and rotary (blow and screw). Continuous-flow compressors (turbomachines) include axial and centrifugal. The compressor designed as part of this work is a reciprocating compressor built by modifying the cylinder and piston assembly of an internal combustion engine.

Compressors are of interest because they are a very costly component of many
energy and chemical engineering systems and, in some cases, dominate total system costs ([40], [33], [9], [1]). The global compressor market is expected to surpass $37 billion by 2022 [4]. The large market potential provides an incentive to explore alternative compressor designs. This thesis proposes a compressor built by retrofitting an internal combustion engine. An engine is selected because it provides a low-cost starting point, even for small scales. The economics of the proposed system versus the commercially-available options are covered in Chapter [7]. The rest of this chapter focuses on the laboratory experimental work, which covers how the engine was retrofitted and the compression performance achieved from various designs. This research finds that an engine can be retrofitted to perform as a gas compressor with few and simple modifications.

4.3 Why retrofit engines?

Throughout this thesis, the motivation for focusing on incorporating internal combustion engines into unconventional applications in chemical engineering and power generation systems is highlighted. For compression technologies specifically, engines are selected because they are inexpensive, ubiquitous, and designed to withstand very high pressures because they are built to handle in-cylinder fuel combustion.

Included below is a brief description of a typical engine cycle from [22]:

The majority of reciprocating engines operate on what is known as the four-stroke cycle. Each cylinder requires four strokes of its piston - two revolutions of the crankshaft - to complete the sequence of events which produces one power stroke. Both SI [spark-ignition] and CI [compression-ignition] engines use this cycle which comprises (see Fig. 1-2 [reproduced in Fig. 4.1]):

1. An intake stroke, which starts with the piston at TC [top center] and ends with the piston at BC [bottom center], which draws fresh mixture into the cylinder. To increase the mass inducted, the inlet valve opens shortly before the stroke starts and closes after it ends.
2. A compression stroke, when both valves are closed and the mixture inside the cylinder is compressed to a small fraction of its initial
volume. Toward the end of the compression stroke, combustion is initiated and the cylinder pressure rises more rapidly.

3. A power stroke, or expansion stroke, which starts with the piston at TC and ends at BC as the high-temperature, high-pressure, gases push the piston down and force the crank to rotate. About five times as much work is done on the piston during the power stroke as the piston had to do during compression. As the piston approaches BC the exhaust valve opens to initiate the exhaust process and drop the cylinder pressure to close to the exhaust pressure.

4. An exhaust stroke, where the remaining burned gases exit the cylinder: first, because the cylinder pressure may be substantially higher than the exhaust pressure: then as they are swept out by the piston as it moves toward TC. As the piston approaches TC the inlet valve opens. Just after TC the exhaust valve closes and the cycle starts again.  

**Figure 4.1:** Four-stroke engine cycle schematic; reproduced from [22]

Although there is no combustion when the engine is retrofitted as a compressor, the basic motion of the piston in the engine cycle, as shown in Fig. 4.1 is very applicable for a compressor design. In addition to the mechanical advantages of retrofitting an engine, this system design is also motivated by economics.
Engine costs are low. As Fig. 2.9 in Chapter 2 shows, in 2016 alone, almost 95 million cars and commercial vehicles were produced around the world [32]. Although not for compression technologies, engines are mass-manufactured as part of the automobile market (in addition to other markets: lawnmowers, motorcycles, etc.). Because so many engines have been manufactured, the cost is low. Some (including a recent example in [46]) posit that internal combustion engines are reaching the end of their lifetime as the penetration of electric vehicles continues to increase. While this notion seems unlikely in the short term, it is worth noting that if it is true, if internal combustion engines are becoming less attractive to the automotive industry, the infrastructure that is in place still highly favors producing engines. In this case, alternative uses of engines, as are proposed in this thesis, become even more attractive to industries currently manufacturing engines.

4.4 General retrofit tasks

An internal combustion engine is designed to create mechanical power output (a motive force) from the chemical energy in a hydrocarbon fuel. One of the largest changes in retrofitting an engine to perform as a reciprocating compressor is in shifting the engine from a power producer to a power consumer. Instead of mechanical shaft power output, the engine as a compressor needs mechanical power input, which requires finding a suitable power source for the engine. The laboratory setup built for this thesis uses an electric motor. Another (non-retrofitted) engine could also be used as the power source. The compression ratio at which and pressure regime in which the engine compressor is being operated will dictate the effective compressor power, which will set the scale for required power input. As the example in Chapter 7 shows, it is reasonable (for at least that specific application) to plan for the same size engine as is being used for the compressor to use as the power source.

In addition to adding the power input, there are a couple of changes that all
retrofit designs have in common. In all designs, a number of engine components can be removed, including the engine fuel tank, air filter, exhaust manifold, and the spark ignition system. In the case of a small, stand alone engine, the spark ignition system includes the spark plug and an ignition magneto (an alternator device using two permanent magnets to create a short current pulse for the spark plug). The spark plug hole must be filled or replaced with another instrument. In the first retrofit, the spark plug hole is used to connect the custom “valvetrain” with the cylinder; in later designs, entirely new cylinder heads are used, now with no spark plug hole. The magneto does not directly affect running the engine in compression mode, but has been removed from the test engine to avoid triggering a high voltage pulse that could create electrical noise in the system. Further modifications range in complexity and creativity, and three main retrofits are explored in this chapter.

4.5 Experimental setup

The experimental engine compressor design consists of three main components, an internal combustion engine, an electric motor, and a variable frequency drive (VFD), and two pressure storage tanks, as shown in Fig. 4.2.

All components were purchased individually and assembled together in the laboratory. A unistrut frame was built to secure the engine, motor, and VFD. A shaft coupler is used to connect the engine shaft with the electric motor shaft. Not shown in Fig. 4.2 is a 1/2” polycarbonate shield that bolts to the unistrut and creates a barrier between the engine/motor and the experimenter. The following subsections give additional details for each of these main components.

4.5.1 Internal combustion engine

The internal combustion engine is a one cylinder, four-stoke, 3 HP, 79 cc OHV (overhead valve) horizontal shaft gas engine, manufactured by Predator Engines and pur-
chased at Harbor Freight. The compression ratio is 8.5:1, maximum RPM is 3,600, and bore \(\times\) stroke is 52.0 mm \(\times\) 37.0 mm (2.0 in. \(\times\) 1.5 in.). See Fig. 4.3a. Compared with a car engine, which is typically about \$10/kW, this engine is more expensive per unit of nominal engine power output (closer to \$40/kW).

### 4.5.2 Electric motor

The electric motor is a 3 HP general purpose motor, 3-phase, 3500 nameplate RPM, voltage 208-230/460, Frame 182T, and is manufactured by Grainger. See Fig. 4.3b.

### 4.5.3 Variable frequency drive

The variable frequency drive is an AC drive, 3 HP, 230 V, single/three-phase input, three-phase output, model GS3-23P0, manufactured by DURApulse and purchased from Automation Direct. See Fig. 4.3c.
4.5.4 Storage tanks

When applicable, a 5 gallon portable air tank with pressure gauge is used for providing inlet air at elevated pressure. In cases where the inlet is air at ambient pressure, there is no inlet pressure tank, and this 5 gallon tank may be used as the exhaust pressure tank. The tank is made of steel and has a maximum pressure of 125 psi. The tank is made by Central Pneumatic and was purchased at Harbor Freight. See Fig. 4.3d.

When the 5 gallon pressure tank is not used for storing exhaust, a 4 liter pressure tank, originally designed as an Argon pressure tank, is used for storing compressed air exhaust. This tank is always used when the intake pressure is above ambient because then exhaust pressures go beyond the limit of the 5 gallon tank. Based on this tank’s argon pressure specifications, the tank holds about 4 liters of air at 1 atm.

Both the experiments and model descriptions in the next sections/chapters designate which exhaust pressure tank was used.

4.5.5 Check valves, connections, and piping

The check valves used are McMaster-Carr part number 7838K530 stainless steel, “Corrosion-Resistant High-Pressure Threaded Backflow-Prevention Valves for Water, Oil, Inert Gas, and Fuel, 1/8 NPT Female.” Two of these valves are used: one for the intake, one for the exhaust. The additional connections, hoses, and pipe adapters were purchased from McMaster-Carr, Ace Hardware, and Summit Racing Equipment. The costs and part numbers are listed in the economic assessment in Chapter 7.

4.5.6 Pressure and temperature sensors

Two pressure transducers were purchased from Omega. One pressure transducer is used at the cylinder head to measure cylinder pressure; the second sensor is used at the inlet to the exhaust pressure tank to measure exhaust pressure. The pressure transducers are both PX409-750A5V, 0 to 5 Vdc output piezoresistive design with
high temperature performance. They are calibrated for pressures ranging from 0 to 750 psia. The pressure transducers record pressure measurements every millisecond.

One thermocouple is used to measure cylinder temperature. In the basic retrofit design (see Section 4.7.1), this thermocouple is a hard, type K thermocouple. In the later retrofits, a soft, hair-like type K thermocouple is used.
4.6 Engine retrofit goals and designs

Internal combustion engines are designed to take fuel as input and use the chemical energy to output mechanical work. An engine functioning as a small gas compressor, therefore, can eliminate some engine components that have no benefits to this new task. After these features are removed or replaced, what remains is a very crude engine compressor.

The most basic goal of this experimental work is to confirm that an inlet gas stream can be compressed inside the engine cylinder and that gas at elevated pressure can be stored as an engine output. Beyond this, the goal is to increase engine compressor performance, which focuses on addressing two main challenges: (1) decreasing cylinder dead volume and (2) facilitating heat exchange during the compression. Decreasing dead volume increases efficiency because it forces more gas volume out of the cylinder during each cycle. The aim of the heat management is to achieve a near-isothermal compression. Without any modifications, the engine will be a polytropic process tending towards adiabatic compression. Isothermal compression is desirable because it requires less work input, as is described in more detail in Section 5.3.2, which raises the system efficiency. Both decreasing clearance volume and facilitating heat exchange are nontrivial challenges. The following sections detail how these goals were pursued in the laboratory.

4.7 Engine compressor performance

Three main engine retrofits were tested. The first retrofit consists of keeping the standard engine cylinder head and using the spark plug hole for intake and exhaust. The second and third retrofits remove the engine cylinder head and replace it with flat plate cylinder heads fabricated from acrylic and then aluminum. In both these flat plate cylinder head applications, the check valves are threaded directly into the cylinder head. The next few subsections show the retrofits and the results of experimental
System efficiency is calculated by finding the work added to the system each cycle via compressed gas and the work requirement per cycle from the variable frequency drive data. The work added to the system per cycle is composed of two parts: (1) the work added to the exhaust tank by adding additional moles of gas compressed from inlet to tank pressure ($P_{in}$ to $P_{tank}$) and (2) the work added by compressing the current tank volume (from $P_{tank}(i-1)$ to $P_{tank}(i)$) to allow room for incoming moles of gas to enter. The two expressions are given in (4.1) and (4.2). In these expressions, $W_{out, added}$ is the work in joules (J) added to the system from adding moles of gas compressed from the inlet conditions to the exhaust pressure tank; $W_{out, in tank}$ is the work output (in J) from further compressing the exhaust pressure tank contents to make room for the additional gas. Here, $m_{inT}$ is the mass in kilograms (kg) added to the tank in one cycle, $m_{tank}$ is the mass of gas in the exhaust pressure tank (in kg), $R$ is the gas constant for air equal to 287 J/kg/K, $T_{tank}$ is the exhaust tank temperature in Kelvin (K), $P_{in}$ is the inlet pressure in psia, and $P_{tank}$ is the exhaust tank pressure in psia. The total work stored in compressed output in a cycle is the sum of the two components, as shown in (4.3). In these equations, $i$ is used as an index for a cycle.

$$dW_{out, added} = m_{inT}(i)RT_{tank}(i) \log \left( \frac{P_{in}(i)}{P_{tank}(i)} \right) \quad (4.1)$$

$$dW_{out, in tank} = m_{tank}(i-1)RT_{tank}(i) \log \left( \frac{P_{tank}(i-1)}{P_{tank}(i)} \right) \quad (4.2)$$

$$dW_{out, total} = dW_{out, added} + dW_{out, in tank} \quad (4.3)$$

The work into the system is calculated from the current, voltage, and phase angle, as shown in (4.4), provided by the variable frequency drive output, as is described more in Section 5.4.5. In (4.4), $V$ is voltage, $I$ is current, and $\angle PF$ is the power factor angle.
\[ P = VI \cos(\angle_{PF}) \]  

(4.4)

The expression in (4.4) is in terms of watts (per cycle). To compare the power input per cycle to the work out of the system in the form of compressed air, the power input is multiplied by the engine compressor cycle length (which is the same as dividing by the engine speed). Now, the power input is expressed as a work input in terms of joules.

The cycle efficiency is calculated as the ratio between total work out of the system and total work into the system.

### 4.7.1 Basic retrofit

As mentioned earlier, the first engine retrofitting steps include removing engine components that are specific to the combustion process, and therefore, have no task in an engine performing as a compressor. These parts include the engine fuel tank, air filter, exhaust manifold, spark plug, and magneto.

The engine used in this experimental setup is four-stroke, but when the engine becomes a vessel for compression only (no combustion), the four-strokes are no longer necessary. Instead, a two-stroke design (intake and exhaust) is more efficient. Converting from four- to two-strokes (while keeping the engine’s intake and exhaust valves) requires designing and machining a new cam lobe to trigger the intake and exhaust valves to open twice as often as they do in the four-stroke. The shape of the lobe will change often, however, because while functioning as a compressor, a final pressure level may be desirable. If, for example, the engine can compress to 100 pressure units but only 80 pressure units are required, the exhaust valve should open before the piston reaches the top of the cylinder. Signaling the valves to open at different pressures requires a new cam lobe to force unique valve timings for each desired case.
To avoid machining various lobe designs, the intake and exhaust valves are altogether abandoned, and the valve pushrods are removed to force the valves closed regardless of the lobe orientation. Intake and exhaust valves are replaced with check valves that are designed to permit flow in one direction only as a function of a pressure differential. In this design, once the cylinder pressure is below inlet pressure, the intake check valve will open and gas will flow into the cylinder. When cylinder pressure exceeds the pressure in the exhaust hose/tank, the exhaust check valve opens and gas flows out of the cylinder. As will be discussed in more detail at the end of this chapter, future designs will likely be most efficient by using mechanical check valves that are actuated based on a prediction of the pressure differences. It may be useful for a future design to be tested with a custom cam lobe to compare system performance; backflow should be evaluated in this case.

As mentioned above, the spark plug is removed since there is no combustion. Instead of plugging this hole, in this basic engine retrofit design, the hole is used to connect the intake and exhaust check valves to the cylinder. See Fig. 4.4 for the Solidworks model of this design and Fig. 4.5 for a picture of the laboratory setup. A spark plug adapter is used to connect to a stainless steel cross. To prevent possible leakage, the adapter was eventually epoxied. The cross holds the intake and exhaust check valves and a tee that connects to a pressure transducer and thermocouple. The exhaust is routed to another tee that leads in one direction through a braided stainless steel line to the exhaust pressure tank and in the other direction to a pressure relief valve. The pressure relief valve is spring loaded and can be adjusted to a particular pressure level. Any gas at a higher pressure will trigger the pressure relief valve to open. It is included as a safety measure to make certain high pressure is never unintentionally built up and trapped inside the system.

This first retrofit was designed for one main reason: to confirm, with minor retrofitting, an internal combustion engine can be driven such that ambient air can
be compressed in cylinder, and an exhaust stream of gas at elevated pressure can be stored in a pressure tank. The results of this system provide a baseline for comparing later retrofit designs. Two of the largest shortcomings of the basic retrofit engine design are the cylinder dead volume and restrictions on heat exchange. These two components are addressed subsequently.

Figure 4.4: Solidworks model of retrofitted “valvetrain” showing intake and exhaust check valves, tee connection to pressure transducer and thermocouple, tee connection to the exhaust hose and pressure relief valve, and pipe fitting to spark plug adapter
Fig. 4.6 shows an experimental run with cylinder pressure and tank pressure (in psia) plotted against elapsed time. This is a sample output from the setup shown in Fig. 4.5. Note in Fig. 4.6 that the exhaust tank pressure data (in blue) is overlaid on top of the cylinder pressure data (in red). The pressure variations are so fast that in this diagram the individual lines are not resolved. In effect the diagram paints the range of the pressure swing in each cycle in its color; the blue is superimposed on the red, which is hidden below the blue. (Other figures show trajectories on the resolution of a single cycle and take advantage of the high resolution of the record.) Note that the peak cylinder pressures are higher than the peak exhaust tank pressures and that the red data does show higher pressures, which can be seen even with the blue overlaid data.

In this experiment, the engine was run at 875 rpm, and the exhaust was routed to the 5 gallon (yellow in Fig. 4.5) exhaust pressure tank. Over the twenty-five minute
run, exhaust tank pressures level off around 65 psia. Because ambient pressure is about 15 psia, the pressure ratio in this single-stage compression is about 4.3.

![Figure 4.6: Original cylinder head with ambient inlet pressure; pressure curves vs. elapsed time](image)

Fig. 4.6 includes a plot of cycle efficiency (calculated from variable frequency drive current, voltage, and phase angle as per (4.4) and shown in Fig. 4.8) versus the exhaust tank pressure (in psia). Efficiency peaks at slightly over 25% around a tank pressure of 30 psia. This suggests that in a multi-stage compressor design, it may be advantageous from an efficiency perspective to limit the pressure in a stage to well below the maximum compression ratio, because as exhaust tank pressure increases, the efficiency decreases as less gas is added to the tank during each cycle.

Fig. 4.7 includes a plot of cycle efficiency (calculated from variable frequency drive current, voltage, and phase angle as per (4.4) and shown in Fig. 4.8) versus the exhaust tank pressure (in psia). Efficiency peaks at slightly over 25% around a tank pressure of 30 psia. This suggests that in a multi-stage compressor design, it may be advantageous from an efficiency perspective to limit the pressure in a stage to well below the maximum compression ratio, because as exhaust tank pressure increases, the efficiency decreases as less gas is added to the tank during each cycle.

Fig. 4.8 includes eight plots summarizing the data collected and the calculations performed. The left column shows all measured data: the pressure in the cylinder (psia), the pressure at the exhaust tank (psia), the cylinder temperature (in degrees Celsius), and the variable frequency drive data, including current (A), voltage (V), power factor angle, and engine speed (rpm). With this data, the power requirement
Figure 4.7: Original cylinder head efficiency vs. exhaust tank pressure (psia)

is calculated (in watts) as per (4.4), and a moving average is applied to smooth the data. The right column shows the calculations for determining the efficiency. The total mass of gas in the tank at each cycle (kg) and the mass of gas added to the tank per cycle (g) are shown. The total energy in the compressed gas per cycle (J/cycle) is plotted with the two components shown individually. The three equations that govern how the energy in the compressed gas is calculated are given in (4.1), (4.2), and (4.3). The two components of the total energy are: the energy added to the exhaust tank by gas that has been compressed from inlet pressure to the tank pressure, (4.1), and the energy added inside the tank from further compressing the mass of gas inside the tank when incoming gas is added, (4.2). The third plot shows the energy requirement per cycle; this is calculated by dividing the power requirement shown in the left column by engine speed (one can also think of this as multiplying by cycle length). Finally, cycle efficiency is the ratio between the energy in the compressed gas and the energy requirement over a cycle.
Figure 4.8: Original cylinder head data summary, including pressures, temperature, and variable frequency drive data
These results are representative of the experimental runs with the engine’s original cylinder head and the custom “valvetrain” as shown in Fig. 4.4 and Fig. 4.5. A few things to note: the compression ratio is less than would be predicted based on the engine specifications alone (4.3 versus 8.5). This is expected, however, because the retrofit has essentially added clearance volume to the cylinder by adding the volume of all the connections between the check valves, pressure transducer, and thermocouple. Based on this simple retrofit, efficiencies as high as 25% provide an encouraging base case. These results are used as a point of reference for gauging the performance of the next retrofits.

### 4.7.2 Reducing dead volume

Engine cylinders are designed with a minimum clearance (dead) volume larger than zero, which is required to accommodate intake and exhaust valves opening without hitting the piston head. It also reflects the fact that the gas at ignition should occupy a final volume and that there is an optimal compression ratio. In the case of the 3 HP engine used in this experimental setup, the total cylinder volume is 79 cc (cubic centimeters) with a compression ratio of 8.5:1. The compression ratio provides a relationship between the maximum and minimum cylinder volumes \[ \frac{79}{8.5} \]. For the engine used here, the dead volume is 9.3 cc (79 cc divided by 8.5).

In the basic engine retrofit described in Section 4.7.1, additional volume is added in the pipe fittings to connect the spark plug adapter to the intake and exhaust check valves (all the volume shown in the cross and tee components in Fig. 4.4). This additional volume becomes part of the cylinder clearance volume, which decreases the compression ratio and lowers the maximum achievable pressure inside the cylinder. The additional volume also decreases system efficiency because the gas that remains in the cylinder occupying the clearance volume cannot exhaust in an engine stroke, which means that compressed gas is added with the next cycle’s intake, expands,
and is compressed again. This inability to empty the cylinder gas lowers the system efficiency.

To enhance performance, the dead volume is reduced by abandoning the engine cylinder head and the additional piping added in the basic retrofit. With the goal of reducing cylinder volume came the idea of a flat cylinder head that would, ideally, be flush with the piston at top dead center (TDC). To minimize additional volume in piping, the design now is such that intake and exhaust check valves and a pressure transducer attach directly to the cylinder head. The new cylinder head is first fabricated with a 1/2” clear acrylic plate. The original cylinder head gasket remains and was used to align the plate with the bolt holes, and three additional holes were drilled and tapped for the two check valves and for the pressure transducer. The acrylic head is shown in Fig. 4.9. Note that in the picture here, the intake check valve is removed to allow a clearer view of the cylinder head. The piston head is visible in the picture.

Data from two experimental runs with the acrylic head are included here. The biggest challenge with the acrylic plate cylinder head experiments was the heat management. The system, to touch, became very warm, so the first experiment, as shown in Fig. 4.10 was stopped after just eight minutes.

To address the heat, a “cooling jacket” was designed and fabricated for the exhaust check valve, the warmest component of the system. The cooling jacket is a 3/8” thick aluminum plate that bolts around the check valve. It can be seen in the picture in Fig. 4.15. With this aluminum piece added, the exhaust check valve is able to shed heat more readily. The experiment was repeated; the results are shown in Fig. 4.11 and in detail in Fig. 4.13. In this repeated example (with the engine running at a speed of 437 rpm), the possibility of leakage through the pipe connections was noted. The data is included here anyway, but note the difference in trajectories of the pressure in these two experiments. The pressure shown for the second run only includes the exhaust tank pressure because the cylinder head pressure transducer
malfunctioned and required replacement.

Notice the efficiency versus exhaust tank pressure curve in Fig. 4.12. Included on this plot is the efficiency curve for the original cylinder head example given in Section 4.7.1. The cycle efficiencies are now higher than those with the engine’s original cylinder head. The increased efficiency is, at least in part, due to decreasing the cylinder clearance volume. Also shown in Fig. 4.10, Fig. 4.11, and Fig. 4.12 are higher exhaust tank pressures compared with those achieved with the original cylinder head, which is also a result of decreasing clearance volume.
Figure 4.10: Acrylic plate cylinder head with ambient inlet pressure; pressure curves vs. elapsed time [Ex. 1]

Figure 4.11: Acrylic plate cylinder head with ambient inlet pressure; exhaust tank pressure vs. elapsed time [Ex. 2]
Note that the data used for plotting the acrylic flat plat cylinder head efficiency in Fig. 4.12 corresponds with the run shown in Fig. 4.11. Variable frequency drive data, which is needed to calculate the system’s work requirement, was not available for the run shown in Fig. 4.10. Because the run shown in Fig. 4.11 likely had a leak, the corresponding efficiency curve shown in Fig. 4.12 is probably an underestimate for what can be expected from the acrylic flat plate cylinder head performance.

![Figure 4.12: Acrylic plate cylinder head and original cylinder head efficiency curves vs. exhaust tank pressures (psia)](image)

To accompany the second acrylic plate experiment shown in Fig. 4.11, the full data collection results are included in Fig. 4.13. The format is the same as the description for Fig. 4.8. The figure includes pressure and variable frequency drive data measurements. Note that cylinder pressure and temperature were not recorded.

Due to the challenge of managing heat with the acrylic flat plate cylinder head, the next retrofit shifted the cylinder head to a metal plate. A 1” aluminum plate was used for the next retrofit. The results of this design are included in Section 4.7.3.
Figure 4.13: Acrylic plate cylinder head data summary, including pressures, temperature, and variable frequency drive data
4.7.3 Facilitating heat exchange

In addition to reducing dead volume, managing heat during compression is the second main challenge in retrofitting an engine to perform as a reciprocating compressor. In Section 5.3.2, the difference between adiabatic and isothermal processes is explored. The laws of thermodynamics show that isothermal compression requires less work input for a given volume ratio compared with that required with adiabatic compression. The reason for this is due to heat. Isothermal and adiabatic processes are at either extreme in terms of heat transfer: either all heat is transferred to/from the system or the system is perfectly insulated and no heat is transferred. In isothermal processes, there is no temperature change ($\Delta T = 0$), so all heat is transferred perfectly in or out of the system; in an adiabatic process, there is no heat transfer ($\Delta Q = 0$), and the system is perfectly insulated.

To achieve the highest efficiencies in this engine compressor, the process should be isothermal rather than adiabatic. A point to add with regards to this, however, is that, as is shown later in modeling output, because the system still has friction, the contribution of friction work as a percentage of total work increases, which can result in a decrease in system efficiency. Another note to add is that, theoretically, a perfectly adiabatic system may also be very efficient because it too is reversible (until the exhaust flows out and into the tank).

The difficulty in pursuing isothermal compression is in facilitating heat exchange to enable the gas to remain the same temperature throughout the entire process. An internal combustion engine is designed with fins creating additional surface area, but is not designed for an isothermal process. Compression closer to isothermal is pursued experimentally here by adding cooling to the cylinder head.

In Section 4.7.2, an acrylic plate cylinder head was fabricated to reduce clearance volume. To further enhance system performance, the next cylinder head design incorporated a method for dealing with the heat produced during compression by
controlling the cylinder head temperature. A 6” square, 1” thick aluminum cylinder head with built-in channels for water flow was fabricated next. Water flows through the head and then is routed to a chiller where the water temperature is maintained at a set point. The water is then recirculated through the cylinder head. The Solidworks design is shown in Fig. 4.14 and a photo from the laboratory setup is included in Fig. 4.15. The 1/2” diameter water channels form an incomplete square around the perimeter of the plate. The eight holes produced in drilling the different sections of the cooling channels are plugged and two 1/8” holes are included on one side of the plate to provide an inlet and outlet for cooling water. There are four holes for bolting the cylinder head down and two guide holes. Three additional holes are drilled and tapped on the surface, which are used for attaching inlet and exhaust check valves and the cylinder head pressure transducer.

Figure 4.14: Solidworks model of aluminum plate cylinder head (transparent and solid views)
4.7.3.1 Ambient inlet pressure

A number of results are included here to characterize the performance of the engine compressor with the aluminum plate cylinder head. The first results all start with ambient inlet conditions. The first example shows data measurements from an experimental setup where the exhaust air is routed to the 5 gallon pressure tank, the same setup used for the previous experiments with the original cylinder head and the acrylic plate cylinder head. In the second and third experiments included in this section, the exhaust pressure tank is switched to the smaller, 4 liter tank. This first data (engine running at 437 rpm) is included to verify that performance can be compared
across the two experimental setups. The cylinder pressure transducer malfunctioned during this run (same issue as with the data shown in Fig. 4.11), so exhaust pressure only is recorded (see Fig. 4.16). Fig. 4.17 shows the full data summary. As shown in this and the following figures, with the aluminum plate cylinder head, tank pressures reach, and in some examples exceed, 150 psia. Ambient air is about 15 psia, so the compression ratio is about 10 with the aluminum plate cylinder head.

Figure 4.16: Aluminum plate cylinder head with ambient inlet pressure; exhaust tank pressure vs. elapsed time [Ex. 1]
Figure 4.17: Aluminum plate cylinder head data summary, including pressures, temperature, and VFD data [Ex. 1]
While the 5 gallon pressure tank was used for exhaust storage in the previous examples, the experimental setup was shifted here to use the smaller 4 liter pressure tank for capturing exhaust (preparing for future experiments where the 5 gallon tank is used to provided elevated inlet pressure). Fig. 4.18 shows the pressure measurements over the hour-long experiment. Fig. 4.19 shows a closer look at the pressure over three engine cycles. Included in Fig. 4.20 is a summary of all data collected and calculations performed to calculate work and efficiency. In this experiment, the engine ran at a speed of 752 rpm.

![Figure 4.18: Aluminum plate cylinder head with ambient inlet pressure; pressure curves vs. elapsed time [Ex. 2]](image)

In the zoomed view in Fig. 4.19, notice the movement in the exhaust tank pressure measurements. Refer back to Fig. 4.2 and notice where the exhaust tank pressure transducer is located: at the end of a hose connecting the exhaust check valve to the exhaust pressure tank. The variability in the pressure measurement is due to the fact that the exhaust pressure transducer is actually measuring the pressure in the exhaust hose, which experiences backflow to the engine cylinder through the exhaust.
check valve.

Figure 4.19: Aluminum plate cylinder head with ambient inlet pressure; pressure curves vs. elapsed time [Ex. 2] (zoomed view)
Figure 4.20: Aluminum plate cylinder head data summary, including pressures, temperature, and VFD data [Ex. 2]
There are two important features to note in comparing the results from these two experiments, with pressure curves shown in Fig. 4.16 and Fig. 4.18. While the final exhaust tank pressure is similar in both experiments (around 150 psia), the pressure rises more rapidly in the second experiment. This is, of course, intuitive because in this experiment, the exhaust pressure tank is about one-fifth of the size of the exhaust pressure tank for the first experiment. It takes a shorter amount of time to fill this tank. Second, note the efficiency curves. Notice in the first experiment (see Fig. 4.17), the efficiency peaks around 40%, whereas in the second experiment (see Fig. 4.20), the efficiency peaks closer to 25%. The reason for this discrepancy is due, at least in part, to the change in speeds between the two runs (437 rpm in the first case; 752 rpm in the second case). The change in speed affects the check valve delays and, therefore, backflow. It is reasonable to expect higher backflow when the speed is increased, which will lower efficiency. (This was also verified with the engine compressor model (which is described in Chapter 5); all else being equal, slower speeds translate to higher efficiencies.)

Numerous experiments were run with the aluminum plate cylinder head to characterize performance over a range of engine speeds, inlet conditions, and cylinder head cooling. Experiments were also run at near-identical conditions in order to characterize the expected variability. Included in Fig. 4.21 are the results of another experiment with the engine running at 752 rpm and the inlet at ambient conditions. Overall, the data largely agrees with the trajectory shown in Fig. 4.18 and Fig. 4.20 but notice that the exhaust tank pressure is now above 150 psia. This experiment was run for almost an hour.
Figure 4.21: Aluminum plate cylinder head data summary, including pressures, temperature, and VFD data [Ex. 3]
These examples show that the engine compressor with the aluminum plate cylinder head retrofit performs the strongest in achieving both high exhaust tank pressures and high efficiencies. The cycle efficiencies for the three aluminum plate cylinder head experiments shown in this section are included in Fig. 4.22 along with the efficiency curves for the acrylic plate cylinder head and the original cylinder head (both of which were shown in Fig. 4.12).

Figure 4.22: Aluminum plate cylinder head, acrylic plate cylinder head, and original cylinder head efficiency curves vs. exhaust tank pressures (psia)

Included in Fig. 4.23 is a plot showing the smoothed exhaust tank pressure curves as a function of time. In comparing the three aluminum plate cylinder head data runs in the same figure, the variability in exhaust tank pressure is now more noticeable. Recall that the purple curve in Fig. 4.23 was for a run using the 5 gallon exhaust pressure tank, whereas the other two aluminum plate runs (plotted in gray and yellow)
routed exhaust to the 4 liter exhaust tank. Still, the final tank pressure varied about 40 psia between the runs, which may in part be due to the varied engine speed across the runs. Even the two experiments running at 752 rpm, however, have noticeably different final tank pressures. This shows the expected variability in the system. Fig. 4.24 and Fig. 4.25 also show the variability across what should be identical experimental runs.

![Figure 4.23](image)

Figure 4.23: Aluminum plate cylinder head, acrylic plate cylinder head, and original cylinder head exhaust tank pressure curves vs. elapsed time

A wide range of additional experiments with the aluminum cylinder head and ambient inlet pressure was explored. Fig. 4.24 summarizes the data collected. Experiments are organized by engine speed; all experiments at one speed are plotted together in the same row. The engine speed is as low as 280 rpm with a high of 1750 rpm. The smoothed tank pressures are plotted, with vertical dotted lines indicating
when the motor was shut off. The right column shows the temperature measurements (in degrees Celsius) from the cylinder head thermocouple.

Figure 4.24: Aluminum plate cylinder head multi-run summary; inlet at ambient pressure

As the experiments in Fig. 4.24 show, achievable exhaust tank pressures range from about 140 to 180 psia. See Fig. 4.25 for a summary of the smoothed exhaust
tank pressure curves across multiple experiments at varying speeds and cylinder head cooling temperatures. The inlet cooling water temperature to the cylinder head is shown clearly on the temperature curves in Fig. 4.24 at the end of each run when the motor is shut off, the temperature returns to the chiller temperature, which is the temperature of the inlet water to the cylinder head. There is no clear trend in how cylinder head cooling impacts the expected compressor performance. For example, a lower cylinder head cooling temperature at 1201 rpm resulted in a slightly higher exhaust pressure. However, the coolest cylinder head experiment at 752 rpm did not result in the highest exhaust tank pressure for that set of experiments. There is also no clear trend in how varying engine speed affects the peak exhaust tank pressures, though this likely affects cycle efficiencies.

Figure 4.25: Aluminum plate cylinder head exhaust tank pressure curves with ambient inlet conditions
4.7.3.2 Elevated inlet pressure

In addition to ambient inlet pressure, the engine compressor was also tested with intake gas at elevated pressures, here, about 60 psia. The engine ran at a speed of 1201 rpm for about 15 minutes. The results are displayed in Fig. 4.26 and a zoomed in view is provided in Fig. 4.27. This experiment actually begins with the engine running first without seeing the elevated pressure in the inlet stream. Once the engine is in motion, a valve is opened to permit gas at elevated inlet pressure to flow into the engine cylinder. As has been mentioned before, the 5 gallon pressure tank is used as a reservoir for inlet gas at elevated pressures. The tank is used as a buffer to maintain a consistent inlet pressure. During the experiment, the inlet pressure tank is filled with shop air as air is lost to the engine cylinder. Notice in Fig. 4.26 the pressure trajectories shift about a minute into the run; this is the point where the engine cylinder sees the inlet gas at elevated pressure. When the run stops, the inlet tank is still connected, so the pressure is constant, but above ambient, in the cylinder, which is visible in this plot. When the inlet pressure tank is disconnected, the cylinder pressure falls to ambient.

In Fig. 4.26, a maximum exhaust tank pressure of 668 psia, or about 46 bar, was recorded, which is approaching the 50 - 60 bar pressure regime that starts to cover many chemical engineering system requirements. This also shows that the compression ratio of 10, identified earlier when ambient inlet pressure is fed into an engine compressor with the same aluminum plate cylinder head retrofit, is maintained even with elevated inlet pressure.
Figure 4.26: Aluminum head with elevated inlet pressure; pressure curves vs. elapsed time

Figure 4.27: Aluminum head with elevated inlet pressure; pressure curves vs. elapsed time (zoomed view)
As shown earlier, efficiency decreases as very high pressures are pushed into the exhaust tank, so while this high pressure was recorded in this experiment (where inlet air began at about 60 psia), it may not be ideal to push the system to achieve this high pressure from 60 psia. It may be optimal to end the experiment before reaching this peak pressure and start a subsequent stage with that pressure as the inlet. Further research should focus on optimizing the compression ratio at each stage.

As before, multiple experiments were run, now with the inlet gas at elevated pressures. Included in Fig. 4.28 are experiments again grouped by engine speed. The left column shows the exhaust tank pressure curves; the right column shows the cylinder head temperature measurements. The lowest engine speed is 750 rpm with a high of 1650 rpm. As this data show, many experiments ended after just a few minutes (the dotted vertical lines show when the motor was shut off in each experiment). The reason for the shorter run times is due to the strain on the engine from compressing gas which is fed into the system at elevated pressure. The shaft coupler, for example, was replaced twice and is an example of one particular component that suffered from how hard the engine was run during these experiments. When comparing the runs in Fig. 4.28, note that although all begin with elevated inlet pressures, there is a range of inlet pressures from about 30 to 60 psia.
Figure 4.28: Aluminum plate cylinder head multi-run summary; inlet at elevated pressure
4.8 Physical footprint

In Section 4.5, the laboratory setup was described. The engine compressor setup (including the motor, variable frequency drive, pressure tanks, and laptop) fit on one laboratory table (about 3 ft. by 6 ft.). For a commercial system, the physical footprint will likely be larger. The engine in a commercial system will likely be bigger and, with multi-stages, additional pressure reservoirs are needed. A commercial system will also require infrastructure for containing the engine compressor, the engine or motor to drive the engine compressor, and the necessary tanks and piping. An estimate for the physical footprint for a commercial-scale system is measured by the size of a few engines plus associated tanks. It would easily fit on a skid mounted stand, that can be containerized.

4.9 Compressor design guidelines

The work reported in this chapter refers to a small engine compressor that was tested in the laboratory. The experimental work focused on compressing air to a maximum of about 650 psia. The design guidelines for this laboratory-scale compressor were framed around first showing proof of concept and then increasing system performance. The engine here is specifically designed for compressing air up to 650 psia.

An engine compressor can be used to compress different gases, but will require additional design guidelines. This technology’s development roadmap includes a better understanding of: the expected performance when moving to a larger engine, how to optimize flow in and out of the engine cylinder, and how to ideally design the full balance of system infrastructure (additional engines/infrastructure for powering the compressor and/or for multi-stage compression). A look at compressing with elevated inlet pressure was included in this experimental work, but a multi-stage optimization will need to be performed in the future. The achievable peak pressure is equal to or higher than what was shown in this experimental work and should be evaluated.
further. A multi-stage compression optimization should be performed by identifying the final outlet pressure requirement and then determining the optimal path to achieving that pressure based on system efficiency versus pressure output at varying inlet pressure conditions. There exists a trade-off between the number of compression stages and engine infrastructure requirements. Additionally, for each stage added, the friction is incurred again.

More specific design guidelines for a retrofitted engine compressor will vary based on the application of a pilot or commercial-scale system. Additional work is required to identify the optimal path to commercialization based on the gas being compressed and the flow and pressure requirements. A cost optimization is also required. In Chapter 7, a techno-economic assessment is included, with a specific section highlighting the differences between the capital and operating costs for the engine compressor and for industrial compressors. This work should be expanded on to understand the costs at commercial-scale.

The experiments suggest that restrictions in the flow into and out of the engine have large impact on the overall performance and should be reduced as much as possible. Large piping, and large bores for the inlet and outlet are important design considerations. A second important design consideration is minimizing check valve backflows, which are due to the fact that check valves show inherent delays that should be minimized. There is a trade-off between these two considerations, as the former drives to larger openings, whereas the second favors smaller openings.

4.10 From pilot to commercial scale
A number of areas need to be addressed in pushing this engine compressor design from the laboratory to the pilot and then commercial scales. Engineering tasks include optimizing flow in and out of the engine cylinder by exploring alternative valve designs, optimizing multi-stage compression, handling lubrication and contamination,
allowing for heat exchange, and sealing and flooding the compressor outer container so that any gas pushed into the cylinder from piston back pressure is the same gas that is being compressed so that mixing does not occur between two different gases inside the cylinder.

In addition to the engineering tasks, further financial analysis is required. The system costs of the laboratory scale engine compressor are explored in Chapter 7. More detailed analysis will be beneficial for quantifying the competitiveness of a commercial-size system.

Future work must also consider engine lubrication. While the engine here uses regular engine oil for lubrication, this could pose a concern for future designs depending on what gas is being compressed. For a very pure gas stream, oil may contaminate the product. This will need to be addressed on a case-by-case basis.

Additionally, attention in future work should be focused on the possibility of varying materials of construction based on the working gas. Some materials are very catalytic and could begin converting the feed gas to undesired products during compression. This applies, in particular, to applications where the engine is used for chemical processes. In many conventional reactors, the internal layer must be passivated with silicon dioxide (SiO$_2$) coatings in order to eliminate activity. The temperature in the engine compressor (or reactor) will likely not be very high due to the built-in cooling jacket, so aluminum may be suitable since it is not generally catalytic. Again based on the specific application, material properties should be considered in designing future engine compressors.

4.11 Summary
The experimental work included in this chapter is used to characterize engine performance across a wide range of varying parameters and retrofits. The first experiments tested performance on the original cylinder head and show proof of concept: gas at
elevated pressure can be stored from the engine exhaust. In this bare-bone retrofit, efficiencies peak at about 25%. The next retrofit focused on decreasing cylinder clearance volume by replacing the original cylinder head with a flat plate acrylic head. The efficiency now peaks at about 30%, and higher exhaust tank pressures are recorded. Finally, an aluminum plate cylinder head was installed with a built-in cooling jacket. The achievable exhaust tank pressures more than double in this scenario. Efficiencies increase as well.

Based on the data collected here, a few modifications are proposed for a future engine retrofit. The aluminum plate cylinder head was efficient in pushing compression to high pressures. What the setup is not efficient with, however, is preventing system backflows. Once gas moves into the exhaust hose, there is evidence of a delay in the check valve closing, which allows gas to return back into the cylinder. See Fig. 4.27 for evidence of this check valve delay and backflow. The check valve delays and corresponding backflow reduce system performance. A future system may be designed with mechanical check valves that are actuated based on a prediction of the pressure differences. In this manner, the check valves can be opened and closed more precisely and so decrease backflow. Continuing to reduce cylinder dead volume is also important. The flat plate cylinder heads are good at decreasing volume, but they likely create a flow resistance inside the cylinder of gas leaving through the exhaust check valve as the piston moves towards top dead center. Built-in ridges in the cylinder head would be beneficial in directing the gas to the valve and out of the cylinder. Finally, larger openings for intake and exhaust should be explored. Based on experimental results, during the piston down stroke towards bottom dead center, pressure drops below ambient (the piston is starting to pull a vacuum), which indicates that the intake valve is too small. Ideally, minimum pressure inside the cylinder should be at inlet conditions; dropping below inlet pressure only decreases the system performance.
Future research and experimentation should focus on decreasing flow resistance by increasing valves sizes in a future retrofit. Optimizing flow rate is a solvable retrofit. An option for achieving better flow in and out of the cylinder is to use valves that have internal electronic controls and can be programmed to open and close based on a prediction when pressure on the two sides will be equal.

In this work, input conditions were varied by increasing the inlet air pressure. In future designs, however, gas could be at even higher inlet pressures or could even be below 1 atm at inlet. Based on the experimental work presented here, the engine compressor can operate with higher inlet pressures, though it does put strain on the system. Testing with inlet conditions below atmospheric pressure is reserved for future work and specific applications where this is desired. It should be noted that the internal piston friction is largely independent of the operating pressure. This likely will reduce the efficiency of compressor stages that operate at low overall pressures.

The data included in this chapter verifies that a small internal combustion engine can, with minimal retrofits, be used as a gas compressor. The maximum pressure achieved with the aluminum cylinder head with elevated inlet pressure was above 650 psia. With varying inlet pressure, the aluminum flat plate cylinder head consistently operates with a pressure ratio of about 10:1. Evaluating the opportunities to push this system beyond its current boundaries are explored by creating a model in Python to simulate the engine compressor performance. Chapter 5 focuses on this model. In Chapter 6, the experimental data from this chapter is compared with the model predictions from Chapter 5.
5.1 Engine model overview

For the analysis and optimizations presented in Chapter 3 and Appendix A, an engine model was created using GT-POWER software to map engine performance. GT-POWER offers many flexible design features, but it is not well-suited for modeling a retrofitted engine as a compressor. For modeling an engine compressor, the engine’s physical constraints are used, but the performance is unconventional; here, there is no combustion. Instead, the engine body is paired with compressor performance. To ensure that the simulation was designed to capture exactly the physical features, and to guarantee a robust understanding of the heat transfer, friction, leakage, and flow, a custom engine compressor model, written in Python, was the preferred method for simulating the experimental data. This provided that the model was designed and system parameters were calculated exactly to match the system built in the laboratory. The Python model created is a robust simulation tool, custom-designed for the retrofitted engine compressor built as per the details in Chapter 4. As is shown in this chapter and particularly in Chapter 6, the Python engine model is accurate (and validated) on the resolution of milliseconds, seconds, and minutes.

The model incorporates engine geometry, material properties, fluid flow, heat transfer, friction, leakage, valve opening/closing delays, pressure, volume, temperature, work performed, work stored as compressed output, and system efficiency. The goal of the model is to capture changes in performance over a range of varying parameters, which include engine speed, engine friction, change in flow rates and flow
resistances, changes in check valve opening/closing delays, cylinder clearance volume, and heat transfer properties. The model is meant to both supplement and inform experimental work. The first model iteration was written by assigning many engine characteristics from values in literature. Laboratory experiments were then specifically designed to determine many individual parameters, including flow rates through valves, friction, leakage, and heat transfer. Results from these experiments were then incorporated into the engine model. In this fashion, engine model parameters were calibrated with the performance of the laboratory engine compressor. The model’s temporal resolution is adjustable, and a comparison is given between the output from identical engine runs at different resolutions to show the output converges (see Section 5.6). For most model output in this chapter and in Chapter 6, the time step resolution is 0.1 ms; the time step is always noted.

The outline of the model is the following: the engine is running at a specified speed (rpm); inlet gas conditions (pressure, temperature) are selected; gas is fed into the engine cylinder through the inlet check valve, which opens as the piston moves toward bottom dead center when the inlet gas pressure becomes greater than the pressure of the gas inside the cylinder; when the piston moves through bottom dead center and now moves towards top dead center, the cylinder gas is compressed; when the gas pressure inside the cylinder is greater than the gas pressure in the exhaust hose, the exhaust check valve opens and air flows out of the cylinder into the exhaust hose; gas in the exhaust hose flows into the exhaust pressure tank when its pressure becomes greater, and vice versa; the cycle repeats with new gas entering the cylinder through the inlet check valve.

The following sections first go through the model parameters and specify inputs and outputs. This is followed by a background on the thermodynamics, which is the backbone of this analysis. The details on the engine model features follow, which describe how the physical characteristics, thermodynamics, and nonidealities (e.g.,
friction) were modeled and calculated. The chapter concludes with how the model was calibrated, which sets the stage for the experimental and model comparisons in Chapter 6.

5.2 Model parameters

Many parameters are calculated in this model to create a map of the engine performance as a compressor. This section describes the model calculations, many of which were calibrated with experimental data. The nomenclature used in the model is provided in Table 5.1.

Table 5.1: Engine model nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P$</td>
<td>Pressure (psia)</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature (K)</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume (cc)</td>
</tr>
<tr>
<td>$v$</td>
<td>Flow (kg/hr)</td>
</tr>
<tr>
<td>$n$</td>
<td>Moles</td>
</tr>
<tr>
<td>$g$</td>
<td>Check valve velocity</td>
</tr>
<tr>
<td>$x$</td>
<td>Check valve position</td>
</tr>
<tr>
<td>$MW$</td>
<td>Molecular weight of air (g/moles)</td>
</tr>
<tr>
<td>$A_p$</td>
<td>Piston head area (m$^2$)</td>
</tr>
<tr>
<td>$H_p$</td>
<td>Piston height (m)</td>
</tr>
<tr>
<td>$dT$</td>
<td>Temperature change (K)</td>
</tr>
<tr>
<td>$W_{in, friction}$</td>
<td>Work requirement due to friction (J)</td>
</tr>
<tr>
<td>$W_{in, cylP}$</td>
<td>Work requirement due to cylinder pressure (J)</td>
</tr>
<tr>
<td>$W_{in, total}$</td>
<td>Total work requirement (J)</td>
</tr>
<tr>
<td>$W_{out, added}$</td>
<td>Work added by adding moles of gas to tank (J)</td>
</tr>
<tr>
<td>$W_{out, in tank}$</td>
<td>Work added by compressing moles of gas in tank (J)</td>
</tr>
<tr>
<td>$W_{out, total}$</td>
<td>Total work added to system (J)</td>
</tr>
<tr>
<td>$SA$</td>
<td>Inner cylinder and piston head surface area (m$^2$)</td>
</tr>
<tr>
<td>$m_{eng}$</td>
<td>Engine mass (kg)</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Efficiency</td>
</tr>
</tbody>
</table>

The engine model has two check valves that permit gas flow into and out of the engine cylinder. The check valve state is modeled as a ball moving from open to close (and vice versa), with a position and velocity. The velocity, $g$, of the ball is based on the pressure difference across the valve, $\Delta P$, and a constant, $C$, as per (5.1), and
the position, $x$, is based on the change in velocity as per (5.2). When the ball is at position equal to 0, the check valve is closed. The check valve is open at a position equal to 1. When the ball is in motion, there is partial flow through the check valve. The direction of flow is always from the higher to lower pressure reservoir. While a valve is in the process of closing, the flow can reverse directions, and there will be backflow.

$$g(i) = g(i - 1) + C \Delta P dt$$  \hspace{1cm} (5.1)

$$x(i) = x(i - 1) + \left( \frac{g(i) + g(i - 1)}{2} \right) dt$$  \hspace{1cm} (5.2)

The check valve delay is on the order of a few milliseconds. How this was estimated is described in Section 5.4.4.1. When the check valve is closed ($x = 0$), no gas flows through the check valve. When the check valve is partially or wide open ($0 < x \leq 1$), flow is permitted through the valve, in either direction based on the pressure difference across the valve. When inlet (which is often the ambient environment) pressure, $P_{in}$, is greater than the cylinder gas pressure, $P_{cyl}$, the intake check valve will begin to open. When the cylinder pressure is greater than the exhaust hose pressure, $P_{hose}$, the exhaust check valve will begin to open.

There are four possible states that result from the pressure differences across the valves, either: both valves are closed ($x = 0$), both valves are partially or wide open ($0 < x \leq 1$), or one valve is partially or wide open while the other is completely closed and vice versa. The engine model is discrete and runs at sub-millisecond intervals. Each time step in the model begins by evaluating the pressure differentials and the valve positions to determine the possible flows in and/or out of the engine cylinder.

The speed of the engine, given as a model input, sets the piston movement in a time step. Based on the piston movement and direction, the cylinder volume and the exposed cylinder surface area (walls, piston head, and cylinder head) either increase
or decrease. This expansion or compression changes the pressure, temperature, and density of the gas inside the cylinder. Based on the pressure of the gas inside the cylinder relative to the inlet pressure and exhaust tank pressure, gas can flow in or out of the cylinder. Incoming gas mixes with the cylinder contents. Gas also leaks in or out of the engine cylinder across the piston rings.

Based on check valve states and piston position, the model calculates gas pressure, temperature, heat flux, mass, and flows across the entire system. Work is stored as compressed gas in the exhaust tank. The work requirement is calculated based on the friction and pressure forces.

The thermodynamic constants used in the engine model calculations are provided in Table 5.2.

Table 5.2: Engine model constants

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Coefficient</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas constant</td>
<td>$R$</td>
<td>8.314</td>
<td>J/mol/K</td>
</tr>
<tr>
<td>Specific gas constant for air</td>
<td>$R_{specific}$</td>
<td>287</td>
<td>J/kg/K</td>
</tr>
<tr>
<td>Molecular weight of air</td>
<td>$MW$</td>
<td>29</td>
<td>g/mol</td>
</tr>
<tr>
<td>Heat capacity ratio</td>
<td>$\gamma$</td>
<td>1.4</td>
<td></td>
</tr>
<tr>
<td>Heat capacity (cast iron)</td>
<td>$c_p$</td>
<td>0.46</td>
<td>kJ/kg/K</td>
</tr>
<tr>
<td>Heat transfer coefficient (cylinder gas to engine block)</td>
<td>$h_1$</td>
<td>200</td>
<td>W/m$^2$/K</td>
</tr>
<tr>
<td>Heat transfer coefficient (engine block to environment)</td>
<td>$h_2$</td>
<td>20</td>
<td>W/m$^2$/K</td>
</tr>
</tbody>
</table>

5.2.1 Calculations in each state

The positions of the inlet and exhaust check valves set the flow rates in and out of the engine cylinder. At the beginning of a time step ($i$), the initial state ($i_0$), which includes the number of moles, pressure, and temperature of gas inside the cylinder, is determined based on the ending conditions of the previous time step. Setting these three parameters is shown in (5.3), (5.4), and (5.5), respectively.
\[ n_{cyl}(i_0) = n_{cyl}(i - 1) \quad (5.3) \]

\[ P_{cyl}(i_0) = \frac{P_{cyl}(i - 1)V_{cyl}(i - 1)^\gamma}{V_{cyl}(i)^\gamma} \quad (5.4) \]

\[ T_{cyl}(i_0) = \frac{P_{cyl}(i_0)V_{cyl}(i)}{n_{cyl}(i_0)R} \quad (5.5) \]

As shown in (5.4), the polytropic index (typically designated as \( n \)) is set to the specific heat ratio, \( \gamma \), and compression is modeled as an adiabatic process. If, instead of \( \gamma \), the polytropic index was set to 1, an isothermal process would be modeled.

All flows through tubes connecting to the cylinder volume or through small gaps in the cylinder seals are calculated based on the relationship shown in (5.6), where the flow, \( v \), is in units of kg/hr. The derivation of this relationship is provided in Section 5.4.4.2. \( \Delta P \) is the pressure difference across the flow interface; \( a \) and \( v_0 \) dictate the flow resistance. The values of \( a \) and \( v_0 \) for the leakage channels are specified in Section 5.4.4.2; the values for \( a \) and \( v_0 \) for the intake flow, exhaust flow, and flow between exhaust hose and tank are provided in Section 5.4.4.3.

\[ v = -\frac{v_0}{2} \pm \sqrt{\frac{v_0^2}{4} + \frac{\Delta P}{a}} \quad (5.6) \]

After the initial states are set at the beginning of the time step, the leakage flow is calculated. Gas can flow either in or out of the cylinder by leaking past the piston rings. When the cylinder pressure is higher than the piston back pressure, gas leaks out of the cylinder. When the piston back pressure is higher than the cylinder pressure, gas leaks into the cylinder.

Based on the pressure of the gas in the exhaust hose and exhaust tank, the flow rate and direction between the hose and tank are calculated (again based on (5.6)). The values of \( a \) and \( v_0 \) are used again to set the flow resistance. The mass of gas in the exhaust pressure tank is updated based on gas flows in or out.
The flow magnitude and direction across both check valves are set based on the check valve positions and the pressure differences between inlet and cylinder pressures and cylinder and exhaust hose pressures. The flow is calculated again with the relationship given in (5.6), where \( a \) and \( v_0 \) are set for each check valve. Based on the flows in and out of the cylinder, the number of moles of gas in the engine cylinder and in the exhaust hose are updated. The flow of moles per unit time, \( \dot{n} \) are calculated by dividing the flow, \( v \), by the molecular weight of the gas, \( MW \), as shown in (5.7).

\[
\dot{n} = \frac{v}{MW} \tag{5.7}
\]

Using the engine cylinder as a frame of reference, the convention is that flow into the cylinder from the inlet is positive and flow out of the cylinder (to leaks and/or to the exhaust hose) is negative. The moles of gas in the cylinder at the end of the time step is given by (5.8).

\[
n_{cyl}(i) = n_{cyl}(i - 1) + n_{toCyl}(i) - n_{toHose}(i) - n_{toLeak}(i) \tag{5.8}
\]

The moles of gas in the exhaust hose and exhaust pressure tank are updated as per (5.9) and (5.10), respectively.

\[
n_{hose}(i) = n_{hose}(i - 1) + n_{toHose}(i) - n_{toTank}(i) \tag{5.9}
\]

\[
n_{tank}(i) = n_{tank}(i - 1) + n_{toTank}(i) \tag{5.10}
\]

The piston position is updated based on the engine speed and the time step resolution. The cylinder volume and exposed surface area are updated based on the piston position. The cylinder gas is either compressed or expanded, and the temperature and pressure are updated. The cylinder gas temperature is calculated as a weighted average by moles of gas flowing in or out of the engine cylinder. This assumes a constant specific heat.
\[
T_{cyl}(i) = \frac{1}{n_{cyl}(i)} [T_{cyl}(i - 1)n_{cyl}(i - 1) + T_{in}(i)n_{toCyl}(i) - T_{hose}n_{toHose}(i) - T_{leak}n_{toLeak}(i)]
\]

(5.11)

In (5.11), if gas is flowing out of the cylinder (to the exhaust hose and to leakage), then \(T_{hose}\) and \(T_{leak}\) are equal to \(T_{cyl}(i - 1)\). If the flow direction is into the cylinder, \(T_{hose}\) and \(T_{leak}\), are determined based on the heat transfer convention set in the model.

The model provides the option to switch between two different temperature conditions across valves and leaks. The model can be set to accommodate either adiabatic or isothermal heat flow. The default is adiabatic, which means that across the check valve (or across the leak), the gas adiabatically expands and the temperature decreases as it enters the receiving reservoir as per the adiabatic relationship given in (5.12). In the isothermal case, the gas remains the same temperature while flowing through the check valve and into the receiving reservoir. The gas mixes with the other moles in the receiving reservoir at the temperature at which it enters.

\[
T = T_0 \left( \frac{P_0}{P} \right)^{(1-\gamma)/\gamma}
\]

(5.12)

The resulting cylinder gas pressure is based on the temperature of gas and number of moles inside the cylinder and the current cylinder volume as shown in (5.13).

\[
P_{cyl}(i) = \frac{n_{cyl}(i)RT_{cyl}(i)}{V_{cyl}}
\]

(5.13)

The pressure in the exhaust hose and exhaust pressure tank are calculated as shown in (5.14) and (5.15), respectively.

\[
P_{hose}(i) = \frac{n_{hose}(i)RT_{hose}(i)}{V_{hose}}
\]

(5.14)

\[
P_{tank}(i) = \frac{n_{tank}(i)RT_{tank}(i)}{V_{tank}}
\]

(5.15)
The hose and tank temperatures, \( T_{\text{hose}} \) and \( T_{\text{tank}} \), remain constant throughout the simulation. This assumes that the required heat flux in or out of the gas is provided by the ambient environment. Therefore, the equations for the temperatures of gas in the exhaust hose and exhaust pressure tank, given in \((5.16)\) and \((5.17)\), always hold.

\[
T_{\text{hose}}(i) = T_{\text{hose}}(i - 1) \quad (5.16)
\]

\[
T_{\text{tank}}(i) = T_{\text{tank}}(i - 1) \quad (5.17)
\]

### 5.2.2 Work requirement and cycle efficiency

The work required to drive the system is a result of overcoming friction, \((5.18)\), and overcoming cylinder pressure, \((5.19)\). At every time step, the work requirements into the system and the work out of the system are calculated.

\[
dW_{\text{in, friction}}(i) = P_{\text{friction}} A_p |H_p(i) - H_p(i - 1)| \quad (5.18)
\]

\[
dW_{\text{in, cyl P}}(i) = (P_{\text{cyl}} - P_{\text{back}}) A_p (H_p(i) - H_p(i - 1)) \quad (5.19)
\]

The total work requirement in joules (J) is the sum of the two work inputs, friction and cylinder pressure, as shown in \((5.20)\).

\[
dW_{\text{in, total}}(i) = dW_{\text{in, friction}} + dW_{\text{in, cyl P}} \quad (5.20)
\]

The work output of the system in each time step is characterized by the energy added to the tank (the compressed air) and the further compression of the gas already inside the tank to accommodate the additional gas added. The work added to the tank is calculated based on the number of moles entering the tank and the ratio of the inlet pressure to the tank pressure, as given in \((5.21)\). The work output from
compressing gas already inside the tank is calculated based on the number of moles inside the tank and the change in pressure from the previous to the current time step, as given in (5.22). The exhaust pressure tank is considered to be isothermal.

\[
dW_{out, \text{added}}(i) = n_{inT}(i)RT_{tank}(i) \log \left( \frac{P_{in}(i)}{P_{tank}(i)} \right) \tag{5.21}
\]

\[
dW_{out, \text{in tank}}(i) = n_{tank}(i-1)RT_{tank}(i) \log \left( \frac{P_{tank}(i-1)}{P_{tank}(i)} \right) \tag{5.22}
\]

The total work out of the system in joules (J) is the sum of the two work outputs from air added to the tank as well as the further compression of the air inside the tank, as shown in (5.23).

\[
dW_{out, \text{total}}(i) = dW_{out, \text{added}}(i) + dW_{out, \text{in tank}}(i) \tag{5.23}
\]

While the efficiency, \( \eta \), can be calculated for every time step, the meaningful calculation is the ratio of work added to the system in the form of compressed product to the work required to drive the engine over an entire cycle; see (5.29). This is an average value for the efficiency over a cycle.

### 5.2.3 Updating temperatures

The last step in the model is to update the temperature of the engine block based on the heat flux due to friction and the heat generated in the adiabatic compression. Because the model uses a constant friction term (as described in Section 5.4.5), the assumption made is that all friction translates to an increase in engine block temperature only, with no contribution to a change in the gas temperature, as given in (5.24).

\[
dT_{eng, friction}(i) = dW_{in, friction}(i) \frac{1}{c_p m_{eng}} \tag{5.24}
\]
There are two additional heat fluxes that result in temperature changes. The first is due to the interaction of cylinder gas with the engine block, (5.25), and the second due to the engine block interacting with the environment, (5.26). In (5.25), $h_1$ is the heat transfer coefficient between the gas inside the cylinder and the engine block; in (5.26), $h_2$ is the heat transfer coefficient between the engine block and the outer environment. More information regarding $h_1$ and $h_2$ is provided in Section 5.4.2. In these equations, $T_{\text{eng}}$ is the engine block temperature and $T_{\text{env}}$ is the outside environment temperature.

\[
dT_{\text{gas, eng}}(i) = h_1SA(i)[T_{\text{cyl}}(i) - T_{\text{eng}}(i-1)]\frac{1}{c_p m_{\text{eng}}}
\]

(5.25)

\[
dT_{\text{eng, env}}(i) = h_2[T_{\text{eng}}(i-1) - T_{\text{env}}(i)]\frac{1}{c_p m_{\text{eng}}}
\]

(5.26)

These three temperature changes ((5.24), (5.25), and (5.26)) provide the input to calculate the temperature of the gas in the cylinder, (5.27), and the temperature of the engine block, (5.28).

\[
T_{\text{cyl, updated}}(i) = T_{\text{cyl}}(i) - dT_{\text{gas, eng}}(i)
\]

(5.27)

\[
T_{\text{eng}}(i) = T_{\text{eng}}(i-1) + dT_{\text{gas, eng}}(i) - dT_{\text{eng, env}}(i) + dT_{\text{eng, friction}}(i)
\]

(5.28)

5.2.4 Cycle calculations

The final feature of the engine model is cycle calculations. The speed (rpm) dictates the cycle length, which allows the engine parameters to be calculated (cycle work, mass of gas into the tank, time intake is open, time exhaust is open, mass of gas leaked, etc.) as a function of cycle rather than time step. The system efficiency over the full cycle is also calculated. The efficiency is the ratio between work out and work in, as shown in (5.29).
\[ \eta = \frac{\Delta W_{out, total}}{\Delta W_{in, total}} \]  

(5.29)

5.3 Thermodynamics

The engine model represents an approximation of physics (mechanics) and thermodynamic relationships to determine the behavior of the engine driven as a compressor and the gas being compressed. This section is focused on the thermodynamics that enter into this model and the motivation behind some of the engine modifications.

5.3.1 Volume, pressure, and temperature

The ideal gas law, shown in (5.30), provides the intrinsic relationship between volume, pressure, and temperature.

\[ PV = nRT \]  

(5.30)

The amount of work to compress a gas is found by integrating pressure over a change in volume as shown in (5.31).

\[ W = \int_{V_i}^{V_f} PdV \]  

(5.31)

This section focuses on the thermodynamics surrounding compression. Compression is a polytropic process that is characterized by the relationship shown in (5.32) where pressure multiplied by volume raised to a polytropic index, \( n \), equals a constant.

\[ pV^n = C \]  

(5.32)

In the next section two cases for selecting a value of \( n \) are explored. The first case is where \( n = 1 \), which is an isothermal process. The second case is where \( n = \gamma \), the adiabatic coefficient; in this case, the process is adiabatic.
5.3.2 Isothermal and adiabatic processes

The two bounds of compression/expansion processes are isothermal and adiabatic conditions. An isothermal process is characterized by having a constant temperature ($\Delta T = 0$); an adiabatic process is characterized by having no heat transfer (it is completely insulated, $\Delta Q = 0$). If an isothermal and adiabatic compression both begin at the same initial volume and pressure ($V_i, P_i$) and end at the same final pressure ($P_f$), the final volume for the isothermal compression will be smaller than the final volume for the adiabatic compression. If the two processes, again beginning at the same initial volume and pressure, end at the same final volume ($V_f$), the pressure will be lower in the case of the isothermal process. See Fig. 5.1 for a plot of these two cases.

![Isothermal and adiabatic processes](image)

Figure 5.1: Isothermal and adiabatic processes to achieve the same pressure ratio (left side) and the same volume ratio (right side); with common initial pressure and volume

The act of compressing a gas performs work on the gas. However, the energy
content of a gas depends little on its density and is mainly a function of its temperature. For an ideal gas, the energy content of the gas is solely a function of temperature. Therefore, in an isothermal compression all the energy added to the gas during compression is removed as heat flow from the reservoir to maintain a constant temperature. In an adiabatic compression this energy is retained in the gas. Consequently, the temperature and pressure both increase. As a result, the gas offers a higher resistance to further compression, and it requires more work to compress a gas adiabatically than isothermally.

In the case of the engine compressor designed as per Chapter 4, initial and final volumes are fixed. The volumetric ratio is set by the cylinder-piston geometry. Because there is an interest in achieving high pressures, adiabatic compression may appear to be desirable because, as shown in Fig. 5.1, the final pressure is higher for this process. Recall, however, that the work requirement is also higher. Normalized by pressure, the work requirement is higher in the adiabatic compression. The goal, then, is to pursue isothermal compression to minimize work requirements.

The isothermal and adiabatic work requirements (the areas under the curves in Fig. 5.1) are given in (5.33) and (5.34), respectively.

Recall that for an isothermal process, temperature is constant and $n$ from (5.32) is 1, which means that $PV = P_i V_i = P_f V_f$.

$$W_{iso} = \int_{V_i}^{V_f} PdV = nRT \int_{V_i}^{V_f} \frac{1}{V} dV = nRT \ln \frac{V_f}{V_i} = P_i V_i \ln \frac{V_f}{V_i} \quad (5.33)$$

For an adiabatic compression, $n$ from (5.32) is equal to $\gamma$, so $PV^\gamma = P_i V_i^\gamma = P_f V_f^\gamma$.

$$W_{adi} = \int_{V_i}^{V_f} PdV = \int_{V_i}^{V_f} \frac{P_i V_i^\gamma}{V^\gamma} dV = P_i V_i^\gamma \int_{V_i}^{V_f} \frac{1}{V^\gamma} dV \quad (5.34)$$

The integral from (5.34) is solved in (5.35).
The solution in (5.35) can be rewritten, as shown in (5.36).

\[ W_{adi} = P_i V_i \left[ \frac{V_f^{1-\gamma} - V_i^{1-\gamma}}{1-\gamma} \right] = \frac{P_f V_f - P_i V_i}{1-\gamma} \]  

(5.35)

The claim made here regarding the magnitude of the work requirements is that for \( \gamma > 1 \), \( W_{adi} > W_{iso} \). This is demonstrated by looking at (5.33) and (5.36). If both equations are divided by \( P_i V_i \), \( V_f/V_i \) is replaced with \( x \), and \( 1 - \gamma \) is replaced with \( \alpha \), the result is that \( W_{adi} = \frac{1}{\alpha} (x^\alpha - 1) \) and \( W_{iso} = \ln x \). If \( W_{adi} > W_{iso} \), the expression shown in (5.37) is correct.

\[ \frac{1}{\alpha} (x^\alpha - 1) > \ln x \]  

(5.37)

The expression in (5.37) is rewritten, as shown in (5.38).

\[ (x^\alpha - 1) > \ln x^\alpha \]  

(5.38)

Replacing \( x^\alpha \) with \( x \) leads to (5.39), which is known to be true because the derivative of \( \ln x \) at \( x = 1 \) is 1 as shown in (5.40), and the slope of \( \ln x \) is decreasing for \( x > 1 \).

\[ (x - 1) > \ln x \]  

(5.39)

\[ \frac{d}{dx} \ln x \bigg|_{x=1} = 1 \]  

(5.40)

The work requirement for an adiabatic process, therefore, is greater than the work requirement for an isothermal process. For this reason, in the experimental work, isothermal processes are desirable to reduce the work input. In practice this is a difficult task because it requires having a means of carrying away all the heat.
produced during compression. However, in an isothermal compression, because the system still has friction, the contribution of the friction work to total work is now higher. Unintuitively, as discussed more in Section 6.3, the efficiency in an isothermal process where there is friction may be lower than in an adiabatic process.

5.4 Fitting model parameters experimentally
The model needs to capture the engine cylinder, but it also needs to include the features of the intake conditions and exhaust pressure tank with which it interacts and to which it is connected. A number of engine features in the model were set directly from the specifications of the actual components used experimentally (e.g., the engine and the pressure tanks). Many model features were fitted from experiments. Because the exact parameters needed in the model are not always easily measured in practice, some assumptions are made to simplify the model while still following experimental results. The parameters informed from experimental results include the check valve open/close delays, which are a feature of the intake and outlet system, the flow through intake and exhaust check valves, engine leakage, engine friction, and engine thermal properties. The following sections include the details of how each model component was defined.

5.4.1 Engine
5.4.1.1 Geometry and material
The engine geometry and material used in the model were set to replicate the engine used for experimental work (see Chapter 4). The internal combustion engine used experimentally is a one cylinder, four-stoke, 3 HP, 79 cc OHV (overhead valve) horizontal shaft gas engine. The compression ratio is 8.5:1, maximum RPM is 3600 RPM, and bore \( \times \) stroke is 52.0 mm \( \times \) 37.0 mm (2.0 in. \( \times \) 1.5 in.). The engine block is made of cast iron. The engine mass is 10 kg. Given the cylinder volume (79
cc) and the compression ratio (8.5:1), a non-modified engine can be expected to have a dead, or clearance, volume, of 79 cc/8.5, or 9.3 cc. These geometric and material specifications are directly put into the engine model.

5.4.1.2 Heat capacity

The next model component that is derived from the physical engine is heat capacity. In an unmodified internal combustion engine, heat transfer is a very complicated parameter. Limiting the heat inside the cylinder is necessary in order to protect the metal engine components from fatigue: “the peak burned gas temperature in the cylinder of an internal combustion engine is of order 2500 K. Maximum metal temperatures for the inside of the combustion chamber space are limited to much lower values by a number of considerations, and cooling for the cylinder head, cylinder, and piston must therefore be provided” [22]. Heat transfer is much more complicated, as Heywood goes on to explain [22]:

Heat transfer affects engine performance, efficiency, and emissions. For a given mass of fuel within the cylinder, higher heat transfer to the combustion chamber walls will lower the average combustion gas temperature and pressure, and reduce the work per cycle transferred to the piston. Thus specific power and efficiency are affected by the magnitude of engine heat transfer. Heat transfer between the unburned charge and the chamber walls in spark-ignition engines affects the onset of knock which, by limiting the compression ratio, also influences power and efficiency. Most critical is heat transfer from the hot exhaust valve and piston to mixture in the end-gas region. Changes in gas temperature due to the heat-transfer impact on emission formation processes, both within the engine’s cylinder and in the exhaust system where afterburning of CO and HC occurs. The exhaust temperature also governs the power that can be obtained from exhaust energy recovery devices such as a turbocharger turbine. Friction is both affected by engine heat transfer and contributes to the coolant load. The cylinder liner temperature governs the piston and ring lubricating oil film temperature, and hence its viscosity. Piston and liner distortion due to temperature non-uniformities have a significant impact on the piston component of engine friction. Some of the mechanical energy dissipated due to friction must be rejected to the atmosphere by the cooling system. The fan and water pump power requirements are determined by the
magnitude of the heat rejected. The importance of engine heat transfer is clear. 22

Understanding an unmodified engine’s heat transfer is difficult. For the engine compressor built and modified here, most of these complications can be avoided. Temperature differences are generally much smaller here because there is no combustion, and the process is relatively slow compared to the speed of heat release in the spark ignition. Therefore, this model dramatically simplifies the heat transfer calculations. Thermal sensitivity analyses are included to justify the simplifications made. The goal has been to maintain a box model, where the entire gas volume and the entire engine block are taken to be homogeneous. The change in the size of the contact area is accounted for, but the heat delivered is distributed throughout the engine block.

During the engine compression cycle in this model, heat transfer occurs between the cylinder gas and the cylinder walls, and further to the engine block and from there to the outside environment. To avoid having a high resolution heat transfer model, which would go beyond the resolution of the calculations for the other model parameters, the heat transfer has been simplified to two components: heat transfer between the gas inside the cylinder and the engine block and heat transfer between the engine block and the outside environment. Setting the model up in this manner requires neglecting the complex thermal profiles that will form in a thin skin layer in the cylinder wall and in the gas temperature profile in the vicinity of the wall. The model is simplified by setting the entire engine block to one temperature and neglecting the thermal gradient in the cylinder skin layer. This simplification is justified by going through the following calculations, which find that the heat capacity of the thermally responsive cylinder skin layer in a single engine cycle is small in comparison to the heat capacity of the engine block.

The engine block wall thickness can be measured and is on the order of a few centimeters. The thickness that is important in this model is the effective thickness of
the layer that responds thermally to temperature changes of the gas inside the cylinder in the time of a single cycle, or a fraction of a cycle. Heat flux, $\dot{q}$, is proportional to thermal conductivity, $k$, and temperature change, $\Delta T$. Using the assumption that $\Delta T$ is constant, the layer thickness from diffusion can be calculated. The result of these calculations is a thickness that results in a cylinder wall heat capacity so much larger than that of the gas inside the cylinder, that the model can neglect heat transfer to the cylinder walls in an individual cycle. Even the boundary layer stays at a roughly constant temperature.

Table 5.3 lists the parameters used in the heat flux and internal energy calculations to determine cylinder skin layer thickness and temperature change. Table 5.4 lists the material properties for aluminum (as a reference), cast iron, and air.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Name</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{q}$</td>
<td>Heat flux</td>
<td>W/m²</td>
</tr>
<tr>
<td>$k$</td>
<td>Thermal conductivity</td>
<td>W/m/K</td>
</tr>
<tr>
<td>$h$</td>
<td>Heat transfer coefficient</td>
<td>W/m²/K</td>
</tr>
<tr>
<td>$dT$</td>
<td>Temperature difference</td>
<td>K</td>
</tr>
<tr>
<td>$dt$</td>
<td>Time change</td>
<td>s</td>
</tr>
<tr>
<td>$dx$</td>
<td>Thickness</td>
<td>m</td>
</tr>
<tr>
<td>$dU$</td>
<td>Internal energy</td>
<td>kJ/m³</td>
</tr>
<tr>
<td>$dH$</td>
<td>Enthalpy</td>
<td>kJ/m³</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
<td>kg/m³</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat</td>
<td>kJ/kg/K</td>
</tr>
</tbody>
</table>

Table 5.4: Material parameters (from online and from [8]); units are in MKSA as given in Table 5.3

<table>
<thead>
<tr>
<th>Variable</th>
<th>Aluminum</th>
<th>Cast iron</th>
<th>Air</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k$</td>
<td>205</td>
<td>58</td>
<td>0.024</td>
</tr>
<tr>
<td>$\rho$</td>
<td>2712</td>
<td>7272</td>
<td>1.169</td>
</tr>
<tr>
<td>$c_p$</td>
<td>0.91</td>
<td>0.42</td>
<td>1.004</td>
</tr>
</tbody>
</table>

Two equations are needed to calculate the relationship between temperature, heat exchange, and material thickness: heat flux (5.41) and enthalpy (5.42).
The equation for heat flux relates heat per unit area to thermal conductivity and the change in temperature over the material thickness.

\[ \dot{q} = -k \frac{dT}{dx} \]  
(5.41)

The equation here for enthalpy can be thought of as an internal energy where volume changes are neglected (i.e., a constant volume is used).

\[ dH = \rho c_p dT \]  
(5.42)

Heat flux multiplied by time is equal to enthalpy multiplied by thickness as shown in (5.43).

\[ \dot{q} dt = dH dx \]  
(5.43)

The thickness of the material, given the material properties, can be found by rearranging (5.43), as show in (5.44).

\[ dx = \left( \frac{k dt}{\rho c_p} \right)^{1/2} \]  
(5.44)

For a cast iron engine block running at a speed of 1,000 rpm, the cycle time is 60 ms \((dt = 0.06 \text{ s})\). Solving (5.44) as shown in (5.45) for cast iron and the cycle time specified, a thickness \((dx)\) equal to 1.02 mm is calculated

\[ dx = \left( \frac{(58)(0.06)}{(7272)(0.46 \times 10^3)} \right)^{1/2} \]  
(5.45)

As engine speed increases, the cycle time \((dt)\) decreases, which means the effective cylinder wall thickness \((dx)\) also decreases. Because the thickness of the effective cylinder skin layer is so small compared with the engine block, the heat capacity of this skin layer over a cycle is negligible.
The heat transfer coefficient in the gas is much lower than in the steel. There is a boundary layer that is laminar and through this heat has to pass. This is a much more severe obstacle than the steel. This analysis makes it possible to ignore the temperature fluctuations in the skin layer and explains why the process follows the adiabat. The heat transfer coefficient between the gas and cylinder is an empirical number and it involves the heat transfer coefficient through the gas to the piston wall. The model’s thermal properties are discussed more in the following section.

5.4.2 Thermal properties

As described earlier, characterizing engine heat transfer is difficult. Fig. 5.2, which is a reproduction from [22], shows a sketch of the components of engine heat transfer between the cylinder gas, the cylinder wall, and the engine coolant. In this engine model, the heat transfer includes just two fluxes.

Because the gas compression inside the engine cylinder cannot shed heat quickly enough to remain isothermal, the gas warms as the pressure increases. Similarly, when expanding, the gas will cool. The heat flux between the gas and the engine block and the engine block and the environment will depend on their temperature
differences. Since the engine warms up slowly, it will eventually transfer heat from
the outside of the block into the ambient environment. There is heat transfer between
the gas and the engine and between the engine and the environment. Heat transfer is
considered positive when moving towards the outward direction, that is, from the gas
inside the cylinder to the engine block to the environment. Heat transfer from the
engine block to the gas inside the cylinder is considered flux in the negative direction.

In the previous section, the thickness of material that would react to a temper-
ature change in an engine cycle while running at 1,000 rpm was found. As a first
approximation, the heat (in kW) per unit of temperature change in a cycle for a
cylinder volume (79 cc) filled with air and for an engine block with mass of 10 kg
can be calculated. These two calculations are given in (5.46) and (5.47), respectively,
where the temperature change during one engine cycle, \( \frac{dT}{dt} \), is equal to 1. The
difference is five orders of magnitude, which means that in a given cycle, temperature
change in the engine block can be neglected because the required heat to change its
temperature is so much greater than the capacity of the gas. Of course, over time,
the engine block can heat up, and this is kept track of in the engine model. The
engine block then will also have a heat flux with the environment.

\[
Q_{air} = \frac{c_p \rho V dT}{dt} = (1.005)(1.169)(7.9 \times 10^{-5}) = 9.3 \times 10^{-5} \text{kW} \quad (5.46)
\]

\[
Q_{eng} = \frac{c_p m_{eng} dT}{dt} = (0.42)(10) = 4.2 \text{kW} \quad (5.47)
\]

In the model, the heat transfer coefficients are estimated. There are two heat
fluxes: one between the gas inside the cylinder and the engine block (see (5.48)) and
the second between the engine block and the environment (see (5.49)).

\[
\dot{q}_1 = h_1 A_1 dT_1 \quad (5.48)
\]
In (5.48), \( h_1 \) is the heat transfer coefficient between the cylinder gas and the engine block in units of \( \text{Wm}^{-2}\text{K}^{-1} \). \( A_1 \) is the area of the exposed cylinder walls, piston head, and cylinder top and varies as a function of piston location. \( dT_1 \) is the difference in temperature between the cylinder gas and the engine block.

The expression for the second heat flux is similar as shown in (5.49).

\[
\dot{q}_2 = h_2 A_2 dT_2 \tag{5.49}
\]

\( h_2 \) is the heat transfer coefficient between the engine block and the outside environment in units of \( \text{Wm}^{-2}\text{K}^{-1} \). \( A_2 \) is the area of the exposed engine block (it is fixed). \( dT_2 \) is the difference in temperature between the engine block and the outside environment.

The two heat transfer coefficients, \( h_1 \) and \( h_2 \), are estimated based on values from literature. \( h_1 \) is assigned a value of 200 \( \text{Wm}^{-2}\text{K}^{-1} \) based on characterizing the heat transfer as forced convection. The heat transfer between the engine block and the surrounding air is considered natural convection, so \( h_2 \) is set at 20 \( \text{Wm}^{-2}\text{K}^{-1} \).

### 5.4.3 Pressure tanks

In the engine model, exhaust from the engine cylinder flows through the exhaust check valve, into an exhaust hose, and into the exhaust pressure tank. Experimentally, a 4 liter pressure tank, originally designed as an Argon pressure tank, is used to contain the engine exhaust. This volume is used in the engine model. In some experimental runs, a 5 gallon exhaust pressure tank was used; when the model simulation is changed to incorporate this larger tank, it is noted. In other experiments, the 5 gallon pressure tank is used as the inlet pressure tank to feed inlet air at elevated pressure. Inlet pressure conditions are always noted.
5.4.4 System flows

5.4.4.1 Valve velocity and position

The engine used experimentally is a four-stroke (intake, compression, power, exhaust) spark-ignition gasoline engine. Because there is no need for the power stroke when the engine is retrofitted as a compressor, the system performs most efficiently with a two-stroke cycle: an intake stroke followed by a compression/exhaust stroke. To make this modification using the engine intake and exhaust valves, the camshaft lobes would need to be redesigned to enable more frequent valve openings. To avoid designing and fabricating new camshaft lobes, the engine intake and exhaust valves are not used; the pushrods are removed completely so the valves remain closed. In place of the traditional valves and valve timing, the intake and exhaust valves used experimentally are pressure-activated check valves (more specifics about the types of valves are included in Chapter [4]). These check valves are built to allow flow in one direction based on a pressure difference. The model mimics this physical parameter by opening valves based on the pressure differences. When the pressure is higher on the intake of a check valve, the valve opens to let gas pass. The valves are 1/8” NPT.

From experimental results, there is evidence of inertia in the valves, which means the valves can be open, closed, or moving in between. The check valves are modeled as ball valves where the position and velocity of the ball are recorded so as to characterize the flow through the valves. The time it takes to move from open to closed or vice versa is a function of the pressure difference pushing on the ball as given in (5.1) and (5.2). The constant, $C$, in (5.1) was calibrated by matching experimental output with model output. Fig. 5.3 shows a sample plot from experimental data, where the first vertical line (in purple) indicates when cylinder pressure is greater than tank pressure, which is the point when the exhaust check valve should open. The second vertical line indicates when tank pressure rises. There is a delay here of about 3 milliseconds. The first gray vertical line shows when cylinder pressure falls below tank pressure,
which is the point when the exhaust check valve should close. The tank pressure, however, appears to decrease for about 6 ms until the pressure becomes more stable, which is when the check valve actually closed, shown with the second vertical gray line in Fig. 5.3.

To show another example, Fig. 5.4 shows the delay at higher pressures (> 300 psia). Here, the open delay is still about 3 ms. Because the tank pressure is so high, by the time the exhaust valve opens between the engine cylinder and the exhaust hose, the cylinder pressure is already falling below the exhaust tank pressure, which should signal the check valve to close. Here it looks like it takes about 9 ms in this example before tank pressure returns to approximately the pressure it should have been when the valve should have closed.

These examples show the order of magnitude of the transition between check valves states and were used in the calibration of $C$.  

Figure 5.3: Check valve open and close delay (from experimental data)
5.4.4.2 Leakage

Gas can leak both in and out of the cylinder via the piston rings. When the cylinder pressure is higher than the piston back pressure, gas will leak out of the cylinder; when the cylinder pressure is lower than the piston back pressure, gas will leak into the cylinder. The engine leakage was measured with the engine piston stationary and in motion.

The relationship from literature [43] shown in (5.50) is used as an approximation for the flow of gas. This relationship includes a friction factor, $f$, and the geometry of the leak, $D$ and $L$, as well as the gas density, $\rho$. In this section, the focus is on the flow of gas that leaks around the engine piston rings.

\[ v = \sqrt{\frac{\Delta P \ D \ L}{2f \ L \ \rho}} \]  

(5.50)

In (5.50), the relationship between pressure difference, $\Delta P$, and flow, $v$, is sim-
plified as a quadratic. In reality, the relationship will depend on the flow rate. A
more robust approximation of the relationship between pressure difference and flow
is provided in (5.51).

\[ \Delta P = a \left( v_0 + \frac{v}{v} \right) v^2 \]  \hspace{1cm} (5.51)

Taking the limits shows that at very low flow rates, or when \( v << v_0 \), \( \Delta P = av_0v \). The relationship is linear at this limit. At very large flow rates, or when \( v_0 << v \), \( \Delta P = av^2 \). The relationship at this limit is quadratic. The system flows, however, may straddle this line. At low speeds viscous forces suggest a linear behavior, at high speed, turbulence may dominate and give a quadratic behavior. In handbooks, this transition is taken care of by assuming that the friction factor makes the transition. The flow parameters for this model are found by using the full form given in (5.51).

The expression in (5.51) can be expanded and rewritten as shown in (5.52).

\[ \Delta P - av_0v - av^2 = 0 \]  \hspace{1cm} (5.52)

Using this, \( v \) can be solved for by rewriting the quadratic as shown in (5.53).

\[ v^2 + v_0v - \frac{\Delta P}{a} = 0 \]  \hspace{1cm} (5.53)

Solving (5.53) results in (5.54), with the positive solution shown in (5.55).

\[ v = \frac{-v_0}{2} \pm \sqrt{\frac{v_0^2}{4} + \frac{\Delta P}{a}} \]  \hspace{1cm} (5.54)

\[ v = \frac{v_0}{2} \left[ -1 + \sqrt{1 + \frac{4\Delta P}{av_0^2}} \right] \]  \hspace{1cm} (5.55)

Experimental data is used to solve for \( a \) and \( v_0 \). There are two experimental protocols for the engine leakage runs. The first experiments were conducted on a stationary piston with the exhaust tank filled with air at a pressure, \( P_E \). The piston is
moved to bottom dead center (BDC). The 5 gallon inlet pressure tank is filled with air at a pressure, $P_I$, where $P_E > P_I > P_{ambient}$. As a result the connection between the engine and the exhaust tank should be closed at all times. Data acquisition is started, which includes digital measurements from the pressure transducer at the cylinder, the pressure transducer at the exhaust tank, and the cylinder head thermocouple. The inlet valve (upstream of the intake check valve) to the engine is opened which causes air to flow from the inlet pressure tank, through the intake check valve, and into the engine cylinder. Because the exhaust pressure tank is at an elevated pressure, air can only flow out of the engine through leaks and not towards the exhaust pressure tank. The experiment was repeated with the piston at top dead center (TDC). These runs, shown in Fig. 5.5, provide the information to calculate the leakage of the engine cylinder, which includes leaks around the piston rings.

For the second leakage experimental protocol, the intake check valve is still connected to the elevated pressure of the 5 gallon pressure tank. Now, the exhaust check valve is routed to an exhaust pressure tank with ambient air pressure. Here, $P_I > P_E = P_{ambient}$. These runs show gas moving from the inlet pressure tank through the engine and into the exhaust tank. The inlet pressure tank decreases in pressure as gas moves into the engine and exhaust tank and to the environment through leaks. Once the pressure in the system falls below the pressure in the exhaust tank, the exhaust check valve closes (and remains closed). For the remaining of these runs, the system behavior is as it was in the previous experimental protocol: the only way for gas to exit the cylinder once the exhaust closes is through leaks.

Data acquisition begins prior to opening the inlet valve, so the first few data points in these runs show a rise in engine cylinder pressure from ambient to the inlet tank pressure after the inlet is opened. Cylinder pressure then begins falling after the exhaust valve closes. To compare multiple runs across the two experiment types on the same plot, the early data points in these runs are ignored, and a smoothing
routine is applied to multiple tests all beginning at just under 45 psia. A summary of these data runs is shown in Fig. 5.5 where the smoothed data of five leakage tests are plotted. This shows only minor differences between runs where the piston is at BDC versus TDC.

![Figure 5.5: Smoothed fits for cylinder pressure (psia) vs. elapsed time for multiple leakage tests](image)

From these runs, the flow velocity as a function of pressure difference can be calculated. The steps are included here. Fig. 5.6 shows the full dataset used for the leakage flow velocity calculation. The run shown in Fig. 5.6 is the raw data for the red smoothed data in Fig. 5.5. For this analysis, the focus is on the portion of data after the inlet tank has been opened to the engine cylinder. That is, the first few points where the engine cylinder pressure rises are disregarded.

As (5.51) shows, the flow, \( v \), is related to pressure difference, \( \Delta P \), not pressure. Pressure difference is the cylinder pressure minus ambient pressure; i.e., the plot in Fig. 5.6 shifted down by 15 psia (atmospheric pressure). The portion of data in Fig. 5.6 where the cylinder is being filled with air from the inlet pressure tank is ignored, and the data is narrowed down to the portion where pressure from the
system begins to decrease.

Now using the pressure difference, the flow rate, $v$, can be determined. The first step in calculating this is to determine the mass of gas (in kg) remaining in the system at each time step as per (5.56), where $P$ is the pressure, $V$ is the system volume (inlet pressure tank plus cylinder volume), $R_{\text{specific}}$ is the specific gas constant (287 J/kg/K for dry air), and $T$ is the temperature (which is kept at ambient, 298.15K).

$$m = \frac{PV}{R_{\text{specific}}T} \quad (5.56)$$

The mass of gas remaining in the system is shown in Fig. 5.7.

The flow of gas out of the system (the leakage), $v$, is the derivative of the mass remaining in the system, $dm/dt$. The data in Fig. 5.7 is fit with a cubic function and the derivative is taken, which gives the leakage flow rate. Finally, $\Delta P$ versus $v$ is plotted and fit with (5.51) as shown in Fig. 5.8.

Notice that the flow rates in Fig. 5.8 are negative. This is a sign convention. Because the mass was measured in terms of gas remaining in the system, the mass
decreases as leakage continues over time. The flow rate, therefore, is negative because it is viewed from the perspective of within the system where gas is flowing outwards.
From the fit in Fig. 5.8, \( a \) and \( v_0 \) in (5.51) are solved for, which gives \( a \) a value of 210.5 and \( v_0 \) a value of -0.34. These parameters are put into the model to characterize the leakage flow. \( a \) is multiplied by a factor that adjusts the flow resistance, which is fit by matching model simulations with experimental data. It takes into account that the system is dynamic.

The experiment was repeated with the engine piston in motion, running at 1003 rpm and 1505 rpm. For these runs data acquisition also includes data from the variable frequency drive (VFD), including motor speed, current, voltage, and phase angle. As shown in Fig. 5.9, there is evidence of backflow (which is addressed as a term different than leakage) from the exhaust pressure tank back into the engine cylinder.

These experiments were critical in calibrating the model. Using these runs, the flow resistances for leakage (the “fudge factor” on \( a \)), flow through the intake and exhaust check valves, and flow from the hose to the exhaust pressure tank were fine tuned. Using these runs, a good approximation for the constant, \( C \), is also found, which is used in determining the check valve velocity (and therefore position) in (5.1).

### 5.4.4.3 Flow through check valves

To determine the flow rate through the intake and exhaust check valves, the data calculated from the second experimental protocol on the stationary engine described in Section 5.4.4.2 is used. Recall that in this case, the exhaust pressure tank begins filled with air at ambient pressure. Now \( P_I > P_E = P_{\text{ambient}} \), so air flows from the inlet tank, into the engine cylinder through the inlet check valve and out the engine through leakage and through the exhaust check valve to the exhaust hose and then the exhaust pressure tank.

Fig. 5.10 shows the results of this data run. The graph shows two very distinct phases. There is a very rapid initial phase in which the pressure in both the cylinder
and the exhaust tank are rising. While not measured digitally, the pressure in the much larger inlet tank is dropping as air expands into the rest of the system (engine cylinder and exhaust pressure tank). Once the pressure in the system has equilibrated, the exhaust tank becomes separated from the system, because the exhaust check valve closes. From here on out, the experiment performed is very similar to before, which showed the pressure in the engine cylinder and inlet tank dropping slowly as leaks
in the system carry the gas away. The leakage results shown in Fig. 5.10 are also plotted in Fig. 5.5 (this experiment is the blue curve in Fig. 5.5), which shows the leakage experiments are consistent.

Recall that this experiment is run with the piston stationary. Because the leakage experiment includes flow through the inlet check valve and flow through the engine leaks, this experiment now isolates the flow through the exhaust check valve. The portion of the data in Fig. 5.10 that is relevant to this analysis is the very first section where gas flowing from the inlet pressure tank through the cylinder and into the exhaust pressure tank is tracked. This zoomed in view is provided in Fig. 5.11 which shows both the cylinder and the exhaust pressure transducer readings.

Notice in Fig. 5.11 that the cylinder pressure is leading, while the exhaust pressure follows. Recall that gas is flowing at elevated pressure from the inlet pressure tank, and that the pressure at the inlet tank can never be less than the final pressure at the exhaust tank. What is clear from the data is that the cylinder pressure is much
closer to the exhaust tank pressure than it is to the inlet tank pressure. The pressure drop between the inlet tank and the cylinder is far larger than the pressure drop between the cylinder and the exhaust system. Fig. 5.11 suggests that the cylinder pressure is indeed between the pressure of the inlet and exhaust pressure tanks, but from the figure, it is clear that the cylinder pressure is very close (within 5 psia) to the pressure at the exhaust pressure tank. This suggests that the flow resistance from the inlet pressure tank to the cylinder is about ten times the flow resistance between the cylinder and the exhaust pressure tank. The gap is closing in Fig. 5.11 because the pressure difference between all three components (the inlet pressure tank, the cylinder, and the exhaust pressure tank) goes away.

Just as was done in Section 5.4.4.2, the parameters in (5.51) are fit to determine the flow characteristics. Because flow through the exhaust check valve is now isolated, $\Delta P$, which is now the pressure difference between the cylinder pressure and the exhaust tank pressure, must first be calculated. The results are shown in Fig. 5.12.
The next step is to solve for the amount of gas (in kg) in the exhaust pressure tank. (5.56) is used again to solve for the mass. The results are provided in Fig. 5.13.
As in Section 5.4.4.2, the data in Fig. 5.13 is fit with a cubic function and the derivative, $dm/dt$, is evaluated to find the flow of gas, $v$, between the engine cylinder and the exhaust pressure tank. Because the curve in Fig. 5.13 is almost linear, the derivative is almost constant. The results from Fig. 5.12 and Fig. 5.13 are put together, and the pressure difference versus the flow rate is plotted as shown in Fig. 5.14.

![Figure 5.14: Pressure difference between exhaust pressure tank and engine cylinder (psia) vs. flow rate (kg/hr)](image)

From the fit in Fig. 5.14, $a$ and $v_0$ in (5.51) are solved for: $a$ is found to be 0.039 and $v_0$ is 1.40. These two parameters are used in the model for three flow rates: flow through the intake check valve, the exhaust check valve, and from the exhaust hose to the exhaust pressure tank. In each of these three cases (as was true in the case for the leakage flow as well), a factor is applied to $a$ to vary the respective resistance of each of the flows. These were calibrated with the experiments shown in Fig. 5.9.

From the experiment shown in Fig. 5.10 (with a zoomed view in Fig. 5.11), the relative resistance of the intake versus the exhaust check valve can be approximated.
The gas flow through the system is driven by a high (65 - 70 psia) pressure in the inlet pressure tank. The gas then moves through the cylinder and into the exhaust tank as described earlier and as shown in Fig. [5.11]. It is clear that the largest resistance in the system is between the inlet pressure tank and the cylinder, and that there is very little resistance between the cylinder and the exhaust pressure tank. The pressure drop between the inlet pressure tank and the cylinder, therefore, is likely higher than the pressure drop between the engine cylinder and the exhaust tank. Because the system does not have a pressure transducer at the inlet pressure tank (only a pressure gauge), the flow rate through the intake check valve is not directly measured. In the model, the same values for \( a \) and \( v_0 \) are used, but different factors are applied to each \( a \) to adjust for the resistances. Another point to note is that in this example, the resistance is traced from the inlet pressure tank, through a hose, and into the engine cylinder. When experiments use ambient air as the inlet gas source, there is no hose and no inlet pressure tank, so a lower resistance (a lower \( a \) factor) is expected for these runs.

5.4.5 Friction

The friction parameters in this model were first approximated from literature, and then adjusted based on experimental data. In a typical engine, friction power accounts for about 10% of the developed engine power [11]. There are three types of friction identified here: boundary, hydrodynamic, and turbulent. The derivation of friction comes from [22], which states:

The work per cycle for each component \( i \) of the total friction is given by integrating the friction force \( F_{f,i} \) times its displacement \( dx \) around the cycle:

\[
W_{f,i} = \int F_{f,i}(\theta)dx
\]

The friction force components are either independent of speed (boundary friction), proportional to speed (hydrodynamic friction) or to speed
squared (turbulent dissipation), or some combination of these. It follows that the total friction work per cycle (and thus the friction mean effective pressure) for a given engine geometry engine will vary with speed according to

\[
W_{tf} \text{ (or tfmep)} = C_1 + C_2N + C_3N^2
\]

Some of the components of hydrodynamic lubrication friction and turbulent dissipation will be dependent on mean piston speed rather than crankshaft rotational speed \(N\). Examples are piston skirt and ring friction, and the pressure losses associated with gas flow through the inlet and exhaust valves. [22]

Boundary friction occurs between two metals, here, the piston and the cylinder walls and is independent of speed. Boundary friction will result in an increase in engine block temperature. Hydrodynamic friction results from the lubricant (oil) between the piston and cylinder walls and is proportional to piston speed. Hydrodynamic friction should add heat to the engine block. Turbulent friction results from the mixing of moving fluids in and out of the engine cylinder and is proportional to the square of piston speed. Turbulent friction will result in an increase in cylinder gas temperature.

To experimentally determine the friction components, four experiments were conducted by running the engine at varying speeds with both inlet and exhaust open. Data was collected from the variable frequency drive and average power was calculated. Fig. 5.15 shows a summary of the data collected in one of the four experiments from the variable frequency drive and pressure sensors. The top figure shows the current, voltage, and phase angle. The second figure shows the calculation for power based on (4.4), where \(P\) is power in watts (W), \(V\) is voltage in volts (V), \(I\) is current in amps (A), and \(\angle_{PF}\) is the power factor angle. The third figure shows the motor speed, and the bottom figure shows the cylinder and exhaust tank pressures.

The power requirement (second plot in Fig. 5.15) is the required work to overcome friction in the motor and engine alone, as no gas is being compressed when both
intake and exhaust are open to the environment. Fig. 5.16 shows the average power calculated from each experiment plotted at the corresponding engine speed. Also
included is a quadratic fit, with the equation provided in (5.57).

![Figure 5.16: Friction test: average power (W) vs. speed (rpm)](image)

\[ P_{\text{average}} = 4.09 \times 10^{-6} x^2 + 5.85 \times 10^{-2} x + 39.26 \]  

(5.57)

The average power requirement as a function of speed is increasing, as illustrated in Fig. 5.16. These figures can be expressed as work per cycle by dividing by speed, which results in joules per cycle. To compare the experimental data with the relationship for friction provided in literature as described above, the work per cycle can be expressed as a pressure. Pressure times volume is work, so the work (J) per cycle is divided by twice the engine cylinder volume (79 cc), as the piston moves twice through the cylinder during one cycle. This gives a measure of the required friction pressure (expressed here in kilopascals (kPa)). These points are plotted in Fig. 5.17.

From the earlier description, the friction mean effective pressure should be quadratic. Fig. 5.17 includes quadratic and linear fits of the four data points. What this shows is that the data is not enough to confidently characterize each component of friction. If friction is constant, there is a 1/16 chance that the result plotted here will occur, with
Figure 5.17: Friction pressure (kPa) vs. engine speed (rpm); experimental data with quadratic and linear fits

the first two points above and the second two points below the constant friction term. Without additional data, it is difficult to say if the quadratic has a minimum around 1500 rpm (as plotted) or at a lower or high speed. If this quadratic is correct, the friction coefficients are as shown in (5.58). Notice that $C_2$ here would be negative, which is unintuitive, but may point to the fact that friction can actually be decreasing with increasing speeds due to lubrication and system inertia (for additional discussions on engine friction, see [26], [42], [51], [37], and [23]).

$$W_{tf} = 78.0 - 5.1 \times 10^{-2} N + 1.46 \times 10^{-5} N^2 \quad (5.58)$$

In Fig. 5.17 it is possible that there exists a downward trend in friction pressure at increasing engine speeds. A linear fit is included, though of course friction pressure does not go to zero as the fit suggests. It is possible, however, that the friction parameters conveyed in the literature described earlier are not quite right. Engines may be fine tuned to result in the lowest friction at or near the optimal design speed. If this is the case, it is reasonable to expect friction to have a minimum at some
mid-range speed.

The experimental results range from about 30 to 60 kPa (0.3 to 0.6 bar). For the engine model, a constant engine friction pressure of 45 kPa is used, which is about the average of the four data points measured experimentally. This is a simplification, because it is known that friction is a function of speed. Some components of friction also increase when maximum cylinder pressure increases, including friction on the main bearing, crankpin, piston rings, and piston body [47]. The piston ring friction accounts for 23% of total engine friction and, with a doubling in cylinder pressure, it increases by 23%. Modeling friction is, therefore, nontrivial and is simplified in this model with a constant term.

5.5 Model calibration

As mentioned in Section 5.4.4, the experiments shown in Fig. 5.9 were used to calibrate the model parameters. The model parameters that are fine-tuned are: cylinder clearance volume, check valve velocity, and the factor on $\alpha$ for four flows: leakage, inlet check valve, exhaust check valve, and flow from exhaust hose to exhaust pressure tank. Table 5.5 shows the results of the model specifications, and Fig. 5.18 shows a side-by-side comparison of the experimental data and the model output. While the simulation does not perfectly match the experimental data trajectory, in particular when the peak pressures are realized, the overall fit is quite good.

The parameters for flow resistances used here are the baseline for fitting other experimental data runs. Additional comparisons between experimental data and model simulations are provided in Chapter 6.
Table 5.5: Calibration model parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>1003</td>
<td>rpm</td>
</tr>
<tr>
<td>Clearance volume</td>
<td>8</td>
<td>cc</td>
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<tr>
<td>Inlet check valve a factor</td>
<td>100</td>
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</tr>
<tr>
<td>Exhaust check valve a factor</td>
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<td></td>
</tr>
<tr>
<td>Hose/tank a factor</td>
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<td></td>
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<tr>
<td>Leakage a factor</td>
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<td>Check valve constant</td>
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</tr>
<tr>
<td>Resolution</td>
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<td>ms</td>
</tr>
<tr>
<td>Exhaust hose volume</td>
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<td>cc</td>
</tr>
<tr>
<td>Exhaust tank volume</td>
<td>4</td>
<td>L</td>
</tr>
<tr>
<td>Inlet gas pressure</td>
<td>45</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust tank pressure</td>
<td>280</td>
<td>psia</td>
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</table>

Figure 5.18: Aluminum plate cylinder head with elevated inlet: experimental (left) and modeled (right) cylinder pressure and hose/exhaust tank pressure (psia) vs. elapsed time
5.6 Model temporal resolution

The engine compressor model’s temporal resolution can be adjusted. In the model calibration in Section 5.5, the model runs on a resolution of 0.1 ms. The temporal resolution is indicated for each model output presented in Chapter 6. A resolution of 0.1 ms is selected for most of the modeling. This time step was chosen because this is where model output converges. A higher resolution does not change the model results. To demonstrate this, a sample model output was run with varying time steps. The results of the exhaust tank pressure outputs are shown in Fig. 5.19. In Fig. 5.19 the right plot shows a zoomed view of the outlined area in the left plot. As is shown here, there is little difference in output at a resolution finer than 0.1 ms time steps (in this particular simulation, the maximum difference between the two curves at 0.1 ms and 0.05 ms is slightly over 0.5 psia).

Figure 5.19: Simulation exhaust tank pressure outputs at varying time steps from 0.05 ms to 1 ms; right plot shows zoomed view of outlined section in left plot

Fig. 5.20 shows the efficiency curves from the same model simulations with varying time steps. At a time step of 1 ms, the efficiency curve has a large spread. At a finer temporal resolution, the spread decreases. The left plot in Fig. 5.20 shows the results from all five runs; the right plot shows the results from a resolution with 0.1 ms and 0.05 ms. The curves for these two cases fall on top of each other. The magnitude of the
greatest difference between the cycle efficiency in any cycle at these two resolutions is less than 0.08%.

![Simulation cycle efficiencies at varying time steps from 0.05 ms to 1 ms; left plot shows all five curves; right plot shows the results at 0.1 ms and 0.05 ms only](image)

**Figure 5.20:** Simulation cycle efficiencies at varying time steps from 0.05 ms to 1 ms; left plot shows all five curves; right plot shows the results at 0.1 ms and 0.05 ms only

### 5.7 Summary

This chapter focused on the design and calibration of an engine model written in Python to simulate engine compressor performance. The compressor model was based on characterizing the physical and thermodynamic boundaries of the engine compressor system. The physical and material specifications were largely derived from the experimental setup used in the laboratory (as per the details in Chapter 4). Laboratory experiments were used to characterize delays in check valve opening/closing, flow and backflow through valves, flow through leaks, and the system’s friction.

Using a full experimental data run, the engine compressor model was then calibrated. The flow resistances, cylinder clearance volume, and check valve velocity were varied in order to match the model simulation with the experimental results. These parameters are used as a starting point to fit additional simulations with experimental output.

The model created here is very robust and provides a tool for comparing the
experimental data presented in Chapter 4 with modeling simulations to enable an understanding of model scenarios beyond what was accomplished in the laboratory. These experimental versus simulations comparisons are provided in Chapter 6. Model simulations that stretch beyond the laboratory experiments are also included in the next chapter.
Chapter 6

Validating the engine model with experimental data

6.1 Introduction
To best understand the Python engine model performance and justify using model simulations to explore scenarios beyond the experimental work completed, included in this chapter are examples of the output the model produces. Specifically, the model is compared with the experimental results and then used to explore simulations that go beyond the experimental setup. These additional simulations run sensitivity analyses on the importance of various design parameters including cylinder dead volume, heat exchange, engine speed, valve size, valve resistance, check valve delays, and engine friction.

6.2 Model and experimental comparisons
Included in this section are model simulations that are specifically designed to match experimental conditions to predict the performance of some of the experimental runs described in Section 4.7. These simulations span all engine retrofits: the original cylinder head, the acrylic plate cylinder head, and the aluminum plate cylinder head.

6.2.1 Standard cylinder head
Recall that the first experiments were run on the retrofitted engine with the original cylinder head as described in Section 4.7.1. The parameters selected for this simulation are given in Table 6.1. The engine speed was set directly from the experimental
conditions. The cylinder clearance, hose and tank volumes were set or approximated based on the experimental setup. The flow resistance factors (these are the factors used to adjust and fine-tune the flow resistance as described in Chapter [5]) were manually adjusted to replicate the flow behavior. The model resolution is set to 0.1 ms. The inlet gas is at ambient pressure. The 5 gallon pressure tank is used to store exhaust air.

Table 6.1: Original cylinder head model parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
<th>Units</th>
</tr>
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<tbody>
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<td>Engine speed</td>
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<td>rpm</td>
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<td>Clearance volume</td>
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<td>cc</td>
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<tr>
<td>Inlet check valve a factor</td>
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<tr>
<td>Exhaust check valve a factor</td>
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<tr>
<td>Hose/tank a factor</td>
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<td></td>
</tr>
<tr>
<td>Leakage a factor</td>
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</tr>
<tr>
<td>Check valve constant</td>
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</tr>
<tr>
<td>Resolution</td>
<td>0.1</td>
<td>ms</td>
</tr>
<tr>
<td>Exhaust hose volume</td>
<td>20</td>
<td>cc</td>
</tr>
<tr>
<td>Exhaust tank volume</td>
<td>5</td>
<td>gallons</td>
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<tr>
<td>Inlet gas pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust tank pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
</tbody>
</table>

Fig. 6.1 shows a side-by-side comparison of the experimental results for measured in-cylinder and exhaust hose/tank pressures (psia) over a twenty-five minute run (left) and the simulation outputs (right).

While the simulation largely agrees with the experimental data, there is a discrepancy in the difference between peak pressure in the exhaust hose/tank in the experimental results versus in the modeling results. In looking more clearly at just the exhaust hose/tank pressures (both experimental and modeling), there is evidence of why this may be. In Fig. 6.2, the experimental and modeled exhaust hose/tank pressures from Fig. 6.1 are plotted together. Fig. 6.2 highlights the large span of pressure measured in the experimental analysis. The model overlaps with the range of pressures measured, but does not vary as much in magnitude. This additional
variation, may be due to experimental dynamics not captured in the simulation. For example, system vibrations may have affected the pressure transducers.

Figure 6.1: Original cylinder head: experimental (left) and modeled (right) cylinder pressure and hose/exhaust tank pressure (psia) vs. elapsed time

Figure 6.2: Original cylinder head: experimental and modeled hose/exhaust tank pressure (psia) vs. elapsed time

The calculated efficiencies based on both experimental data and model output were also compared. Fig. 6.3 shows the efficiencies versus the exhaust hose/tank
pressures. Here notice that the experimental work shows a higher efficiency. This is largely due to what Fig. 6.2 shows: that the model does not get as high pressure into the exhaust tank.

![Efficiency vs. Exhaust Tank Pressure](image)

**Figure 6.3:** Original cylinder head: experimental and modeled efficiency vs. hose/exhaust tank pressure (psia)

### 6.2.2 Acrylic plate cylinder head

To compare the model simulation with the acrylic plate cylinder head, the data from two experimental runs are used due to gaps in data collection from sensor malfunction. The two comparisons are used to build an understanding of the acrylic plate cylinder head system.

#### 6.2.2.1 Short run

The first model is an 8 minute-long experiment where pressure data was collected from both the cylinder and exhaust hose/tank pressure transducers. This run does not have temperature and variable frequency drive data. The engine is modeled at
a speed of 437 rpm; however, because there is no VFD data, this speed may not be accurate for matching experimental conditions. The full parameter list is given in Table 6.2. Notice that the flow factors vary from those in Table 6.1. Here, the inlet $a$ factor is higher, which translates to an increased inlet flow resistance (decreased flow rate). Notice also that the exhaust check valve flow factor is smaller here, which signifies decreased exhaust flow resistance (increased flow rate). The leakage rate is also much larger in this example, but the clearance volume much smaller (4 cc versus 14 cc). The estimated clearance volume is likely too low. Again, the 5 gallon pressure tank is used to store exhaust.

Table 6.2: Acrylic plate cylinder head model parameters [Ex. 1]

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
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<td>Hose/tank $a$ factor</td>
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<td>Exhaust hose volume</td>
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<td>cc</td>
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<td>Exhaust tank volume</td>
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<td>gallons</td>
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<tr>
<td>Inlet gas pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust tank pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
</tbody>
</table>

Fig. 6.4 shows the experimental and simulation pressure measurements and calculations side-by-side. Fig. 6.5 shows the experimental and modeled exhaust hose/tank pressures plotted together.

In Fig. 6.5, the model output falls lower than the experimental data, which suggests that the flows are not quite matched to the experimental data.

Because there is no variable frequency drive data, the cycle efficiency is not calculated because the power requirement is unknown.
Figure 6.4: Acrylic plate cylinder head: experimental (left) and modeled (right) cylinder pressure and hose/exhaust tank pressure (psia) vs. elapsed time [Ex. 1]

Figure 6.5: Acrylic plate cylinder head: experimental and modeled hose/exhaust tank pressure (psia) vs. elapsed time [Ex. 1]

6.2.2.2 Long run

A second experimental and simulated acrylic plate cylinder head run is also included. Recall from Section 4.7.2 that this run likely had a leak during the experiment. To incorporate leakage into the model, the leakage flow factor is decreased, which reduces
the leakage flow resistance and increases the leakage flow. The cylinder clearance volume is also increased. The inlet flow factor is unchanged from Table 6.2 but the exhaust check valve and hose/tank flow factors are increased, which also reduces flow to the tank, which may have resulted from the suspected leak in the pipe nipple connection at the exhaust check valve. Because the pressure data in this run is accompanied by variable frequency drive data, this experimental run is included to compare the predicted model efficiency. The in-cylinder pressure measurement is not included here because the cylinder head pressure transducer malfunctioned, so the only experimental pressure data recorded is at the exhaust hose/tank. The model specifications are included in Table 6.3.

Table 6.3: Acrylic plate cylinder head model parameters [Ex. 2]

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>437</td>
<td>rpm</td>
</tr>
<tr>
<td>Clearance volume</td>
<td>14</td>
<td>cc</td>
</tr>
<tr>
<td>Inlet check valve $a$ factor</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>Exhaust check valve $a$ factor</td>
<td>1.5</td>
<td></td>
</tr>
<tr>
<td>Hose/tank $a$ factor</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Leakage $a$ factor</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>Check valve constant</td>
<td>0.015</td>
<td></td>
</tr>
<tr>
<td>Resolution</td>
<td>0.1</td>
<td>ms</td>
</tr>
<tr>
<td>Exhaust hose volume</td>
<td>20</td>
<td>cc</td>
</tr>
<tr>
<td>Exhaust tank volume</td>
<td>5</td>
<td>gallons</td>
</tr>
<tr>
<td>Inlet gas pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust tank pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
</tbody>
</table>

Fig. 6.6 shows a comparison of experimental and simulated data. Notice in particular how noisy the pressure transducer data is. This is likely due to the fact that there was a leak and the system was not able to seal off its pressure. The data spans about 40 psia in a cycle. Many of the model parameters provided in Table 6.3, therefore, may be adjusted and the model still calculates exhaust hose/tank pressure within the boundary of the experimental data. The selected parameters are meant to aim for the middle of the experimental data as shown in Fig. 6.7.
The experimental and simulated efficiency curves versus exhaust hose/tank pressure are shown in Fig. 6.8. The results are aligned within a few percentage points.
6.2.3 Aluminum plate cylinder head

The next retrofit modeled is the aluminum plate cylinder head. Two different examples are included: one with ambient inlet pressure and one with elevated inlet pressure. These experiments refer back to those presented in Chapter 4.

6.2.3.1 Ambient inlet pressure

While the experimental data spans a range of engine speeds, a mid-range speed of 752 rpm was chosen to compare with the model output. The model parameters are given in Table 6.4. Notice the clearance volume, flow factors, and check valve constant are exactly matched to the ones selected in the model calibration in Section 5.5. Notice that here the exhaust pressure tank is switched to the small, 4 liter tank.

Fig. 6.9 shows the experimental results for measured in-cylinder and exhaust hose/tank pressures (psia) over a twenty-five minute run and the model results. The agreement between the experiment and the simulation is very strong.
Table 6.4: Aluminum plate cylinder head with ambient inlet model parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>752</td>
<td>rpm</td>
</tr>
<tr>
<td>Clearance volume</td>
<td>8</td>
<td>cc</td>
</tr>
<tr>
<td>Inlet check valve $a$ factor</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>Exhaust check valve $a$ factor</td>
<td>1.5</td>
<td></td>
</tr>
<tr>
<td>Hose/tank $a$ factor</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Leakage $a$ factor</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>Check valve constant</td>
<td>0.015</td>
<td></td>
</tr>
<tr>
<td>Resolution</td>
<td>0.1</td>
<td>ms</td>
</tr>
<tr>
<td>Exhaust hose volume</td>
<td>20</td>
<td>cc</td>
</tr>
<tr>
<td>Exhaust tank volume</td>
<td>4</td>
<td>L</td>
</tr>
<tr>
<td>Inlet gas pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust tank pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
</tbody>
</table>

Fig. 6.9 shows the experimental and modeled exhaust hose/tank pressures from Fig. 6.9 plotted together. The simulation results perfectly fall within the measured experimental exhaust hose/tank pressure range over the twenty-five minute run.

Included in Fig. 6.11 are the experimental and modeled efficiency curves versus exhaust hose/tank pressures. The curves are in agreement.
Figure 6.10: Aluminum plate cylinder head with ambient inlet: experimental and modeled hose/exhaust tank pressure (psia) vs. elapsed time

Figure 6.11: Aluminum plate cylinder head with ambient inlet: experimental and modeled efficiencies vs. exhaust hose/tank pressure (psia)
As Fig. 6.9 and Fig. 6.10 show, the model is a very close approximation for the experimental results. Fig. 6.12 shows a zoomed view of the experimental cylinder and exhaust hose/tank pressures overlaid with the model predictions. This plot shows that the Python engine model is accurate and reliable in predicting model output on the minute level as well as the (sub) millisecond level. On a single crankshaft revolution, the engine model reliably follows the experimental pressure measurements. The ability for the model to capture the experiments at this fine a resolution whilst remaining accurate over the course of the entire run provides the necessary tool to go beyond the experiments. This shows the model is fully validated, particularly for this specific experimental setup.

Figure 6.12: Aluminum plate cylinder head with ambient inlet: experimental (left) and modeled (right) cylinder pressure and hose/exhaust tank pressure (psia) vs. elapsed time (zoomed view)
6.2.3.2 Elevated inlet pressure

In addition to the experiments with ambient inlet pressure, an example where the inlet pressure is at elevated pressure is also modeled. In this example, the engine is run at a faster speed (1201 rpm), and the clearance volume is decreased. This is physically the same setup as the previous aluminum plate cylinder head run, so the clearance volume does not physically change, but is adjusted for the simulation to achieve the high pressures measured experimentally. The full parameter list is provided in Table 6.5. Notice the tank is again the smaller, 4 liter tank. The starting inlet pressure is set at 60 psia.

Table 6.5: Aluminum plate cylinder head with elevated inlet model parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>1201</td>
<td>rpm</td>
</tr>
<tr>
<td>Clearance volume</td>
<td>5</td>
<td>cc</td>
</tr>
<tr>
<td>Inlet check valve a factor</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td>Exhaust check valve a factor</td>
<td>1.5</td>
<td></td>
</tr>
<tr>
<td>Hose/tank a factor</td>
<td>5</td>
<td></td>
</tr>
<tr>
<td>Leakage a factor</td>
<td>0.3</td>
<td></td>
</tr>
<tr>
<td>Check valve constant</td>
<td>0.015</td>
<td></td>
</tr>
<tr>
<td>Resolution</td>
<td>0.1</td>
<td>ms</td>
</tr>
<tr>
<td>Exhaust hose volume</td>
<td>20</td>
<td>cc</td>
</tr>
<tr>
<td>Exhaust tank volume</td>
<td>4</td>
<td>L</td>
</tr>
<tr>
<td>Inlet gas pressure</td>
<td>60</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust tank pressure</td>
<td>14.7</td>
<td>psia</td>
</tr>
</tbody>
</table>

Fig. 6.13 and Fig. 6.14 show the modeling results. Note that experimentally, the engine began compressing air at ambient pressure when the motor was turned on. About a minute into the run, the inlet valve connection to the inlet pressure tank was opened, which began the flow of inlet pressure at about 60 psia. The kink in the data can be seen in Fig. 6.13 (left plot). This detail was not incorporated in the model. The simulation is set with elevated inlet pressure from the beginning of the run, hence why the entire pressure curve is smoother in the simulation output than in the experimental data.
6.3 Additional model simulations

As this chapter has shown, the engine model dependably predicts experimental data within a close range. The model has been verified across engine retrofits and engine
speeds. The experimental data provides a lot of information about the performance of a retrofitted engine compressor, and the model is now used to go beyond the experimental work. Included in this section are a number of sensitivity analyses that explore the results of reducing cylinder clearance volume, increased engine heat exchange, forcing an isothermal compression, changing engine speed, increasing check valve size (flow rates), reducing check valve delay (backflow), and decreasing engine friction. Pushing the model beyond the limits of the experimental data revealed how sensitive the system is to clearance volume and flow rates. Therefore, focusing on improving these two parameters experimentally is worthwhile to achieve higher system performance on future designs.

In the following simulations, all model adjustments in parameters are made using the aluminum plate cylinder head with ambient inlet pressure as the base case. This simulation aligned best with experimental data, so it is used as the base case for additional systems to measure against. Refer to Table 6.4 for the base case parameters. All models in this section are run for a twenty-five minute experiment at a 0.1 ms resolution.

Fig. 6.15 shows the tank pressure curves as a function of time for engine retrofits explored one at a time. Fig. 6.16 adds three curves to Fig. 6.15 which show the resulting performance when more than one retrofit is incorporated into the same run.

A summary of the efficiencies for each simulation shown in Fig. 6.15 is shown in Fig. 6.17. Fig. 6.18 includes the additional runs that result when multiple retrofits are pursued in parallel.

As Fig. 6.16 shows, performance can be greatly improved by decreasing clearance volume, check valve delays, flow resistance, and leakage. When all these retrofits are combined in one system (the brown curve in Fig. 6.16), the model predicts a peak pressure of over 800 psia in one compression stage. When just clearance volume, flow resistance, and leakage are decreased (pink curve), the model still predicts about 600
psia peak pressure. When flow resistance and leakage only are reduced (gray curve), the performance is similar to what results when only one parameter is adjusted at a time. This shows how important it is to reduce cylinder clearance volume and check valve delays (and therefore backflow) to increase engine compressor performance.

The simulations summarized in Fig. 6.16 with efficiency curves shown in Fig. 6.18 push the boundaries of the experimental work and may be pushing the limits of the engine compressor Python model. Claiming that 800 psia can be achieved in one compression stage beginning with ambient inlet pressure is a bold claim, even with the proposed retrofits. This would be a compression ratio of over 50:1. Therefore, the limits of this engine compressor model must be validated with more experimental data to ensure how to model improved system designs without predicting the impossible.

The curves shown in these four figures are described more following the figures.

![Exhaust tank pressures (psia) vs. elapsed time for multiple sensitivity analyses](chart)

Figure 6.15: Exhaust tank pressures (psia) vs. elapsed time for multiple sensitivity analyses
Figure 6.16: Exhaust tank pressures (psia) vs. elapsed time for multiple sensitivity analyses combining retrofits

Figure 6.17: Cycle efficiency vs. exhaust tank pressure (psia) for multiple sensitivity analyses
6.3.1 Reduced dead volume

In this simulation, all parameters remain the same as described earlier except for the cylinder clearance, volume, which is now reduced from 8 cc to 4 cc. As shown in Fig. 6.15, the final tank pressure is now over 270 psia, twice as high as in the base case. This shows the importance of clearance volume and how large an increase in performance a reduction in clearance volume has. As shown in Fig. 6.17, this simulation did not have the highest peak efficiency, but it resulted in the highest exhaust tank pressure for a single retrofit.

6.3.2 Reduced check valve delay

Reducing check valve delay (increasing the check valve constant) lowers the system backflow because the valves open and close more rapidly. This increases system
efficiency and results in a higher tank pressure as shown in Fig. 6.17 and Fig. 6.15 respectively.

### 6.3.3 Increased flow through check valves

To simulate increased flow through check valves, the system flow resistances (inlet and exhaust check valves and hose/tank flow) are reduced to 5 or 10% of the base case (purple and orange curves, respectively, in Fig. 6.15). Reduced flow resistance increases flow rates. As Fig. 6.15 shows, higher exhaust tank pressures result from decreasing flow resistance, and system efficiencies peak between 35 and 40% rather than 20% in the base case as shown in Fig. 6.17. Interestingly, decreasing the resistances by 5% results in a higher peak efficiency but lower final exhaust tank pressure.

### 6.3.4 Decreased engine leakage

Engine leakage reduces system efficiency because it decreases the amount of gas in the system that can reach the exhaust pressure tank. To simulate decreased engine leakage, the leak flow resistance was increased by twenty fold. The results of this simulation are included in Fig. 6.15 (yellow curve). As this figure (as well as Fig. 6.17) shows, the peak exhaust tank pressure increases as leakage decreases. The efficiency is also higher.

### 6.3.5 Decreased engine friction

Although not included in the figures, another sensitivity analysis focused on decreasing engine friction to 75% its current value (now 33.75 kPa rather than 45 kPa). With decreased engine friction, there is no evidence of being able to achieve an increased tank pressure. Reducing friction does, however, increase system efficiencies.
6.3.6 Increased engine heat exchange

The next sensitivity analysis explored the effect of changing engine heat transfer properties. Here, the heat transfer coefficient is adjusted and a system where the heat transfer between the environment and the cast iron engine block is 100 times larger (i.e., \( h_2 \) is 100 times larger) is modeled. Changing the heat transfer has little effect on the system performance. The efficiency curve for this simulation is not included in Fig. 6.17 because it is virtually the same as in the base case. The result shows that although heat transfer matters conceptually, the numbers are too small to significantly affect the behavior of the system.

6.3.7 Isothermal compression

Another simulation focused on the compression thermodynamics by changing the system from an adiabatic to an isothermal process. This changes the polytropic index, \( n \), from \( \gamma \) (a value of 1.4 for air) to a value of 1. In this simulation, the efficiency actually decreases; the reason for this is, at least in part, due to the fact that friction has not changed. As peak tank pressures decrease, therefore, the friction component of the work requirement increases, which lowers efficiency. This hints at the complex relationship between isothermal and adiabatic compressions. While the compression in both cases is performed reversibly (i.e., at optimal efficiency), the adiabatic compression creates an irreversible cooling step when hot gas from the cylinder mixes with the low temperature gas in the storage tank. This irreversible loss must be compared to losses from friction, which are also irreversible, and are more pronounced under isothermal compression.

6.3.8 Change in engine speed

Engine speed is lowered to 500 rpm and increased to 1,000 and 1,500 rpm as shown in Fig. 6.19.
As these three examples show, increasing engine speed, with all else equal, reduces the final exhaust tank pressure. This is intuitive because backflow is more pronounced at higher speeds (recall a similar discussion on this topic from Chapter 4). The runs with increased speeds correspond to lower efficiency curves. This simulation confirms the same trends shown in the experimental data in Chapter 4.

6.3.9 Change in gas

In addition to air compression, compressing synthesis gas ("syngas") and compressing oxygen were also simulated. Changing the gas requires changing the molecular weight and the heat capacity. Because all three of these gases have similar attributes, the change in performance is very small as shown in Fig. 6.20.

Unlike with air, handling leaks with other gases means the compressor has to be built into an external chamber to avoid contamination and mixing with unlike gases. Depending on the gas, a mixture inside the engine cylinder with air leaking into the
system can be dangerous. Additionally, leakage into the surroundings can be harmful.

### 6.4 Summary

In this chapter the Python engine model was successfully validated by comparing model predictions across various engine retrofits with experimental output. This analysis finds that reducing cylinder clearance volume greatly increases system performance. This work also shows the system’s sensitivity to flow rates. Based on these results, a future design should focus engine retrofits on reducing cylinder clearance volume (as the flat plate cylinder heads tested experimentally do) and on improving the intake and exhaust valves by replacing them with mechanical devices that predict pressure differences and actuate valves more precisely to reduce backflow and flow resistance.

Finally, the bounds of the engine model need to be established. In the sensitivity analyses presented, a single-stage, given a particular combination of engine retrofits, yielded an exhaust tank pressure of 800 psia. This is likely extremely difficult, if not impossible, to achieve in practice. The limits of the model should be further established.
7.1 Introduction

A techno-economic assessment is a tool used to evaluate the economic viability of a project. In this thesis, a high-level techno-economic assessment is used to quantify the value of the small-scale gas compressor built and tested as per the details in Chapter 4. As a basis for grounding the engine compressor’s financial feasibility, state-of-the-art commercially-available compressor costs are provided first. Next, this chapter includes an analysis of the costs that comprise the engine to compressor retrofit. First, the costs of the actual system built are itemized. From this first-of-its-kind system, the costs of future systems are extrapolated. A discussion regarding operational and capital costs is included, as these are different between the industrial compressors and the engine compressor. Finally, a case study is included to give an example of one project where a retrofitted engine compressor can substantially reduce system costs.

7.2 Industrial compressor costs

In Product and Process Design Principles: Synthesis, Analysis, and Evaluation, Sedier et al. include cost functions for centrifugal, reciprocating, and screw compressors as functions of horsepower \[41\]. Each equation is reproduced below and plotted in Fig. 7.1. Seider et al. define the general form for the compressor costs as shows in
\[ C_P = F_D F_M C_B \] (7.1)

where \( C_P \) is the compressor purchasing cost; \( F_D \) is the drive factor (equal to 1.15 for a steam turbine drive; 1.25 for a gas turbine drive); \( F_M \) is the material factor (equal to 2.5 for stainless steel; 5.0 for nickel alloy); and \( C_B \) is the base cost, which includes an electric motor drive. The base cost is a function of consumed horsepower, \( P_C \).

For centrifugal compressors with \( P_C \) ranging from 200 to 30,000 HP, costs are as shown in (7.2).

\[ C_B = \exp(7.5800 + 0.8 \ln(P_C)) = 1958.63 P_C^{0.8} \] (7.2)

For reciprocating compressors with \( P_C \) ranging from 100 to 20,000 HP, costs are as shown in (7.3).

\[ C_B = \exp(7.9661 + 0.8 \ln(P_C)) = 2881.60 P_C^{0.8} \] (7.3)

For screw compressors with \( P_C \) ranging from 10 to 750 HP, costs are as shown in (7.4).

\[ C_B = \exp(8.1238 + 0.7243 \ln(P_C)) = 3373.82 P_C^{0.7243} \] (7.4)

Seider et al.’s costs are based on a 2006 chemical engineering plant cost index (CEPCI) of 500. The annual 2014 CEPCI index was 576.1 [24]. A conversion factor of 1.1522 (576.1/500), therefore, was used to convert these costs to 2014 USD. Fig. 7.1 is a log-log plot showing the cost per horsepower versus horsepower for screw, centrifugal, and reciprocating compressors.
Figure 7.1: Log-log plot of compressor costs per unit of power output ($/HP) by type of compressor vs. horsepower as per the cost equations provided in [41].

7.3 Engine compressor costs

The engine compressor costs comprise the engine and all components needed for retrofitting the engine to a compressor, including but not limited to the electric motor, variable frequency drive, piping, check valves, pressure relief valve, and storage tanks. These components are described in detail in Chapter 4. The engine retrofit with the best compressor performance is the aluminum plate cylinder head. The costs of each component required for building this engine compressor are provided in Table 7.1.

The original cylinder head retrofit and the acrylic plate cylinder head shared many of the components in Table 7.1 and use additional components as listed in Table 7.2.

In addition to the costs of all the components listed in Table 7.1 and Table 7.2, additional sensors were used for testing. The costs of these components are included in Table 7.3.
Table 7.1: Engine compressor component costs for engine with aluminum plate cylinder head retrofit

<table>
<thead>
<tr>
<th>Component</th>
<th>Manufacturing information</th>
<th>Quantity</th>
<th>Unit cost ($)</th>
<th>Total cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>Predator Engines&lt;br&gt;3 HP (79 cc) OHV Horizontal Shaft Gas Engine</td>
<td>1</td>
<td>110</td>
<td>110</td>
</tr>
<tr>
<td>Motor</td>
<td>3 HP General Purpose Motor, 3-Phase, 3500 Nameplate RPM, Voltage 208-230/460</td>
<td>1</td>
<td>402</td>
<td>402</td>
</tr>
<tr>
<td>Variable frequency drive</td>
<td>DURApulse GS3 series&lt;br&gt;AC general purpose drive, 230 VAC, 3 HP with 3-phase and 1-phase input</td>
<td>1</td>
<td>347</td>
<td>347</td>
</tr>
<tr>
<td>Intake pressure tank</td>
<td>Central Pneumatic 5 gallon Portable Air Tank</td>
<td>1</td>
<td>30</td>
<td>30</td>
</tr>
<tr>
<td>Exhaust pressure tank</td>
<td>20 cubic ft argon gas cylinder</td>
<td>1</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>Aluminum block</td>
<td>1” thick, 6” × 6” square plate</td>
<td>1</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Check valves</td>
<td>McMaster-Carr High-Pressure Backflow-Prevention Valve, Corrosion-Resistant, 1/8 NPT Female Part #7838K53</td>
<td>2</td>
<td>67</td>
<td>134</td>
</tr>
<tr>
<td>Intake and exhaust lines</td>
<td>Summit Racing Equipment&lt;br&gt;Earl’s Performance Speed-Flex Brake Lines 64151518ERL</td>
<td>2</td>
<td>16</td>
<td>32</td>
</tr>
<tr>
<td>Unistrut (+ bolts)</td>
<td>Measured in 1’ increments</td>
<td>12</td>
<td>4</td>
<td>48</td>
</tr>
<tr>
<td>Shaft coupler</td>
<td>Lovejoy shaft coupler with flexible spider insert</td>
<td>1</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Polycarbonate shield</td>
<td>1/2” thick</td>
<td>1</td>
<td>80</td>
<td>80</td>
</tr>
<tr>
<td>Miscellaneous additional connections</td>
<td></td>
<td></td>
<td></td>
<td>200</td>
</tr>
</tbody>
</table>

The total system cost for each retrofit is on the order of $1,600. (Note that the cumulative costs for all retrofits is not three times this figure. Many components (the engine, motor, VFD, valves, etc.) are used across all three retrofits.) In addition to the extra piping required in the case of the original cylinder head, this system also used a pressure relief valve that was removed in later retrofits, but that is a costly component, which pulls this cost above the others. The cost of the additional components for testing, as itemized in Table 7.3, is on the order of $1,200. Therefore, total costs incurred for the retrofits and testing were under $4,000. Without the
Table 7.2: Engine compressor additional component costs for engine with acrylic plate cylinder head and original cylinder head retrofits

<table>
<thead>
<tr>
<th>Component</th>
<th>Manufacturing information</th>
<th>Quantity</th>
<th>Unit cost ($)</th>
<th>Total cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acrylic block</td>
<td>1/2” thick</td>
<td>1</td>
<td>20</td>
<td>20</td>
</tr>
<tr>
<td>Valve connections: Cross</td>
<td>McMaster-Carr Cross Connectors Part #45525K551</td>
<td>1</td>
<td>23</td>
<td>23</td>
</tr>
<tr>
<td>Valve connections: Pipe nipple</td>
<td>McMaster-Carr Stainless Steel Pipe Nipples Part #4830K111</td>
<td>10</td>
<td>2</td>
<td>20</td>
</tr>
<tr>
<td>Spark plug adapter</td>
<td>Summit Racing Equipment</td>
<td>1</td>
<td>10</td>
<td>10</td>
</tr>
<tr>
<td>Pressure/temperature sensor connections: Tee</td>
<td>McMaster-Carr Tee Connectors Part #45525K542</td>
<td>1</td>
<td>19</td>
<td>19</td>
</tr>
<tr>
<td>Exhaust tee</td>
<td>McMaster-Carr Tee Connectors Part #45525K541</td>
<td>1</td>
<td>19</td>
<td>19</td>
</tr>
<tr>
<td>Pressure/temperature sensor connections: Pipe adapter</td>
<td>McMaster-Carr Male × male adapter Part #51205K191</td>
<td>1</td>
<td>11</td>
<td>11</td>
</tr>
<tr>
<td>Pressure relief valve</td>
<td>Swagelok Stainless Steel High Pressure Proportional Relief Valve, 1/2 in. MNPT x 1/2 in. FNPT</td>
<td>1</td>
<td>270</td>
<td>270</td>
</tr>
<tr>
<td>Adapter to Pressure relief valve</td>
<td>McMaster-Carr Part #51205K315</td>
<td>1</td>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>Miscellaneous additional connections</td>
<td></td>
<td></td>
<td></td>
<td>200</td>
</tr>
</tbody>
</table>

Table 7.3: Engine compressor instrumentation component costs

<table>
<thead>
<tr>
<th>Component</th>
<th>Manufacturing information</th>
<th>Quantity</th>
<th>Unit cost ($)</th>
<th>Total cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure transducer</td>
<td>Omega PX409-750A5V 0 to 5 Vdc output Piezoresistive design with high temperature performance</td>
<td>2</td>
<td>555</td>
<td>1,110</td>
</tr>
<tr>
<td>Thermocouple (hard)</td>
<td>Type K</td>
<td>1</td>
<td>24</td>
<td>24</td>
</tr>
<tr>
<td>Thermocouple (soft)</td>
<td>Type K</td>
<td>1</td>
<td>22</td>
<td>22</td>
</tr>
</tbody>
</table>

182
pressure transducers, the current engine system per HP of nominal engine power is about $550. A very quick way to decrease the system costs is to replace the electric motor and the variable frequency drive with another unmodified, power-producing engine. Changing the system’s power source replaces $750 worth of infrastructure with a $110 engine. Now the system costs under $1,000, or $340/HP of nominal engine power. This figure will fall even lower as more systems are built and retrofits are streamlined. In terms of cost per compressor power output, this will vary based on the system’s compression ratio. If the effective compressor power is just half of the nominal engine power, the cost per unit power output will be double. As is shown in the example later in this chapter, however, effective compressor power can be on par with nominal engine power, so using this figure is justified here.

7.4 Engine compressor projected cost

Recall the scaling background provided in Chapter 2. Scaling in numbers shows that as the number of units produced increases, costs decrease as per the progress ratio. Because the engine is not likely to come further down its learning curve, for this analysis it is set as $7.5/HP for future systems. The rest of the engine compressor assembly, which costs about $250/HP, is subjected to learning. If a conservative progress ratio of 85% (i.e., a learning rate of 15%) is used, the initial system cost (excluding the engine used as the compressor and the engine used to drive the engine compressor) of about $250/HP will yield a cost of $12/HP at a cumulative production of about half a million (or $2^{19}$) units as per (7.5). This figure is two orders of magnitude lower than state-of-the-art compression technologies. Recall that the raw engine cost is on the order of $10/kW [29], or about $7.5/HP (this in terms of nominal engine output). The engine compressor system here, therefore, is projected to drop to about $20 - $30/HP. Now the engine compressor is the same order of magnitude as the original engine. An additional cost not included here is
the assembly cost. This cost will need to be addressed in future work.

\[ k_n = \$250/\text{HP}(0.85)^{19} \]  

(7.5)

### 7.5 Compressor capital and operational costs

In this chapter, capital costs of the engine compressor have been evaluated. The lifetime of the engine compressor is likely shorter than the lifetime of an industrial compressor, which requires that a cost comparison take this into account. In normal use, an internal combustion engine may last up to 10,000 hours \[28\]. A retrofitted engine performing as a compressor has no combustion, so the engine lifetime may be double that figure. The engine running as a compressor may last three years (this estimate should be verified with further experimental and modeling work in future research). An industrial compressor typically lasts ten years \[36\]. Conservatively, the engine compressor may need to be replaced four times to match the lifetime of the commercial compressor. Based on the costs reported in this analysis, increasing the engine compressor costs by four times has little affect on the overall capital costs competitiveness of this system compared with current state-of-the-art compressor costs.

So far this chapter has focused on capital costs. Operational costs also need to be evaluated. Based on the experimental work presented in Chapter \[4\], the efficiency of the engine compressor is, at a maximum (as of right now), about 40%. Industrial compressors typically have efficiencies above 80%. This difference means that the engine compressor consumes significantly more power per unit of compressed output. At the current, laboratory scale, this is not as large a difference in magnitude in power input, but for big (thousands of horsepower) commercial compressors that currently consumer MWs of power input, having an engine compressor with less than half the efficiency really dramatically increases the required power input.
Another point about capital versus operational costs for the engine compressor is that, due to the very low capital costs combined with the need to more frequently replace engines, the engine compressor replacement costs can actually be absorbed into the operational costs. An example of how this plays out is included in Chapter 3 and Appendix A with the engine gensets committing to power in the energy and reserve markets. Here, the first unit is a capital cost and all replacement units are absorbed into operational costs. Depending on the engine compressor system lifetime, this could become a hybrid case where the infrastructure can be viewed as either a capital or an operational cost.

7.6 Case study
An Advanced Research Projects Agency-Energy (ARPA-E)-funded project (DE-AR-0000506) that concluded in December 2016 provides a case study for an industrial application of gas compression technology. The project, “Compact, Inexpensive Micro-Reformers for Distributed GTL,” was a joint effort between Research Triangle Institute (RTI) International, Columbia University, and MIT. The project summary as per ARPA-E [1] is as follows:

RTI is leveraging existing engine technology to develop a compact reformer for natural gas conversion. Reformers produce synthesis gas—the first step in the commercial process of converting natural gas to liquid fuels. As a major component of any gas-to-liquid plant, the reformer represents a substantial cost. RTI’s re-designed reformer would be compact, inexpensive, and easily integrated with small-scale chemical reactors. RTI’s technology allows for significant cost savings by harnessing equipment that is already manufactured and readily available. Unlike other systems that are too large to be deployed remotely, RTI’s reformer could be used for small, remote sources of gas. [1]

In this ARPA-E project, compressing synthesis gas (“syngas”) from 4.65 to 45 bar is a required task. The system’s single compressor, however, accounts for 50% of the entire small-scale distributed gas-to-methanol system equipment costs. Their current
pilot system has a compressor cost of $287k for a 75 HP reciprocating compressor, which translates to $3,831/HP. The commercial system, which still uses small compressors compared to industrial practice, requires 525 HP, which, based on the scaling relationship in Fig. 7.1, would result in a compressor cost of $1.24M, or $2,373/HP. To combat this large cost per horsepower, scaling up to larger systems with much higher power is currently the only option. Using the fundamentals this thesis provides, this case study proposes a new approach to this compressor need that targets the small scale. A cost efficient, stationary, small-scale compressor could decentralize many chemical processes, and in this ARPA-E project, significantly reduce system costs.

Syngas is a mixture of methane (CH$_4$), water (H$_2$O), nitrogen (N$_2$), oxygen (O$_2$), hydrogen (H$_2$), carbon monoxide (CO), and carbon dioxide (CO$_2$). The proposed, commercial-scale compressor for the distributed modular systems requires a syngas flow rate of 2533 kg/hr. At a molecular weight of 21.6 g/mol, the required syngas throughput is 32.5 mol/s. The syngas incoming pressure and temperature are 4.65 bar and 129°C, respectively. The required outlet pressure is 45 bar [9].

The engine model described in Chapter 5 was used for estimating the performance of a small engine compressing air. This model is adapted to estimate the performance of an engine specifically redesigned for syngas compression. Instead of the properties for air, the model now takes in syngas properties, including molecular weight (21.6 g/mol) and density (0.95 kg/m$^3$). The inlet conditions are now syngas at 4.65 bar and 129°C. Instead of the one-cylinder, 79 cc engine used in the lab experiments described in Chapter 4, a larger engine, in particular an aluminum 5 L V8 Ford F-150 with peak power of 385 HP at 5750 rpm, was selected. The F-150 has eight cylinders each with bore $\times$ stroke of 3.63 in. $\times$ 3.65 in. and a displacement of 302 cubic inches, or 618 cc. The compression ratio is 10.5. Using this engine in the engine compressor model requires updating these geometric parameters as well as the flow rates. The flow rate
equations in the original engine model described in Chapter 5 were fitted using a 79 cc engine and 1/8” NPT check valves. Because the cylinders are now almost eight times as large in volume with a longer bore, the intake and exhaust valves can be larger in this design, which enables larger flow rates in and out of the engine cylinders.

As before, the engine model performs the calculations on one engine cylinder. The flow out of the engine is an average of the flow from all eight cylinders. The model runs at 0.001 ms time steps, and the flows are calculated at that resolution. Compared with the examples shown in Chapter 6, the temporal resolution in this simulation is much finer. The reason for this is because this simulation is at a much higher speed with higher pressures. Before, a cycle may have taken 80 ms (when running at a speed of 752 rpm); here a cycle is just 25 ms, which means a finer resolution is needed to capture enough points in each cycle to understand the system behavior.

For this case study, a flow of 32.5 mol/s of syngas at a pressure of 45 bar (650 psi) is needed as the exhaust. This simulation finds that one 5 L, 8 cylinder F-150 engine can accommodate these compression and flow requirements. At a speed of 2400 rpm, the flow of exhaust syngas at 650 psi is almost 35 mol/s. The model is structured by assuming a back pressure on the exhaust hose (a buffer tank) that is modeled as an exhaust pressure tank that remains at a constant pressure of 650 psi. The model parameters are given in Table 7.4 and a summary of the system performance is shown in Fig. 7.2.

The engine has an efficiency of about 56% in this setup. The work requirement, therefore, is about 330 kW, or about 440 HP. A second, non-retrofitted, power-producing engine, therefore, is needed to drive the compressor engine.
Table 7.4: Case study syngas compression model parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine speed</td>
<td>2400</td>
<td>rpm</td>
</tr>
<tr>
<td>Clearance volume</td>
<td>10</td>
<td>cc</td>
</tr>
<tr>
<td>Inlet check valve $a$ factor</td>
<td>$5 \times 10^{-5}$</td>
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</tr>
<tr>
<td>Exhaust check valve $a$ factor</td>
<td>$1.5 \times 10^{-4}$</td>
<td></td>
</tr>
<tr>
<td>Hose/tank $a$ factor</td>
<td>$2.5 \times 10^{-5}$</td>
<td></td>
</tr>
<tr>
<td>Leakage $a$ factor</td>
<td>30</td>
<td></td>
</tr>
<tr>
<td>Check valve constant</td>
<td>150</td>
<td></td>
</tr>
<tr>
<td>Resolution</td>
<td>0.001</td>
<td>ms</td>
</tr>
<tr>
<td>Exhaust hose volume</td>
<td>20</td>
<td>cc</td>
</tr>
<tr>
<td>Inlet gas pressure</td>
<td>67.5</td>
<td>psia</td>
</tr>
<tr>
<td>Starting exhaust buffer tank pressure</td>
<td>650</td>
<td>psia</td>
</tr>
</tbody>
</table>

Figure 7.2: Case study engine compressor simulation showing cylinder, exhaust hose, and exhaust tank pressures (psia) vs. time (in seconds)
Each F-150 engine costs $5,111, which was determined by starting with a Ford F-150 truck price of $27,625. Manufacturing costs account for 50% of MSRP (manufacturer’s suggested retail price) and the engine unit accounts for 18.5% of the manufacturing cost. Therefore, $27,625(50%)(18.5%) = $2,555.31. Because the assumption is that the engine is purchased at MSRP, this figure should be doubled, so the engine cost is $5,111.

A total of two engines results in a cost of $10,222. The cost per unit of power out of the system, therefore, is $10,222/250 HP, or $40/HP. This cost estimate does not include any costs for actually retrofitting the engine. If a similar retrofit to the aluminum head described in Chapter 4 was used here to reduce clearance volume and facilitate greater heat exchange, an additional $40/HP may be a reasonable estimate for the retrofits and assembly (when many are produced; a first of its kind, as shown in the previous section, is an order of magnitude higher cost per unit of output). The estimate of $80/HP for the engine compressor is drastically lower than industrial compressor options. Keep in mind that this figure is not a cost estimate for the first unit built. The first unit built at the commercial scale will likely require more costs than have been included here, but as many systems are produced, the costs decrease. The cost estimate of $80/HP is an expectation for engine compressor costs when many are built.

Going back to the initial industrial cost curves in Fig. 7.1 and now adding this new data point along with the pilot system and commercial system compressor costs, as shown in Fig. 7.3, there is an order of magnitude difference in costs per HP. A $1.24 million compressor is replaced with a $25,000 system. The engine compressor is a fraction of the cost of the commercially-available compressor. The retrofitted engine system would have to cost fifty times more than is estimated here for it to be more expensive than what is available in industry. This is a large buffer to stay within in order to remain competitive. However, the engine compressor will use more
power.

Figure 7.3: Log-log plot of compressor costs per unit of power output ($/HP) by type of compressor vs. horsepower with engine compressor cost shown.

In the cost comparison above, the difference in system lifetimes have been neglected between the industrial compressor and the engine compressor. If the ratio of the lifetime of the industrial compressor to the lifetime of the engine compressor is on the order of fifty, the cost competitiveness goes away because this would require fifty engine systems to be purchased throughout the lifetime of the standard compressor. The difference in lifetimes, however, is likely closer to five years. As per the conclusions found in Chapter 3 and Appendix A, short lifetime is not, by nature, prohibitive, and the cost of replacement engine compressors would be discounted. The need to replace the infrastructure can be an asset; this provides a built-in flexibility to upgrade. The lifetime and engine wear needs to be evaluated for this application.
7.7 Conclusions

Based on this techno-economic assessment, the system costs for a first-of-its-kind retrofitted engine compressor are extremely attractive and competitive and warrant further attention. This high-level techno-economic assessment sets the scale and potential of this technology by looking at a particular application for small-scale compressors. This system, however, is not limited to compressing for syngas production (nor limited to air compression as the experiments in Chapter 4 focused on). The goal of this work is to lay the foundation for further research and later deployment of small-scale engine compressors, with the aim of meeting small-scale, stationary compression needs across various industries.

Markets for the engine compressor include air and carbon dioxide (CO$_2$) liquefaction as well as hydrogen and synthesis gas compression. The market potential needs to be explored in more detail, and the costs of a pilot and commercial scale system need to be quantified by building out this system at these larger scales. A more robust analysis of the full balance of system must be quantified, as the engine compressor will require additional infrastructure for packaging the system in such a way that it can be easily sold, transported, and installed.
8.1 Conclusions

In this thesis, two specific applications are explored in detail and highlight the potential for incorporating internal combustion engine technology into energy and chemical engineering industries. This work is motivated by the low cost of engine infrastructure, which has resulted, to a high degree, from automotive production. Global annual production of cars and commercial vehicles is on the order of 100 million. The automobile mass-manufacturing infrastructure has enabled not only very high production numbers of engines, but correspondingly low prices. This research specifically aims to take advantage of these low costs. Although engines are not originally designed for the tasks proposed in this thesis, which focus on power generation and compression, the low cost of mass-manufactured infrastructure, in particular internal combustion engines, provides an incentive for retrofitting these units. While this thesis has focused on exploiting internal combustion engine technology, this work at large is not limited to this unit. Mass-manufacturing brings prices down, and small-scale, modular systems can be designed by building off of other mass-produced systems, beyond or in addition to engines.

The claims regarding the potential for small-scale, mass-manufactured modular designs are justified by first exploring scaling laws and the path dependence of total costs. This thesis explores the potential for engines to be used in power generation and chemical engineering industries. This work includes optimizations, modeling and simulations in Python, laboratory experimentation, and a techno-economic assess-
ment with a case study to quantify the value of engine technology in non-automotive applications. The main contributions of this thesis are included below:

- Prior to this work, scaling in individual unit size and scaling in number of units produced have been analyzed in isolation. This thesis builds the mathematical framework for understanding how the two scaling forces interact in conjunction with each other. The result is that the path to a cumulative capacity, as a function of unit size and number of units produced, affects total costs. There is path dependency in scaling, and there are benefits to focusing on small units at the beginning to take advantage of learning. For all practical applications, however, the underlying scaling laws seem unaffected, and the path dependency observed is fairly weak. In practice, the path dependency may be overwhelmed by details not incorporated into the simple model presented here. This analysis, therefore, does not detract from the conclusion that small modular systems provide a novel and interesting approach to scaling up in numbers rather than in individual unit size.

- Small-scale, modular designs enable increased flexibility, risk aversion, and distributed operation in energy and chemical engineering industries, fields often dominated by centralized, long-lived systems. This research highlights the opportunity for disruptive change in replacing archaic systems with infrastructure based on mass-manufactured, low-cost internal combustion engines. Energy systems today must solve constantly evolving optimizations driven by everything from advances in power grids to tightening environmental constraints. This thesis finds a competitive advantage in being small, modular, and low-cost, all characteristics that promote flexibility, especially compared with the incumbent technology. In particular, the focus here has been on engines.

- Power plants cost about $1,000/kW, whereas internal combustion engines are
about $10/kW \textsuperscript{[29]}. This cost disparity motivates the interest in exploiting the inexpensive engines for non-automotive applications. This thesis shows how designs based on the indivisible unit of an internal combustion engine provide a route for promoting small-scale, low-cost infrastructure.

- Specifically applied to energy generation, an optimization was designed and written in Python to maximize profits by determining the optimal power commitments for engine and generator units (engine gensets) simultaneously participating in the day-ahead energy and ancillary services (spinning reserve) markets. The optimization finds expected annual profits for the engine genset as a price-taker on the order of $30/kW. Typically, over 80% of the time, the engine commits to the reserve market, where it is not always called on to deliver power. The majority of profits come from the reserve market. The study includes sensitivity analyses, which highlight the increased profit in scenarios where prices for electricity in reserve markets are more expensive. With the increased penetration of renewables on the grid, it is probable that reserve prices will increase, making the engine genset system even more financially attractive.

- Gas compression is often cited as a critical component in many chemical engineering and refining industries that resists cost-efficient downscaling. This thesis proposes using retrofitted internal combustion piston engines as small-scale gas compressors to bypass the financial barrier to achieving small-scale compressors. Based on laboratory experiments, a model written in Python, and a techno-economic assessment, this research shows how small-scale compression technology can be revolutionized with retrofitted internal combustion engines. Modeling and laboratory experiments show the potential for single- and multi-stage compression, with exhaust gas pressures surpassing 650 psia. Efficiencies vary as a function of exhaust tank pressure, engine speed, and
retrofitting techniques. Although the engine compressor system efficiency is lower than commercially-available compressors, this system costs an order of magnitude lower on a per horsepower basis than the state-of-the-art industrial compressors. The inherently low system costs based on the foundation of an internal combustion engine provide flexibility in retrofitting to meet the new system needs. Experimentally, three main retrofits were tested, with an optimal design based on a custom-designed and fabricated aluminum cylinder head with reduced clearance volume and increased heat exchange.

- A one-dimensional model, with high temporal resolution for an engine used as a compressor was written in Python. The model was calibrated and validated by experimental data. Model simulation output matches experimental data so well that the model is used to explore experimental conditions beyond those tested in the laboratory. These simulations highlight the importance of reducing flow resistance, check valve delays, and cylinder clearance volume to increase system performance.

- The experiments and engine model show that retrofitting an internal combustion engine to a gas compressor is limited by the valves and hose connections, not by the engine and cylinders. Getting gas in/out of the system is the limiting factor.

- Finally, this thesis includes a case study focused on a tangential topic: onsite, small-scale energy storage optimized for peak demand shaving. This study finds that, for all energy storage technologies considered, a positive net present value is realized over each system’s lifetime. The storage technologies considered can be expanded to include an engine genset, which will change the optimization’s boundary conditions, and may be worth exploring further.

This thesis has provided a robust analysis of small-scale, modular infrastructure
built around an internal combustion engine. This research has explored using an engine genset to commit to power generation in energy and reserve markets as well as a retrofitted engine to perform as a small-scale gas compressor. The analyses in this thesis include a mathematical foundation on scaling, models, simulations, optimizations, laboratory experiments, and a techno-economic assessment. This work builds on the foundation others have established in terms of both understanding laws of scaling as well as giving examples of systems that benefit from small-scale technology. The hope is that future work continues to build on these findings and that a revolution in energy, chemical engineering, and refining industries based on modularity and mass-manufacturing gains attention. Pushing the boundaries on large versus small infrastructure can bring about unique and revolutionary system designs.

8.2 Future research

A recent article, “The death of the internal combustion engine,” published in The Economist [46], points out, as the article title suggests, that the internal combustion engine is being pushed out of the automotive industry and replaced with electric vehicles. This statement is difficult to believe. Yes, batteries and electric vehicles are coming up alongside conventional liquid fuels and engines, but this does not mean the engine is dead. The work in this thesis shows applications for engines beyond automotive applications. Additionally, engines can still be used in the future with methanol-based synthetic liquid fuels and with energy storage. The case study presented in Chapter 7 looked specifically at a project using internal combustion engines as small-scale gas reformers. As part of this thesis, a small-scale gas compressor was built by retrofitting an internal combustion engine. This research supports the view that engines are still in the game, both in the automotive industry and in unconventional applications in energy, chemical engineering, and refining industries.
If, however, this view is incorrect and the internal combustion is soon to be eliminated in new automotive production, it is encouraging for the applications proposed in this thesis, that the infrastructure for engine manufacturing is so broad and so plentiful that these industries will resist closing their doors and shutting off the lights. This research offers unconventional applications for engine technology, and this may be a very attractive proposition to the many engine manufacturing plants. This work provides value for engines at a time when the mainstream thinks they are dying.

As a follow-up to this thesis, a few areas are highlighted for future work:

- Exploring energy markets in finer detail to determine energy generation scales needed to enter the market and specifics of committing power simultaneously in energy and reserve markets in real-time and day ahead markets. The results in the study here suggest profitable returns, and this should be explored further in light of how execution and market entry would occur in practice.

- With regards to the engine compressor modeled, designed, and built as part of this work, the potential of this technology is highlighted as well as the need to develop it further on larger engines. Engine compressor retrofits can be optimized beyond the scope of this project, with a focus on decreasing cylinder clearance volume, increasing flow rates, and decreasing check valve delays. The economics presented here are so attractive, that a commercially-viable system should be explored in more detail. The first step on this path is to drive efficiency higher than it is currently.

- The theme of this thesis has been in promoting small-scale, modular, mass-manufactured designs, particularly with a focus on internal combustion engine technology. The possibilities in this field are plentiful. By taking advantage of automation and mass-manufacturing, many modular systems may be well-suited to compete with traditionally large infrastructure. An example of another
system that may be well suited to be built from internal combustion engines is a Stirling engine.

This research builds on the foundation others have established in continuing to resist the conventional notion that “bigger is better,” particularly with regards to energy infrastructure. The hope is that this work is used to encourage further development of the themes, examples, modeling, experimentation, and case studies included in this work. The landscape in today’s energy infrastructure must be agile and evolve quickly in order to solve constantly changing optimizations driven by many forces including advances in smart power grids and environmental constraints driven by pollution and climate change. This research shows how systems based on principles of small-scale, modular, mass-manufactured designs are the most likely to succeed in this landscape.
Bibliography


[4] Air Compressor Market Analysis By Product (Stationary, Portable), By Technology (Reciprocating, Rotary, Centrifugal), By Lubrication (Oil Filled, Oil Free), By Application (Semiconductors and Electronics, Food and Beverage, Healthcare, Home Appliances, Energy, Oil and Gas, Manufacturing) And Segment Forecasts To 2022. Web Page. 2016.


The value of small-scale internal combustion engine gensets in energy and reserve markets

This chapter is a draft of a paper currently under review for publication. Authors are Zara E. L’Heureux and Klaus S. Lackner.

Abstract

When normalized by power output, internal combustion engines are one hundred times less expensive than conventional, large power plants. Engines are mass-manufactured, low-cost, short-lived power generators that can rapidly change output from zero to full power. In this analysis we treat them as consumables, as an operational rather than capital cost. This paper briefly reviews the economics of scaling, modularity, and centralization vs. distribution in the context of power generation and then presents a model that optimizes how an internal combustion engine and electric generator (“engine genset”) could participate simultaneously in the day-ahead energy and ancillary services (spinning reserve) markets. The paper quantifies the value of small-scale, modular, mass-manufactured components in energy markets, with an example focused on internal combustion engine gensets for electricity generation. The optimization shows that net annual profits from committing power output from an engine genset in the day-ahead energy and spinning reserve markets are positive in all cases analyzed.
Nomenclature

\[ \dot{W} \quad \text{Wear rate} \]

\[ S \quad \text{Engine speed} \]

\[ \text{AF} \quad \text{Engine air-to-fuel ratio} \]

\[ \gamma_f \quad \text{Engine hourly fuel cost as a function of power output} \]

\[ \mu \quad \text{Engine brake specific fuel consumption} \]

\[ P_{\text{out}} \quad \text{Engine power output} \]

\[ C_{\text{opt}} \quad \text{Optimized engine hourly total operating cost} \]

\[ \rho_{\text{CH}_4} \quad \text{Density of natural gas} \]

\[ E_C \quad \text{Engine capital cost} \]

\[ \pi \quad \text{Engine profit function} \]

\[ P_{\text{peak}} \quad \text{Peak engine power} \]

\[ P_1 \quad \text{Energy committed to day-ahead energy market} \]

\[ P_2 \quad \text{Energy committed to the total of day-ahead energy and ancillary services markets} \]

\[ \theta_e \quad \text{Day-ahead energy market price} \]

\[ \theta_f \quad \text{Fuel cost} \]

\[ \theta_s \quad \text{Day-ahead spinning reserve price to commit power} \]

\[ \theta_{s,d} \quad \text{Day-ahead spinning reserve price to deliver power} \]

\[ \eta \quad \text{Mechanical to electrical power efficiency} \]

A.1 Introduction

Over the last century, conventional energy generation systems have largely focused on increasing individual unit sizes and centralizing production. One challenge with this “bigger is better” concept is its ability to evolve. Large systems are often designed with long lifetimes, innovate slowly, and necessitate high upfront costs. Breaking away from this cycle is essential for promoting change, especially change happening quickly in the energy and chemical engineering industries. The ability to accommodate rapid change provides a competitive advantage in a field dominated by large and often very old equipment that cannot respond to technology change shaping the wider energy infrastructure from advances in power grids to environmental constraints driven by pollution and climate change.
A.1.1 The case for small

Systems engineering across many sectors has a common goal: scaling up to large unit size and centralized production. The idea that bigger is better is often associated with capitalizing on economies of unit scale, or the “benefits that are directly attributable to building and operating larger individual units of technology” (as opposed to economies of scale, which typically encompasses all “benefits associated with increasing total firm-wide output”) [1]. Increasing the size of individual production units is one, but not the only, way of achieving economies of scale (increasing firm-wide output); building small, massively parallel production systems (economies of numbers) can achieve the same [1]. In fact, based on empirical rules for estimating fixed costs, scaling up an individual unit size is mathematically as effective as scaling a small unit size in numbers to meet the capacity of the larger, single unit.

Modular, small-scale units enable decentralization and diversification. They generate location, investment, and operating flexibility [1]. In theory, small-unit-scale systems can not only compete with large systems, but also offer benefits beyond those offered by monolithic units. Particularly in the energy and chemical engineering industries, small-scale systems enable autonomy, provide a buffer against risks, and, in some cases, eliminate the need for transportation and long-range distribution. Modular and small-scale units allow for distributed operation while retaining the option for centralization. They provide increased flexibility to engage markets due to shorter lead times. Small and short-lived units allow firms to more easily disengage from unprofitable ventures without stranding a significant investment meant to last for decades. Finally, small unit size provides redundancy whereby failure of a single component translates to partial (as opposed to total) outage. Small systems also face their own challenges. Difficulties with small-scale units include lower efficiency due to friction and heat retention, complications due to contamination from wall materials and corrosion, and costly control systems. However, these challenges are
case-dependent. Some systems scale down more easily than others.

One design challenge with small systems is finding appropriate ways to minimize both economic and physical losses whilst maintaining high system efficiencies. Low capital costs can in some cases compensate for low efficiency. The ultimate aim for small-scale systems is to achieve low costs while maintaining high efficiency. Unfortunately, these optimizations are very dependent on the details of each particular implementation. Deciding when to invest in large vs. small-scale units requires analysis on a case-by-case basis.

A.1.2 United States electricity generation

The United States electricity generating sector provides an example of why historically, bigger systems were favored. U.S. electricity generation data from the 1970s to 2010s show average capacity size increased in nuclear, coal, combined cycle, and gas turbines. In coal-fired plants alone, the average generator size increased more than 10 fold from 1950 to 1980 (from 50 to 600 MW) [1]. Dahlgren et al. found that when labor costs are removed, however, none of the technologies show any cost reductions as a function of increasing unit size [1]. Reducing labor costs has been the driving force over the last 70+ years in building bigger systems. If every generator needs a worker overseeing its performance, fewer, larger units lower the required personnel. The human labor constraint of the past, however, is not as applicable today because of greatly improved computing resources. With automation and controls, the labor constraint largely falls apart.

The question we pose in this paper is: if operational costs are independent of unit size, can we decrease capital costs by building small units? We think yes. Even though scaling in size and scaling in numbers results in similar costs to achieve the same total production capacity, we find examples where small units are still a fraction of the cost of the larger units. Internal combustion engines, for example, are 100 times
less expensive per kW than a coal-fired power plant [2]. We have investigated two possible reasons why car engines are so inexpensive compared to power plants and why, in general, small systems can end up orders of magnitude less expensive than their large counterparts. The first reason is internal combustion engines and smaller equipment in general, do not need to be built for long life times. As a result, they can be built with lower durability, which saves capital costs. Short lifetimes in turn lead to more units being manufactured, which in turn pushes the technology further down its learning curve, which results in more declines in cost. The second reason is, as Dahlgren points out, that smaller machinery tends to follow steeper learning curves, which again points in the direction of lower costs for smaller scales [3]. These pieces together motivate this analysis to use mass production as a novel avenue for cost reductions. In the end we expect to not just match the cost reductions of the economies of scales, but by taking advantage of shorter life times, we expect to beat them.

Mass production means making many individual units, and as cumulative output increases, the process optimization leads to cost reductions (commonly referred to as “learning-by-doing” or the “learning curve”) [4]. There are many examples across numerous industries over the last hundred years that show that costs tend to decrease by a fixed percentage with each doubling in cumulative output. As an example, analysis of production numbers and costs of the Model T found that with every doubling in production volume, costs declined by 15% [5]. (For a longer list of learning rates, see [3].) Mass production leads to cost declines because producing large volumes uncovers faster and less expensive methods for producing output.

New small-scale systems have the challenge of competing with incumbent, often large-scale systems. Beginning at the top of a new learning curve with a first-of-its-kind unit requires much time and iteration before costs are competitive with current technology. To overcome this barrier, we focus on importing developed technologies
from other, much larger markets that have already come down their learning curves. If such modules can be used without large modifications, one can take advantage of their already low cost in a new market. Because these modules were designed for different tasks without this new market in mind, we accept lower efficiencies. By incorporating existing learning from developed technologies into the design of new small-scale systems, we can accelerate the move towards novel, competitive systems. The challenge is to avoid losing much of the learning during the system redesign.

**A.1.3 Internal combustion engine gensets in the energy and reserve markets**

As electric-drive vehicles become more prevalent, researchers investigate opportunities for incorporating vehicle batteries or fuel-cells into electricity markets for use as power or storage; see [6] and [7]. These authors focus on vehicles because they are accessible electricity hubs, but even more so because “the relatively lower capital costs of vehicle power systems and the low incremental costs to adapt EDVs [electric-drive vehicles] to produce grid power suggest economic competitiveness with centralized power generation” [6]

In this paper, we look at a related but different application. We are interested in engines as power producers, apart from their conventional motive-power application (i.e., the engine without the vehicle). We are interested in engines because they are inexpensive, and they can rapidly vary the amount of power delivered. Internal combustion engines (ICEs) (found in vehicles, motorcycles, lawnmowers, etc.) are inexpensive because they are mass-manufactured for applications designed to use fuel as an input and use its chemical energy to generate mechanical power. As a whole, on a per kW basis, vehicles are over fifteen times cheaper than conventional power plant generators [7]. Engine costs account for 18.5% of vehicle manufacturing costs [8], which makes the engine 100 times cheaper than the power plant as a power
producer. The cost comes to $10/kW (for engines) vs. $1,000/kW (for power plants) [2].

This work is, in part, motivated by an earlier study by Malco Parola who, for a masters thesis with one of the authors, developed a model of power plants built from small automobile engines [9]. His results suggested that low cost engines viewed as consumables offered a promising economic framework. This paper includes a more detailed engine model that accounts for performance change with operational conditions and optimization under different pricing structures. It offers a differentiated product for different markets. ICEs are designed to quickly ramp up to full power. The fuel consumption and engine wear, however, are nonlinear with power; both have minima that lie below full engine power output. At peak power output, fuel consumption will be costlier and engine wear will be higher, which will decrease the engine lifetime.

We are interested in electricity markets and how small systems can compete with conventional, centralized, large power plants. This paper evaluates how an engine paired with an electric generator (“engine genset”) should be operated to deliver electricity in the day-ahead energy market and in the day-ahead ancillary services (spinning reserve) market. Here, an engine takes natural gas as fuel and produces mechanical power; the generator takes the engines mechanical work and generates electricity.

Because spinning reserves require a fast response, we hypothesize that engines will be well-suited for delivering in this market. Furthermore, due to the pricing structure, spinning reserve markets pay for standby capacity, and a call on that capacity occurs only a small percentage of the time. We hypothesize that the profit gained by offering a quick ramp-up from optimal running conditions to full throttle and driving the engine well beyond optimal conditions (defined by fuel consumption and engine wear) when occasionally called upon, outweighs the costs associated with
fuel, engine wear, and decreasing engine lifetime. Quantifying this potential is part of the work presented here.

In addition to the spinning reserve market, we look at delivering power to the day-ahead market. The price for delivering in this market is more lucrative than being on call in the spinning reserve market, but here, when committing to power, delivery is required 100% of the time. The cost of input fuel to the engine genset is the dominant cost in selling in the energy market.

The model presented here optimizes power commitments from an engine genset in both day-ahead energy and day-ahead spinning reserve markets, while considering fuel cost, engine performance (efficiency and power output as a function of speed and air-to-fuel ratio), engine cost, wear, and lifetime.

A.2 Data and Methods

The objective of this model is to determine the optimal power output of an internal combustion engine genset participating in day-ahead energy and day-ahead ancillary services markets. The optimization is two-fold and requires finding the power output to the day-ahead market and the power output dedicated to spinning reserve capacity. The model considers the electricity price, spinning reserve price, likelihood of delivering spinning reserve capacity, price of fuel (methane), fuel consumption, engine wear, engine cost, and engine lifetime.

A.2.1 Natural gas price

Data from the U.S. Energy Information Administration report of U.S. Natural Gas Electric Power Prices is used for the price of natural gas [10]. There are a number of different natural prices quoted in the literature. This price reflects the cost of natural gas to a regulated electric utility and unregulated members of the electric power sector. Fig. A.1 shows the monthly average natural gas electric power price in
dollars per thousand cubic feet from January 2002 through October 2016. This data is used to select fuel prices in the engine model.

![Figure A.1: U.S. Natural Gas Electric Power Prices as monthly averages from January 2002 through October 2016 in dollars per thousand cubic feet](image)

**A.2.2 Electricity markets and prices**

New York Independent Service Operator (NYISO) prices are used for both the day-ahead electricity market and the ancillary services (spinning reserve) market [11]. The electricity market prices are from the day-ahead location based marginal pricing (LBMP) Zone West; the spinning reserve prices are from the day-ahead ancillary services prices for Zone West. See Fig. A.2 for an example of one month (April 2016) of day-ahead electricity and day-ahead spinning reserve prices.

**A.2.3 Engine performance data**

Engine performance data, including brake power (kW), brake specific fuel consumption (bsfc) (g/kWh), and air-fuel ratio, is collected from a one-cylinder engine modeled in GT-POWER Engine Simulation Software [12]. The simulation is run for air-fuel
ratios ranging from 12 to 21 and speeds ranging from less than 1,000 to 8,000 rpm. See Fig. A.3 for a schematic of the GT-Power engine model. The components of the GT-POWER one-cylinder engine model shown in Fig. A.3 are as follows:

- env-inlet/env-outlet: These components set the inlet and outlet boundary conditions.

- intrunner/exhrunner: These components set the geometry of the intake and exhaust runner pipes that are used to connect the environment to the intake and exhaust ports.

- intport/exhport: These components set the geometry of the intake and exhaust ports to the engine.
- **si-inject**: This component is the fuel injector and includes air mass flow rate and injection timing.

- **intvalve/exhvalve**: These components set the geometry of the intake and exhaust valves to the engine.

- **cylinder**: This is the engine cylinder, which includes subcomponents defining cylinder geometry, wall temperature, heat transfer, and combustion.

- **Engine**: This component is the engine crank train, which sets engine type (2 vs. 4-stroke), speed, friction, and volumetric efficiency.

![Figure A.3: Schematic of GT-POWER one-cylinder engine model showing environmental inlet/outlet, intake/exhaust runner piping and ports, intake and exhaust valves, fuel injection, cylinder, and engine crank train](image)

Fuel consumption and brake power are functions of both speed and air-fuel ratio. Fig. A.4 shows this relationship: the left column shows power (kW) and brake specific fuel consumption (bsfc) curves at constant air-to-fuel ratios (defined in the legend); the right column shows cross-sectional views of the curves in the left column. Here, power and brake specific fuel consumption are plotted as functions of air-to-fuel ratio at 4,000 and 6,000 rpm.
A.2.4 Engine wear

The rate of engine wear is approximated by (A.1). The wear is normalized from 0 to 1 such that running the engine at optimal power output for the duration of the engine lifetime (in hours) results in a wear equal to one (i.e., $\int W \, dt = 1$). The rate of wear and the equivalent lifetime hours at constant wear are reciprocals. This function was developed to approximate how engine deterioration is related to power output. Literature is sparse for calculating engine lifetime, but one paper uses engine size, load factor, and mean piston velocity for gauging lifetime [13]. Here, we use engine speed to capture the forces acting on piston and cylinder walls due to the systems inertia. The air-to-fuel ratio is included to capture the wear from engine pressure. When the engine runs lean, increasing the pressure in the cylinder causes increased wear; when running rich, the cylinder also sees byproducts of incomplete combustion.
that also affect the engine wear. In this model, the wear curve becomes steeper as a function of increasing power output.

\[
\dot{W}(S, AF) = (\epsilon S)^2 \left( \frac{S}{S_0} \right) \left( \frac{AF_{sr}}{AF} \right)
\]  

(A.1)

\(\dot{W}\) is the rate of engine wear (units of 1/hour), \(\epsilon\) is a dimensional fit parameter, \(S\) is speed, \(S_0\) is used to normalize engine speed, \(AF\) is air-fuel ratio, and \(AF_{sr}\) is the stoichiometric air-fuel ratio. The wear rate is normalized such that \(1/\dot{W}\) is the engine lifetime in hours.

Fig. A.5 shows engine wear rate and engine lifetime hours as functions of power output.

![Figure A.5: Engine wear rate and engine lifetime hours as functions of power output (kW)](image)

A.2.5 Engine replacement cost

The engine replacement cost is incorporated into the wear rate such that when \(\int W dt=1\), the engine is replaced. We note, however, that the engine cost is an engine replacement cost and the first capital investment into the plant includes the first set
of engines. In this way, engine replacement is considered an operational rather than a capital cost. Each hour of engine operation incurs a cost towards replacing the engine.

Engines cost around $10/kW [2]. The base case for the one-cylinder engine in this model is $200 because, as shown in Fig. A.4, the engine peak power is 20 kW.

### A.2.6 Engine fuel cost

Engine hourly fuel costs ($\gamma_f$) as a function of power output are found by multiplying the cost of fuel ($\theta_f$) by the brake specific fuel consumption ($\mu$) at a particular power output ($P_{out}$). (A.2) shows the calculation for engine fuel costs where $\rho_{CH_4}$ is the density of methane; Fig. A.6 shows the cost in dollars per kWh for fuel as a function of power output. For the base case, a natural gas price of $3.24/MCF is used (this was the monthly average natural gas price in October 2016; see Fig. A.1).

$$\gamma_f = \mu P_{out} \theta_f \frac{1}{\rho_{CH_4}}$$  \hspace{1cm} (A.2)

### A.2.7 Total engine operating costs

Total engine operating costs, shown in (A.3), are a function of both engine wear rate, engine cost, fuel consumption, and fuel price (a combination of (A.1) and (A.2)).

$$C_{opt} = \dot{W} E_c + \gamma_f$$  \hspace{1cm} (A.3)

where $C_{opt}$ is the engine hourly operational cost to provide a constant power output, $W$ is the rate of engine wear as a function of speed and air-to-fuel ratio, $E_c$ is the cost of the engine, and $\gamma_f$ is the engine hourly fuel cost.

Prior to the power optimization, an optimization is first performed for evaluating costs. As Fig. A.4 shows, a particular power output can result from a range of speeds when the engine operates at different air-to-fuel ratios. Which combination is best?
To achieve higher output, is it better to change the air-to-fuel ratio, the speed, or both? The first optimization finds this solution by solving for the speed and air-to-fuel ratio that results in the minimum total cost for a given power output. The result is a smooth total cost curve as a function of power output. Both speed and air-to-fuel ratio are varying along the range of power output: sometimes it is best to ramp up the engine in speed to achieve a higher power output, other times it is better to keep speed constant and instead lower the air-to-fuel ratio. The optimization shows that the engine stays lean or stoichiometric (for methane, stoichiometric is just over 17 [14]), which is intuitive because fuel costs dominate (so minimizing fuel consumption is key).

Fig. A.6 shows the total operating costs (in $/kWh) to deliver a particular power output with the two components of cost, wear and fuel consumption, shown. In the base case, fuel dominates the operating costs, and the wear cost (replacing engines) is almost negligible.

Figure A.6: Total cost ($/kWh) of the engine with fuel and wear components as functions of power output (kW)
A.2.8 Generator cost

The engine genset comprises an engine and a standalone electric generator. A standalone electric generator manufacturers suggested retail price (MSRP) is on the order of $100/kW (as an example, a 5HP (3.7 kW) NorthStar Belt-Driven Generator Head costs around $400 [15]). For vehicles, manufacturing costs account for 50% of MSRP [8]. Assuming generators have similar MSRP to manufacturing cost ratios, the manufacturing cost of the generator is about $50/kW. Generators also last much longer than engines; in a study focused on small wind electric generators, the system was estimated to have a twenty-year lifetime [16]. The generators long lifetime compared with the engine suggests that it should be viewed as a capital expense rather than an operating cost.

A.3 Calculation

The model presented here is designed to maximize profit for an engine genset providing power in the day-ahead energy and day-ahead ancillary services (spinning reserve) markets. A single engine is considered a price-taker; its addition to the market does not alter prices.

A.3.1 Profit function

Goal: maximize profit (max[\pi(t)]) where:

\[
\pi(t) = P_1 \theta_e \eta + (P_2 - P_1)(\theta_s + \theta_{s,d}\beta)\eta - (1 - \beta)C_{opt}(P_1) - \beta C_{opt}(P_2) \quad (A.4)
\]

such that 0 \leq P_1 \leq P_2 \leq P_{peak} where P_1 is power delivered to the day-ahead energy market, (P_2-P_1) is power committed to spinning reserves, P_{peak} is peak engine power, \theta_e is the electricity price per kWh, \theta_s is the spinning reserve price per kWh for being on standby, \theta_{s,d} is the spinning reserve price per kWh when called upon
to deliver power, is the probability of being required to deliver the spinning reserve commitment, $C_{opt}$ is the optimal total engine operational cost (wear and fuel) at a given power ($P_1$ or $P_2$), and $\eta$ is the mechanical to electrical efficiency (the loss between mechanical engine output and electric generator output). The profit function in (A.4) can be divided into four pieces: Energy profit

$$\pi(t) = P_1 \theta_e \eta$$  \hspace{1cm} (A.5)

Spinning profit

$$\pi(t) = (P_2 - P_1)(\theta_s + \theta_{s,d} \beta) \eta$$  \hspace{1cm} (A.6)

Energy cost

$$\pi(t) = C_{opt}(P_1)$$  \hspace{1cm} (A.7)

Spinning cost

$$\pi(t) = (C_{opt}(P_2) - C_{opt}(P_1)) \beta$$  \hspace{1cm} (A.8)

As (A.5), (A.6), (A.7), and (A.8) show, there are profit and cost components to each market, and the optimization determines $P_1$ and $P_2$.

The profit model was developed in Python 2.7, and the scipy package optimization method used is L-BFGS-B, which is “a limited-memory quasi-Newton code for bound-constrained optimization” [17].

A.4 Results

The optimization is run once every hour for twelve months from June 2015 through May 2016. At every hour, the day-ahead energy market price and the day-ahead spinning reserve price is input to the optimizer; the system has no hysteresis and is not forward thinking. Each hourly choice is independent.

A week’s worth of results is shown in Fig. A.7 for an example of how the model responds to price signals. As is evident, when electricity market prices are high
enough, energy is delivered to the day-ahead energy market. In Fig. A.7 energy prices are so high on June 4 and 5 that the engine power output is at times, at a maximum, which shows an example of when it is more beneficial to drive the engine beyond optimal operating conditions to capitalize on high market prices. \(P_2\) largely follows the trends of the spinning reserve market. The engine commits to neither market when both prices are too low to be profitable.

The optimization is run for every hour of the year from 1 June 2015 through 31 May 2016. At every hour, profit is maximized as per (A.4). Twelve cases in total were evaluated, including the base case and sensitivity analyses; Table A.1 describes the main features of each case. The results for the base case (1A) and sensitivity analyses are provided in Table A.2. In addition to the results, Table A.2 also lists the fuel price, spinning reserve price multiplier, engine replacement cost, total cost curve, and probability of delivering in the spinning reserve market. The annual profit (\$/kW-yr) results are labeled “Co-optimized,” which provides the profit resulting from participating in both day-ahead energy and reserve markets. This column also provides the breakdown between day-ahead and spinning reserve profits,
which together result in the total profit. The last two columns in Table A.1 show the profit ($/kW-yr) if the engine competes in spinning reserves only or the day-ahead energy market only (but not both together).

<table>
<thead>
<tr>
<th>Case ID</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1A</td>
<td>Base case</td>
</tr>
<tr>
<td>1B</td>
<td>Low fuel cost</td>
</tr>
<tr>
<td>2A</td>
<td>Base fuel cost; low spinning prices</td>
</tr>
<tr>
<td>2B</td>
<td>Low fuel cost; low spinning prices</td>
</tr>
<tr>
<td>3A</td>
<td>Base fuel cost; high spinning prices</td>
</tr>
<tr>
<td>3B</td>
<td>Low fuel cost; high spinning prices</td>
</tr>
<tr>
<td>4A</td>
<td>Base fuel cost; low engine replacement cost</td>
</tr>
<tr>
<td>4B</td>
<td>Low fuel cost; low engine replacement cost</td>
</tr>
<tr>
<td>5A</td>
<td>Base fuel cost; high engine replacement cost</td>
</tr>
<tr>
<td>5B</td>
<td>Low fuel cost; high engine replacement cost</td>
</tr>
<tr>
<td>6A</td>
<td>Base fuel cost; high deliver probability</td>
</tr>
<tr>
<td>6B</td>
<td>Low fuel cost; high deliver probability</td>
</tr>
</tbody>
</table>

Costs are based on wear (engine cost) and fuel consumption (fuel price). These two components are varied so as to produce six possible curves for the base case and sensitivity analyses given in Table A.1. Table A.3 summarizes the fuel price and engine cost selections for each cost curve, and Fig. A.8 shows a plot of the six curves. Fig. A.8 is separated into three subplots by the three different engine replacement costs. The bands in each subplot show the range of costs as a function of fuel price, bounded by the total cost curves at the high (h) and low (l) fuel prices.

The sensitivity analyses performed are as follows:

- All “B” cases are optimized with a lower fuel price ($2.33/MCF). This fuel price was the minimum price of natural gas over the last decade (occurred in March 2016).

- Cases 2A and 2B are optimized with spinning reserve prices halved. These cases are included as examples of what may happen if the engine genset becomes a
Table A.2: Model results and choice of fuel price, spinning reserve price multiplier, engine replacement cost, total costs curve, and probability of delivering spinning reserves. Net annual profit ($/kW-yr) from operating in both day-ahead energy and day-ahead spinning reserve markets and from operating in each market separately are also listed.

<table>
<thead>
<tr>
<th>Case ID</th>
<th>Fuel price ($/MCF)</th>
<th>Spinning reserve price multiplier</th>
<th>Engine replacement cost ($)</th>
<th>Total costs curve β</th>
<th>Co-optimized ($/kW-yr) Spin / Energy</th>
<th>Spinning reserve only ($/kW-yr)</th>
<th>Day-ahead energy market only ($/kW-yr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1A</td>
<td>3.24</td>
<td>1x</td>
<td>200</td>
<td>hM 0.03</td>
<td>27.55 / 23.43 / 4.12</td>
<td>24.59</td>
<td>4.26</td>
</tr>
<tr>
<td>1B</td>
<td>2.33</td>
<td>1x</td>
<td>200</td>
<td>lM 0.03</td>
<td>35.28 / 23.77 / 11.50</td>
<td>27.25</td>
<td>12.11</td>
</tr>
<tr>
<td>2A</td>
<td>3.24</td>
<td>1/2x</td>
<td>200</td>
<td>hM 0.03</td>
<td>12.92 / 8.57 / 4.34</td>
<td>9.45</td>
<td>4.26</td>
</tr>
<tr>
<td>2B</td>
<td>2.33</td>
<td>1/2x</td>
<td>200</td>
<td>lM 0.03</td>
<td>21.50 / 9.41 / 12.09</td>
<td>11.84</td>
<td>12.11</td>
</tr>
<tr>
<td>3A</td>
<td>3.24</td>
<td>2x</td>
<td>200</td>
<td>hM 0.03</td>
<td>58.71 / 55.12 / 3.59</td>
<td>55.71</td>
<td>4.26</td>
</tr>
<tr>
<td>3B</td>
<td>2.33</td>
<td>2x</td>
<td>200</td>
<td>lM 0.03</td>
<td>65.01 / 55.01 / 9.99</td>
<td>58.43</td>
<td>12.11</td>
</tr>
<tr>
<td>4A</td>
<td>3.24</td>
<td>1x</td>
<td>100</td>
<td>hL 0.03</td>
<td>29.84 / 24.30 / 5.54</td>
<td>25.75</td>
<td>5.86</td>
</tr>
<tr>
<td>4B</td>
<td>2.33</td>
<td>1x</td>
<td>100</td>
<td>lL 0.03</td>
<td>39.60 / 23.67 / 15.93</td>
<td>28.44</td>
<td>16.81</td>
</tr>
<tr>
<td>5A</td>
<td>3.24</td>
<td>1x</td>
<td>400</td>
<td>hH 0.03</td>
<td>24.11 / 21.67 / 2.43</td>
<td>22.37</td>
<td>2.38</td>
</tr>
<tr>
<td>5B</td>
<td>2.33</td>
<td>1x</td>
<td>400</td>
<td>lH 0.03</td>
<td>29.44 / 22.93 / 6.51</td>
<td>24.94</td>
<td>6.74</td>
</tr>
<tr>
<td>6A</td>
<td>3.24</td>
<td>1x</td>
<td>200</td>
<td>hM 0.1</td>
<td>15.98 / 11.96 / 4.03</td>
<td>13.56</td>
<td>4.26</td>
</tr>
<tr>
<td>6B</td>
<td>2.33</td>
<td>1x</td>
<td>200</td>
<td>lM 0.1</td>
<td>27.00 / 15.74 / 11.26</td>
<td>20.03</td>
<td>12.11</td>
</tr>
</tbody>
</table>

Table A.3: Total cost curve summary; fuel price is either high (h) or low (l); engine cost is either high (H), medium (M), or low (L)

<table>
<thead>
<tr>
<th>Total costs curve</th>
<th>Fuel price (h/l)</th>
<th>Fuel price ($/MCF)</th>
<th>Engine replacement cost (H/M/L)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Engine replacement cost ($)</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>hM 3.24</td>
<td>200</td>
<td></td>
</tr>
<tr>
<td></td>
<td>lM 2.33</td>
<td>200</td>
<td></td>
</tr>
<tr>
<td></td>
<td>hL 3.24</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td></td>
<td>lL 2.33</td>
<td>100</td>
<td></td>
</tr>
<tr>
<td></td>
<td>hH 3.24</td>
<td>400</td>
<td></td>
</tr>
<tr>
<td></td>
<td>lH 2.33</td>
<td>400</td>
<td></td>
</tr>
</tbody>
</table>
price setter rather than a price taker.

- Cases 3A and 3B are optimized with spinning reserve prices doubled. This case may occur if baseload power becomes more unreliable, for example because of higher penetration of renewable energy technologies.

- Cases 4A and 4B are optimized with the engine replacement cost halved.

- Cases 5A and 5B are optimized with the engine replacement cost doubled.

- Cases 6A and 6B are optimized with the probability of delivering on commitments in the spinning reserve market increased to 10%.
The profit results from Table A.2 are shown graphically in Fig. A.9. Fig. A.10 displays the results in percentages showing the contribution from the day-ahead energy and spinning markets to total annual profit for each case. Fig. A.11 shows the amount of time (as percentages) over the year that the engine spends delivering in the day-ahead spinning market, day-ahead energy market, both markets, or neither market. Finally, Fig. A.12 shows the cumulative probability function for net profit in each case. This plot shows the percent of time net hourly profit is at or below a particular value.

These results show the importance of fuel cost and furthermore, the sensitivity to spinning reserve prices. Case 2A shows that, while holding fuel price unchanged from the base case (1A), cutting spinning reserve prices in half cuts annual profit by more than half. This case is included to demonstrate what could happen if engine gensets become price setters rather than price takers. When enough engine gensets enter the market, they will start influencing (and lowering) spinning reserve prices. The profit is still positive in the case presented here, but not as high as in the base case. Case 2B shows that even when spinning reserve prices are halved, if fuel costs are lowered, the toll on annual profit is not as severe. While annual profits in case 2A are just 47% of profits in case 1A, case 2B profits are still 61% of case 1B profits. Fuel prices, therefore, can compensate for a decrease in spinning reserve prices. In fact, the contribution of the energy market in case 2B outweighs the contribution from the spinning reserves. Therefore, in a situation where this energy technology is a price-setter, low fuel costs may shift the focus to the energy market rather than in the spinning reserves, although the number of hours spent in the spinning reserve market may still dominate. Case 2B is the only scenario that shows the energy market profit higher than the spinning profit (the horizontal dotted line in Fig. A.10 shows the 50% mark).

Case 3A and 3B are included to show the result of an increase in spinning reserve
prices, which would be a likely scenario if there is less confidence in baseload power with increased penetration of renewable energy into the market. Here, profits are much higher than in the base case, which shows a strong case for pursuing this system in combination with higher renewable penetration. Cases 4A and 4B show a less expensive engine replacement cost; cases 5A and 5B show a more expensive engine replacement cost. Decreasing or increasing engine replacement costs has a modest impact on annual profit compared with the impact of changing spinning reserve prices as shown in Fig. A.9. Finally, cases 6A and 6B show the results of increasing the probability of being called upon to delivering a spinning reserve commitment from 3 to 10%. This yields a similar (but not as severe) result to halving spinning reserve prices.

Fig. A.11 shows that across all sensitivity analyses, the time (number of hours of the year) spent in spinning mode-only dominates.

Recall from Appendix A.2.5 and Appendix A.2.8 that the first engine cost ($10/kW) and the generator cost ($50/kW) are together an upfront capital expense on the order of $60/kW. Fig. A.9 shows that in the best scenario (case 3B), this expense is covered by profits in the first year. In the worst case (case 2A), it will take six years to break even from the upfront capital expenses (assuming a discount rate of 7%).

A.5 Discussion and conclusions

The analysis presented in this paper shows the potential for positive net annual revenues for a methane-fueled internal combustion engine genset participating concurrently in day-ahead energy and spinning reserve markets. The results show the largest contribution to annual profit often comes from the spinning reserve market, but that operating in both markets is more profitable than operating in only one market at a time. In all scenarios, the majority of the year (in terms of hours) is spent committing engine capacity to the spinning reserve market. Profit is often dominated by
Figure A.9: Annual profits ($/kW-yr) from using an engine to commit to electricity generation in both the day-ahead spinning reserve and day-ahead energy markets as a function of case; the breakdown of profit contributions from each market are shown.

Figure A.10: Percentage of annual profits from using an engine to commit to electricity generation in both the day-ahead spinning reserve and day-ahead energy markets as a function of case; the contribution to total profit from each market is indicated.
Figure A.11: Percentage of the hours in the year the engine commits to the day-ahead spinning reserve market, the day-ahead energy market, both markets, or neither market as a function of case.

Figure A.12: Cumulative distribution functions showing percentage of the time the net hourly profit ($/hr) is at or below a particular value; showing results for all sensitivity cases.

the spinning reserve market, but can, under one scenario shown here, be dominated by the energy market. The positive results indicate this system design competes well
against existing power markets, and has a strong incentive for competing in spinning reserve markets, especially when baseload power is less dependable.

The system design in this analysis takes advantage of two features of readily available automobile engines: low cost and short lifetime. Internal combustion engines have a very low cost because they are mass-manufactured. As this paper shows, the costs are so low that the short lifetime has little impact on the overall system cost, and we treat the engine replacement cost as an operational rather than a capital cost. The cost of fuel overshadows the engine replacement cost. When committing engine capacity to the spinning reserve market, the engine replacement cost is very low because the likelihood of being called upon to deliver on the commitment is low. When prices are high in the day-ahead energy market, the low engine replacement cost and short lifetime make it possible to view the engine as a consumable that can be burned up rapidly. This ability to trade engine wear against power cost is very different from what a large power plant can do because the plant is viewed as a capital investment.

The low cost and short lifetime combination will also require engines to be replaced frequently. This continuous introduction of new and improved primary plant components cannot be achieved in a conventional, large power plant. Here, it allows introduction of new technology and higher efficiency throughout the plant's lifetime. The electric motor in the current design reduces some of the flexibility provided by the combustion engine, as it introduces a long-lived component into the system. However, it may be possible to move to lower cost, mass-produced replacements. We notice that a shop-vac available on the internet for about $60 has an electric engine that can deliver a peak power of 6HP or 4kW. While these are engines rather than generators, certainly not designed for long-life, they hint at how to lower the cost of the electric generator.

The importance of spinning reserve capacity increases as renewables become a
greater portion of the grid. If the average cost of renewables is sufficiently low, fuel-based power cannot compete at times when renewable energy is available. In this scenario, continuous baseload power using fuels will become obsolete. Fuel-based systems will have to limit operations to times renewable energy availability is limited. In such a scenario, the system proposed in this paper becomes even more valuable, as engine replacement costs are such a small component of operational costs. Large plants with high upfront capital costs will not be able to survive staying off while renewables are providing power, but our system has no issue staying idle until renewables fail to meet demand. Engine gensets have very small capital costs, and therefore can fill an important niche, current designs will have a harder time filling.

Finally, in this paper, when engines are called upon in the spinning reserve market, they must deliver within ten minutes. The engine genset response time, however, is far faster than the ten-minute requirement, which may make this system applicable for highly responsive power delivery applications.

The positive results in this paper show the potential for small, mass manufactured, inexpensive components to compete against traditionally large energy infrastructure. We realize there is still a gap between such a first analysis and a detailed costing of a specific design, but the analysis points toward a way of taking advantage of such systems. The concepts and the conclusion that taking advantage of existing mass production with modular designs likely extends beyond energy production, and we challenge chemical engineering industries to move beyond the continued emphasis on monolithic plants. Mass manufacturing and automation open the possibility for revolutionary changes in energy and chemical engineering industries. This paper shows how internal combustion engines can help clear the way for a transformation in the energy, chemical, and refining industries that is akin to the transition computer technology experienced with the shift from mainframes to small personal computers.
A.6 Acknowledgements
This research did not receive any specific grant from funding agencies in the public, commercial, or not-for-profit sectors. The authors are, in full or in part, funded by the Center for Negative Carbon Emissions at Arizona State University, paid for by the State of Arizona.

A.7 References


Appendix B

Small scale energy storage for peak demand shaving

This chapter is verbatim from \cite{27} with permission to include the paper obtained from ASME on August 29, 2017.
ABSTRACT
Utilities in regulated energy markets manage power generation, transmission, and delivery to consumers. Matching peak demand with peak generation is costly, and the increasing penetration of renewable energy into the grid adds complexity due to fluctuations in supply. A few options exist for addressing the task of balancing supply and demand, including demand response, energy storage, and time-varying pricing (tariffs).

Arizona Public Service (APS), the largest electric utility company in Arizona, employs tariffs that charge more for electricity at certain times (on-peak periods) and a demand charge for the highest power demand throughout the billing period. Such tariffs incentivize end users to lower peak demand. Arizona State University (ASU), a public university with its largest campus in Tempe, AZ, participates in a time-of-use tariff structure with APS. Analysis in this paper shows that ASU’s 16MWdc of onsite solar capacity alone can lower its monthly electricity bills by over 10% by decreasing on-peak power demand.

A novel contribution of the paper is the analysis of the value of small scale, on-campus energy storage in lowering the demand charge. Most analyses consider savings from transferring off-peak electric power to peak-electric power, but this paper considers using stored electricity solely to reduce peak demand and thus lower the demand charge. Small amounts of electricity could greatly reduce overall cost. An algorithm was developed and executed in Python to decide when on-campus storage should be charged and discharged. The critical part of the algorithm is to decide when to discharge. Deploying too early, or too late, will not change peak demand.

The paper’s storage dispatch model is implemented alongside a financial model that calculates the savings in electricity bills and determines the net present value (NPV) of different storage technologies as a function of storage lifetime and installed capacity (kWh). The results show that, for all storage technologies considered, a positive NPV is realized. NPVs are very sensitive to actual power demand and thus vary from year to year. This is to be expected because the storage dispatch strategy operates on extreme values, which tend to include very rare events.

This analysis uses actual data from ASU, which allows us to extend the results to other universities and commercial customers. The favorable results suggest that a smarter dispatch algorithm based on machine learning would enable further cost savings by determining what can be thought of as a shadow price of electricity.

INTRODUCTION
Standard utility company electricity rates are charged per unit of energy generation and tend to be fixed. Some U.S. utility companies, however, offer rate structures that instead vary electricity prices per unit of energy generation with the hour of the day and/or day of the week. These structures are designed to offer lower electricity prices during off-peak periods with higher electricity during on-peak periods. Some utilities also charge for maximum power consumption during a pay-period both on- and off-peak. This means that a customer is charged for the energy used as well as the peak power demanded during a specified interval during on- and off-peak periods.

Based on these time-varying pricing structures, it can be financially beneficial for an entity to shift its demand profile. Entities can shift their demand profiles by (1) adjusting how much energy is purchased during on-peak and off-peak periods and (2) decreasing the peak power demand. These changes can be achieved (at least partially) by active change in behavior, which means that entities must purposefully shift their energy consumption and power demand. For many consumers shifting energy consumption and power demand is not an option. When purposefully shifting energy use is not desirable, alternative methods may be used to decrease energy use and power demand from the grid; this can be achieved (at least in part) by supplementing grid purchases with renewable energy resources and/or with energy storage.

Arizona Public Service (APS) is the largest utility in Arizona and, for some commercial and residential contracts, uses tariffs that charge different electricity prices and peak power charges as a function of hour of the day and day of the week. Arizona State
As the plot in Fig. 1 shows, on-campus photovoltaic solar power cuts on-peak power demand at times by over 25%. Due to the nature of the electricity price structure ASU has established with APS (see Data and Methods (a)), cuts in on peak power demand result in significant decreases in electricity bills. On-peak power demand is the costliest component of the electricity bill (why this is the case is explained further in Data and Methods (a)) and by lowering peak power demand (measured as the average in 15 minute intervals, in a given pay-period, during on-peak hours) by incorporating solar power, the university decreases electricity bills. Further cost decreases can be realized if on peak power demand is further reduced, namely by deploying on-site energy storage.

This paper provides an example whereby strategic action based on shaving peak power demand further decreases energy bills. This is achieved by deploying small-scale energy storage. Deploying stored energy such that peak power demands are decreased requires a well-calculated approach to ensure the finite storage reservoir is used to resist the largest peak in power. Arizona State University provides a unique opportunity to test the potential for on-site, small-scale energy storage as a means to decrease the electricity bill.

The research presented here focuses on determining if on-campus energy storage can be cost-effectively implemented on Arizona State University’s Tempe campus with the school’s current electricity purchasing agreement with Arizona Public Service. We determine the net present value of different storage technologies and analyze how revenue varies as a function of kilowatt-hours (kWh) of installed storage capacity.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_m$</td>
<td>Monthly electricity cost ($)</td>
</tr>
<tr>
<td>$E_{off}$</td>
<td>Off-peak electricity use (kWh)</td>
</tr>
<tr>
<td>$E_{on}$</td>
<td>On-peak electricity use (kWh)</td>
</tr>
<tr>
<td>$P_{off}$</td>
<td>Off-peak peak power demand (kW)</td>
</tr>
<tr>
<td>$P_{on}$</td>
<td>On-peak peak power demand (kW)</td>
</tr>
<tr>
<td>$R_{offE}$</td>
<td>Rate for off-peak energy ($/kWh)</td>
</tr>
<tr>
<td>$R_{offP}$</td>
<td>Rate for off-peak peak power demand ($/kW)</td>
</tr>
<tr>
<td>$R_{onE}$</td>
<td>Rate for on-peak energy ($/kWh)</td>
</tr>
<tr>
<td>$R_{onP}$</td>
<td>Rate for on-peak peak power demand ($/kW)</td>
</tr>
</tbody>
</table>

**DATA AND METHODS**

**(a) ASU’s APS electricity tariff pricing**

Arizona State University uses APS Rate Schedule E-35: Extra Large General Service Time of Use [2]. This rate schedule consists of basic service, demand, and energy charges [3]. The demand and energy charges vary as a function of time of day and day of week separated into on- and off-peak hours. On-peak ranges from 11:00 – 21:00 Monday through Friday; off-peak comprises all other hours.

The energy and power rates are shown in Table 1. The energy rates apply to all energy (kWh) used throughout the month. The power demand rates apply to the maximum power demand in a 15-minute interval (i.e., from hh:00-hh:15, hh:15-hh:30, hh:30-hh:45, hh:45-hh:00 where the beginning and end of the quarter hour are set and do not slide) in the month.

<table>
<thead>
<tr>
<th>Table 1: ASU’s APS electricity rate structure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy ($/kWh)</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>On-peak hours</td>
</tr>
<tr>
<td>Off-peak hours</td>
</tr>
</tbody>
</table>

According to APS [3]:

“For billing purposes, the On-Peak kW used in this rate schedule shall be the greater of the following:

1. The average On-Peak kW supplied during the 15-minute period (or other period as specified by an individual customer contract) of maximum use during the On-Peak hours of the month, as determined from readings of the Company’s meter.”
2. 80% of the highest On-Peak kW measured during the six (6) summer billing months (May-October) of the twelve (12) months ending with the current month. The Off-Peak kW used in this rate schedule shall be the average kW supplied during the 15-minute period (or other period as specified by individual customer contract) of maximum use during the Off-Peak hours of the month as determined from readings of the Company’s meter."

The monthly electricity bill, therefore, is determined by the energy purchased both on- and off-peak as well as the peak power purchased (as described above) (see Eq. 1).

\[ C_m = E_{on} R_{on,E} + P_{on} R_{on,P} + E_{off} R_{off,E} + P_{off} R_{off,P} \]  \hspace{1cm} (I)

Where \( C_m \) ($\text{}/\text{month}$) is the monthly electricity cost, \( E \) (kWh) is the sum of the monthly on-/off-peak energy demand, \( P \) (kW) is the peak 15-minute power demand during on-/off-peak times and \( R \) is the rate for each component as provided in Table 1.

For simplicity, this analysis leaves out the second component of the APS rate schedule, which stipulates a minimum demand charge of 80 percent of the last six summer months. Including 12 months of history would limit the available data even further. Inclusion of this feature in the tariff structure would mean an even larger impact of demand peaks in the summer and raise the threshold at which storage is used to shave peaks, because any peak below the high set by 80% of May through October peaks in the previous 12 months has no impact on the cost. This means that accurately defending against summer peaks becomes even more valuable financially. For example, if in October the highest peak was 30 MW, anything below 24 MW should not be defended against in the following months. When the 80% ratchet is higher than any peaks in the current month, there is no way to save money in that month with energy storage.

As is evident from this pricing structure, the on-peak power demand component is critical, as it accounts for a large portion of the monthly charges due to its high price. On-peak power demand is the costliest component, which can be seen easily with the following example:

Let’s assume a constant power demand of 1 kW throughout an entire month comprising 21 weekdays and 9 weekend days. Each weekday has 10 on-peak hours (from 11:00 – 21:00). So the month will have 210 on-peak hours. Total hours in the month are 720 leaving 510 off-peak hours. The bill has four parts:

1) On-peak energy:
\[ E_{on} R_{on,E} = \text{(1 kW)-(210 hrs.)} \times \text{($0.04076/kWh$)} = \text{$8.56$} \]  \hspace{1cm} (II)

2) On-peak demand charge:
\[ P_{on} R_{on,P} = \text{(1 kW)($15.792/kW$)} = \text{$15.79$} \]  \hspace{1cm} (III)

3) Off-peak energy:
\[ E_{off} R_{off,E} = \text{(1 kW)-(510 hrs.)} \times \text{($0.03219/kWh$)} = \text{$16.42$} \]  \hspace{1cm} (IV)

4) Off-peak demand charge:
\[ P_{off} R_{off,P} = \text{(1 kW)($2.966/kW$)} = \text{$2.97$} \]  \hspace{1cm} (V)

The total bill (the sum of Eqs. II – V; following Eq. I) would amount to $43.73. More than a third (36%) of the bill comes from the on-peak peak power demand. Additionally, the on-peak power demand is almost double the price of the entire on-peak electricity charge ($15.79$ vs. $8.56$). In reality, the contribution of the demand charges will be higher, because consumption is typically less and never more than peak consumption. The fifteen minutes where average power demand is highest over the month during on-peak hours is extremely costly.

Figure 2 shows the breakdown of a monthly electricity bill in two cases: with and without the campus’s installed solar capacity. These financial results are not directly from ASU electricity bills, but rather from the financial model developed in this paper. As this bar graph shows, on-campus solar power decreased the monthly bill by over 10%. The graph also shows agreement with the sample calculation from above in terms of the relationships between the electricity bill components (notice specifically how costly on-peak power is).

\[ \text{Fig. 2: Monthly electricity bill price breakdown (March 2014) with and without on-campus solar power} \]

(b) Data source

ASU’s Energy Information Systems (EIS) stores 15-minute power, energy, and weather data for buildings across the Tempe campus [4]. For this study, fifteen-minute total building load (kW) and solar power (kW) data was downloaded beginning with 1 January 2012 and continuing through 30 April 2016. The model looks at three years’ worth of data: 2013, 2014, and 2015. 2012 is excluded because building load data did not follow the trends seen in the other years (confirmed by the early 2013 data not aligning with the rest of the year). 2016 is excluded because it is not a complete year’s data yet.
Figure 3 shows the daily maximum and minimum for total load (MW) and solar power (MW) for the three years considered.

![Graph showing daily maximum and minimum for total load (MW) and solar power (MW) for the three years considered.]

Figure 3: ASU Tempe campus maximum and minimum daily total building load (MW) and solar power (MW); January 2013 – December 2015

From Fig. 3, annual trends are evident, including high demand in the summer, coinciding with higher air conditioning loads. A gradual increase in overall total power demand suggests a connection to the increase in building construction on the campus. Finally, there is evidence of brief gaps in reliable data, though the reason for such is unknown.

The amount of power imported from the electricity grid is the difference between the total building load and the available solar power. The recorded grid purchases (total load minus solar power) are the base case for our study; we compare our results (i.e., the grid import required when on-campus small-scale energy storage is available) with the power that was actually purchased from Arizona Public Service.

(c) Energy storage overview

Figure 4 provides a sketch of the proposed relationship between building loads, on-campus solar power, APS grid import, and on-campus energy storage. During on-peak periods, storage can be dispatched to defend against peak power demand. Total load is satisfied first by solar electricity and then by the grid. During off-peak periods, storage is recharged and total load is satisfied first by solar electricity and then by the grid.

The schematic in Fig. 4 shows that on-campus energy storage can be dispatched to defend against peak power demands and is recharged by the grid during off-peak hours. The goal is to decrease the on-peak peak power demand over the course of each one-month long billing period.

(d) Energy storage dispatch algorithm and model

The motivation for adding small scale, on-campus energy storage is rooted in the hypothesis that by doing so, on-peak power demands can be mitigated. As shown in Fig. 2, the highest power demand during any 15-minute on-peak period accounts for a large portion of the University’s monthly electricity bill, and shaving this peak may be possible and cost-effective even with small amounts of on-campus storage. Deploying small storage capacity to reduce the peak demand in the month is cost effective because the savings created by lowering demand in a single fifteen-minute time-interval are very large and thus can easily support the cost of storage.

![Diagram of on-campus storage dispatch.]

Figure 4: Overview of on-campus storage dispatch

To determine the cost-effectiveness of storage at the ASU Tempe campus, a dispatch algorithm was developed. Billing periods were set as each calendar month. The first step in the model is to determine an estimate for an on-peak power demand threshold at the beginning of each month. When the month begins, storage will be charged during off-peak hours (the month will always begin with off-peak hours), and, during on-peak hours, storage will begin to discharge during any 15-minute time period when power demand is above the established threshold. A month’s initial threshold is determined by looking at the previous month’s last weekday and determining the threshold on that day for which defending against would have resulted in a full depletion of storage capacity.

If storage fails to fully compensate for the higher power demand, the threshold is increased to the new on-peak peak power demand. Any power demand below the current threshold will not be defended against by storage, regardless of how much storage is available. Storage is never discharged during off-peak hours. Storage is only recharged during off-peak hours and in a manner such that it will never increase the off-peak peak power demand.

A model was written in Python 2.7 to organize an energy storage dispatch algorithm with total building load, available solar power, and grid purchases. Alongside the storage dispatch model designed to handle energy and power figures, a financial model was incorporated to calculate the savings in electricity bills (namely from reducing on-peak peak power demand) and determine the net present value of different storage sizes as a function of lifetime and installed storage capacity (kWh). Figure 5 describes the decision making process employed by the algorithm.
The metric used for determining cost-effectiveness is net present value (NPV). For each storage technology described in Data and Methods (e), a net present value was calculated based on the technology’s cost (applied today, in year 0) and the annual savings realized by having the storage participate in shaving on-peak peak power demands. A discount rate of 7% and a lifetime of 20 years was used for the NPV calculations. The net present value was also normalized to storage size (in kWh) calculated as a function of kWh of installed storage capacity. The Results section shows the outcomes of these analyses.

(e) Storage technologies and costs

The storage dispatch model described in Data and Methods section (d) is applied to various technologies. Three technologies were chosen for this model based on available commercial energy storage options. Tesla battery packs have been gaining popularity as residential and commercial energy storage options. Both the Tesla Powerwall 2 (marketed more for residential applications) and the Tesla Powerpack (marketed more for commercial applications) are modeled here. A flywheel is also used because it is a technology that responds well for short periods of discharge.

Tesla markets the Powerwall 2 battery system as having the capacity to “power a two-bedroom home for a full day. Compact, stackable and with a built-in inverter, installation is simple, either indoor or outdoor” [5]. Information and specifications for the Tesla Powerwall 2 come directly from Tesla’s website [5]. Note, Tesla quotes 90% roundtrip efficiency. An efficiency of 92.5% was used in this model from previous information about the technology.

The Tesla Powerpack battery systems are marketed as being “fully integrated energy storage solutions for commercial and industrial sites. They include one or more battery modules and supporting hardware, including inverters, combiners and cabling” [6]. Powerpack systems begin as small as 50 kW and can be scaled to over 2.5 MW [6]. Information and specifications for the Tesla Powerpack come directly from Tesla’s website [6].

A flywheel is a mechanical energy storage that uses the input of electrical energy to spin up a rotating mass; when electricity is required, kinetic energy from the inertia of the rotating mass spins a motor and generator and outputs electrical power. A flywheel can vary in capacity and power output as well as roundtrip efficiency. The values chosen here were selected from a range of parameters given in [8].

A summary of each technology’s energy, power capacity, and lifetime is included in Table 2. Because the three technologies may be purchased in different sizes, the table shows the specific sizes and specifications chosen for the model here.

<table>
<thead>
<tr>
<th>Technology</th>
<th>Capacity (kWh)</th>
<th>Power (kW)</th>
<th>Roundtrip efficiency (%)</th>
<th>Lifetime cycles</th>
</tr>
</thead>
</table>

Table 3 shows purchasing costs of each technology and the costs normalized to energy size (kWh).

<table>
<thead>
<tr>
<th>Technology</th>
<th>Cost ($)</th>
<th>Cost ($/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tesla Powerwall 2</td>
<td>5,500</td>
<td>407</td>
</tr>
<tr>
<td>Tesla Powerpack</td>
<td>105,280</td>
<td>554</td>
</tr>
<tr>
<td>Flywheel</td>
<td>25,000</td>
<td>1,000</td>
</tr>
</tbody>
</table>

RESULTS

The storage dispatch algorithm was applied to three datasets, each comprising one year’s worth of 15-minute power data beginning in 2013. Net present values (NPVs) were calculated for each year under the assumption that all years of the life of the equipment were the same as the representative year. In all three cases (2013, 2014, and 2015), a positive net present value (NPV) is realized for each technology described in Data and Methods (e). These results are shown in Fig. 6. Net present values vary in magnitude, as is expected because the storage dispatch strategy operates on extreme values which tend to include very rare events. Figure 7 shows the distribution functions for four different months in 2013, 2014, and 2015. Figure 7 shows that predicting peaks is difficult because the distribution of on-peak power from one year may not help predict the distribution for the following year(s). Additionally, the storage is operating on the tail of the distribution.

Figure 6 shows that all technologies reach a positive net present value within a twenty-year technology lifetime. While these net present values are conservative in terms of profit (most hover around $500/kWh), positive net present values do signify profitable investments.

In 2014, the flywheel hit the highest NPV of all three technologies in all three years. In this scenario, a positive NPV occurred within the first four years of technology deployment and the technology’s twenty-year lifetime NPV was over $1,700/kWh. Outside of this result, most NPVs tend to become positive in under ten years and hover around $500/kWh for a
twenty-year technology lifetime. The lowest NPV occurred in 2015 with the Tesla Powerpack with a value of $203/kWh. All of these results, however, show that even very small amounts of energy storage are profitable because they are specifically being targeted to attack the costliest component of the University’s electricity bill, where even a small decrease in on-peak peak power demand makes a large difference in costs.

The next two sections look at specific examples of the storage dispatch algorithm to demonstrate how the model performs.

(a) Single storage installation

To understand how the model’s dispatch algorithm governs the proposed storage capacity, peak power is recorded and the amount of grid purchases required in the model (with energy storage) is compared to the grid purchases in the current ASU system (with no energy storage). The amount of grid purchases in the model with energy storage can also vary as a function of size of storage capacity. To illustrate how the model works, results from 2014 are presented here for the 25kWh/100kW flywheel. This is the scenario from Fig. 6 that yielded the highest NPV. Figure 8 shows a snapshot of the entire 2014 year. The top panel shows the average daily total load and daily solar power. The bottom panel shows the energy storage capacity. As the plot reveals, storage is not dispatched daily, but rather only a few times a month, specifically geared towards lessening on-peak power demands.

Figure 8: ASU Tempe campus 2014 average daily total load and average daily solar power (top); 25 kWh flywheel storage level (bottom)

Figure 8 is helpful for an annual summary of the system’s behavior, but it does not show the intricacies of the storage dispatch model. To understand the model, we first zoom in on a month (Fig. 9), and then on a day (Fig. 10), and ultimately individual hours (Fig. 11) where the month’s on-peak peak power is reduced by dispatching on-campus energy storage. The final view in Fig. 11 includes the 15-minute interval where cost savings were realized. Displayed in all top panels (for Figs. 9 – 14 and 16 – 24) are the total load, the grid purchases without storage, the calculated grid purchases with storage, the on-peak power threshold that the model aims to defend against with storage, the solar power, and a star to show where the month’s on-peak peak power demand occurred. The bottom panels on these figures show the energy storage capacity (in kWh).

Figure 9 shows that storage was dispatched three times in August 2014. Figure 10 zooms in on the third dispatch, which occurs on 25 August 2014. Here, the on-peak hours are shaded in gray. In Fig. 11, the interval between 17:45 and 18:00 shows where the most important storage dispatch occurred that month.
and ultimately how the peak was reduced. Notice that the grid purchases with storage (the peak shown with a star) is below the grid import without storage. Notice also that the threshold adjusts itself to a higher level.

While these figures show an example in which flywheel storage was perfectly prepared to defend against the month’s highest on-peak peak power demand, the current storage dispatch model does not perform this successfully each month. An example of a month in which on-campus energy storage was not able to shave the highest on-peak power demand is May 2014. As before, the data is displayed in three views: monthly (Fig. 12), the day with peak power (Fig. 13), and the hours with peak power (Fig. 14). Again, notice in Fig. 12 how infrequently the storage is dispatched throughout the month.

Figure 13 shows 27 May 2014, the day where the month’s highest on-peak power demand occurred. The day began with a power threshold just over 20MW. At 16:45, however, power demand began increasing. Once the energy storage saw demand above the established threshold, the flywheel began discharging. The error here, best seen in the zoomed in view in Fig. 14, is that the storage was completely depleted by defending against the first occurrence above the threshold even when power demand was continuing to rise. By the time the highest demand occurred at 18:00, the storage level was empty. In this scenario, storage was depleted prior to reaching the highest on-peak power demand, so there was no option to defend against the highest peak when the time came. This resulted in $0 saved towards lowering on-peak peak power demand in this month, even with on-campus storage (here the energy storage essentially shifted peak electricity to off-peak electricity minus the inefficiencies of the storage device). This example shows the imperfections of the current model, as well as the importance of strategically predicting when the highest on-peak power demand will occur.
The next section shows examples of incorporating larger energy storage capacities and the corresponding results.

(b) Double installation

In Results (a), results from a single installation of a flywheel energy storage technology are presented. In this section, we expand the analysis to consider increased storage capacity. Figure 15 shows net present values for both the Tesla Powerwall and for flywheel energy storage as a function of kWh of installed capacity. The Powerwall NPV remains almost completely unchanged (with a slightly negative slope) as a function of kWh of installed capacity, which says that the additional savings due to increased storage capacity are almost exactly balanced by the increase in storage costs over the technology lifetime. The flywheel NPV, however, tends to decrease as installed capacity increases (for the year 2013, increasing capacity may have helped, to a point), which says here that small is actually better. An increase in size in this case does not help. Because this model uses a strategy that operates on extreme values, the larger size does not guarantee that the threshold will be reduced. Further, the most important component of the model is identifying the highest peak and using the installed capacity to defend against it. This works when the highest peak is avoided, but as the storage size increases, the number of peaks at the next highest threshold increases and it is not possible to continuously defend against them (see Fig. 7 for a distribution of peaks). In this case, additional storage capacity does not help.

To understand these trends, we look back at the two examples from the Results (a) highlighting August and May 2014. In this section, we see how the results change when more storage capacity is available (50 kWh instead of 25 kWh). Figure 16 shows August 2014 (similar to Fig. 9, but notice now storage capacity is 50 kWh), Fig. 17 zooms in on 25 August 2014, and finally Fig. 18 show the hours where storage is dispatched. Notice in Fig. 18 that storage is dispatched at the same time as before to defend against the peak at 18:00. Because the storage capacity is twice as large (2 x 25 kWh flywheel), the peak is reduced exactly twice as much as before.
The next example using a double installation of energy storage looks again at May 2014. Figure 19 shows the month view; Fig. 20 shows 27 May 2014; Fig. 21 zooms in on the hours surrounding storage dispatch. Notice in this example, the same fate is realized as in a single installation: storage is still depleted prior to hitting the highest on-peak peak, so the additional storage capacity does not change the peak in that month. Notice that the star showing the peak in Fig. 21 is at exactly the same level as the star in Fig. 14.

In the previous two examples, results show that where the model worked with a single storage installation, it worked twice as well with a double installation and where it did not work with a single, it still failed with a double. This is not always the case. In some instances, a double installation can defend against a peak that a single installation would miss. To see this, the following example includes an instance where the monthly peak is successfully reduced with more storage, and in such a manner that the peak occurs on a different day.

January 2014 provides an example where, unlike in August where two times the storage capacity decreased the peak by twice as much and unlike in May where two times the storage capacity did not change the month’s on-peak peak, two times the storage capacity decreases on-peak peak power demand, but not by twice as much. Figure 22 and Fig. 23 show 28 – 29 January 2014 with one flywheel; Fig. 24 shows the same January days now with two flywheels. Here the plots show that the first flywheel decreased the highest on-peak from 19,260 kW down to 19,181 kW and the additional capacity from the second flywheel further decreased the peak down to 19,125 kW. This additional peak shaving was achieved by applying additional defense to finish defending the same peak (on 28 January) that was previously only partially defended against. Notice the change in location of the stars in these figures showing where the month’s peak occurred.
be based on machine learning and may incorporate weather data as well as time of year (because this example uses university data, in-session versus out-of-session for the academic year has a strong effect on power demand). An example of smarter predictive modeling specifically geared at electricity loads can be found in [9] and a move towards machine learning is advantageous for this application.

With a smarter dispatch algorithm, we imagine this scheme being even more profitable and being an option for other universities and commercial entities to use. The most important piece is predicting peaks. The next step for this project then is to create a smarter decision algorithm to elucidate what can be thought of as a shadow price of electricity. For each 15-minute period, the system must make a decision to answer “Is it worth dispatching storage at this time?” It is likely profitable to recharge storage throughout the day as opposed to restricting recharge to off-peak hours. Again, the focus is on being prepared to defend against the highest peak. These and other options should be further explored in future work.

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REFERENCES
Appendix C

Python code

The Python code written for the engine optimization described in Chapter 3 and Appendix A and the Python code written for the engine model described in Chapter 5 can be accessed at github.com/zlheureux