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Tools & techniques for reduced energy consumption with residential energy system example application

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by

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LIST OF SYMBOLS & ABBREVIATIONS

Symbol	**Meaning**
A	area (m^2)
Bi	Biot number
c	specific heat ($kJ\ kg^{-1}\ K^{-1}$)
c_p	specific heat at constant pressure ($kJ\ kg^{-1}\ K^{-1}$)
c_v	specific heat at constant volume ($kJ\ kg^{-1}\ K^{-1}$)
D	diameter (m)
E	energy (kJ)
e	specific energy ($kJ\ kg^{-1}$)
$\dot{E}$	energy rate (kW)
Ex	exergy (kJ)
ex	specific exergy ($kJ\ kg^{-1}$)
$\dot{E}x$	exergy rate (kW)
Gr	Grashof number
g	gravitational acceleration/constant ($m\ s^{-2}$)
h	specific enthalpy ($kJ\ kg^{-1}$)
h	convection heat transfer coefficient ($kW\ m^{-2}\ K^{-1}$)
k	thermal conductivity ($kW\ m^{-1}\ K^{-1}$)
L	characteristic length (m)
M	molar mass ($kg\ kmol^{-1}$)
$\dot{m}$	mass flow rate ($kg\ s^{-1}$)
Nu	Nusselt number
NTU	number of transfer units
P	perimeter (m)
Pr	Prandtl number
p	pressure (kPa)
Q	heat (kJ)
$\dot{Q}$	heat transfer rate (kW)
R	thermal resistance ($K\ W^{-1}$)
Ra	Rayleigh number
Re	Reynolds number
R_{eq}	equivalent thermal resistance ($K\ W^{-1}$)
R-value	resistance to heat flux ($m^2\ K\ W^{-1}$)
S	entropy ($kJ\ K^{-1}$)
s	specific entropy ($kJ\ kg^{-1}\ K^{-1}$)
t	Time (s)
T	temperature (K or °C)
U	overall heat transfer coefficient ($W\ m^{-2}$), internal energy (kJ)
u	specific internal energy ($kJ\ kg^{-1}$)
V	volume (m^3)
v	
W	work (kJ)
$\dot{W}$	power (kW)

x	quality, mass fraction

Greek letters

α	thermal diffusivity (m^2 s^{-1})
β	coefficient of volume expansion (K^{-1})
ε	effectiveness
η	efficiency
μ	viscosity (kg s^{-1} m^{-1})
ν	kinematic viscosity (m^2 s^{-1})
ρ	mass density (kg m^{-3})
ϕ	efficiency comparison

Indicies

air	Air
cond	condenser, conduction
conv	convection
cs	Cold storage (refrigerator)
env	environment
DHW	domestic hot water heater
evap	evaporator
hs	heat sink
ref	Refrigerant, refrigerator
s	surface
t	Tank
w	water
∞	free stream conditions

Abbreviations	**Meaning**
COP	coefficient of performance
gpm	gallon per minute
kWh	kilowatt-hour
psi	pound force per square inch
psia	pound force per square inch absolute
psig	pound force per square inch gauge

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ABSTRACT

Finite supply and increasing demand characterize our modern energy landscape. Pressure from growing populations, increasing standards of living, and industrializing nations has continued to push energy demand upward. Our societies preferred energy sources are based on fossil fuels that have a finite supply. Although debate continues over the remaining levels of fossil fuel supply it is widely agreed that the sources that are easy to collect are reducing. With the easy to reach sources already in production, sources once thought to be not economically viable due to extreme environments and low-quality or diluted energy are being explored. In the case of the Athabasca oil sands in Alberta Canada, new processes have been developed to extract smaller amounts of oil from larger areas that would have been considered lost in the past.

What is happening on the supply side in the Canadian oil sands is also happening on the demand side with cogeneration (using waste heat in power generation) and diurnal cold storage (capturing cold at night for use in day time space cooling). These are examples of getting useful work from previously discarded (cogeneration), unutilized (oil sands), and under-utilized (cold storage) energy sources.

This thesis focuses on the demand side of the energy equation in residential buildings. Specifically the paper focuses on conversion and use of energy in residential energy systems; space heating, space cooling, water heating, and refrigeration with the goal to reduce domestic energy consumption by sharing resources and combining components.

This research evaluates the feasibility of combining refrigeration and hot water production in a single heat pump system including a steady-state model of a residential vapor-compression refrigerator (heat pump) and energy and exergy analyses. The refrigerant cycle is modeled as steady-state while the cold and hot sink are dynamically modeled. Simulation duration is one day with a time step for dynamic calculations of ten seconds.

CHAPTER 1. OVERVIEW

1.1 Introduction

Public interest in energy is increasing due to a number of factors that range from high utility bills to carbon emissions, global warming, political unrest, national security, and international policy. World-wide energy consumption continues to increase as more and more nations industrialize, trading human labor for the energy in wood, coal, oil, and natural gas. One the demand side, society has been challenged to reduce the consumption of energy and on the supply side scientists and engineers have been challenged to find new energy sources and more efficient ways to utilize energy.

No new energy source has been commercialized since nuclear energy (nuclear fission) was developed in the 1950's. The hope for new energy sources rests solidly on nuclear fusion, but the science behind it remains elusive even in a laboratory setting. The fields of energy conversion and storage on the other hand have seen steady improvement by the scientific community. Improving current energy conversion processes has room for improvement and the development of new energy conversion processes are seemingly endless.

1.1.1 Efficiency vs. Conservation

Even though global warming is becoming a concern of the public, there is little evidence that the average person, household, or business is making large-scale changes to their behavior in order to reduce energy consumption. To attract the majority of the population we will have to have use less energy through “energy use efficiency” not though “energy conservation”. To the media and public, energy conservation usually implies using less energy by a reduction of services or doing without. In engineering practice, the term conservation is used in fundamental laws that physically cannot be violated such as the conservation of mass and energy. In the fields of biology and environmental sciences the term conservation is understood as preserving or protecting species or habitat. To avoid confusion and negative stereotypes the term *energy use efficiency* or *increased energy use efficiency* is used to describe the reduction in energy consumption through the use of new technologies or systems. By the first definition, conservation is a technique to reduce energy consumption that can be applied to any process or end use if the user is willing. An often-used example is turning down the thermostat in the winter. For most of the US populous, this has the negative and unacceptable side effect of an uncomfortable home. The social challenge is to make *conservation* acceptable. The engineering challenge is to develop new strategies for *improved energy use efficiency.*

1.2 Scientific Tools

A partial list of tools that can help engineers improve energy use efficiency range from Thermodynamics, Systems Engineering, Industrial Ecology, Biology to Business.

Tools for improved energy use efficiency:

1. First-Law Analysis (Energy Balance)
2. Second-Law Analysis (Entropy Balance)
3. Exergy Analysis
4. Reduction of irreversible process
5. Life-cycle Analysis

6. Economies of Scale
7. Learning from Natural Systems
    a. Biomimetics
    b. Combining elements to eliminate waste
    c. Constructal Theory
    d. Maximum Power Principle

1.2.1 First-Law Analysis

One of the fundamental concepts in engineering is stated in the first-law of thermodynamics that energy can be converted or stored but cannot be created or destroyed. Using this principle, energy can be tracked through a system or accounted for entering and leaving with an energy balance. The general expression for an open system energy balance is,

$$\frac{dE}{dt} = \dot{Q} - \dot{W} + \dot{m}_i\left(u_i + p_i v_i + \frac{V_i^2}{2} + gz_i\right) - \dot{m}_e\left(u_e + p_e v_e + \frac{V_e^2}{2} + gz_e\right)$$

where $\frac{dE}{dt}$ is the change in stored energy, $\dot{Q}$ is the rate of heat transfer in to the system, $\dot{W}$ is the rate of work done by the system, and the last two terms represent the energy transfer by flows of matter at the inlet (i) and exit (e) of the system boundary respectively. The final matter energy transfer term is comprised of the properties of the exit flow including the mass transfer rate $\dot{m}_e$, the internal energy u_e, the product of the pressure and specific volume $p_e v_e$, the velocity of the stream V_e, and the product of the gravitational constant and the height gz_e. Sign conventions were established from early steam engine development where the work done by a system is defined as positive and heat added to the system is positive. A system is considered closed when there is no exchange of matter with the environment and isolated when there is no matter or energy exchange with the environment. In a closed system there is no mass flow rate across the system boundary so the general expression for the energy balance reduces to,

$$\frac{dE}{dt} = \dot{Q} - \dot{W}$$

An isolated system has no energy exchange across the system boundary so the general expression for the energy balance reduces further to,

$$\frac{dE}{dt} = 0$$

In order to understand and reduce the energy usage buildings, a type of energy balance known as an energy audit would be performed cataloging the power (wattage) and duty-cycle of every energy-consuming device. Although a typical energy audit focuses on large equipment and/or other devices known to consume substantial amounts of energy, even a simple residential energy audit includes the flow of matter such as air is not ignored completely. Air-flow in or out of a residence is evaluated with a blower doorway test where a doorway is sealed up with a special fan that is capable of generating a negative pressure inside the residence. In addition to measuring the flow rate of air required to maintain the pressure differential and calculating the number of air changes per hour to determine the air-tightness of the housing envelope, the negative pressure makes it easier to find gaps in the housing envelope through which heat can escape more easily. The negative pressure increases the flow of air through a gap in the housing envelope which is located by observing the trail of smoke from a smoke pencil.

1.2.2 Second-Law Analysis

The first-law provides a basis to understand energy conservation and conversion but the second-law of thermodynamics is required to know if the energy conversion can occur and describes the extent of the energy conversion that is possible. The second-law states that entropy increases as a system approaches equilibrium. The defining equation for the second-law is,

$$dS = \left(\frac{\delta Q}{T}\right) + s_i\,\delta m_i - s_e\,\delta m_e + \sigma$$

where dS, δQ, and T denote the change in entropy, the heat transferred across the boundary, and the temperature over which the heat transfer occurs respectively. The two terms with subscripts i and e are from matter transfer at the inlet and exit respectively equal to the product of the entropy and change in mass. The final term, sigma, represents entropy production in the control volume.

1.2.3 Exergy Analysis

Exergy is defined as the maximum theoretical useful work obtainable by a system that interacts only with a second system that is suitably large, referred to as the environment, as they come to equilibrium. The general equation for the change in exergy including exergy transfer due to the transfer of matter across the system boundary is,

$$\frac{dEx}{dt} = \sum_j \left(1 - \frac{T_o}{T_j}\right)\dot{Q}_j - \left(\dot{W} - p_o \frac{dV}{dt}\right) + \sum_i \dot{m}_i e_{x_i} - \sum_e \dot{m}_e e_{x_e} - \dot{E}_D$$

where T_o and p_o are the temperature and pressure of the environment respectively, T_j and $\dot{Q}_j$ are the instantaneous temperature and heat transfer rate at the boundary, $\dot{W}$ is the work in the system other than flow work, and $\frac{dV}{dt}$ the rate of change in the volume of the control volume. The term $\dot{m}_i ex_i$ represents the exergy transfer rate at the inlet, $\dot{m}_e ex_e$ similarly at the exit. The final term, $\dot{E}_D$, accounts for the rate of exergy destruction in the system.

Exergy can be divided into components following the way energy can be divided into components,

$$Ex = Ex^{PH} + Ex^{KN} + Ex^{PT} + Ex^{CH} + \cdots$$

where the superscripts PH, KN, PT, and CH stand for the physical, kinetic, potential, and chemical components of exergy respectively. Notably absent in this definition is the exergy contribution from nuclear, magnetic, electrical, and surface tension effects [Kondepudi 1998].

Two useful states of exergy are the *dead state* and the *restricted dead state*. When mechanical, thermal, and chemical equilibrium between the system and the environment are satisfied, the state is referred to as the *dead state* since no useful energy can be extracted. When only the mechanical and thermal equilibrium are met, the state is referred to as the *restricted dead state*.

First-Law or Energy analyses are common practice especially in business optimizations because utility bills come in dollars per unit of energy ($/kWh) not exergy. A brief comparison of energy and exergy analyses is shown in Table 1.

Table 1. Comparison of energy and exergy.

	Energy Analysis	Exergy Analysis
Units:	$/kWh	$/kWh
Use:	Selection amongst similar	Selection from different sources
Level:	Component selection & optimization	System optimization
Minimizes:	Energy inputs	Available work losses

Exergy takes into consideration the quality of the energy not just the quantity to differentiate between energy sources. For most end uses electricity is more useful than hot water or cut wood even if they contain the same amount of energy because the electricity can not only do more work but is easier to do the work. A common system goal is to reduce exergy destruction (available energy) to keep energy in a high quality state. As an example, heat pumps show promise in being able to move heat with less destruction of exergy when compared to resistance heating.

1.2.4 Reduction of Irreversible Process

Although no completely reversible processes can exist in the physical world, they are valuable as standards of comparisons as they represent the limiting cases for maximum work or efficiency that can be obtained. For heat engines the maximum theoretical efficiency comes in fully reversible processes such as the Carnot cycle. Even though the Carnot cycle cannot be implemented due to physical limitations, their efficiency can be approached. An example of a more efficient, less reversible cycle in practice is a heat pump. When implemented correctly, heat pumps can supply heat at two to three times the work that drives them. Resistance heating on the other hand has large exergy destruction by taking high-quality energy and converting it into low-quality heat. Other irreversible processes are listed below in Table 2 and paired with a more reversible alternative:

Table 2. Reversible alternatives for irreversible processes.

Irreversible Process	**More Reversible Process**
Free expansion	Expansion in a turbine
Resistance heating (current through a resistance)	Heat pump
Heat transfer across a finite temperature difference	Highly conductive materials with thin walls and small temperature gradients
Inelastic deformation	Elastic deformation (storage)
Friction	Rolling friction instead of sliding friction or reduce the number of moving parts
Mixing	
Spontaneous chemical reaction, combustion of fossil fuels	Energy conversion by electrolysis and/or fuel cell

1.2.5 Life-Cycle Analysis

Since annual operating cost for much of residential, commercial, and industrial equipment can be much larger than the initial purchase cost it makes sense to consider both initial "capital" costs and expected operating costs when making purchasing decisions. Life-cycle analysis is a Systems Engineering tool used to look at the whole problem from purchasing through operation to the eventual disposal and select the *best* option. The *best* option is subjective but typically is the lowest total life-cycle cost.

Selecting more efficient equipment or optimally sized equipment such as motors to match loads are two possible results of a life-cycle analysis. Over-sizing of equipment can increase purchase costs, maintenance costs, and operating costs, all without improving performance. If only part of the life-cycle is evaluated, such as the initial equipment expense, you could be committing to large and unknown operating costs. With tighter environmental regulations some equipment can be expensive to remove and dispose of making it is not only appropriate but a necessity to include disposal costs.

A frequent scenario that restricts the application of a complete life-cycle analysis occurs when there are different goals and the purchaser is not responsible for operation of the equipment. In many commercial environments equipment is purchased and installed by someone who will not operate & maintain the same equipment. The purchaser only looks at the up-front cost where the operator may be stuck with the subsequent utility bills. In the case of a short term renter or leaser, they have no incentive to replace equipment with more efficient models and many times they do not have the ability too either.

1.2.6 Economies of Scale

Many modern systems are built around the economies of scale. From retail businesses to power plants, increasing the size of the service and combining similar services increases the efficiency. An energy system example of this is combined space and water heating also known as combi systems. Central air conditioning is another example of economies of scale in practice. A larger compressor and heat exchanger replace multiple window units and the air handling unit is shared with the space heating system.

1.2.7 Biometrics

Biomimetics is the application of methods and systems found in nature to the study and design of engineering systems and modern technology. It is also known as bionics or biomimicry. Biomimetics is based on the philosophy that man-made systems can benefit by observing lifeforms and applying natural solutions. Evolutionary pressure forces natural systems to develop desirable traits such as minimal energy use, adaptive shapes to minimize material use (bones grow stronger in response to increased levels of stress), miniaturization, and integration. Classic examples of biomimetics are Velcro (imitating the barbs on a thistle or bur) and Lotus paint (imitating the non-stick surface of a lotus leaf).

Discussion of biomimetics often leads to ecology and ecosystems. Ecosystems, like civilizations and products, are often described in a four-stage life cycle beginning with birth (introduction) that leads to growth (rapid rates of development), maturity (stabilization or plateau), and finally death (decline) as shown in Figure 1.

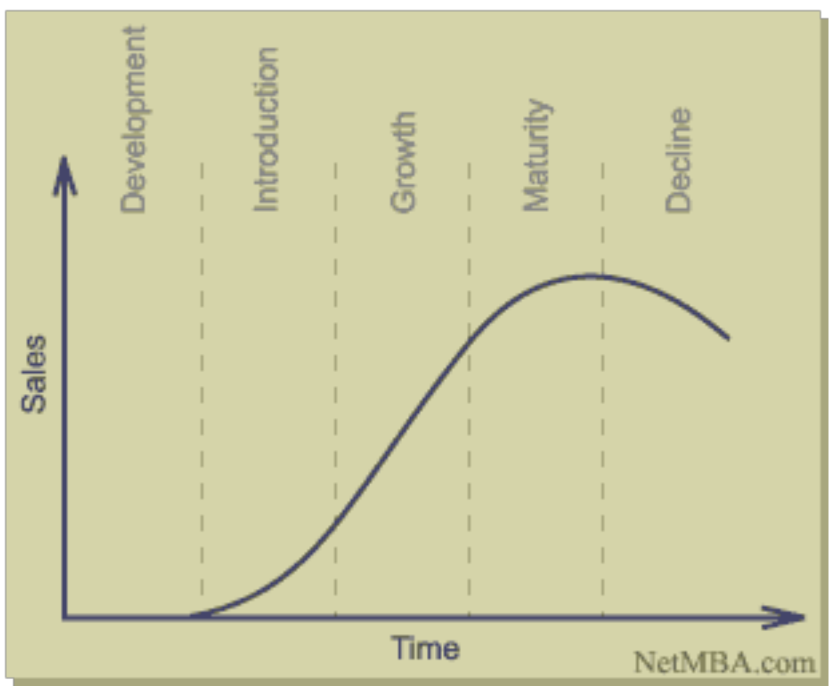


Figure 1. Typical stages of products & ecosystems [NetMBA].

Identifying the correct stage of the *environment* is important to find the best solution to the problem given the constraints of the system. Here the word *environment* can represent a variety of levels and categories such as political, social, religious, geological, climate, energy, or ambient temperature.

In a growth stage there are many competing designs that are rapidly changing in an attempt to figure out what will work best for a given environment. The "dot-com" boom of the 1990's is an example of explosive and volatile growth just like the mental and physical growth that children experience, much exploration and experimenting occurs before the *best* path is discovered.

Once a near optimum, or *best*, path has been discovered, there is little incentive to experiment and similar designs are adopted by many. High degree of convergence in methods or designs indicates a stable or mature environment as designs are close to optimality as allowed by the environment. Slow and incremental changes are common in this phase since "the best design" has already been discovered. Appliances that are differentiated by color and minor features exemplify a stable environment. In a mature ecosystem the following rules apply: [Benyus 1997]

1. Use waste as a resource
2. Diversify and cooperate to fully use the habitat
3. Gather and use energy efficiently
4. Optimize rather than maximize
5. Use materials sparingly
6. Don't foul their nests
7. Don't draw down resources
8. Remain in balance with the biosphere
9. Run on information
10. Shop locally

1.2.8 Combining Elements to Reduce Waste

Nature provides many examples of relationships from the complimentary relationship (mutualism) seen in moss and lichen or pollination to the detrimental (parasitic) relationship of fleas or mosquitoes and mammals.

Under the best circumstances not only are resources shared, but the waste stream from one element or service becomes the fuel for another. Co-generation is an example of connecting a source (waste heat from fossil fuel combustion) to a sink (cold water) to yield combined power generation and heat. In some locations *district heating* takes advantage of would-be waste steam from a power-plant to heat local buildings. Co-generation yielding power and refrigeration or combined heating and cooling are other options to turn waste into fuel.

If we were to envision a scenario in which a residential house had no waste leave the system, every appliance would be connected to one or more complimentary devices. Heating would be connected with cooling. Waste heat would be absorbed by something else that could use it. The refrigerator would not pump heat into the living space in the summer, as that would be detrimental to the space cooling process. The heat exchanger from the refrigerator would be outside like a modern air conditioner or transfer heat to pre-heat air or water. This would likely reduce the energy need in cooler months and keep the interior cooler in warmer months.

It is likely that by following nature we could reduce energy requirements and save money at the same time. Possible benefits in energy reduction would have to be weighed against the added complexity and added cost associated with it. Modern home environments isolate services as much as possible so that each service can operate on it’s own with minimal reliance or connection to another service. When the water heater breaks in a typical house the furnace and oven still work. Connecting all heating devices increases the magnitude of a failure, suddenly all services are shut-down. It seems that there would be a range of possibilities for a heavily interconnected system.

1. Interdependence creates shortages of one service while providing sufficient levels of another.
2. Interdependence creates surpluses of one service while providing sufficient levels of another.
3. Interdependence with no shortage or surplus results in reduced efficiency.
4. Interdependence with moderate shortage or surplus results in improved efficiency.

1.2.9 Constructal Theory

Constructal Theory is the development of a principle that is developed and then observed in nature. This is the opposite of Biomimetics, where the idea is observed first in nature and then applied to a man-made system. One constructal theory is that global optimization can be achieved by balancing and rearranging flow resistances. Resistances imply irreversibility and since irreversibility reduces efficiency, flow architectures that are the least irreversible are the most desired. "Optimal distribution of imperfection" as used by A. Bejan and S. Lorente describes another system goal [Bejan 2007]. This is similar to a method that Buckminster Fuller routinely used to improve his designs. He would start with an intentionally under-designed structure in order to see it's flaws and weak spots. Once identified, the weak spots would be reinforced until the failure is shifted elsewhere and the processes could be repeated indefinitely.

1.2.10 Maximum Power Principle

In the fields of Ecological Sciences such as Ecological Economics and Evolutionary Biology two names are linked to the development of the Maximum Power Principle, Alfred James Lotka (1880-1949) and Howard T. Odum (1924-2004). Lotka took an energetic perspective of evolution and proposed that natural selection was a struggle for available energy. In the engineering disciplines available energy is known as the property *exergy*. Odum continued Lotka's work on ecology while incorporating thermodynamic principles and introducing the concept of *emergy* or the amount of solar energy imbedded in plants, animals, and human society. Emergy is a contraction of the term "embodied energy" and is defined as the product of Energy Quality (expressed by Transformity) by Energy Quantitiy (expressed by Exergy). Although debate within the ecological sciences continues over *emergy* and *empower* (flow rate of emergy) regarding the definitions and units their use in selected fields is growing. In an effort to reduce confusion with other engineering terms, nomenclature such as eMergy and EMERGY are often used.

The Maximum Power Principle has been stated in a number of different ways including the simple,

"Self-organizing systems tend towards the maximization of useful power" [Odum 1983]

but generally states that designs prevail that maximize power. Like the maximum power theorem, Odum's statement of the maximum power principle relies on the notion of 'matching', such that high-quality energy maximizes power by matching and amplifying energy (1994, pp. 262, 541): "in surviving designs a matching of high-quality energy with larger amounts of low-quality energy is likely to occur" (1994, p. 260). The Maximum Power Principle suggests that the most evolutionarily advantageous system function occurs when the environmental load matches the internal resistance of the system.

Although the Maximum Power Principle and its derivatives are not widely acknowledged as governing laws, they do provide, (1) another energy accounting method, and (2) that in its self can provide useful insight to existing system and engineering problems.

1.3 Goals of study

This study's goal is to analyze energy consumption in combined domestic energy systems including hot water generation and refrigeration. Since we must understand the nature of the parts before we can understand the behavior of the whole system and the interaction of the parts in the whole system, domestic hot water and refrigeration are investigated independently before investigating combined systems. The details of component function are looked at from a *systems* perspective to identify details that make the greatest impact on system efficiency. Of greatest interest are possible energy savings that have already been identified and rejected because of an existing system constraint, because in a new system configuration the constraint may no longer be active.

CHAPTER 2. REVIEW OF LITERATURE

2.1 Refrigeration

The goal of the literature review for refrigeration is to establish a baseline energy consumption and cost for current technology, determine efficiency standards and test procedures, and then look at proposed efficiency improvements. Typically energy consumption in a domestic refrigerator is influenced by (i) the ambient temperature, (ii) the refrigerator type including size, efficiency, and insulation levels, (iii) operation including number and duration of door openings, (iv) cold storage temperature, and (v) food loading.

2.1.1 Types of Refrigeration

As refrigeration has moved from the very first refrigeration systems that used frozen water or evaporating ether, refrigeration for space cooling and food refrigeration has increasingly been based on a closed cyclic process. Modern refrigeration cycles can be classified as either a *vapor* cycle or a *gas* cycle with the vapor cycle being further divided into *vapor-compression* and *gas absorption* cycles. Typical domestic refrigerators use the *vapor-compression* cycle to produce cooing effect. The vapor-compression cycle is minimally composed of a compressor, a high temperature heat exchanger, an expansion device, and a low temperature heat exchanger. In US households, alternating current (AC) is typically used to operate the refrigerator and primarily to power the compressor, which increases the pressure and temperature of the refrigerant in the vapor phase. The compressed refrigerant enters the high temperature heat exchanger known as the condenser where the refrigerant looses both latent and sensible heat to the environment as it turns from vapor to liquid. An expansion device, typically in the form of a valve or a small diameter tube (capillary tube), allows the refrigerant to expand in a controlled manner, and cool as it does so. The cooled liquid refrigerant enters the low temperature heat exchanger known as the evaporator and gains both latent and sensible heat from the cold storage cabinet before returning as a vapor to the compressor to complete the cycle.

Gas absorption refrigeration uses thermal energy to dissolve refrigerant in an absorbent liquid, where the refrigerant is expelled from the absorbent liquid creating the cooling effect, when the thermal energy is removed. These typically have 30 percent lower efficiencies than a vapor-compression system and cannot provide continuous cooling [Radermacher 1996]. Absorption refrigeration systems have benefits over vapor-compression refrigeration systems, by not using ozone-depleting refrigerants and being able to operate on lower quality energy such as thermal energy instead of electricity. Absorption refrigerators are used where electricity is not available or is expensive, as is typical in motor-homes and off-grid homes where refrigeration is often powered by propane (LP).

Gas cycles utilize the *Reverse Brayton cycle* for refrigeration where the working fluid such as air is compressed and expanded but does not change phase. Although this gas cycle is typically less efficient and more bulky than the vapor-compression cycle, a variation called an *air cycle machine* is commonly used to create cool air in turbine-powered aircraft [Hunt 1995]. The refrigeration effect of the gas cycle is described as,

$$R_{effect} = c\Delta T_{gas,low}$$

where c is the specific heat of the refrigerant and $\Delta T_{gas,low}$ is the temperature increase in the low temperature heat exchanger as the refrigerant extracts sensible heat from a heat source.

A modern exception to the closed cyclic refrigeration cycles that use a working fluid to transport heat, thermoelectric cooling takes advantage of the Peltier effect to create a heat flux at the junction of two different materials. Although these devices are more robust than a typical vapor-compression system they are not as efficient because they create the heat flux through a non-reversible resistance process. Applications such as computer chip cooling that require small package sizes use thermoelectric cooling because they can be easily scaled to small sizes. Other modern uses include portable coolers in the commercial market to electronic coolers for the military.

2.1.2 Configurations of Domestic Refrigeration Units

There are many configurations available in the current domestic refrigeration marketplace but they all can be reduced to three major categories defined by DOE; refrigerators, refrigerator-freezers, and freezers (which includes compact refrigerators). *Refrigerators* are appliances designed for fresh food storage at temperatures above 0°C (32°F) and below 3.9°C (39°F). A refrigerator may include a compartment for the storage of food at temperatures below 0°C (32°F), but does not provide a separate low temperature compartment designed for freezing and storage of food at temperatures below -13.3°C (8°F). *Refrigerator-freezers* consist of at least one compartment designed for the refrigerated storage of fresh food at temperatures above 0°C (32°F) and at least one other compartment designed for the freezing and storage of food at temperatures below -13.3°C (8°F). Most commercially available units have the cold storage volume split into two compartments with the larger one dedicated to refrigeration and the smaller one to freezing. The cabinets can be split vertically so that they are in a side-by-side arrangement or split horizontally in a top-mounted or bottom-mounted arrangement where the freezer is above and below the refrigerator respectively. To enable direct comparison of one refrigerator-freezer model to another with different proportions of refrigerating and freezing volumes DOE introduced the term *adjusted volume* (AV). AV is used to determine the federal energy conservation standards for refrigerators and freezers and is defined as,

Refrigerator Adjusted Volume = Fresh Volume + (1.63 x Freezer Volume) [DOE 2005]

A *freezer* has a cabinet for the storage and freezing of foods at -17.8°C (0°F) or below. The AV for freezers is defined as,

Freezer Adjusted Volume = 1.73 x Freezer Volume [DOE 2005]

Compact refrigerators are defined as refrigerators, freezers, or refrigerator-freezers with a total volume of less than 220 liters (7.75 cubic feet) and 0.91 meters (36 inches) or less in height. These compact appliances are often used as second refrigerators in a household or as the primary refrigeration appliance in apartments, hotels, or dormitories.

Like many modern appliances the options available commercially are many, including the increasingly popular though-the-door (TTD) service where chilled water and ice can be dispensed to the user (outside the cabinet) from an internal compartment. Less popular options include refrigerator-freezers with three compartments where one can be configured as a refrigerator or freezer based on the needs of the user. The two most popular configurations in the US are both refrigerator-freezers; (1) refrigerator-freezers with top-mount freezers and without TTD service and (2) refrigerator-freezers with side-mount freezers and with TDD service [DOE 2005]. Table 3 shows the number of refrigerator-freezers shipped including those with and without TDD service.

Table 3. Residential Refrigerator and Refrigerator-Freezer Unit Shipment Data [DOE 2005].

Year	Refrigerator One-Door	Refrigerator-Freezers Top- and Bottom-Mount	Refrigerator-Freezers Side-Mount
2004	164,614	6,925,454	3,832,129
2003	180,128	6,383,096	3,457,797
2002	61,880	6,488,361	3,194,103
2001	36,245	6,283,725	2,985,467
2000	33,151	6,297,553	2,885,902
1999	46,662	6,252,716	2,799,194
1998	75,535	6,077,185	2,624,970

2.1.3 Characteristic Sizes

DOE reports in the 1995 Technical Support Document that typical sizes for the two most popular refrigerator-freezers are 515 liters total volume or 606 liters AV (18.2 or 21.4 cubic feet AV) and 614 liters total volume or 742 liters AV (21.7 or 26.2 cubic feet AV) for top-mount without through-the-door features and for side-mount with through-the-door features respectively [DOE 1995b, DOE 2005].

2.1.4 Cost

The cost of full-size domestic refrigerators and freezers can range from as low $400 to over $5000 while compact refrigerators can cost as little as $100. The cost is influenced by the size, insulation level, energy efficiency, type and number of features, quality of construction, and to some degree the style of the appliance. Increasingly kitchens and the appliances in them are being treated as society status symbols. A continuing trend reported by the American Institute of Architects (AIA) is that kitchens remain a focus in overall home design with almost a third of residential architects reporting that the number of separate kitchen facilities or secondary food storage or food preparation areas is increasing in homes [AIA 2008]. What was once a simple functional appliance for storage of fresh and frozen food has now become a vehicle for modern features such as filtered and chilled water, cubed or crushed ice, and even internet-connected kiosks.

The most efficient commercially available refrigerator is the *SunFrost R-19* model, which is 53 percent better than the US standards [Energy Star 2008]. Even at a suggested retail cost of $2,710 the SunFrost model is significantly less expensive than high-end refrigerators such as the *SubZero 650* at a suggested retail cost of $5,700 which is only 15 percent better than the US maximum standard [Energy Star 2008, SunFrost, Consumer Reports 2002, Consumer Reports 2007]. Although the SubZero model is larger, the high-end refrigerator has a cost premium that cannot be explained only by its size and energy-efficiency rating. Materials, style, and warranty are largely what account for the cost premium for these two models. Since there are an abundance of features that are available on different refrigerator models, some that affect the energy-efficiency and others that don't, it is necessary to focus on the incremental cost of adding features and their impact on energy consumption. Although difficult to measure the value in the marketplace, new and unique features are of interest for manufacturers as this enables product differentiation and can help grow their market share.

2.1.5 Standards

The history of appliance standards in the US began with the National Appliance Energy Conservation Act (NAECA) that was signed into law in 1987. This act gave the US Department of Energy the power to set federal standards for maximum energy consumption on household appliances including refrigerators. DOE mandatory standards for refrigerators and refrigerator-freezers were initiated in 1993 and most recently updated in 2001. Refrigerator specifications are based on the unit energy consumption (UEC) that depends on product class and AV. The energy standards apply to refrigerators and refrigerator-freezers with a total refrigerated volume of less than 1104 liters (39 cubic feet) and to freezers with a total refrigerated volume of less than 850 liters (30 cubic feet). A full list of refrigeration product classes governed by the NAECA and their respective energy standards are shown in Table 4.

Energy Star was formed in 1992 as a voluntary joint DOE and Environmental Protection Agency (EPA) program, that seeks to reduce air pollution through increased energy efficiency. The Energy Star criteria for residential refrigerators and freezers are based on the NAECA appliance standards. Active Energy Star criteria for refrigerators became active January 1st, 2004 that require full-size refrigerators to be at least 15 percent more energy efficient than current (2001) federal energy efficiency standards. New Energy Star criteria that would require energy efficiency to be at least 20 percent greater that the federal efficiency standards was announced in 2007 and is scheduled to go into effect April 28th, 2008. Like the NAECA specification that they are based on, Energy Star criteria does not apply to commercial models, refrigerator-freezers and refrigerators with total refrigerated volume exceeding 1,104 liters (39 cubic feet), and freezers with total refrigerated volume exceeding 850 liters (30 cubic feet). A full list of refrigeration product classes supported and Energy Star criteria is shown in Table 4.

Other voluntary standards in the US for refrigerators include the Federal Energy Management Program (FEMP), which sets standards for US Federal Government purchases and the Super-Efficient Home Appliances Initiative (SEHA) promoted by the Consortium for Energy Efficiency (CEE). Table 5 shows the maximum annual energy consumption for the DOE standard and the criteria for various voluntary standards.

Table 4. Maximum UEC values for refrigerator-Freezers with different specifications [DOE 2005].

Specification	UEC (kWh/year) Top-Mount*	UEC (kWh/year) Side-Mount**	Source
2001 DOE Efficiency Standard	486	671	Energy Star (2004)
FEMP (5-8% decrease)	460	620	FEMP (2005)
Current Energy Star (15% decrease)	413	570	Energy Star (2004)
CEE Tier 1 (20% decrease)	389	537	
CEE Tier 2 (25% decrease)	364	503	CEE (2004)
CEE Tier 3 (30% decrease)	340	469	

* Auto defrost, no through-the-door features, 515.4 liters (18.2 ft^3) total volume, and 606 liters (21.4 ft^3) adjusted volume.
** Auto defrost, through-the-door features, 614.5 liters (21.7 ft^3) total volume, and 741.9 liters (26.2 ft^3) adjusted volume.

It is worth noting that Energy Star maintains a list of products that meet or exceed its criteria with currently 2206 active products that range from the minimum 15 percent less energy usage to a maximum 53 percent less energy usage [Energy Star 2008a].

Table 5. NAECA Criteria for Refrigerators and/or Freezers [Energy Star 2007a].

<table>
<tr><th>Product Class</th><th>NAECA energy use (kWh/year) (2001)</th><th>Previous ENERGY STAR Criteria</th><th>Current ENERGY STAR Criteria (as of January 1, 2004)</th></tr>
<tr><td>1) Refrigerator & Refrigerator-Freezer with manual defrost</td><td rowspan="2">8.82AV + 248.4
0.31av + 248.4</td><td rowspan="10">10% less energy than NAECA maximum</td><td rowspan="7">15% less energy than NAECA maximum</td></tr>
<tr><td>2) Refrigerator-Freezer with partial automatic defrost</td></tr>
<tr><td>3) Top Mount Freezer without through-the-door ice</td><td>9.80AV + 276.0
0.35av + 276.0</td></tr>
<tr><td>4) Side Mount Freezer without through-the-door ice</td><td>4.91AV + 507.5
0.17av + 507.5</td></tr>
<tr><td>5) Bottom Mount Freezer without through-the-door ice</td><td>4.60AV + 459.0
0.16av + 459.0</td></tr>
<tr><td>6) Top Mount Freezer with through-the-door ice</td><td>10.2AV + 356.0
0.36av + 356.0</td></tr>
<tr><td>7) Side Mount Freezer with through-the-door ice</td><td>10.1AV + 406.0
0.36av + 406.0</td></tr>
<tr><td>8) Upright freezer with manual defrost</td><td>7.55AV + 258.3
0.27av + 258.3</td><td rowspan="3">10% less energy than NAECA maximum</td></tr>
<tr><td>9) Upright freezer with automatic defrost</td><td>12.43AV + 326.1
0.44av + 326.1</td></tr>
<tr><td>10) Chest Freezers</td><td>9.88AV + 143.7
0.35av + 143.7</td></tr>
<tr><td>11) Compact Refrigerator and Refrigerator-Freezer with manual defrost</td><td>10.7AV + 299.0
0.38av + 299.0</td><td colspan="2" rowspan="8">20% less energy than NAECA maximum</td></tr>
<tr><td>12) Compact Refrigerator and Refrigerator-Freezer with partial automatic defrost</td><td>7.0AV + 398.0
0.25av + 398.0</td></tr>
<tr><td>13) Compact Refrigerator-Freezer-automatic defrost with top freezer</td><td>12.7AV + 355.0
0.45av + 355.0</td></tr>
<tr><td>14) Compact Refrigerator-side mounted freezer with automatic defrost</td><td>7.6AV + 501.0
0.27av + 501.0</td></tr>
<tr><td>15) Compact Refrigerator-bottom mount Freezer with automatic defrost</td><td>12.1AV + 367.0
0.46av + 367.0</td></tr>
<tr><td>16) Compact Upright Freezers with manual defrost</td><td>9.78AV + 250.8
0.35av + 250.8</td></tr>
<tr><td>17) Compact upright freezers with automatic defrost</td><td>11.4AV + 391.0
0.40av + 391.0</td></tr>
<tr><td>18) Compact Chest Freezer</td><td>10.45AV + 152.0
0.37av + 152.0</td></tr>
</table>

AV = Adjusted Volume (cubic feet)

av = Adjusted Volume (liters)

Refrigerators Adjusted Volume = Fresh Volume + (1.63 x Freezer Volume)

Freezers Adjusted Volume = 1.73 x Freezer Volume

The Department of Energy (DOE) is the primary body for refrigeration standards in the United States but there are other ratings and test conditions initiated by independent groups. Other relevant organizations in the US include; Association of Home Appliance Manufacturers (AHAM), Air-Conditioning and Refrigeration Institute (ARI), American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE), and American Society of Mechanical Engineers (ASME).

Internationally there are many more independent groups that have combinations of standards, ratings, and test conditions. The most influential international organizations include the International Organization for Standardization (ISO), International Electrotechnical Commission (IEC), European Committee for Standardization (CEN), and Japanese Industrial Standard (JIS).

2.1.5.1 Test conditions

This paper focuses on the ISO and DOE test specifications because the DOE test governs appliances sold in the United States and the ISO test is the primary international specification. Problems with current standards are outlined well by Bansal [Bansal 2003]. Some of these problems with different standards are that they represent a barrier to international trade, energy consumption results can vary significantly depending on the test performed, labeled energy consumption of a cabinet differs from the ‘‘in-field’’ data by up to 25%, standards quickly become ‘‘out of date’’, and are often unable to accommodate new product innovations.

Energy Star requires that residential refrigerator and freezer manufacturers self-test their equipment according to DOE's test procedure defined in 10 CFR 430, Subpart B, Appendix A1 and Appendix B1 respectively. DOE continues to use a procedure based on ANSI/AHAM HRF-1-1979, even though the ANSI/AHAM HRF-1 test specification that as been revised in 2001, 2004, and most recently in 2007. Internationally ISO 5155, ISO 7371, ISO 8187, and ISO 8561 are the relevant standards for testing the energy consumption of freezers, all refrigerators, refrigerator–freezers, and forced air frost-free units respectively.

2.1.5.2 Temperature of environment

The refrigeration appliance is placed in a temperature-controlled chamber to maintain a constant ambient temperature. DOE test conditions maintain an ambient temperature of 32.2°±0.6°C (90°F) with no specification of the relative humidity. Rationale for the environment temperature is to simulate typical room conditions of approximately 70°F with door openings, by testing at 90°F without door openings. All major test standards noted including DOE and ISO specify that the door remain closed the entire test duration with the exception of the JIS standard. The ISO test specification is intended to cover a wide range of products over the globe so it allows for two different environmental temperatures and four different cold storage temperatures. ISO specifies the following four climatic zones and their ambient test temperatures of either 25°C (77°F) or 32°C (90°F), each held at a relative humidity of 45%–75%,

Extended Temperature zone: 25°±0.5°C
Temperate zone: 25°±0.5°C
Subtropical zone: 25°±0.5°C
Tropical zone: 32°±0.5°C

2.1.5.3 Temperature of cold storage

Since maintaining an exact temperature can be difficult with standard commercial controls, both test standards require two tests to be performed bracketing the temperature requirement with the final energy consumption value an interpolation of the two test results. DOE requires the temperature for a refrigerator only appliance to be 3.3°C and for a freezer only appliance to be -17.8°C. Refrigerator-freezers are tested at slightly warmer temperatures than the individual appliances, the temperature for the fresh-food cabinet in the refrigerator-freezer is 7.22°C and the freezer temperature -9.4°C or -15°C for 1 or 2 star product ratings. ISO requires a fresh food cabinet (refrigerator) temperature 5°C while the freezer temperature is maintained at -6, -12, or -18°C for 1, 2, or 3 star product ratings respectively. Refrigerator or freezer only appliances under ISO are tested at 5°C and -18°C respectively, the same temperatures as the three star refrigerator-freezer.

2.1.5.4 Test duration

DOE allows for test durations to vary from three to 24 hours but must include two or more full compressor and defrost cycles. According to the ISO standard, the test period shall be at least 24 hours long with no door openings. The test is run without automatic defrosting if the appliance does not have this feature or with a whole number of defrost cycles for appliances with automatic defrosting.

2.1.5.5 Test loading

DOE specifies that refrigerators and refrigerator-freezers not be loaded during testing, but the freezer only appliances be loaded to fill 75% of available space. The packages are used to simulate food load in the freezer compartment. Their function is to provide thermal ballast and fill up space. The ISO standard requires loading of the freezer compartment with test packages that take up 100% of the available space and touch the walls for energy-consumption tests. The thermal characteristics of the ISO packages are different from DOE and correspond to those of lean beef. The chemical composition of the ISO packages per 1000 g is: 764.2 g of water; 230.0 g of oxyethylmethylcellulose; 5.0 g of sodium chloride; 0.8g of parachioromethacresol. The freezing point of this material is - 1°C. Although not specified in these tests, loading can be also be more simply by simulated by placing a specified mass of water at ambient temperature in to the cold storage [DOE 2005]. Table 6 summarizes the DOE and ISO test specifications for refrigerators, refrigerator-freezers, and freezers.

2.1.6 Reduction in energy use not just efficiency improvements

Typically energy consumption in a domestic refrigerator is influenced by (i) the ambient temperature, (ii) the refrigerator type including efficiency and insulation levels, (iii) operation including number and duration of door openings, (iv) cold storage temperature, and (v) food loading. For refrigerators that have been in-use for some time, additional factors that can reduce the efficiency include, worn and leaking door seals, fouled heat exchangers, worn compressor and fan bushings, damaged thermostats, and reduced refrigerant quantity from leaks. Although user and operational conditions cannot be controlled by the manufacturer or government standards they do represent often missed opportunities for reduced energy consumption with little or no expense.

Table 6. DOE & ISO Refrigeration Test Specification.

		ANSI/AHAM (DOE [a])	ISO
Ambient Temperature		32.2°±0.6°C	25/32°±0.5°C
Ambient Humidity		-	45-75%
Door Openings		none	none
Cabinet Temperature			
Refrigerator only	Fresh-food	3.3°C	5°C
Refrigerator-Freezer [b]	Fresh-food	7.22°C	5°C
	Freezer *	-9.4°C	-6°C
	Freezer **	-15°C	-12°C
	Freezer ***	-	-18°C
Freezer only	Freezer	-17.8°C	-18°C
Loads			
Refrigerator-Freezer	Freezer	Frost Free - 75% Loaded [c]	100% Loaded [d]
Freezer only	Freezer	75% Loaded [c]	
Volume Adjustment			
Refrigerator-Freezer	Freezer	1.63	2.15
Freezer only	Freezer	0.7 / 0.85 [e]	-
Test Period [f]		3h<t<24h 2 or more cycles	≥ 24h

[a] In North America (Mexico, USA and Canada), energy consumption testing is carried out under US DOE Code of Federal Regulations (Part 430, Sub Part B, Appendices A1 and B1), which is based on AHAM HRF-1-1979.
[b] As per ISO - one, two and three star (*, **, ***) compartments are defined by their respective storage temperature being not higher than -6, -12, and -18C.
[c] The freezer temperature is taken to be the air temperature (contrary to ISO). Freezer compartments that are frost free (forced air) are generally unloaded. However, separate freezers in ANSI/AHAM are always loaded irrespective of defrost type.
[d] The freezer temperature is defined by the warmest test package temperature that must be lower than -18C.
[e] The US DOE code specifies that the measured energy consumption is adjusted by a factor of 0.7 for chest freezers and 0.85 for vertical freezers ‘to adjust for average household usage’.
[f] Note that for cyclic products, the test period consists of a whole number of compressor cycles. For frost free models, the test period consists of a whole number of defrost cycles.

Another opportunity for increased energy efficiency with little or no cost to the end-user is through optimal sizing of components. The manufacturer would spend more design and testing time but may result in smaller components whose reduced cost would offset the development time. In practice the optimal design may not be achievable as there is not a continuously variable spectrum of components to choose from but instead a discrete number of options.

2.1.6.1 End-user choices

There are efficiency gains to be had among widely available models by making simple not high-technology choices. Energy Star recommends purchasing an appropriate-sized refrigerator or freezer model, since larger refrigerators or freezers generally result in greater energy consumption. The most energy-efficient refrigerator models are typically 16-20 cubic feet [Energy Star 2008b]. If more space is required, a single larger appliance is typically more efficient than two smaller ones. The configuration of the refrigerator-freezer also is important, top freezer models typically use 10-25% less energy than side-by-side models [Energy Star 2008b]. Automatic ice-makers and through-the-door dispensers, that can often be found on side-by-side models, increase energy use by 14-20% and raise the purchase price by about $75-250 [Energy Star 2008b]. In standalone freezers, the most energy-efficient models are typically chest freezers because they prevent the cold air from spilling out of the cabinet when the doors are open. Manual defrost models use half the energy of automatic defrost models but must be defrosted periodically to remove frost from the evaporator and realize energy savings. Other energy consumers include “anti-sweat” heaters that will consume 5% to 10% more energy than models without this feature [Energy

Star 2008b]. Models with an automatic moisture control feature prevent moisture accumulation on the cabinet exterior without the addition of a heater. Even after the initial purchase decision, the end-user can significantly reduce the appliances energy consumption by altering their behavior. Minimizing the time the door is open, allowing hot food to come to room temperature before putting it in the refrigerator or freezer, minimizing the need for defrosting by covering food to reduce moisture from escaping, and thawing frozen food in the refrigerator are all simple behaviors that can minimize the energy consumption. The end-user can to a degree control and minimize the ambient temperature by placing the refrigerator appliance in a cool household location, or in the kitchen away from heating appliances such as an oven, out of direct sunlight, and by making sure there is adequate space around the back and top for convection to easily remove heat from the condenser.

2.1.6.2 Energy consumption

Technologies used to increase the energy efficiency of refrigerator-freezers include: high-efficiency compressors; variable-capacity compressors; high-efficiency evaporator and condenser fans; high-efficiency evaporator and condenser fan motors; eliminating thermal shorts; improved door face frame/gasket design; smart defrost technology; added cabinet insulation; lower-conductivity insulation; and vacuum panel insulation [DOE 2005].

A reasonable approach to reducing energy consumption is to target systems (components and mechanisms) that are responsible for the greatest energy consumption or conversion fraction. This is known as an *energy analysis*. Another approach to reducing energy consumption is to target systems that consume the most available energy. Available energy is also called exergy, and exergy is said to be destroyed when energy is converted (consumed) in a non-reversible process such as heat transfer. When the distinction of energy quality is made, that is a distinction between low temperature heat, high temperature heat, mechanical work, and electricity is made, this is known as an *exergy analysis*.

Experimental data of a small household refrigerator-only unit has shown that the greatest exergy destruction occurs in the compressor, followed by the condenser, capillary tube, evaporator and superheating coil [Hepbasli 2007]. From the same study, Table 7 details the energy and exergy rates for the main refrigeration components.

The thermal load on a refrigerator is a combination of cooling food and then keeping it cold. ASHRAE states most of the thermal load in a refrigerator–freezer, about 60%–70% of the total, comes from conduction of the cabinet walls [Masjuki 2001]. Since heat transfer in conduction is proportional to the difference between the ambient temperature and the internal cold storage temperature, the higher the difference, the higher the load imposed on a refrigerator–freezer. Figure 2 shows annual energy consumption of three refrigerator-only units at various ambient temperatures [Bansal 2003]. An ambient temperature reduction from 32.2°C to 25°C results in approximately 20% less annual energy consumption, while a reduction from 32.2°C to 10°C results in 44% less annual energy consumption. Since compressor efficiency also declines as the ambient temperature rises, a refrigerator–freezer's electricity use is very sensitive to the ambient temperature.

Table 7. Exergy destruction and EXCEM parameters provided for one representative unit of the refrigerator considered [Hepbasli 2007].

Component (device)		Exergetic product rate (kW)	Exergetic fuel rate (kW)	Exergy destruction (irreversibility) rate (kW)	Power use	Exergy efficiency (%)	Relative irreversibility (%)	EXCEM parameters				
No.	Name	$\dot{P}$	$\dot{F}$	$\dot{E}x_{dest}$	(kW)	ε	RI	K (USD)*	$\dot{L}_{en}$ (kW)	$\dot{R}_{en}$ (kW USD^{-1}) $\times 10^{-4}$	$\dot{L}_{ex}$ (kW)	$\dot{R}_{ex}$ (kW USD^{-1}) $\times 10^{-4}$
I	Compressor	0.0462	0.065	0.0188	0.0490	71.08	33.40	123.5	0	0	0.0188	1.522
II	Condenser	0.036	0.053	0.017	0.2036	67.92	30.26	19.6	0	0	0.017	8.673
III	Capillary tube	0.0269	0.036	0.0091	—	74.72	16.16	11.2	0	0	0.0091	8.125
IV	Evaporator (cold chamber)	0.0087	0.0182	0.0095	0.1190	47.80	16.87	25.4	0.119	46.85	0.0095	3.740
V	Superheating coil	0.0068	0.0087	0.0019	0.0350	78.16	3.37	11.2	0	0	0.0019	1.696
I–V	Overall system	0.1246	0.1809	0.0563		68.88	100.00	190.9	0.119	46.85	0.0563	2.949

Note: Reference state temperature and the atmospheric pressure are 20°C and 101.325 kPa, respectively.
*1 US$ = 1.4765 YTL (New Turkish Lira) on 6 September 2006.

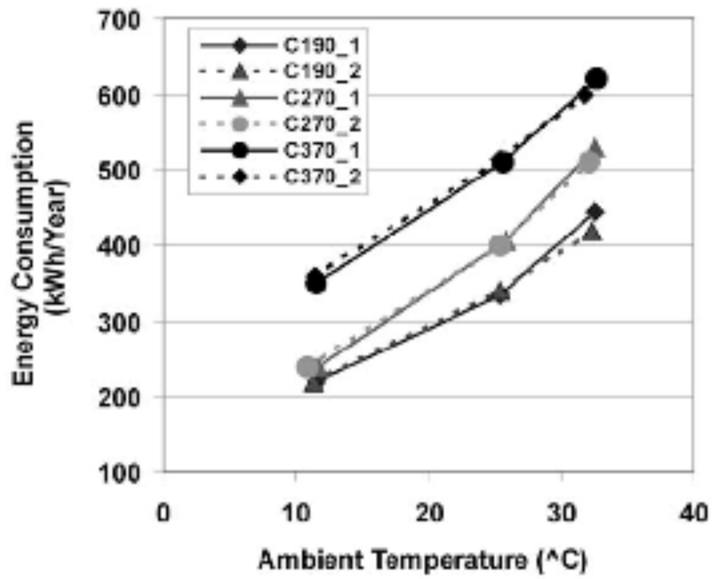


Figure 2. Annual energy consumption with ambient various temperatures [Bansal 2003].

2.1.6.3 Department Of Energy (DOE) proposal

DOE Technical support document (TSD) analyzed two cases of improved efficiency, the 15 percent reduction is equal to existing Energy Star standards, and the 25 percent reduction corresponds to expected new Energy Star standards assuming that the new efficiency standards are equal to the existing Energy Star standards. Table 8 shows the baseline UEC and UEC for various reduced standards. The 1995 TSD provided two paths for the top-mount refrigerator-freezer to achieve the minimum 422 kWh energy use which is two percent higher than current Energy Star. The proposed paths included six common design changes and two options for reducing the thermal losses shown in Table 9.

Table 8. Maximum UEC Values for Refrigerator-Freezers with Different Specifications [Masjuki 2001].

Specification	UEC (kWh/year) Top-Mount*	Side-Mount**	Source
1993 DOE Efficiency Standard	697	954	DOE TSD (2005)
2001 DOE Efficiency Standard	486	671	Energy Star (2004)
Current Energy Star (15% decrease)	413	570	Energy Star (2004)
20% decrease	389	537	
25% decrease	364	503	
1995 TSD Analysis Baseline	701	800	DOE TSD (2005)
1995 TSD Analysis Minimum	422	508	DOE TSD (2005)

* Auto defrost, no through-the-door features, 515.4 liters (18.2 ft^3) total volume, and 606 liters (21.4 ft^3) adjusted volume.
** Auto defrost, through-the-door features, 614.5 liters (21.7 ft^3) total volume, and 741.9 liters (26.2 ft^3) adjusted volume.

Table 9. Design options from 1995 DOE TSD [DOE 1995b].

1	Higher-efficiency compressor (increase from 4.68 to 5.45 EER)
2	Higher-efficiency evaporator fan
3	Higher-efficiency evaporator and condenser fan motors (motor input wattage reduction from 9.1W for the evaporator fan and 12W for the condenser fan to 4.5W for each)
4	Improved gaskets
5	Increased condenser and evaporator areas
6	Adaptive defrost control (ADC)
7a	2.54cm (one-inch) Wall and door insulation thickness increases*
7b	Vacuum insulated panels (VIP)**
* Baseline insulation thicknesses were 3.8 cm (1.5 inch) for the doors, 5.5 cm (2.2 inch) average for the freezer walls, and 4.3 cm (1.7 inch) average for the fresh food walls.	
** The vacuum panel option assumed that half of the total wall and door surfaces would be covered with 1-inch thick vacuum panels.	

The first path uses an approximate one-third increase in insulation thickness that would be relatively easy and inexpensive for manufacturers but would negatively impact the interior volume of the refrigerators. The second path proposes vacuum insulated panels (VIP's) covering half of the total area of the refrigerator that could increase the interior volume but has yet to be incorporated in mainstream refrigerators or freezers. In the United States VIP refrigerators can be found for mobile uses such as medical applications or in motor-homes. In Europe a small number of refrigeration appliances built with VIP's can be found including the award winning *Blomberg CT 1300A* with an Energy Efficiency Index (EEI) of 19.81 and annual energy consumption of 137 kWh for 288 liters of net volume [Eksert 2004]. Individual panels, such as the Barrier Ultra-r product that utilizes aerogel, are available in the US with thermal resistance values of R-50 per inch [Glacier Bay]. This gives the VIP three to seven times more thermal resistance than the standard polyurethane (PU) used in typical refrigerators at a typical thermal resistance value of R-7 to R-8 [EERE 2008e]. Drawbacks to current VIP technology include, reduced insulation value over time as air infiltrates the core, limited shapes and sizes available, less thermal resistance around the edges than in the center due to the conduction in the enclosing membrane, and high manufacturing cost.

The 1995 TSD analysis does not provide a summary of the design options that would be required to achieve energy use 25 percent below the 2001 Standard for top-mount refrigerators. The reader could easily be led to believe that technology and cost in 1995 did not support a detailed analysis or that this required significant extrapolation from existing products. DOE 2004 states that a 25% reduction could *generally* not be achieved by just switching to more efficient components. A reduction in thermal load through the walls and doors would be required through more insulation (thicker walls) or better insulation (VIP).

2.1.6.4 Consortium for Energy Efficiency (CEE) proposal

The CEE investigated near-term options concluding that the combination of five options in Table 10 would have an incremental cost of $106 (at retail) with a $16 annual savings resulting in a simple payback of 6-7 years [CEE]. Most of the design options proposed by DOE and CEE to increase near-term energy efficiency are the same, the exception being that DOE proposed increasing heat exchanger size while CEE proposed modulating variable speed compressors.

Table 10. CEE Near-term energy efficiency options.

1	Variable speed compressor with a potential 10% efficiency gain
2	Variable speed evaporator fans, which use only 2 watts compared to 10-12 with a conventional unit
3	Modulating compressor speeds that adjust to cooling demand
4	Adaptive defrosting that monitors the defrost cycle. Refrigerator defrosts only when blocked by ice, resulting in defrost every 3-4 days (compared to daily defrost in conventional models).
5	Additional insulation

Longer-term options for increasing refrigerator efficiency explored by the CEE includes vacuum insulated panels, which can significantly reduce heat gain in a refrigerated cabinet and potentially result in 10-20% efficiency gain. Secondly, improving fan motors could result in a 9% efficiency gain by switching to electronically commutated motors (ECMs) which typically require less than half as much power as shaded pole motors currently in use. Electronically commutated motors are brushless DC motors that can be programmed to efficiently match the speed required by the application. Additional cost of ECM's, when compared to the simple AC induction machines like shaded pole motors, is due to the required power conversion and electronic controls.

2.1.6.5 Compressors

DOE reports that Energy Efficiency Ratios (EER's) of commercially available, high efficiency compressors used in standard-size refrigerator-freezers can range from about 5 to about 6.1 (Btu/hr-W) at standard rating conditions resulting in a 23% improvement from the 1995 TSD baseline (4.68 EER). Increased compressor efficiency can be accomplished with higher efficiency motors, variable speed compressors, or more efficient compressor designs. The variable speed compressors reduce energy consumption by minimizing cycling losses and allowing the compressor to typically operate at lower capacity. Scroll compressors can result in less energy consumption than the typical reciprocating compressors. Since compressor power depends strongly on the inlet and outlet pressures, any heat exchanger improvements that reduce the temperature difference will reduce compressor power by bringing the condensing and evaporating temperatures closer together [Hepbasli 2007]. More efficient compressors also generate less heat while doing the same amount of work on the refrigerant, which reduces the amount that the refrigerant is super-heated, increasing the cooling effect.

2.1.6.6 Alternate refrigeration cycles

Dual-loop refrigeration systems reduce the thermodynamic irreversibility by employing two separate refrigeration cycles for the refrigerator and freezer. In practice this configuration is not as efficient as theoretically expected because two compressors are used instead of one, and they are typically smaller and less efficient. Cost and space are additional concerns as the doubling of components requires more volume and is achieved at an increased cost. Theoretical analysis predicts energy savings of 20% while experimental results ranged from 4-16% energy savings [Radermacher 1996].

Another variation of the vapor-compression refrigeration cycle is the *Two-stage* system shown in Figure 3. Although the two-stage system has two compressors, just as the dual-loop does, the two-stage has the thermodynamic benefit of low-pressure ratio for both compressors, which allows the compressors to work more efficiently and under reduced power requirements. Compressors are configured in a series arrangement with a heat exchanger in between for intercooling using cool vapor from a flash chamber or suction line. The two-stage system achieves a reported theoretical improvement in performance of 48.6% [Radermacher 1996]. Two patents [Jaster 1990a, Jaster 1990b] have been issued in the US for a two-stage system with intercooling.

Yet another system configuration is the *Cascade refrigeration system* described in thermodynamic textbooks, which has two or more refrigerant loops that have no direct connection except by a heat exchanger [Jones 1996]. In a system with two fluid loops, the condenser for the low temperature loop is the evaporator for the high temperature loop as shown in Figure 4. Large system temperature differences are possible with this type of refrigerator vapor-compression cycle by specifying compressors and refrigerants optimally suited to the small temperature range they are required to operate in. Compared to a conventional single loop system, the cascade system demands only half the temperature difference in each loop.

2.1.6.7 New configurations

The DOE test plan has a definition of and test procedures for an “externally vented refrigerator or refrigerator-freezer” that has air ducts for transferring the exterior air from outside the building envelope into, through, and out of compartments of the refrigerator such as the condenser or condenser/compressor compartment [DOE 2000b]. A literature review resulted in no papers reviewing this configuration and an internet search resulted in no websites mentioning the idea.

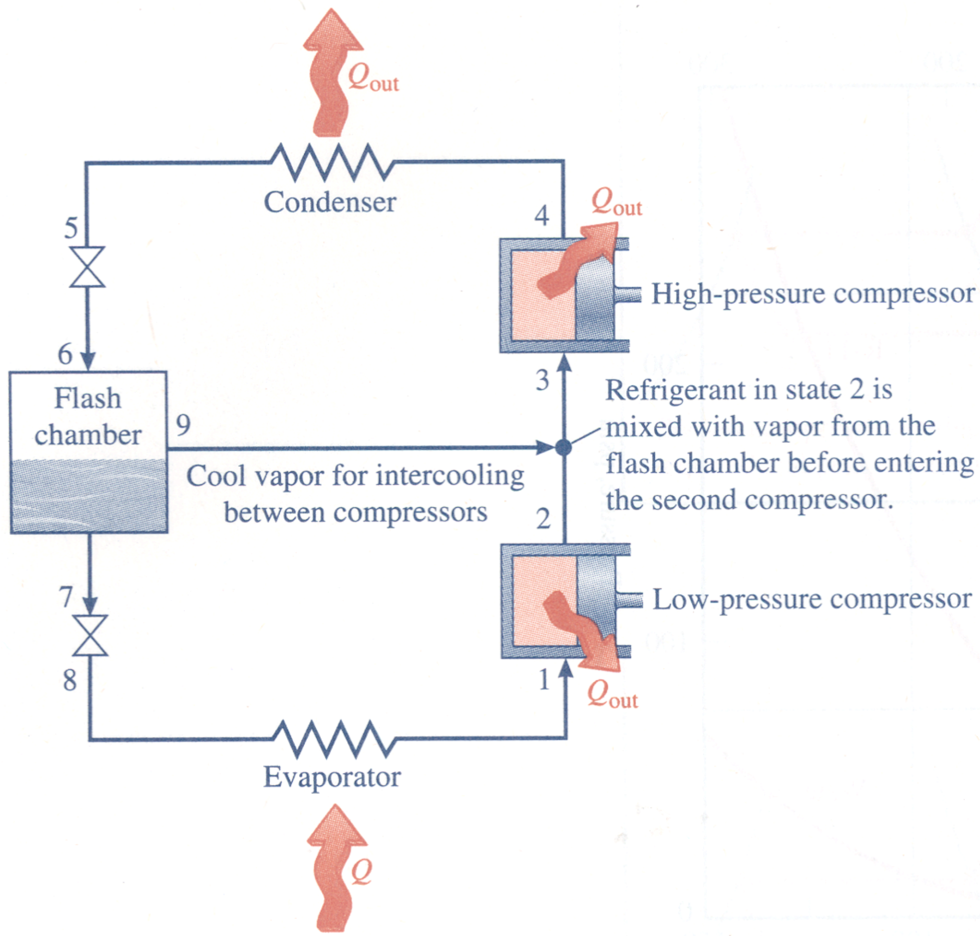


Figure 3. Two-stage refrigeration loop.

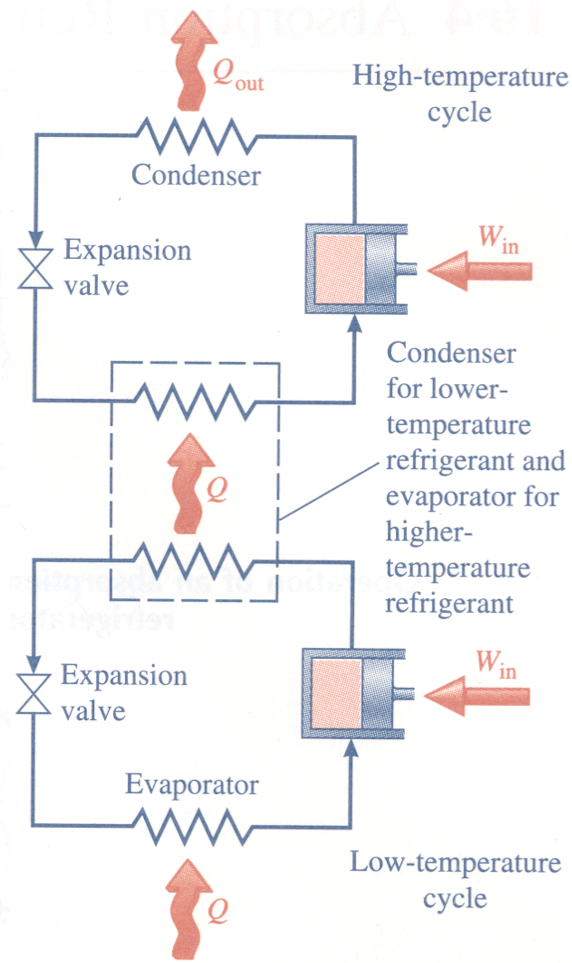


Figure 4. Cascading refrigeration loop.

The inclusion of the externally vented refrigerator in the DOE test plan in 1997 was at the request of Edward Schulak Equities, Inc. (ESE). DOE acknowledged the gap in testing standards and the possible energy savings that could result from this new configuration. Figure 5 from Schulak's 1998 US patent (number 5,743,109) clearly shows the overall concept of taking in cool air from outside the building envelope (28), passing the air over and removing heat from the condenser of the refrigerator (16), and expelling the warmed air outside the building envelope (30) [Schulak 1998].

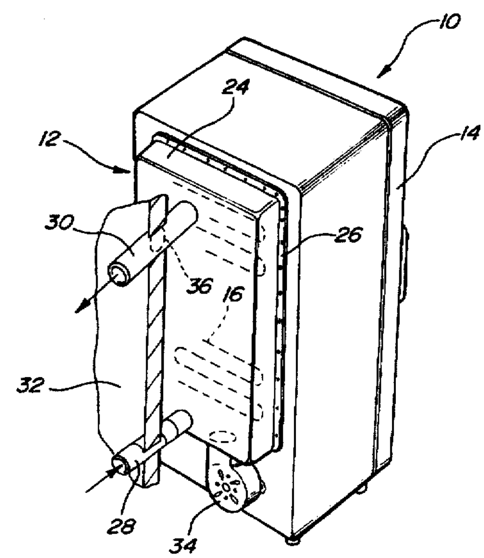


Figure 5. Schulak's 1998 US patent for an Energy efficient domestic refrigeration system.

Figure 6 shows the proposed refrigeration cycle from the 1998 patent overlaid on a conventional refrigeration cycle. The solid line is the "typical" system and the dotted line the "proposed" externally vented system. The lower ambient air temperature available in the externally vented system results in greater COP from two phenomenon, first less compressor power is required because the condenser operates at a lower saturation temperature and pressure, and secondly the refrigerant arrives in the evaporator at a lower quality meaning that more latent heat is available to contribute to the refrigeration effect.

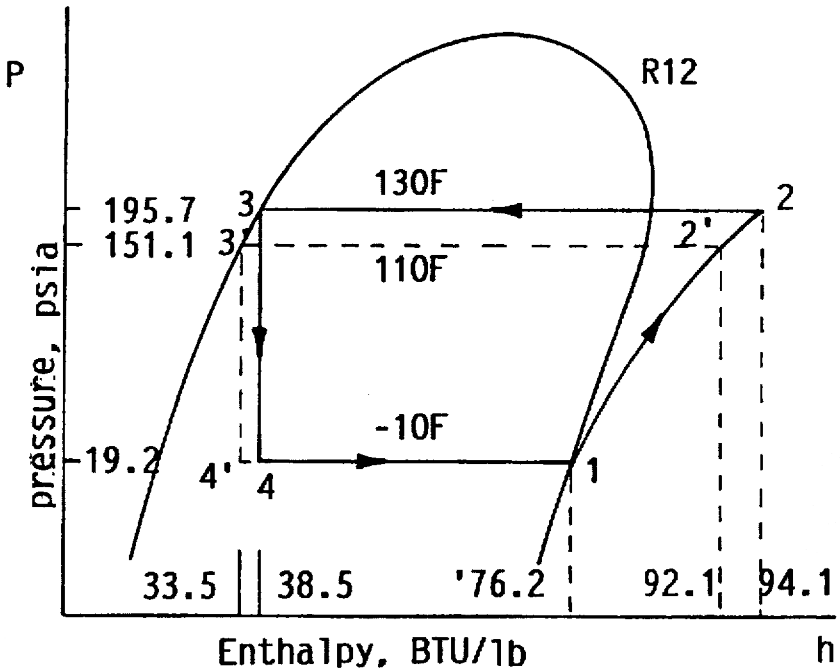


Figure 6. Refrigerant cycle in Schulak's 1998 US patent for an "Energy efficient domestic refrigeration system".

2.2 DHW

The goal of the literature review for domestic hot water heaters (DHW) is to establish a baseline case for current technology including hot water usage, energy consumption, and cost, determine efficiency standards and test procedures, and then look at proposed efficiency improvements and the role of renewable energy. Typically energy consumption in a DHW is influenced by (i) the volume of hot water drawn off, (ii) the DHW type and configuration including efficiency and insulation levels, and (iii) environmental and operating conditions including storage temperature, make-up water temperature, and environment temperature.

2.2.1 Types of DHW

Domestic hot water heaters can be divided into three classifications, *storage*, *instantaneous*, and *batch*. *Storage* is the conventional arrangement that stores several uses worth of hot water at an elevated temperature in an insulated tank. When water is not being drawn from the tank, the water is held at semi-constant elevated temperature so it is ready to be used. Energy in the form of heat is added to the water in the tank either directly or indirectly when the temperature of the water drops from heat transfer to the environment or when cold water replaces the hot water drawn out by the user. *Instantaneous* water heaters are also known as tankless or on-demand water heaters where there is no storage of water at an elevated temperature. Instead, water is heated only as it is used which reduces the energy consumption compared to storage tanks because the stand-by energy loss has been eliminated. The third classification of hot water heaters is a *Batch* water heater were a batch or a fixed volume of water is held and warmed until the water is used, then only after use is the water refilled. Although rarely used in domestic water heating, the batch concept is important when considering renewable energy and cyclic loads.

2.2.2 Fuels for DHW

Water heating in the United States is dominated by the use of electricity and natural gas fuels. Combined, these two fuels account for 93 percent of US household water heating and 89.5 percent of US energy consumption by water heaters among the main water heating fuels as shown in Table 11 [EIA 2004].

Table 11. Number of households and energy consumption for main water-heating fuels [EIA 2004].

	Number of Households		Energy Consumption		
			Household average	Total US	
	Millions of households	% of total	Million Btu (10^6 Btu)	Quadrillion Btu (10^{15} Btu)	% of total
Electricity	40.8	38.4%	8.7	0.355	21.2%
Natural Gas	58	54.6%	19.7	1.143	68.3%
Fuel Oil	4.6	4.3%	28.1	0.129	7.7%
LPG	2.9	2.7%	15.9	0.046	2.8%
total	106.3	100.0%		1.673	100.0%

The type of fuel used in the US for water heating can alternately be viewed though new water heater sales to view consumers' trends. In 2006, the 9.8 million water heater shipments in the U.S were almost evenly split between electric and gas storage water heaters. Conventional electric-resistance accounted for 4.8 million, conventional gas storage 4.7 million, and gas on-demand water heaters accounted for 254,600 shipments. Solar and electric heat pump water heater shipments were an estimated 2,430 and 2,000 units per year respectively [Energy Star 2007b]. Table 12 summarizes the numbers of water heaters sold in the US,

Table 12. Number of water heater units shipped in 2006.

	Units (million)	% of total
Electric Resistance	4.8	49.2%
Gas Storage	4.7	48.2%
Gas On-Demand	0.2546	2.6%
Solar	0.00243	0.02%
Electric Heat Pump	0.002	0.02%
total	9.8	100%

2.2.3 Numerical Models for DHW

The Department of Energy (DOE) Technical Support Document provides detailed information about the water consumption patterns in the US that is the basis of much of the baseline case [DOE 2000a]. As stated in the Technical Support Documnet, the scope of the document is,

> "The assessment includes analysis of: the water heater market; retail prices, manufacturing costs, and markups for water heaters; design options to improve water heater energy efficiency; and costs and benefits of efficiency standards to consumers (including subgroups of consumers who might be particularly affected), manufacturers, utilities, and the nation as a whole, including effects on employment."

In the DOE (2000a) engineering analysis of water heaters, three numerical models are used to investigate the efficiency and payback periods of possible design options and combinations of design options with the intent of establishing manufacturing (energy) standards. These models are, (i) the WATSIM computer simulation model for electric storage water heaters, (ii) the TANK computer simulation model for gas-fired storage water heaters, and (iii) WHAM, a simplified water heater analysis model that calculates average daily energy consumption based on a small number of variables that describe the water heater and its operating conditions. A desirable numerical model is both accurate and easy to use. Due to the use of only seven basic variables to determine energy consumption, four operating condition variables and three water heater efficiency variables, the Water Heater Analysis Model (WHAM) is easy to use. DOE (2000a) reports that residential water heater energy usage is accurately estimated, within 3% to 5% compared to TANK or WATSIM making it accurate enough for parameter studies. The required inputs for WHAM are,

Operating Conditions Characteristics	Water Heater Characteristics
o daily draw volume	o rated input (P_{on})
o thermostat setpoint temperature	o stand-by heat loss coefficient (UA)
o inlet water temperature	o recovery efficiency (η_{RE})
o ambient air temperature	

2.2.3.1 Water Heater Analysis Model (WHAM)

WHAM calculates the average daily energy consumption (Q_{in}) of a water heater as the sum of energy required to heat water and standby losses,

$$Q_{in} = \left(\frac{Q_{out}}{\eta_{RE}}\right) * \left[1 - \frac{UA(T_{\tan k} - T_{\infty})}{P_{on}}\right] + 24\ UA(T_{\tan k} - T_{\infty})$$

where 24 is the number of hours of the test (one day) and Q_{out} is the heat content of the water being drawn from the water heater (kWh/day). Q_{out} can be expressed as,

$$Q_{out} = Vol\, \rho\, c_p \left(T_{\tan k} - T_{in}\right)$$

Vol = volume of water that is drawn (m^3/day)

ρ = density of water (kg/m^3)

c_p= specific heat of water (kJ/kg*K)

η_{RE}= recovery efficiency.

A water heater's η_{RE} value is the ratio of energy added to the water compared to the energy input to the water heater. It is a measure of how efficiently energy is transferred to the water when the heating element is on or the burner is firing [DOE 2000a].

UA = stand-by heat loss coefficient (kW/°K). When unknown, the heat loss coefficient can be derived from the energy factor (EF) and the recovery efficiency η_{RE},

$$UA^* = \frac{\dfrac{1}{EF} - \dfrac{1}{\eta_{RE}}}{\left(T_{\tan k} - T_{\infty}\right)\left|\dfrac{24}{Q_{out}} - \dfrac{1}{\eta_{RE}\, P_{on}}\right|}$$

where 24 is the number of hours of the test (one day)

$T_{\tan k}$ = thermostat setpoint temperature (°C)

T_{in} = inlet water temperature (°C)

P_{on} = manufacturer's rated input power (kW)

T_{∞} = temperature of the ambient air (°C)

*The equation for stand-by heat loss is from DOE (2000a), Appendix D-2. Earlier in Chapter 8 of the same document, the recovery efficiency (η_{RE}) in the numerator was incorrectly listed as the power on (P_{on}).

Alternately from fundamental heat-transfer calculations,

$$UA = \frac{Q_{loss}}{T_{\tan k} - T_{\infty}} = \frac{1}{R_{eq}}$$

where Q_{loss} is the heat loss to the environment is and R_{eq} is the equivalent thermal resistance.

2.2.3.2 Advanced Models

Transient Systems Simulation (TRNSYS) is a popular computer program developed at the University of Wisconsin-Madison with a modular structure that simulates transient thermal systems. A wide variety of standard components can be assembled such as vertical cylindrical tanks, rectangular tanks with heat exchangers, constant speed pumps, air-to-air heat recovery system, geothermal components, buildings, and occupancy loads to name just a few. The ability to simulate stratification in thermal stores makes TRNSYS a powerful modeling tool. Johannes et al. compared

TRNSYS to experimental data and found the upper most and lower most regions of the tank were the closest to the experimental data with the middle varying significantly [Johannes 2005]. Computational Fluid Dynamics (CFD) software FLUENT was used to visualize the fluid flow and understand the limitations of TRNSYS. Johannes et al. believe that the water inlet design influenced the results by inducing a stream at the top and creating layers with non-uniform temperature. The discovery of non-uniform temperature layers help explain why there would be a difference in experimental data that was collected at only one point per height (layer) from the multi-nodal TRNSYS model.

2.2.4 Characteristic Values

Baseline characteristics for storage water heaters were developed for the DOE Technical Support Document [DOE 2000a] using simulation models and customized calculation tools created at the Lawrence Berkeley National Laboratory (LBNL), information from water heater manufacturer and retail contacts, and independent sources. These values are summarized in Table 13,

Table 13. Storage water heater design characteristic values [DOE 2000a].

	Rated Volume		*UA*		η_{RE}	P_{on}	
	Gal	liter	Btu/hr-F	W/K		Btu/hr	kW
Electric	30	114	2.92	1.54	0.972	15,354	4.50
	40	151	3.4	1.79	0.968	15,354	4.50
	50	189	3.64	1.92	0.967	15,354	4.50
	65	246	3.98	2.10	0.966	15,354	4.50
	80	303	4.42	2.33	0.965	15,354	4.50
Natural Gas and LPG	30	114	11.56	6.10	0.758	30,000	8.79
	40	151	13.86	7.31	0.756	40,000	11.72
	50	189	16.14	8.51	0.723	50,000	14.65
	75	284	21.8	11.50	0.672	75,000	21.98
Oil-Fired	32	121	14.93	7.88	0.76	90,000	26.38
	50	189	18.26	9.63	0.76	104,000	30.48

In industry, materials such as insulation are rated by the resistance to heat flux in what is known as the *R-value* of the material. The *R-value* is equal to the inverse of the overall heat transfer coefficient U, and should not be confused with the equivalent thermal resistance R_{eq}. Conventions, associated units, and conversion values for *R-value* are shown below for SI and Imperial unit systems,

$$R_{SI}\left(\frac{m^2\ {}^\circ K}{W}\right)$$
$$R-xx = R_{imperial}\left(\frac{ft^2\ {}^\circ F\ hr}{Btu}\right)$$
$$R_{SI} = R-xx * 0.17612$$

A range of *R-values* provided by the DOE Energy Efficiency and Renewable Energy office [EERE 2008b] for a water heater is 8-25 (ft^2*°F*hr/Btu).

$$R-value = \frac{A}{UA}$$

2.2.5 Hot Water Usage

A variable of DHW energy consumption is the volume of hot water drawn. During the development of the DOE test specifications that call for a daily draw of 64.3 gallons (243.4 l), several government, industry, and utility groups proposed reducing the daily hot water usage to 50 gallons (189.3 l). Just as many groups proposed no change was necessary as the current value of 64.3 gallons per day was close to actual values. The fact that each backed their proposal with study results underlines the reality that household behavior can vary significantly.

Average domestic hot water usage and fuel consumption in the US is from DOE (2000a) data. The analysis uses as its underlying data source the *Residential Energy Consumption Survey* (RECS) from 1997. The average daily hot water draw is less than 61 percent of the DOE test specification. This appears to be a significant difference but is consistent with the common approach in the US of buying not for the average use but for the occasional use. Underlying values for DOE calculations are shown in Table 14.

Table 14. Household Values Underlying DOE Calculations [DOE 2000a].

Water Heater Fuel Type	Average Household Size	Average Hot Water Use		Water Heater Energy Consumption by Fuel	
		Gallons/day	Liters/day	kWh/yr	GJ/yr
Electricity	2.45	37.7	171.5	3460	12.5
Natural Gas	2.82	41.6	188.9	6856	24.7
LPG	2.58	38.1	173	6680	24
Oil	2.87	39.4	179	7517	27.1
U.S. Average	2.68	39.2	178.1		

Average hot water use values supplied by DOE are appropriate for a model like WHAM that (i) simulate simple storage tanks, (ii) have relatively close load following capabilities, (iii) have essentially fixed power supplies, and (iv) calculate energy consumption for periods of time on the order of days or greater not minutes or hours. For studies of daily profiles that include varying temperature thermal stores realistic daily profiles (RDP) are required [Jordan 2001b]. Because there is such variety in the daily draw-off profiles, even when they are used, there can be significant differences in system behavior. In general, the greater the possibility for a system to vary from a known state, such as constant temperature thermal storage, the more detailed the draw-off profile needs to be.

Typical storage tank water heaters that are thermostatically controlled with a never-ending supply of heat source such as natural gas or electricity require less detailed draw-off profiles (loads) than water heaters heated by external sources that are not always possible to control such as cyclic solar energy (renewable energy), heat from electricity generation (co-generation), or waste heat from cooling applications (de-superheaters).

So many profiles exist, not only due to the different studies that generated them, but also because user behavior varies dramatically from one household to another and even for a single household throughout the year. Existing profiles range from as few as three draw-offs in a day (EN 12977) to as many as forty-three draw-offs in a day (RDP3). Three RDP's with draw volumes, flow rates, and times were created by Spur from the Jordan and Vajen model with total daily draw volumes of 100, 180, and 320 liters for RDP1, RDP2, and RDP3 respectively [Jordan 2001b, Spur 2006]. The model created by Jordan and Vajen is based on statistically compiled field measurements in Europe corresponding to real user behavior and takes into account draw volume, draw rate, probability of occurrence throughout the day, and probability of occurrence for each day of the year. For the three RDP's, hot water draw-offs from 11:00pm to 5:00am were excluded due to their low probabilities of occurrence. The three RDP's created by Spur are shown in Figure 7.

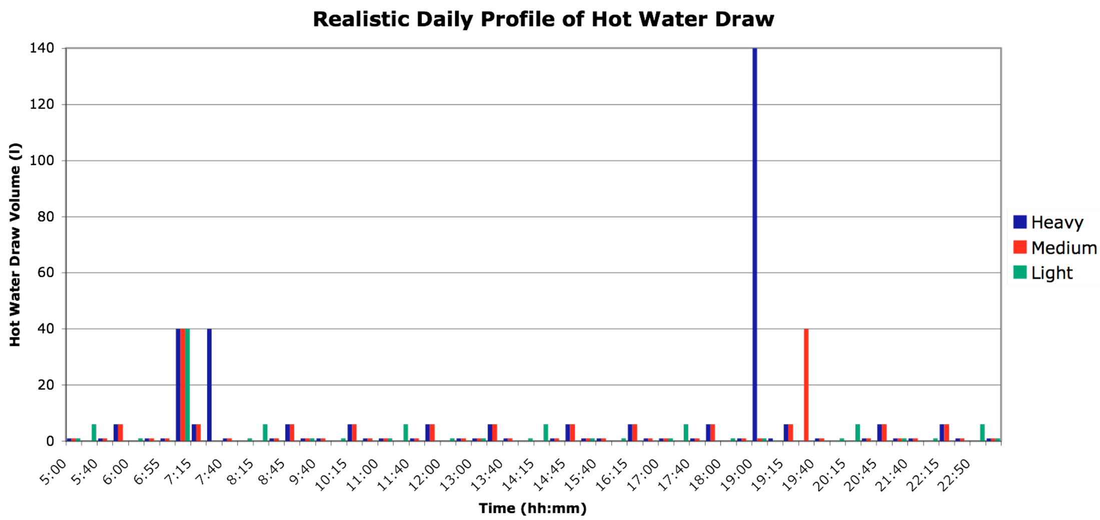


Figure 7. Realistic Daily Profiles of hot water draw.

2.2.6 Storage Tank Temperature

Although energy consumption is reduced by lowering the tank set-point, this has a limit as there is a minimum temperature that is acceptable by the user. Other than personal preference, the lower limit to the hot water temperature is also influenced by health and safety factors. Temperatures lower than 120°F (48.9°C) can allow unhealthy bacteria to develop and prevent some dishwashers from reaching sanitary cleaning levels [Paxton 2007]. The primary concern is the bacteria known as Legionella pneumophila. There are a number of factors that foster the survival and proliferation of Legionella. Water temperature along with the presence of organic matter (sediment) and other micro-organisms are the primary contributing factors [Lacroix 1999]. This bacteria has the highest growth potential at 37°C and starts to die above 46°C as shown in Figure 8.

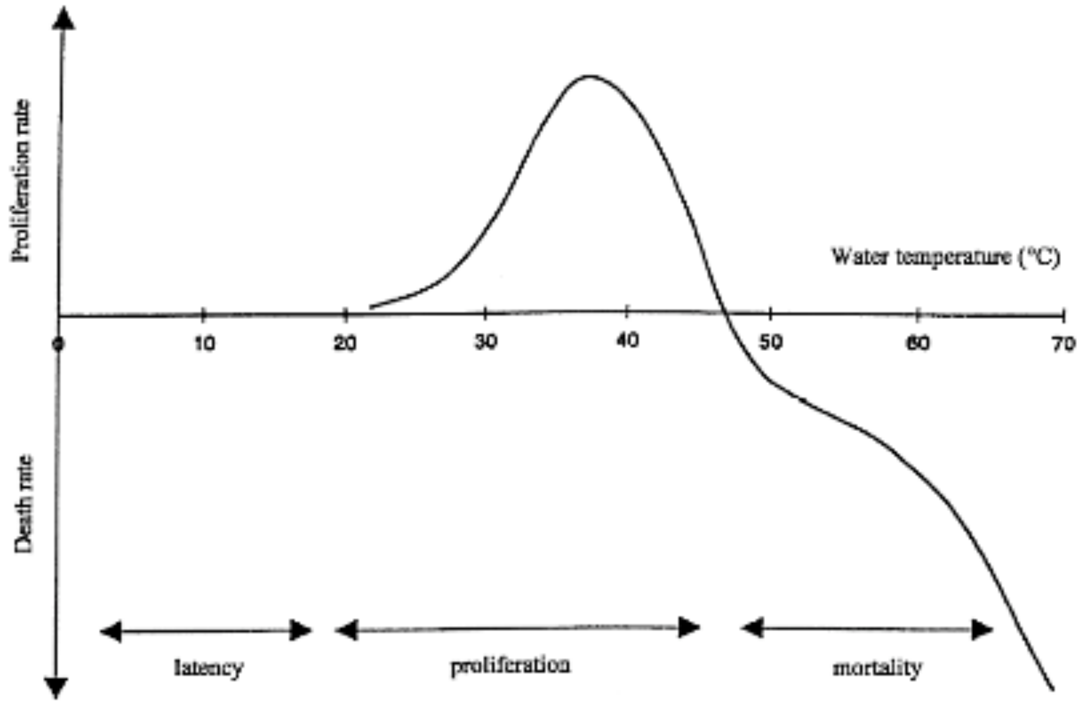


Figure 8. Effect of temperature on the survival of Legionella.

In the development of the DOE test standards that call for a tank setting of 135°F ± 5°F (57.2°C ± 2.8°C), several groups commented and recommended lowering the thermostat setting to 120°F ± 5°F (48.9°C ± 2.8°C), and an almost equal number of groups proposed keeping the standard as-is. Each side claimed in turn that 135°F (57.2°C) matched and did not match what customers actually set their water heaters to. Among the reasons for lowering the tank set-point are,

1) Certain local codes restrict the thermostat setting to be no higher than 120°F (48.9°C) to preventing potential scalding
2) Most energy-related organizations advocate a setting of 120°F (48.9°C) when promoting energy efficiency and safety

Combined responses from the six groups who proposed not changing the standard from 135°F are,

1) A setting at 120°F (48.9°C) could pose a potential health risk (e.g., legionella) to consumers.
2) A setting at 135°F (57.2°C) is necessary to meet consumers' expected hot water needs (as with machine-use for washing clothes)
3) Changing the thermostat setting from 135°F (57.2°C) will not alter the comparative ranking of water heaters but would result in a substantial cost to industry in retesting and re-labeling.

Some reports of greater acceptability of hot water temperature of 45°C (113°F) in Europe are especially beneficial for lower power heating applications that are common for renewable energy applications [Spur 2006]. Reducing the storage tank temperature results in a reduction in energy consumption in both conditions of use, first by reducing the amount of heat that is required to raise the water temperature and second by reducing the stand-by heat loss through a reduced temperature differential with the ambient air. The DOE test standard temperature should be the starting point for new water heater system development since direct comparisons can then be made to existing models and technologies. Sensitivity studies with varying tank set-points should also be performed to identify potential energy savings that could be had with different temperatures. Without other safety measures in place, set-points below 45°C (113°F) should not be encouraged due to the possible bacteria growth.

2.2.7 Ambient Air Temperature

The specification for the DOE water heater test is based on a typical room temperature of a conditioned space with a temperature between 18.3°C (65°F) and

21.1°C (70°F). If the water heater is in a location that is not maintained at a constant, conditioned, temperature, water heater performance will vary from the DOE test results. The actual ambient air temperature around a water heater depends on the geographic location, location in residence, day of year, and even the time of day. Water heaters can be located in many different locations of a residence including; conditioned living space, closets, basements either conditioned or not, crawl spaces, garages, or even outside in moderate climates. Even conditioned living spaces can be at temperatures below the DOE test specification. A 1996 survey in England found that 28% had living rooms at or below 16°C (60.8°F) [ECI 2006]. Placing the water heater in warmer locations will reduce the energy consumption and should be utilized as much as possible.

2.2.8 Supply Water Temperature

Knowing the temperature of the incoming water into the water heater is important to the function and performance of the system. Warmer incoming water is desired in a DHW system since the temperature rise is reduced and the associated required energy to reach the desired DHW use temperature. Incoming water temperature is dependent on the geographic location and day of the year. The mains temperature varies from a low of 1.8°C in February to a high of 23°C in August for Montreal [Biaoua 2007]. In the case of Los Angeles, the temperature fluctuates between 17°C in February and 25°C in August [Biaoua 2007], while incoming water temperatures in Europe range from 5-15°C [Spur 2006]. The DOE test specification is for storage tank make-up water to be 58°F ± 2°F (14.4°C ± 1.1°C) but clearly a range should be expected in actual operation.

The DOE Water Heater Test Conditions are (DOE Final Water Heater Rule, 1998):

a. Daily Hot Water Usage. The current test procedure prescribes water heater testing to determine the energy factor must be based on a daily hot water usage of 64.3 gallons (243.4 l) per day (gpd).

b. Storage Tank Temperature. The existing test procedure uses a thermostat setting of 135°F ± 5°F (57.2°C ± 2.8°C).

c. Ambient Air Temperature. The current DOE test procedure specifies ambient air dry-bulb temperature for heat pump water heaters to be 67.5°F ± 1°F (19.7°C ± 0.6°C) and for all other water heater types to be between 65° F (18.3°C) and 70° F (21.1°C).

d. Supply Water Temperature. The current DOE test procedure specifies supply water temperature to be 58°F ± 2°F (14.4°C ± 1.1°C).

e. Relative Humidity. The current DOE test procedure specifies relative humidity for heat pump water heaters to be between 49% and 51%.

2.2.9 Stand-by Heat Loss

Energy consumed by a storage type water heater is a combination of the energy required to heat the water drawn off and the energy required to keep the store at an elevated temperature. The energy required to keep the store at an elevated temperature is known as the stand-by loss and should be minimized as it results in increased cost and represents no benefit to the consumer. A Canadian study based on both US and Canada statistics reports standby losses for a standard storage tank water heater range from a low of 23% to a high of 28% of the total hot water heater energy consumption [Aguilar 2005]. Rearranging terms from WHAM to get the standby loss,

$$Q_{stdby} = UA\left(T_{\tan k} - T_{\infty}\right)\left(24 - \frac{Vol\,\rho\,c_p\left(T_{\tan k} - T_{in}\right)}{\eta_{RE} \cdot P_{on}}\right)$$

Checking the stand-by loss values using WHAM, average US hot water use, characteristic values for a 40 gallon natural gas water heater, and representative temperatures,

Vol = 178.1 (l/day) = 0.1781 (m^3/day)
$T_{\tan k}$ = 55 (°C)
T_{in} = 13 (°C)
T_{∞} = 25 (°C)
UA = 7.31 (W/°K) = 0.00731 (kW/°K)
η_{RE} = 0.756
P_{on} = 11.72 (kW)
for water at 315°K
ρ = 10^3/1.009 = 991.08 (kg/m^3)
c_p = 4.179 (kJ/kg*°K)

$$Q_{in} = 16433 \frac{W \cdot hr}{day}$$

$$Q_{stdby} = 4496 \frac{W \cdot hr}{day}$$

and the percentage of energy loss through standby as a function of the total energy in,

$$\frac{Q_{stdby}}{Q_{in}} = 27.4\%$$

which matches very well the reported values of 23-28%.

Water heater standby energy consumption values are reported for different fuels in Table 15.

Table 15. Water heater standby energy losses [DOE 2000a, Aguilar 2005].

Water Heater Type	Standby Heat Loss Coefficient (UA)[1] (Btu/hr - °F)	Standby Energy Loss per year (kWh / yr - °C)	Standby Energy Loss per year for a 49°C temperature rise (GJ / yr)
Electric	3.64	16.82	2.97
Natural Gas	13.99	64.65	11.40
Oil	14.49	66.96	11.81

2.2.10 DHW Cost

This study uses the lifecycle cost as the metric for system worth. The total lifecycle cost of a domestic hot water heater includes the upfront purchase plus installation, operation, and maintenance minus rebates or tax credits. In practice, the lifecycle cost is not always the goal or, in some cases, it is not even considered. One market analysis found that half of all customers of retrofit purchases considered only one water heater during the purchase process while the other half only considered an average of 1.9 different water heaters [EIA 2004]. For the customers in the study who "considered only one water heater", they must have at least decided on a size and fuel type and just not compared multiple brands or models. This statistic is reasonable when you consider that 41 percent of retrofits are emergency replacements where there is little time to learn about new technologies and evaluate lifecycle costs. Of the remaining retrofit purchases, 32% are working replacements and 28% are non-emergency, non-working replacements [EIA 2004].

The same analysis goes on to say that the most important purchase factors are energy efficiency or lower operating cost, size of the unit, warranty, and tank life. Despite the stated importance of energy efficiency by the consumer, only 26% of respondents said they would be willing to pay more ($119 on average) for a more efficient water heater (heat pump) that would save them $125 per year [EIA 2004]. That means that only a quarter of consumers with electric water heaters are willing to accept a one-year simple payback. There is a significant reported difference in behavior between electric and gas water heater consumers. The gas consumers pay a little more for their replacement water heaters, $581 compared to an average $353 for an electric unit, and are more willing to invest in energy efficient models with 32 percent willing to pay an average of $247 more for a tankless unit with $75 annual savings. This results in a simple payback period of less than 3-1/2 years. Typical upfront cost, operating cost, life, and normalized life-cycle cost for several current technologies are listed below,

Table 16. Life-cycle costs for different types of water heaters [ACEEE 2007].

Water heater type	Efficiency	Cost[1]	Annual energy cost[2]	Life (years)	Cost over 13 years[3]
Conventional gas storage	57%	$380	$179	13	$2,707
High-eff. gas storage	65%	$525	$157	13	$2,566
Conventional oil storage	55%	$950	$220	8	$4,760
High-eff. oil storage	66%	$1,400	$180	8	$5,140
Conventional electric storage	90%	$350	$410	13	$5,680
High-eff. electric storage	95%	$440	$380	13	$5,380
Demand gas	70%	$650	$160	20	$2,730
High-eff., pilotless demand gas	84%	$1,200	$90	20	$2,370
Demand electric (2 units)	100%	$600	$414	20	$5,982
Electric heat pump	220%	$1,200	$140	13	$3,220
Indirect water heater with efficient gas or oil boiler	79%	$600	$100	30	$1,900
Solar with electric back-up	n/a	$2,500	$125	20	$4,125

1 Approximate cost. Includes installation.

2 Energy costs based on hot water needs for typical family of four and energy costs of 8.28¢/kWh for electricity, 68.8¢/therm for gas, $1.09/gallon for oil.

3 Future operation costs are neither discounted nor adjusted for inflation.

When the purchaser is not responsible for the resulting utility bills they do not have the incentive to select a more expensive but more efficient water heater. Even when the purchaser is responsible for paying the resulting utility bills they may not be able to buy a more efficient water heater because the upfront cost is prohibitive or because the more efficient water heater is not available. The Energy Star office reports that the nature of water heater replacement poses the most significant barrier to market penetration of advanced technologies that can provide increased efficiencies [Energy Star 2007b]. Sudden failure is the reason that two-thirds of the water heaters are replaced.

Retrofit, or the replacement of an existing water heater whether it is operating or has failed, and new installation are two very different categories of water heater purchases and may require different engineering solutions due to replacement urgency, market structure, and consumer behavior (education).

2.2.11 US DHW Standards

Energy conservation standards for domestic water heating were most recently revised to take effect in 2004 [DOE 2001a]. When revising these standards the agency is required to design them to achieve the maximum improvement in energy efficiency that has been determined to be technologically feasible and economically justified. The minimum energy factor specified by the standard is dependent on water heater type and size. U.S. manufacturers are required by federal law to determine the Energy Factor (*EF*) for all products and to label all products with this information [DOE 2001b]. Along with these federal minimum standards US DOE manages the Energy Star rating system to easily identify the appliances that are among the most efficient in their category and size. With the Energy Star label, manufacturers are encouraged to produce more efficient appliances through the recognition of the Energy Star emblem in the marketplace and consumers have confidence that they are purchasing more efficient appliances. Until this year water heaters were the only major domestic appliance not to have an Energy Star rating system. In 2004, the Energy Star Program Manager released a letter indicating the reasons that there would not be an Energy Star rating for domestic water heaters. Among the major reasons were:

1. The differences between the top and bottom performing water heater in a category is small.
2. Conventional gas and electric storage water heaters are approaching the physical limits of energy performance, particularly electric water heaters with energy factors ranging to nearly 0.95. For electric water heaters, significant gains are only possible by using heat pump technology yielding energy factors greater than 1.0. Gas storage water heaters are near their physical limits as well. In order to achieve significant energy efficiency gains, manufacturers will have to pursue condensing or tankless technologies."

3. Non-conventional products do not have proven reliability, depend too heavily on external factors (climate or infrastructure), unreasonable payback timeframes (3.6 to 19 years), and pricing and availability vary widely.

Beginning in 2007, the Energy Star program for domestic water heaters opened again and new standards are expected to go in to effect September 2008. At this time only the draft standards are available for review. The following are not included in the second draft of the domestic water heater standard:

1. Electric-Resistance Storage Water Heaters – Savings from top models in this category are not significant and do not represent meaningful differentiation.
2. Electric-Resistance Tankless Water Heaters – In addition to small energy savings the high current draw requires a dedicated 100 Amp service which is equal to the entire household service for many making it a costly and impractical retrofit.

And, the following are included in the second draft of the domestic water heater standard:

1. High-Performance Gas Storage Water Heaters (under a limited three year inclusion) includes; gas condensing, advanced non-condensing, and near-condensing.
    a. A minimum Energy Factor (EF) of 0.65.
    b. A minimum First-Hour Rating requirement of 67 gallons-per-hour. This is to ensure models earning the label provide sufficient hot water delivery.
    c. A minimum six-year warranty. This is to ensure models earning the label are reliable and perform properly.

2. Whole-Home Tankless Water Heaters (gas only)
    a. A minimum Energy Factor (EF) of 0.82.
    b. A minimum gallons-per-minute (gpm) requirement of 3.0 gpm at a 77°F rise. This is to ensure models earning the label provide sufficient hot water delivery.
    c. A minimum ten-year warranty.

3. Heat Pump Water Heaters
    a. A minimum Energy Factor (EF) of 2.0.
    b. A minimum First-Hour Rating requirement of 50 gallons-per-hour.
    c. A minimum six-year warranty.

4. Solar Water Heaters
    a. A minimum Solar Fraction of 0.50.
    b. OG-300 certification from the SRCC.
    c. A minimum ten-year warranty.

5. Gas Condensing Storage Water Heaters
    a. A minimum Energy Factor of 0.80.
    b. A minimum First-Hour Rating of 67 gallons-per-hour.
    c. A minimum eight-year warranty.

2.2.12 Heat Pump Water Heaters

A heat pump water heater uses a closed refrigeration cycle to transfer heat from the air to the water in a storage tank at a higher temperature. Most heat pumps operate on the vapor-compression cycle with four main components (compressor, evaporator, condenser, and an expansion device) used to circulate refrigerant as it changes state from a vapor to a liquid and back. Just as with a refrigerator or air conditioner, the heat pump makes it possible to move heat against a thermal gradient. That is, to move heat from a lower temperature source to a higher temperature source. Heat pump water heaters were eliminated by DOE as a design option in 1998 and again in 2000 in its *Technology Assessment and Screening Analysis* while understanding its advantages over conventional electric resistance water heaters is apparent in their statement:

> "With current technology, a heat pump can achieve the greatest efficiency improvement for heating water."

DOE continued to state that a two-fold increase in the energy factor (EF) was possible when compared to the conventional electric resistance storage water heater. Looking back at the 1998 and 2000 DOE documents, one can gain an understanding of not only the technology but also the market and consumer response for the reasons this type of water heater was excluded. The following issues form the basis of DOE's decision to screen out heat pump water heaters as a design option pursuant:

- Practicability to Manufacture (on the scale necessary to serve the relevant market at the time of the effective date of the standard),
- Practicability to Install (reliably on the scale necessary to serve the relevant market at the time of the effective date of the standard),
- Practicability to Service (reliably on the scale necessary to serve the relevant market at the time of the effective date of the standard), and
- Impacts on Product Utility to Consumers (significant adverse impact to significant subgroups of consumers).

These issues did not stop the Energy Star office from including heat pump water heaters in the latest *Draft Criteria Analysis*. In fact, the influence can be seen in the inclusion of a six-year warranty in their draft criteria. In the 1998 Final Rule, DOE proposed two categories for heat pump water heaters, *add-on* for units sold without a tank and *integral* for units sold with a tank. The Energy Star *Draft Criteria Analysis* only recognizes *integral* units presumably to satisfy the previously stated issue with installation and reliability. The performance of an *add-on* heat pump water heater would also depend too highly on the system in which it was installed on to give consumers confidence in the Energy Star label.

Energy factors of a continuously run heat pump water heater can be 2.5 to 3.0. In actual operation, energy factors closer of 2.0 are observed [DOE 2000a]. The difference is due to cycling losses, stand-by losses, and the use of less efficient electrical resistance backup heat. A concern when using and testing heat pump water heaters is that the heat that is removed from the air needs to be accounted for. In warmer climates there would be no penalty to the household as the heat pump water heater's exhaust would be a welcome cool air. In colder climates when space heating is operating the heat extracted from the air by the heat pump water heater would need to be offset by an increased load on the space heating equipment. Based on current technology with an energy factor of 2.0, DOE calculates that heat pump water heaters have a payback of three years without tax credits or rebates and 2.5 years with the current 10% tax credit [Energy Star 2007b].

2.2.13 Efficiency Improvements

Much work has been done by the Department of Energy and the Energy Star office on energy consumption, upfront cost, life cycle cost, and expected life span of domestic water heaters in preparation for the minimum efficiency standards and the Energy Star criteria. The final report by the Energy Star office describes energy efficiency criteria for categories and types of water heaters but excludes detailed information about the means to achieve the criteria. At a more detailed level of water heater features the DOE Technical Support Document [DOE 2000a] analyzes individual options that can increase the efficiency of water heaters. The analysis includes the following options that passed the initial screening process based on the economic, efficiency, and implementation feasibility criteria.

1. Increased Insulation (side and bottom)
2. Blowing agents for insulation
3. Heat Traps
4. Plastic Tank
5. Improved Flue Baffle/Forced Draft

6. Increased Heat Exchanger Surface Area
7. Flue Damper (Electromechanical)
8. Side Arm Heater
9. Electronic (or Interrupted) Ignition

Although the DOE energy efficiency standards do not specify any specific means, the level of the standard its self is responsible for dictating much of the design direction manufacturers must follow. Because of this influence, DOE must consider the impact on all involved parties from utilities to installers to the end users. Design options must be more than just commercially viable to be included in the DOE analysis. The following are excluded from the DOE analysis based on the initial screening process.

i. Flue Damper (Buoyancy Operated)
ii. Submerged Combustion
iii. Directly Fired
iv. Condensing Option
v. Condensing Pulse Combustion
vi. Advanced Forms of Insulation
vii. U-Tube Flue
viii. Thermophotovoltaic and Thermoelectronic Generators
ix. Reduced Burner Size (Slow Recovery)
x. Heat Pump Water Heater Options
xi. Timer Controlled
xii. System Application Options
xiii. Sediment Removal Features
xiv. Two-Phase Thermosiphon (TPTS) Design
xv. Air-Atomized Burner (Oil-Fired Only)

Of specific interest of the excluded design options that will be investigated further are the *Heat Pump Water Heater* and the *System Application Options*.

2.2.13.1 System Application Options

DOE recognizes three system application options that are excluded from the analysis on the grounds that they are each installation options that are independent of the water heater design. Each of these three application options pre-heat the mains water before entering the water heater, which reduces the temperature rise required and thus the energy required. *Solar Pre-Heat* uses solar collectors as pre-heaters for a standard electric or gas storage type water heater instead of relying on the solar collectors as the primary energy source. *Drain Water Heat Recovery System* uses a heat exchanger to recover waste heat from the drain becoming a pre-heater for a standard storage type water heater. *Tempering Tank* is an un-insulated storage tank plumbed in the water line before the water heater used to raise the inlet water temperature to the ambient temperature when installed in conditioned or semi-conditioned space.

2.2.13.2 Insulation

Because the useful hot water is always warmer than the environment, the hot water storage tank and pipes should be insulated to prevent the loss of heat to the surroundings. The amount of thermal resistance provided by the insulation is dependent on the type and the thickness. Typical tank insulation material is polyurethane foam that requires a blowing agent to expand and reduce thermal conductivity. Just as some aerosol propellants and refrigerants have been phased out in favor of less ozone-depleting chemicals, insulation-blowing agents such as HCFC–141b, have been recently eliminated from insulation manufacture. There are many blowing options available to manufacturers but only a few that meet the requirements of cost, thermal efficiency, processability, safety, and environmental impact. Because selecting a compound for use in mass manufacuture requires capital investment, companies must not only consider the costs and benefits but also potential future regulation when selecting a blowing compound. Current top choices summarized by DOE are water, cyclopentane, HFC–134a, and HFC–245fa or blends of these [DOE 2001a]. The last blowing agent, HFC-245fa, is a new and patented blowing agent that has only 3 percent greater thermal conductivity than the now banned HCFC-141b, but with the required zero Ozone Depleting Potential (ODP). Table 17 shows the thermal conductivity of the likely blowing agents along with

Table 17. Insulation blowing agents characteristics [DOE 2000a].

	Thermal Conductivity [1]				Cost [2]	
	$\frac{W}{m\,°K}$	$\frac{Btu\ in}{hr\ ft^2\ °F}$	Ozone Depletion Potential	Global Warming Potential	$/lb	Comment
HCFC-141b	0.0219	0.152	0.11	0.1400	$1.00	Current blowing agent [3]
HFC-245fa	0.0227	0.157	0.00	0.2400	$1.32	Not commercially available[3]
Water-Blown	0.0321	0.222	0.00	0.0003	$1.00	Used by (2) small mfrs [3]
HFC-356mfc	0.0240	0.166	0.00	0.2100	n/a	Similar to HFC-245fa; not yet available in the U.S. [3]
HFC-134a	0.0237	0.164	0.00	0.2400	$1.50	Limited solubility in polyols
Cyclopentane	0.0228	0.158	0.00	0.0030	$0.80	Flammable

[1] at 37.8 °C (100 °F).

[2] This cost covers the blowing agent and all other components of the insulation.

[3] As of 2001.

Although this new compound appears to be the best choice for making environmentally responsible insulation, the US Department Of Justice (DOJ) recommended against basing a standard on the patented blowing agent HFC–245fa, as it would restrict customers' choices and potentially increase consumer cost unnecessarily by reducing competition. DOE decided that this concern was unwarranted in part based on NIST's testing with various blowing agents. In this testing, NIST showed that even a large change in insulation effectiveness results in a small change in energy factor. NIST equated a 50 percent drop in insulation effectiveness to only a 0.06EF reduction in the energy factor [EERE 2003]. The tests performed by NIST were on electric storage water heaters with a typical thickness of 51mm (2.0 inches) of insulation. Commercially available water heaters have between one and three inches of insulation [DOE 2000a]. Figure 9 shows the Energy Factor results from NIST testing of electric water heaters outfitted with insulation with three blowing agents.

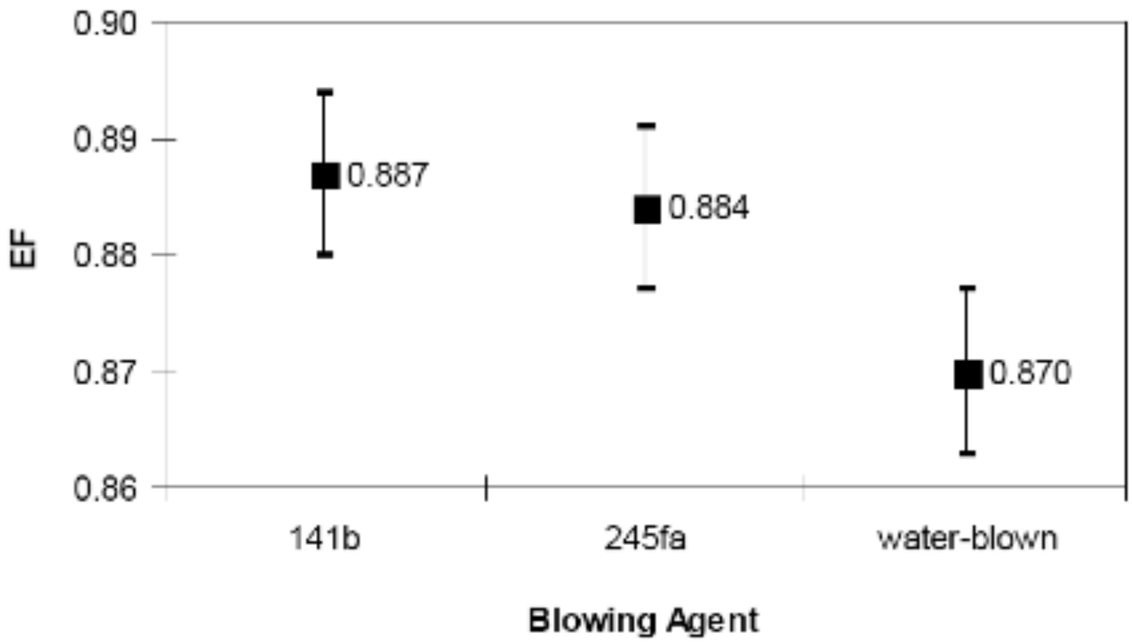


Figure 9. Average Energy Factor (EF) for DOE 24hr water heater test for three blowing agents [DOE 2000a].

While the insulation in most commercial water heaters is foam, the typical retrofit insulation is fiberglass. Pre-cut blankets are widely available commercially and can be found in thicknesses ranging from one to three inches with insulation values of R-3 per inch. Although most fiberglass insulation for water heaters has a vinyl covering, some insulation is available with a reflective covering to reduce the radiation heat loss. It is estimated that 2 inches of insulation will save about $25 a year on an 80-gallon tank, and with four inches, more than $30. These savings are reduced with a smaller tank. With a 40-gallon tank the annual savings are estimated $15 to $20 with two to three inches of insulation respectively [Paxton 2007]. Although there is no technical limit to the amount of insulation that can surround the storage tank, there is a practical limit that prevents much space from being used and economically there are diminishing returns. DOE reports that 27 percent of gas water heating households would need to remove the closet door for water heaters with 3 inch thick insulation [DOE 2001a].

Pipe insulation typically comes in two forms, foam (closed or open cell) and fiberglass. To reduce energy consumption hot water pipes should be insulated, reducing the heat loss during use and to keep the water in the pipes warm in between use. Even through there is no energy savings in doing so, cold water pipes are typically insulated to prevent condensation from forming on the outside of the pipes and dripping in the house. As the delivery pipes are site-specific and therefore not included in the DOE test specification, little attention is devoted to them.

2.2.13.3 Heat Traps

Heat traps prevent water in the pipes at the top of the water heater from leaving the tank due to its buoyancy, where it can cool and return to the tank. In this way the heat trap can reduce the stand-by heat loss inherent in the thermal store. DOE defines a heat trap as,

> *Heat Trap* means a device which can be integrally connected or

independently attached to the hot and/or cold water pipe connections of a water heater such that the device will develop a thermal or mechanical seal to minimize the recirculation of water due to thermal convection between the water heater tank and its connecting pipes.

DOE performed computer simulations and tests at NIST with heat traps and pipe insulation and found EF improvements of 0.005 and 0.007 respectively [DOE 2001a]. Although these are small energy efficiency improvements and the heat traps are typically required to be fitted at the time of installation, which at the discretion of the installer, may not be installed at all, this design option was included in the DOE TSD analysis [DOE 2000a]. AGA reported that manufacturers would use the heat traps in order to achieve the 0.01EF improvement required by the new federal standards [DOE 2001a].

2.2.13.4 Stratification

Heat can be more easily rejected to a cold sink (the environment) than it can to a hot sink. This is in part why an air conditioner works harder, i.e. requires more electricity for the same amount of cooling reducing its COP, in the peak heat of summer than on moderate days. The air conditioner has a greater cooling load due to the increased temperature in the building but the COP is also reduced. Stratification is a natural effect in fluids and gasses where there is a temperature difference between the upper and lower regions. In hot water storage tanks the more buoyant hot water naturally rises and collects at the top while the cooler water falls to the bottom. Thermal stores can encourage stratification by limiting mixing of volumes, minimizing internal convection, minimizing the dead water volume, and minimizing heat loss from the tank. Figure 10 shows the minimized dead water volume by positioning inlet and outlet near the vertical extents of the thermal storage tank.

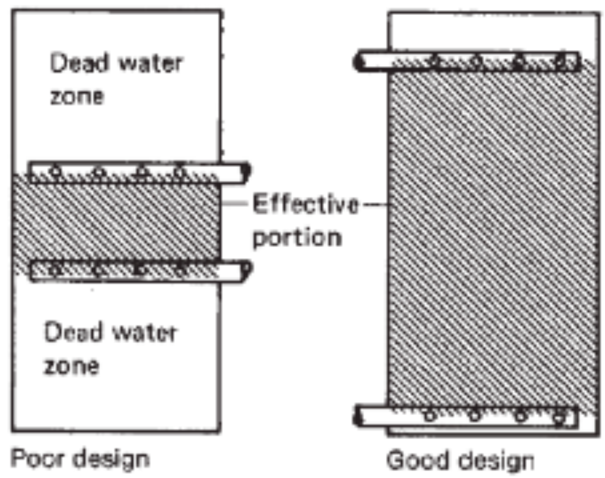


Figure 10. Position of inlet and outlet, effective quantity of water (hatched regions), for a thermally stratified TES [Rosen 2004].

Storage tanks are typically greater in height than they are in diameter to encourage stratification. Devices and strategies can also be employed to promote stratification to a greater extent. Renewable energy systems especially benefit from stratification, increasing efficiency by as much as 15 percent with a 50°C temperature difference by selectively heating the coldest water [Jensen 1984]. With hot water storage temperatures of 50-60°C and minimum inlet water temperatures of 5°C, this represents the maximum temperature difference that could be achieved in most domestic water heater applications. In addition to tank volume and tank aspect ratio, other important factors in determining the level of stratification in a tank are the water inlet and outlet geometry. Special devices inside the tank can improve thermal performance by maintaining stratification by minimizing mixing while adding make-up water for a conventional DHW arrangement or while circulating water in and out of the tank for a solar water heater. Figure 11 shows three different water inlet geometries tested, with test results in Table 18 with the best combination being the tall tank (H/D=2), slower draw rates (5 L/min), and the slotted inlet geometry. Andersen et al. investigated several stratification enhancing designs including a fabric inlet tube and two rigid vertical pipes, one with only two one-way openings and the other with many circular openings in the laboratory and numerical [Andersen 2007]. These devices use buoyancy to minimize mixing by introducing incoming water into a thermal layer of the stratified tank that is at the same temperature as the incoming water. The theory can be clearly seen in the computational fluid dynamics (CFD) results of the rigid vertical pipe with circular openings in Figure 12 [Andersen 2007].

Table 18. Effect of inlet design on DHW performance measures [Hegazy 2007].

	Inlet design					
	Wedged		Perforated		Slotted	
Electrical water heater	Tank A, $V_{st} = 50$ L, $H/D = 1$, $H = 40$ cm					
Draw rate (L/min)	5	10	5	10	5	10
Discharge efficiency (%)	82.4	75.9	86.2	79.8	91.3	86.4
Extraction efficiency (%)	56.3	49.1	68.7	62.4	81.2	74.7
Heat recoverable, F (%)	61.4	53.3	73.2	64.5	82.6	79.3
Electrical water heater	Tank B, $V_{st} = 50$ L, $H/D = 2$, $H = 63.4$ cm					
Draw rate (L/min)	5	10	5	10	5	10
Discharge Efficiency %	91.3	85.2	92.2	88.2	93.2	90.6
Extraction efficiency %	79.9	66.4	81.3	77.7	85.5	82.5
Heat recoverable, F (%)	82.4	69.5	83.1	79.1	86.5	83.2

For *fully mixed* tank: $\eta_{dis} = 63.2\%$, $\eta_{extrac} = 10.5\%$, $F = 20\%$.

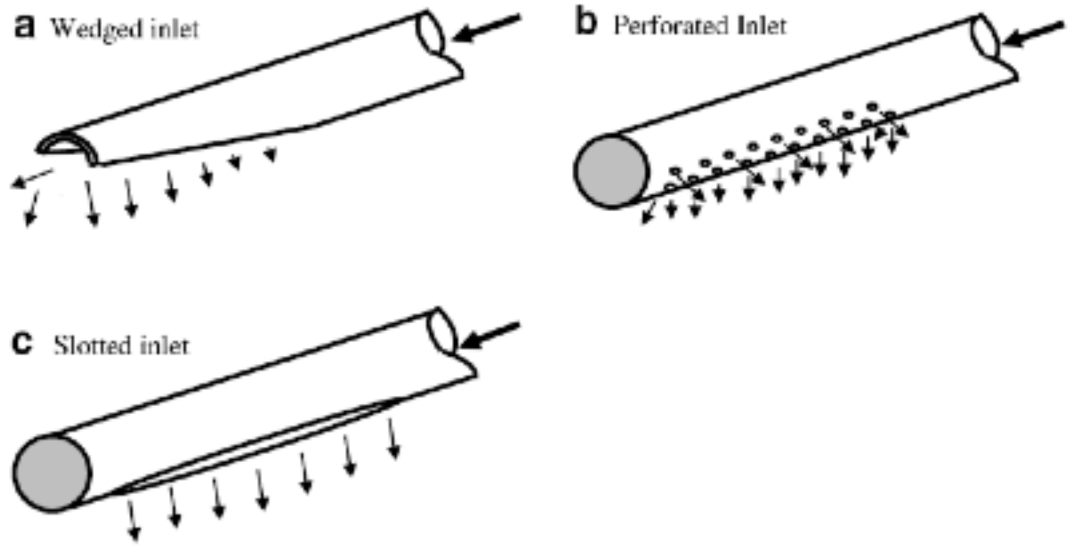


Figure 11. Schematics of inlet designs tested in Table 18.

The figure shows the axis-symmetric (2D) CFD model with warm water flowing up the center (red on right side), only leaving the vertical inlet tube radially once it encounters similarly buoyant (similar temperature) water.

Numerical methods commonly used to simulate stratification in a storage tank include Finite Volume Method and Multi-Node. The Finite Volume Method divides the storage tank into a discrete number of non-overlapping control volumes commonly known as elements. Three-dimensional temperature variation can be accounted for with the Finite Volume Method. Energy and continuum equations for each element are solved simultaneously for simple problems or iteratively for more complex problems. As the number of elements increases, accuracy typically increases but required computing resources necessarily increases. Multi-Node model is achieved by dividing the tank in a number of cross-sectional segments or nodes and performing energy balance on each node. This is a one-dimensional simplification of the Finite Volume Method where temperature can only vary in the vertical direction. Each segment is comprised of uniform temperature fluid. In *plug flow*, the model keeps track of temperature, size, and position of segments of fluid that move through the tank in axial plug flow. A specific type of Multi-Node model is the Three-Zone Temperature-Distribution Model. In the general three-zone temperature-distribution model, there are three horizontal zones where the temperature varies linearly within each zone, and continuously across each zone. This model calculates temperatures and energies within 0.4% of actual values and is considered suitable for engineering analysis and design due to the balanced trade-off of precision results and computational effort [Rosen 2004]. Close approximation of the stratification distribution is achieved because of the upper and lower zones vary in temperature only slightly with the middle zone varying substantially. Figure 13 shows the temperature distributions for a number of models including the continuous-linear model that is equivalent to the three-zone model.

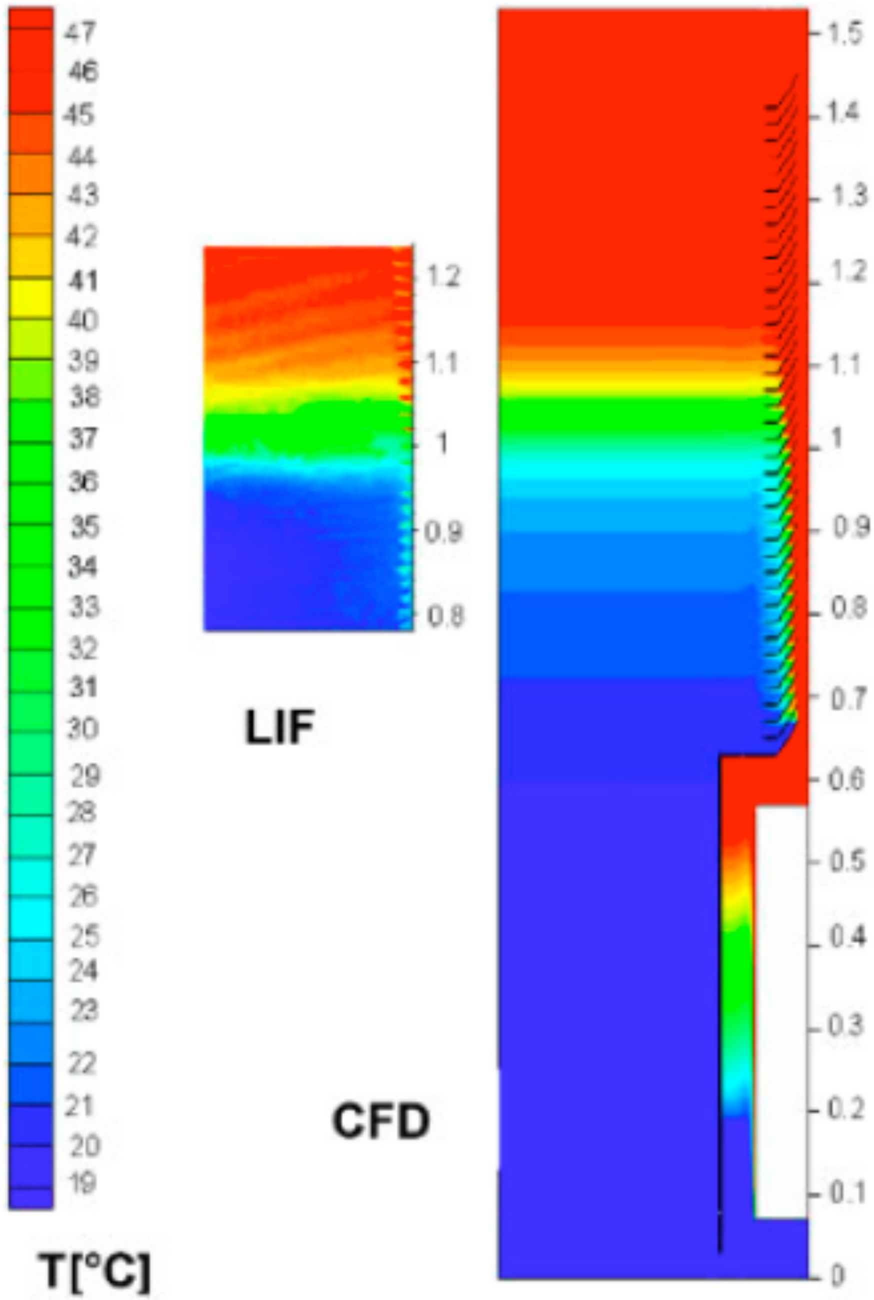


Figure 12. Circulation of hot water with a stratification enhancing device [Andersen 2007].

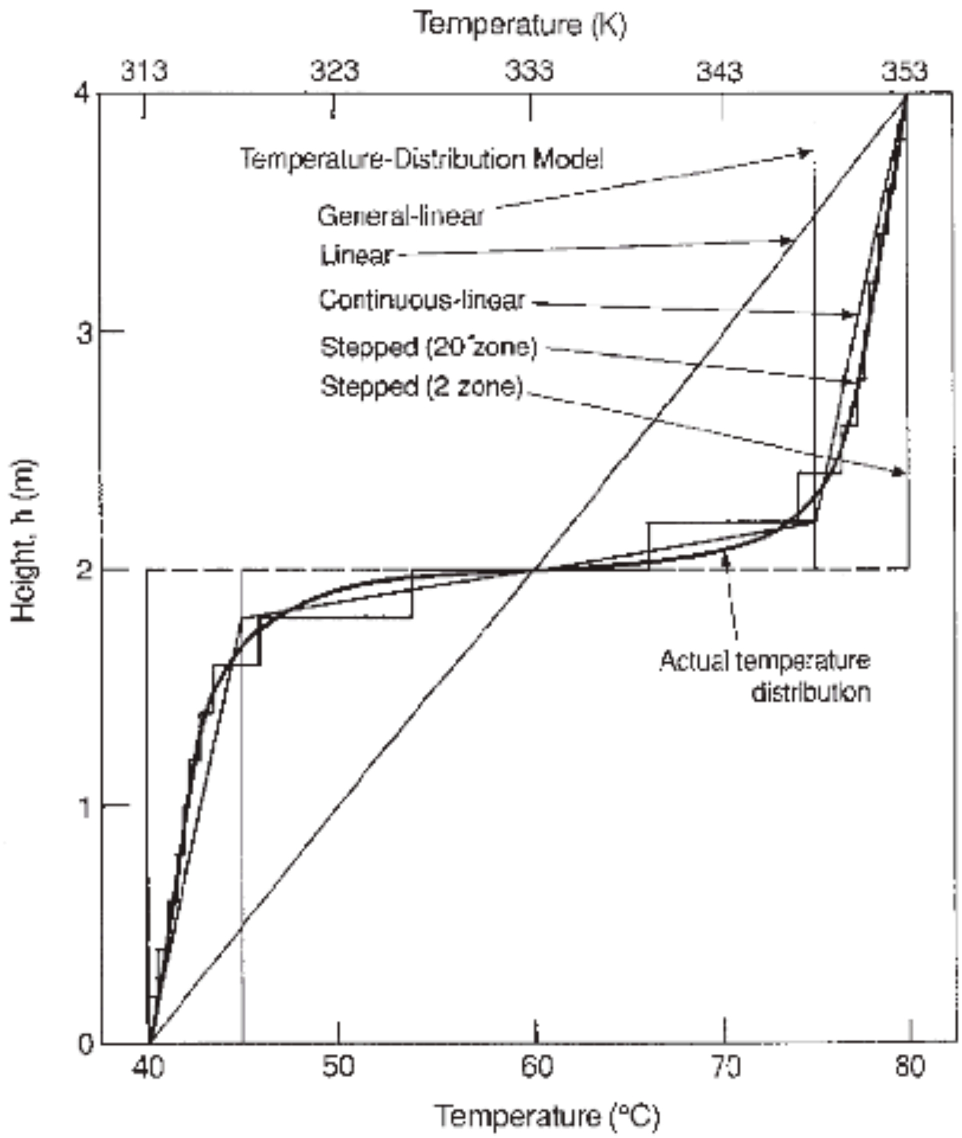


Figure 13. Approximate models of the vertically stratified temperature distribution and the shown continuous-linear distribution that is equivalent to the three-zone distribution model [Rosen 2004].

2.2.14 Renewable Energy

In *Achieving Total Domestic Hot Water Production with Renewable Energy* [Biaoua 2007], the authors investigate four options for total domestic water heating with renewable energy, using photovoltaic panels to provide electrical power. In this comparison of four alternatives for hot water production, thermal solar collectors had the lowest make-up energy requirement with the other alternatives, heat pump water heater, ground source heat pump (GSHP) De-superheater, and electrical resistance in order of increasing make-up energy [Biaoua 2007]. The lowest overall cost was a solar thermal system backed up with electric resistance heat from PV panels. Without the renewable energy requirement we saw earlier that a solar thermal system had a cost over twice the least expensive option, which was an indirect water heater paired with efficient gas or oil boiler [ACEEE 2007]. DOE estimates the amortized cost over the life of a residential grid-connected PV system at $0.25 per kWh making it two to three times more expensive than typical utility supplied power [EERE 2003].

2.2.14.1 Solar assisted heat pump water heater

Solar-assisted heat-pump water heaters increase the coefficient of performance (COP) by increasing the heat gain of the condenser by exposing it to direct sunlight and making it also a collector. Although a distinction is sometimes made between Integral-Type Solar-Assisted Heat Pumps (ISAHP) that are built into a single unit and the more general Solar-Assisted Air-Source Heat Pumps (SASHP or SAHP) that may have more than one interconnected component, for the purpose of this review they are considered the same and are referred to as a SASHP. The COP is dependent on the ambient temperature, inlet water temperature, final water temperature, and amount of solar radiation the condenser receives. Simulated yearly average COP for a SASHP water heater of 4 was reported by Guoying et al. [Guoying 2006]. This is a significant improvement of a regular heat pump water heater's COP of 2.0 to 2.2 [Energy Star

2007b, ACEEE 2007]. Table 19 summarizes air source and solar assisted heat pump COP for a range of operating conditions.

Table 19. Experimental coefficient of performance (COP) values.

	Water Delta T °C	Winter		Spring/Autumn		Summer	
		Amb.T °C	COP	Amb. T °C	COP	Amb. T °C	COP
Air source heat pump water heater (ASHP)							
Applied Thermal Engineering 27 (2007) 1029-1035	40	0	2.61	25	4.817	35	5.66
Solar-assisted air-source heat pump water heater (SASHP)							
Applied Thermal Engineering 26 (2006) 1257-1265	20						4.69
	40			15	4.32		
	50	5	3.83 [1]				
	50	5	3.3 [2]				
Solar Energy 74 (2003) 33-44	42.2	15	1.7	20		25	2.5

[1] Sunny winter day
[2] Overcast winter day

Since the COP is dependent on so many factors, Huang et al. proposed evaluating solar-assisted heat pumps at a standard temperature difference of,

$$T_f - T_{a,ave} = 15°C$$

where T_f is equal to the daily mean water temperature and $T_{a,ave}$ is equal to the average daily air temperature [Rosen 2004]. Based on long-term outdoor field testing of four SASHP, COP was found to be proportional to this temperature difference and so proposed by the authors that it be used as the performance comparison for SASHP's. The authors summarize the general function of a SASHP and review basic trends in operation to explain why the simple temperature difference is appropriate as a performance indicator. As is common with solar thermal collectors, increasing the water temperature decreases the COP and thermal efficiency of the collector. So with colder intake water as would be common in winter, the SASHP must raise the water temperature more but the condenser/collector has the greatest thermal efficiency. Conversely, in the summer when intake water is warmer, the condenser/collector is less thermally efficient but less temperature increase is required. Figure 14 shows the COP decreasing with linearly with an increase in the temperature difference for four of the long-term SASHP water heaters.

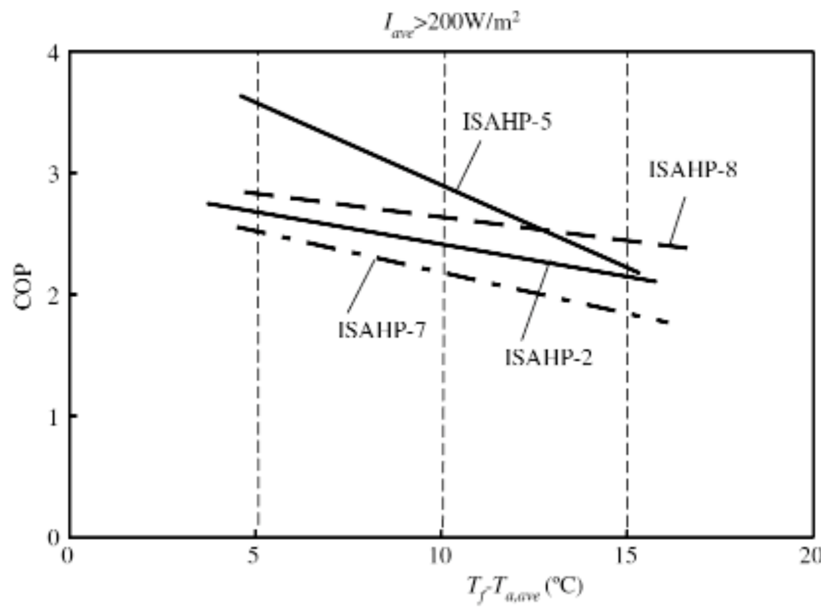


Figure 14. Coefficient of Performance (COP) for Solar-Assisted Air-Source Heat Pump Water Heaters for various temperature differences [Rose 2004].

2.2.14.2 Solar

Solar Thermal Systems can be divided into two types based on their collectors, non-concentrating and concentrating. A very common type of non-concentrating collector is a flat-plate collector. This is an insulated box with a black metal sheet with built-in pipes. Rigid foam insulation insulates the back and sides against heat loss with insulated glazing on the front to allow solar radiation through. At the bottom of the collector a single pipe feeds a manifold of smaller vertical pipes that increase the surface area of the water inside the collector. The metal sheet is attached to the pipes to increase the surface area even further without reducing the cross-sectional flow path of the water. In commercial units the black metal sheet has been carefully designed as a selective surface. Special attention is paid to the material and surface of the black metal sheet to increase the solar absorption (alpha) and to reduce the infrared emission (epsilon). Typical values for a selective surface are 0.90 for absorption and 0.10 for emissivity. An increase in efficiency over the flat plate can be accomplished by using evacuated tubes. Either inside of an enclosure or exposed, evacuated tube collectors increase the insulative value by creating a partial vacuum between the air and the water that is being heated. Concentrating technologies such as a Parabolic Trough, Parabolic Dish, a Power Tower can create very high temperatures but are almost completely excluded from residential installations because the high temperatures are not required and the size and cost is higher than with the non-concentration systems.

2.3 Cogeneration

Cogeneration in general, describes systems that are capable of utilizing more than just the primary output in order to increase the overall efficiency of the system. The most common application is combined heat and power (CHP), where thermal power is co-generated with electricity, the primary output. The term trigeneration is an extension of cogeneration and is most often used to describe a combined heating, cooling, and power

(CHCP) process. No energy conversion process is perfectly efficient with heat inevitably being generated as a waste by-product. Here the term *waste* is considered to be anything that is created but that is not desired. In large-scale utility electrical power generation, thermal power generation is an undesired by-product that requires the construction of cooling towers or a site near a large body of water to dissipate the exhaust heat. If a by-product provides a benefit and can be utilized, it is no longer considered to be waste. In power plants, heat is the most obvious "waste" by-product to capture and put to good use, but industrial facilities often have chemical by-products in addition to heat that can be captured and sold as a commodity. Steam heat or hot water is commonly co-generated with the electrical power in a utility scale power plant to increase the overall efficiency by offsetting auxiliary heating bills and can be used on-site or off-site for heating of local residences in what is known as district heating. Figure 15 compares the overall efficiency of conventional power generation with separate heat generation to cogenerated heat and power. The example shown in Figure 15 is a representative example of separately generated heat and power, clearly specific applications may require different ratios of heat and power which will change the overall efficiency. For cogeneration to be effective the by-products must be useful. To be useful they must be among other things,

Close in proximity to end use (short distance)

Reliable in quantity and quality (high or low temperature)

Match the load profile (daily / annual)

Not be contaminated (clean water or chemical by-products)

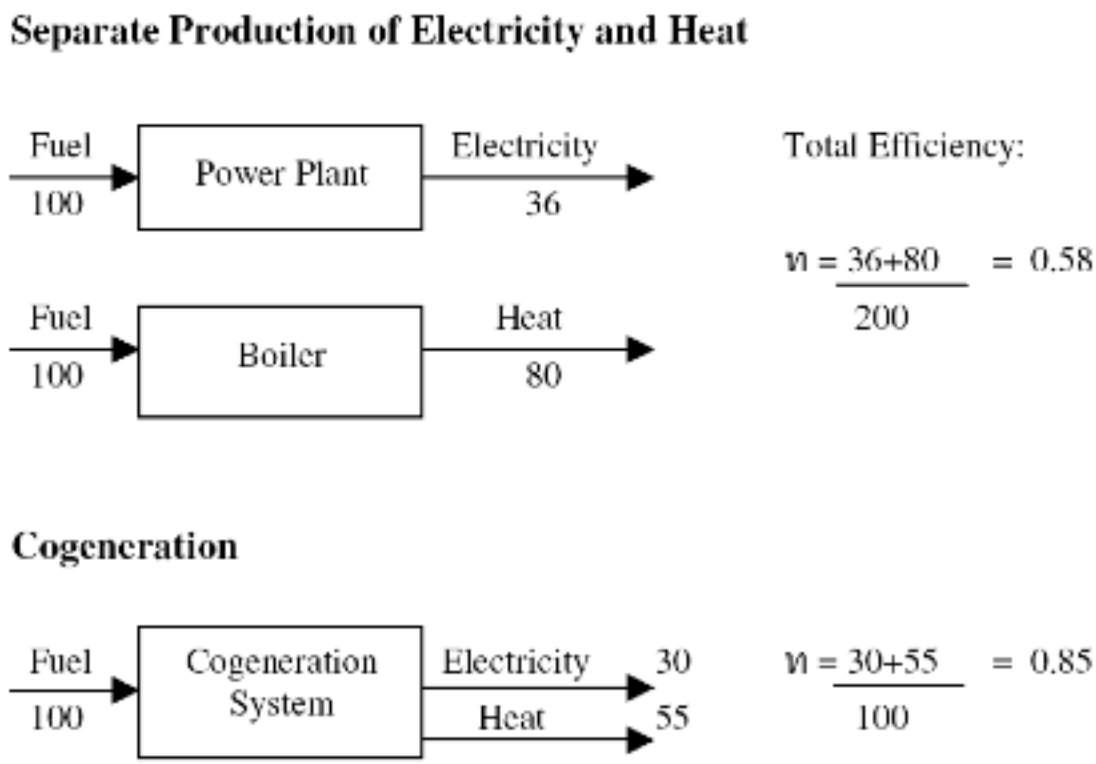


Figure 15. Conventional power and cogeneration compared.

For the analysis of cogeneration systems all products need to be taken in account including both the electricity and the heat. When the heat was part of the waste stream this was not necessary, but now the heat can reduce the amount of energy purchased or be sold to a second party, it is providing a tangible and measurable benefit and must be accounted for. This is valid for both economic and environmental analysis, with measures such as cost per kWh ($/kWh) of electricity or heat, or ton of greenhouse gas per kWh (CO_2/kWh). While cogeneration can be financially profitable, it typically excels in environmental analysis because little to no fuel is consumed to generate and utilize the by-products such as heat. The capturing of by-products does require investment in specialized equipment, but if capital is available for investment it can be used to offset the purchase of energy.

2.3.1 Distributed Cogeneration

In utility scale electrical generation and cogeneration, there are losses associated with the transmission and distribution of both the electricity and the steam heat or hot water. The electrical losses are due to resistance in the high-voltage transmission lines and in the step-up and step-down transformers and the thermal losses are due to heat transfer to the environment. By moving the generation site closer to the customer, in what is known as *Distributed Generation*, some of these losses can be eliminated. In distributed generation, the central power generation facilities are replaced or enhanced by adding multiple smaller generation facilities closer to the customers. Although the multiple smaller generation sites may have lower efficiency in electrical generation due to the loss of economies-of-scale, the overall efficiency is higher without transmission losses and even higher yet when the thermal power can be utilized. Figure 16 shows a comparison between utility generated electricity with 30% efficiency and an on-site (distributed) electricity and heat cogeneration facility with 90% efficiency.

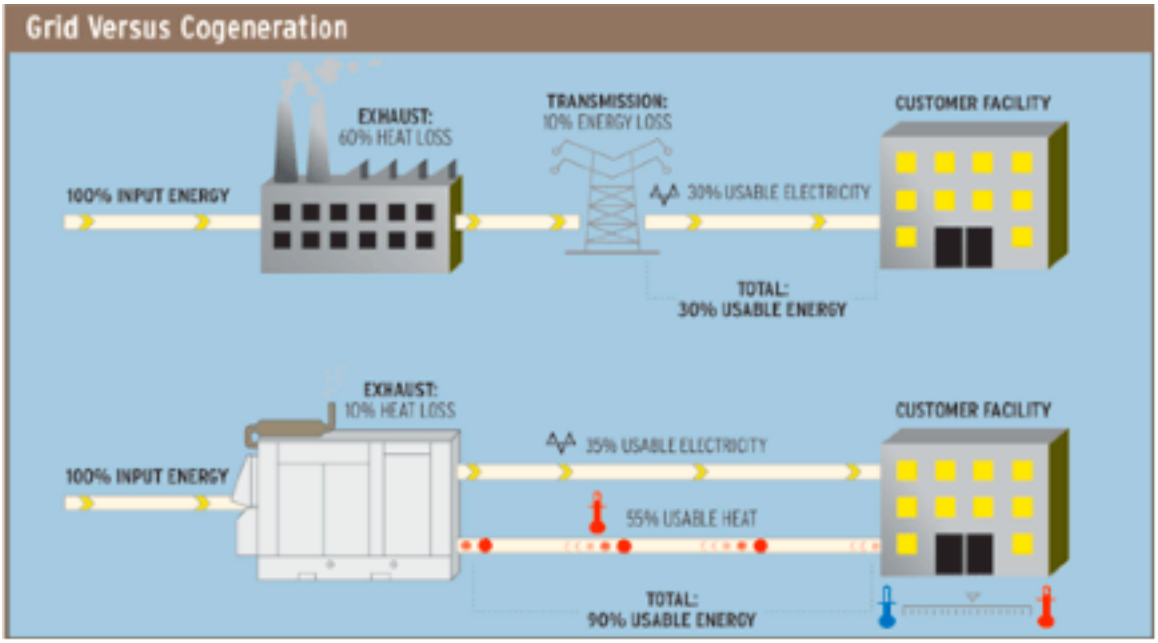


Figure 16. Efficiency of grid generated electricity compared to on-site cogeneration [Leposky 2003].

Although the co-generated thermal power can be most easily utilized directly in heating applications with a heat exchanger, it can also be indirectly utilized to create a cooling or refrigeration effect when paired with an absorption chiller as shown in Figure 17.

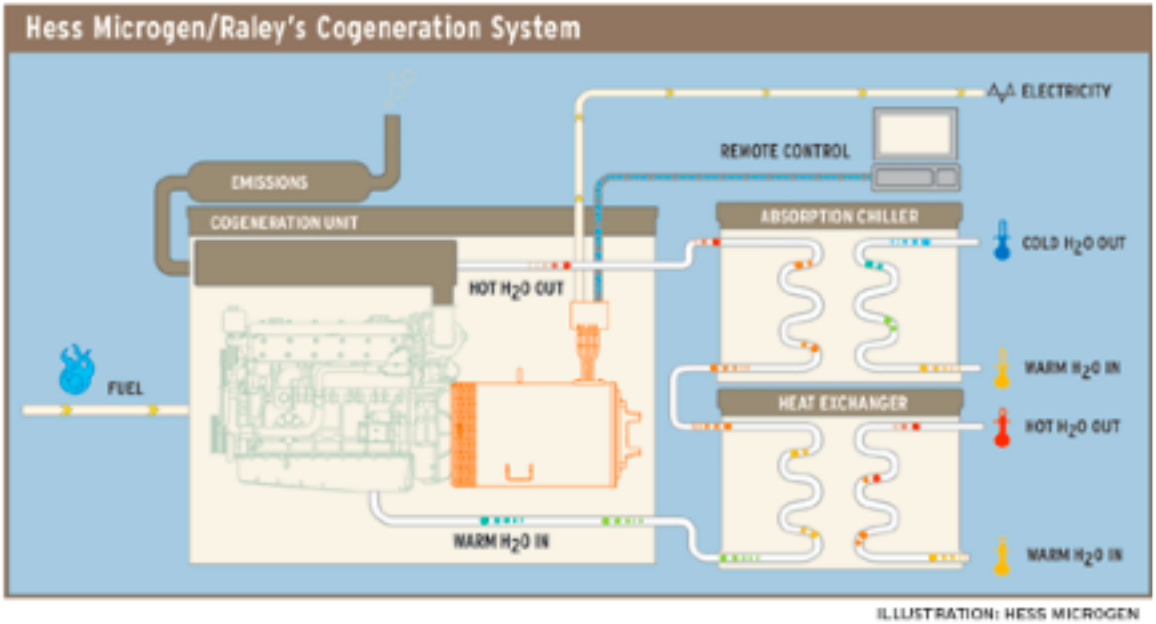


Figure 17. Cogeneration system from grocery store creates electricity, chilled water, and hot water [Leposky 2003].

One cogeneration system that is commercially available is the Polar Power Micro-Cogeneration unit, with a retail price of $9,995 it can be configured to provide either heating or cooling along with electrical generation and hot water,

Cooling Mode	**Heating Mode**
34,000 Btu/hr of air conditioning	36,000 Btu/hr of space heating
Up to,	Up to,
6 kW of power for battery charging &	6 kW of power for battery charging &
30,000 Btu/hr for heating hot water	30,000 Btu/hr for heating hot water

2.3.2 Refrigeration Heat Reclaim (RHR)

Refrigeration Heat Reclaim (RHR) systems capture heat that would normally be rejected from a vapor-compression refrigeration system and apply it to heating water. Any type of vapor-compression refrigeration system can be used, such as food refrigeration or air conditioning. The chief drawback to the RHR system is that heat is only generated when the refrigeration system is operating. This can be overcome by optimally pairing usage or by storing the heat for offset refrigeration and water heating loads. An office building provides a preferable environment where air conditioning and hot water are only required during the day. Even with demand generally coinciding during the daylight hours, the actual load profiles are not matched for hot water has a nearly continuous load with the air conditioning load peaking mid to early afternoon.

There are two types of RHR systems, *de-superheater* and *condensation*, the differentiation being in the refrigerant phase where the heat is removed. The de-superheater transfers the most readily available heat from the refrigerant in the super-heat region and returns saturated vapor to the refrigeration system. Condensation types transfer heat from the refrigerant primarily in the two-state region but can include the heat transfer from the super-heat and sub-cooled liquid regions as well. By including condensation of the refrigerant in the heat recovery, significantly more energy can be reclaimed. The amount of heat that can be reclaimed from 1lbs of a low temperature refrigerant, R404a, entering the heat reclaim coil at 180°F for a refrigeration system that is working on 120°F condensing pressure is almost five times more in a condensing RHR than a de-superheating RHR [Jaster 1990b].

by de-superheating with only sensible heat alone,

$$(180-120)°F * 1\,lb * \frac{0.365\,Btu}{°F\ \ lb} = 21.9\,Btu$$

by condensing with sensible and latent heat,

$$(180-120)°F * 1\,lb * \frac{0.365\,Btu}{°F\ \ lb} + 1\,lb * \frac{87\,Btu}{lb} = 108.9\,Btu$$

The two basic types of RHR can also be described by their configuration, where the heat reclaim coil is in series or in parallel with the normal condenser. With the heat reclaim coil in series with the normal condenser the system is acting as a de-superheater. Arranged in this series configuration the RHR can recover up to 50% of the total rejected heat [KeepRite 2003]. Limiting factors to recovering more energy are that the pressure drop must be kept at a minimum to ensure proper function through both condensers and that condensation should not occur. When the heat reclaim coil is in parallel with the normal condenser it acts as a condensing RHR and each coil is typically designed to be able to recover 100% of the rejected heat individually. Figure 18 shows a sample for a series condenser on the left and a parallel condenser with a control valve to switch from normal mode to recovery mode on the right.

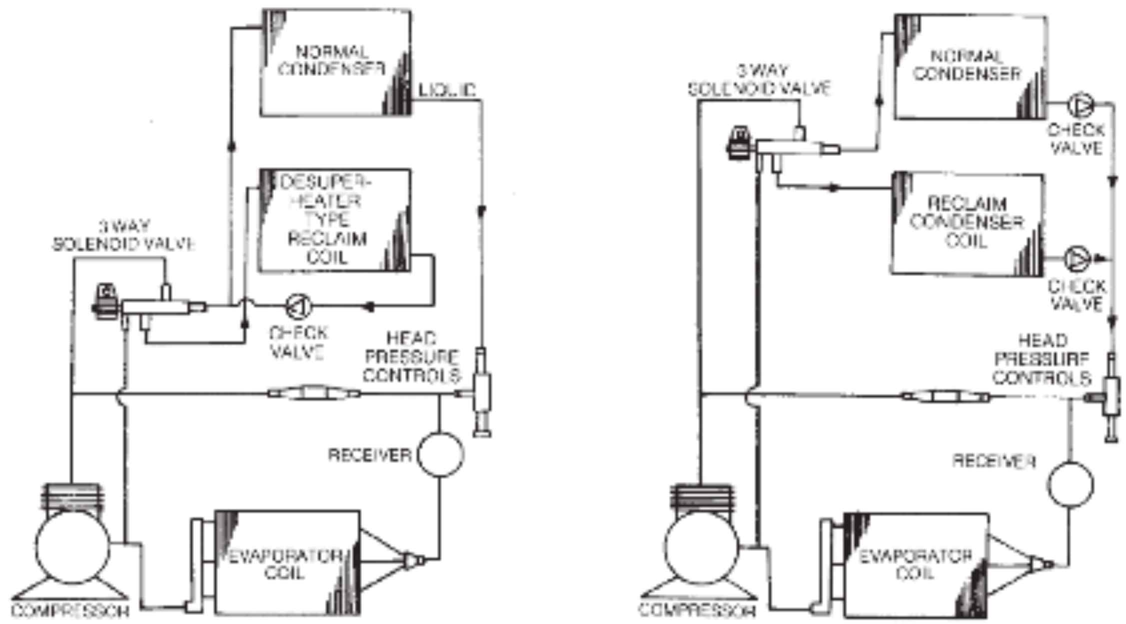


Figure 18. RHR System configurations, series condenser (left) and parallel condenser (right) [Jaster 1990b].

Low temperature or less efficient refrigeration systems are especially attractive for RHR as they release more heat per ton of cooling effect. The utility Mississippi Power states that under typical conditions a de-superheater can remove about 10 to 30% of the total heat that would have been rejected by the condenser [Mississippi Power]. Just as the size, efficiency, and condenser temperature will determine the amount of waste heat that is recoverable by a RHR, so too does the type of refrigerant. Refrigeration heat recovery from an HCFC-22 air-conditioning system will provide approximately 2.5 times the refrigeration effect, while an HFC-134a air-conditioning system will provide approximately 1.8 times the refrigeration effect [Mississippi Power]. If designed optimally and then run at the designed conditions, it could be reasoned that RHR systems can improve the host system efficiency, but in reality the demand for a high degree of reliability coupled with unknown operational variables such as weather necessitate designs that are built with extra capacity.

The majority of RHR examples are found in industrial or commercial applications such as ice rinks, food service, grocery stores, and restaurants because the energy costs from large amounts of refrigeration can justify investment in equipment to capture the waste heat. In the literature review one proposal for a residential system is found, dubbed the Home Energy Center (HEC) by the inventor. The authors of the paper test the HEC described as a combined refrigeration and water heating system for domestic use. "An Integrated Domestic Refrigerator and Hot Water System" [O'Brien 1998]. In the design of the HEC, an additional condenser, referred to as the *primary* condenser, is placed in a heat sink to reclaim heat before the refrigerant enters the *secondary* condenser attached to the refrigerator. The secondary condenser rejects the remaining heat to the environment that could not be reclaimed by the primary condenser. Reclaimed heat is transferred from the heat sink to the water heater storage tank by the heat recovery coil. In tests the HEC does provide benefit although not as much as was hoped for. In practice, the heat sink does not transfer energy to the water heater storage tank because the heat sink remains at a lower temperature than the storage tank, but the heat sink does reduce the heat loss of the storage tank by surrounding it with a warmer environment. Conversely, the heat sink is too warm for the refrigerant to reject too much heat to the heat sink. Figure 19 shows the configuration of the HEC with the polyethylene tank encompassing the compressor, primary (HEC) condenser, water heater storage tank, and heat recovery coil.

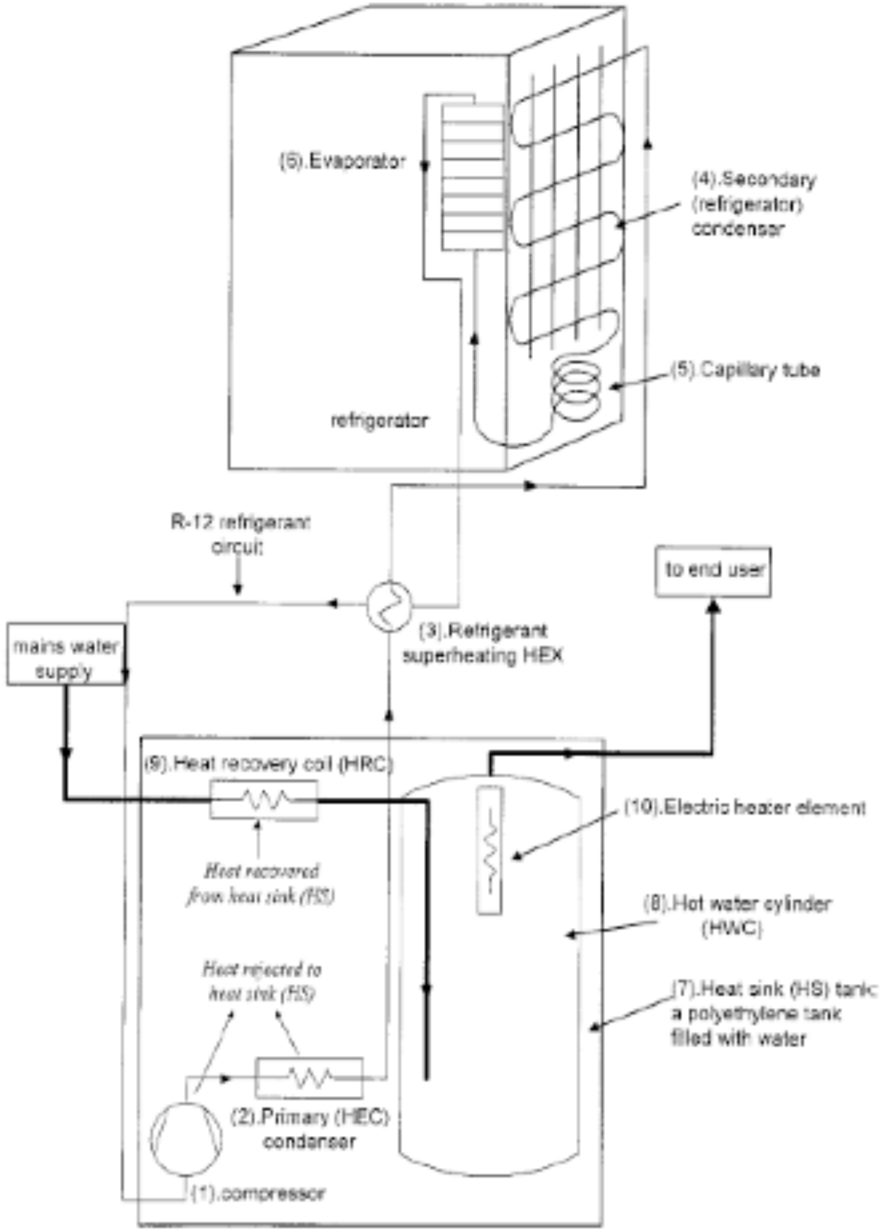


Figure 19. Configuration of the Home Energy Center (HEC).

2.3.3 Combisystems

Combisystems are so named because they have space and water heating combined into one central unit. Although not technically cogeneration since thermal power is the only power output, combisystems improve overall efficiency by utilizing economies-of-scale and combining similar energy conversions. Referring back to Table 16 of water heater life-cycle costs (LCC), the indirect water heater with an efficient boiler had the lowest LCC. This type of system has several characteristics that increase the energy efficiency, including low temperature heating and thermal mass in the water/steam used for heating. These characteristics also make it a good candidate for the use of renewable energy as a solar combisystem does. Several investigations into solar combisystems are underway and have been completed by the International Energy Agency (IEA), including Task 26, which from 2001 to 2003 monitored 200 installed solar combisystems thought Europe [IEA 2003b]. These combined space and water heating residential systems gathered between 10 and 100% of their energy from the sun. Much effort has been put into making these systems less complex, smaller, and less expensive while retaining the functionality and efficiency. Figure 20 shows an example of a single tank combisystem with heat exchangers for solar input on the bottom, space heating in the middle, and hot water on the top.

Figure 20. Sample solar combisystem [IEA 2003b].

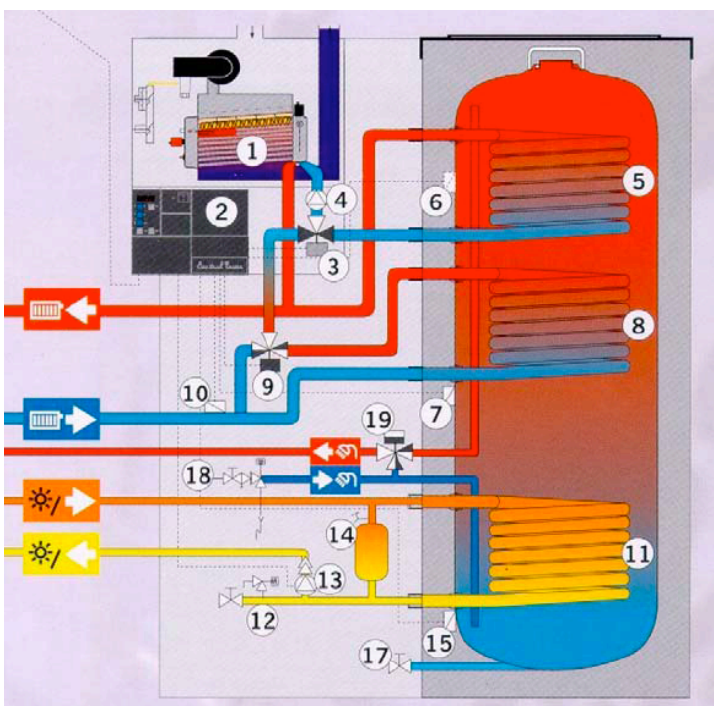


Figure 21. Sample solar combisystem schematic [IEA 2003b].

The main components of this combisystem, shown in Figure 21, are the auxiliary heater, thermal store, heat exchangers, and control equipment. The auxiliary heater is comprised of (1) stainless steel heat exchanger, (2) control management system, and (4) system pump. Other main components of the heat store, (5) hot water heat exchanger, (8) space heating heat exchanger, (11) collector loop heat exchanger, (13) collector pump, (18) cold water inlet [IEA 2003b].

It is important to clearly define the efficiency of solar combisystems and solar thermal systems in general because they often use fuel with cost (electricity or gas) and without cost (solar). Typical thermal efficiency is defined as,

$$\eta = \frac{Q_{out}}{Q_{in}}$$

where Q_{in} is the sum of all energy consumed by the system and Q_{out} is the output energy in the useful heat. In the example of a water heater, Q_{in} is the electricity or gas in and Q_{out} is the hot water out. For a heat pump water heater, the term coefficient of performance (COP) is used because the amount of useful heat energy out of the system is often more than the energy in. The COP is greater than one because only the energy that must be paid for (electricity or gas) is accounted for on the *input* side. Thermal energy that is extracted from the ambient air for an air-source heat pump or from the ground for a ground-source heat pump is typically not accounted for. These heat sources are typically designed to be interacted with in a way that they can be thought of as constant temperature sources. Just as the ground can provide as much heat energy as we would like to extract, so to can the sun as long as enough ground volume or solar surface area is provided. Just as heat pumps use COP and air conditioners use *Energy Efficiency Ratio* (EER) and *Seasonal Energy Efficiency Ratio* (SEER), solar combisystems must be evaluated based on several different but not unique measures. In particular there are two useful measures of efficiency for solar thermal systems like solar combisystems. The two efficiencies are the fuel specific efficiency and the system overall efficiency defined as,

$$\eta_{fuel} = \frac{Q_{space\ heating} + Q_{DHW}}{Q_{fuel}} \qquad \eta_{overall} = \frac{Q_{space\ heating} + Q_{DHW}}{Q_{solar} + Q_{fuel}}$$

It can be seen that the fuel specific efficiency, η_{fuel}, is based on the input fuel for the auxiliary heater and neglects the solar input contribution. The second, the system overall efficiency, includes the heat energy from the sun and the fuel on the input side. As the fraction of solar energy input increases, both the fuel and overall efficiency increases because low temperature solar heat is replacing wood, oil, or natural gas combustion, but only up to a point. As the solar fraction continues to increase, so to does the amount of system loss, resulting in an overall system efficiency decrease but a fuel efficiency increase. This overall system efficiency reduction comes exclusively from the solar energy component. Increasing amounts of solar energy are captured which reduces the auxiliary fuel requirement, but at the same time thermal losses in the storage tank are increasing from the excess solar energy captured. If the solar thermal components are sufficiently inexpensive, then the reduction in efficiency does not have a negative result.

2.3.3 Domestic Cogeneration

Historically there has been little commercial interest in domestic cogeneration due to low energy cost, added cost and complexity of cogeneration systems, and small amounts of waste heat available to reclaim (in comparison to industrial scale systems). Efficiency improvements in modern appliances to meet Department of Energy (DOE) standards have further reduced the amount of waste heat available to reclaim and forced domestic cogeneration operate on smaller margins. This trend to increase energy efficiency however has however, been offset by the greater energy demands of modern homes including more and larger appliances.

To develop a domestic cogeneration configuration it is reasonable to look at the top domestic energy consumption end uses and attempt to match complimentary services. Space heating has the highest fraction of the residential energy consumption but is a winter seasonal load. Domestic hot water (DHW) is a semi-continuous load with low to moderate peaks for a typical storage tank style heater. Tankless water heaters are more efficient, requiring less energy over a year, but have peak loads that are three to five times greater. Space cooling load is characterized by summer seasonal loading with high peaks in the early afternoon corresponding with the day high temperatures outside. Domestic refrigeration is a near continuous cooling load with low to moderate peaks as long as the refrigeration appliance is in a conditioned space. If the appliance is not in conditioned space, but subjected to the outside weather, the peak loading can be high and will coincide with the peak load of the space cooling. Table 20 summarizes the profile, amount of energy consumed, and the maximum amount of energy that can be recovered based on condenser heat rejection.

Table 20. Summary of domestic energy system appliances with heat recovery potentials.

	Space Heating	Hot Water	Space Cooling	Refrigeration
Annual profile	Seasonal (Winter)	Semi-continuous	Seasonal (Summer)	Semi-continuous
Daily profile	Semi-continuous	Intermittent	Mid-day peak	Semi-continuous
Energy consumed (kWh/yr) [1]	12,866	4,631	2,263	1,465
Efficiency	90%	65%	SEER 12 [2]	COP 2.4 [3]
Energy available for recovery (kWh/yr) [4]	N/A	N/A	6,632 [5]	6,047 [6]
[1] RECS 2001 [EIA 2004]				
[2] [KSU 2003]				
[3] [Hepbasli 2007]				
[4] Condenser heat rejection				
[5] 100kWh of electricity required for 1MBtu cooling for SEER of 12 [KSU 2003], results in 22.63MBtu cooling from RECS 2001 annual energy consumption [EIA 2004]. Assuming 1 to 1 cooling to heat rejection based on steady state experimental data [Techarungpaisan 2007].				
[6] Based on COP from experimental test [Hepbasli 2007] and RECS 2001 annual energy consumption [EIA 2004].				

Although space heating has the greatest energy requirement and space cooling has the greatest heat rejection, these two loads would not typically overlap and would therefore require low temperature thermal storage to utilize the waste heat. Water heating is better suited than space heating for accepting waste heat since it has a nearly continuous annual heating load. Both refrigeration and space cooling have annual condenser heat rejection values that exceed the annual energy consumption of the water heater by 1.30 and 1.43 times respectively. Either of these waste heat sources could provide a significant portion of the thermal energy consumed by the water heater. Based on Table 20, it can be seen that hot water production and refrigeration loads coincide favorably of the top residential energy end-uses.

2.4 Engineering Economics

Scarcity is noted as a fundamental condition of humanity, not just of modern society, resulting from having limited resources available to satisfy the unlimited wants and needs of humanity. The idea of scarcity sits at the root of economics and guides all economic analysis. We have a finite amount of resources since we can only either have access to or use so many resources at any one time. Another fundamental of economics is that we all want to maximize the benefits we receive, the result of satisfying a want or need. Since we have a finite amount of resources and we want to maximize our benefits, we also want to minimize the cost or amount of resources required to achieve those benefits. This becomes an important consideration when choosing between different options, how much benefit will we receive at what cost. Options, all of which it is assumed have a cost, either in real dollars or in opportunity cost, which is the sacrifice of not doing something else. It is important to note that the option of *doing nothing* should be considered when applicable, as the best option may be doing as little as possible at a certain time.

There are a number of economic methods that can be used to compare two or more options including *benefit-cost ratio, life-cycle cost* (LCC)*, rate of return* (ROR)*,* and *payback period* (PBP). Because one dollar today does not (typically) have the same value as one dollar a year from now, the *time value of money* should be accounted for when the analysis time is longer than one year in duration. *Nominal* dollars are not adjusted for the time value of money where as *real* dollars are adjusted for the time value of money and as such must be specified with respect to a specific fixed time. To calculate the time value of money, the interest rate and the inflation rate must be known. The interest rate is the rate at which loaned money is paid for or the fee for borrowing money and is also called the discount rate when calculating the present worth of future costs and benefits. The inflation rate is the rate at which the cost of goods and services increase over time. Both of these rates can and typically do vary over time, from week to week, or even hour to hour. The interest rate for simple financial structures such as bonds and certificates of deposit can be exactly known into the future where as complex financial structures such as stocks and the inflation rate can only be estimated from a combination of past information and current trends. Table 21 shows discount rate projections from January 2008 and December 2008 from the United States (U.S.) Office of Management and Budget (OMB), who's primary responsibility is overseeing the preparation of the federal budget as well as to assess competing funding demands among agencies and set funding priorities [Energy Star 2006, OMB 2008b]. The OMB is responsible for establishing financial projections such as the discount rate, which affects all monetary analyses.

Table 21. Discount rates from the U.S. Office of Management and Budget 2008-2009 [Energy Star 2006, OMB 2008a, OMB 2008b].

	3-Year	5-Year	7-Year	10-Year	20-Year	30-Year
2008 Nominal Interest Rate*	4.1	4.3	4.4	4.6	4.9	4.9
2009 Nominal Interest Rate*	2.7	3.3	3.7	4.2	4.7	4.5
2008 Real Interest Rate**	2.1	2.3	2.4	2.6	2.8	2.8
2009 Real Interest Rate**	0.9	1.6	1.9	2.4	2.9	2.7

* Nominal interest rates on treasury notes and bonds of specified maturities.
** Real interest rate on treasury notes and bonds of specified maturities, from which the inflation premium has been removed.

Discount rates used for economic planning can vary depending on the level of risk that is acceptable and depending on the type of investor or borrower as seen in Table 22.

Table 22. Discount Rates for Types of Costs, year 2004 [Rosenquist 2004].

Type of Cost	Discount Rate (%)	Basis for Rate
Residential equipment standards *	5.6	Opportunity cost for households of investment in energy efficiency
Commercial equipment standards **	6.1	Weighted cost of capital for typical commercial sector enterprises
* Based on weighted average of 1998 after-tax real interest rates.		
** Based on 1999-2001 interest rates.		

A *simple* payback period is so named because it does not take into consideration the time value of money and is defined as,

$$PBP_{simple} = \frac{Cost}{Savings}$$

where, *Cost* is the initial price and *Savings* is the net savings per year.

In order to compare two or more options with different cash flows over time, each cash flow must be converted to an equivalent cash flow at the same time. Although there are a number of ways to calculate equivalent cash flows of money, the OMB requires that net present worth be used for analysis, that is that all costs (disbursements) and benefits (receipts) be discounted and expressed in equivalent terms of current (present) dollars. The present worth of a future payment or receipt is,

$$P = F(1+i)^{-n}$$

where P is the present value, F is the future value, i is the interest or discount rate, and n is the number of interest periods. The present worth can also be calculated using tabulated compound interest factors where the nomenclature is,

$$P = F(P/F, i, n)$$

where $(P/F, i, n)$ is the factor to find present value (P) given future value (F). Life cycle cost (LCC) is the same as net present value in that they are both the sum of present values. LCC typically places emphasis on the entire life cycle including research and development costs (R&D), maintenance costs, and disposal costs in addition to initial purchase and operation costs. Other common methods to compare two or more options include equivalent cash flow for some time in the future (net future worth) and yearly average cost (annual cost).

The U.S. Department of Energy (DOE) performed a life-cycle cost (LCC) and payback period (PBP) analysis to help determine whether the operating cost savings of potential new standards for refrigerator-freezers are sufficient to justify the higher purchase price. Increasing the energy efficiency of a product to comply with a standard affects the costs of purchasing and operating the product. Higher-efficiency products usually have higher installed costs and lower operating costs. If the decrease in operating costs are not enough to offset the increase in installed cost, the PBP can extend past the life of the appliance resulting in a negative net present worth (net cost). This would be an example of when an analysis of doing nothing is required. If no action is required, then no options should be chosen, provided the option to maintain the status quo has a net present worth of greater than or equal to zero. In the event that an action must be taken, such as complying with changing standards, the option that has the least cost (least negative net present worth) would be the best option.

Figure 23 from the DOE 2005 TECHNICAL REPORT: *Analysis of Amended Energy Conservation Standards for Residential Refrigerator-Freezers*, shows factors used by DOE in determining life-cycle cost [DOE 2005].

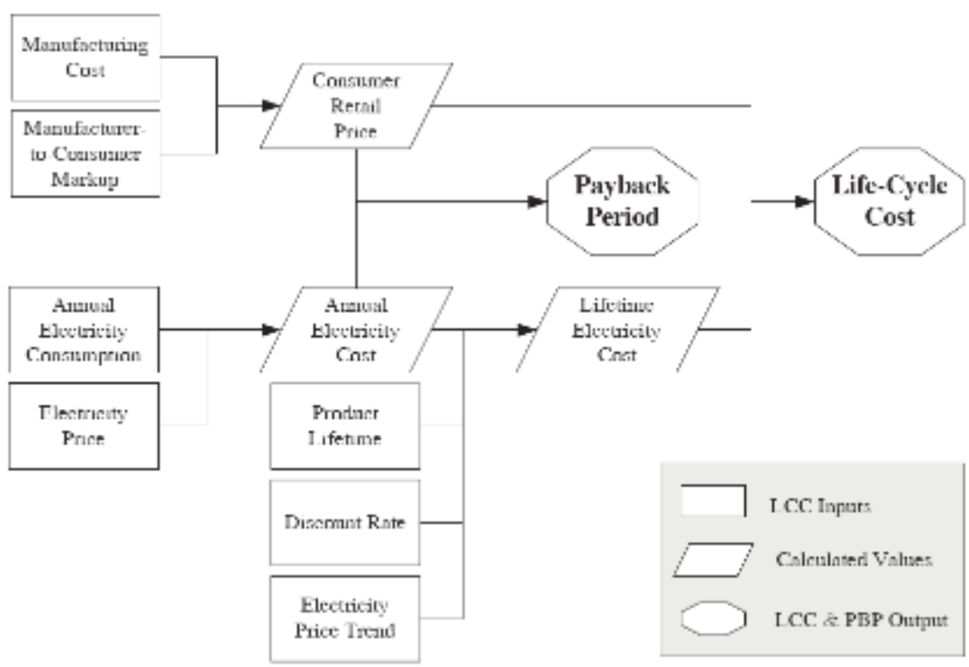


Table 23. Flow diagram of life-cycle cost and payback period analysis [DOE 2005].

For short-term policy and decision-making important factors omitted are rebates, tax credits, and tax deductions. Perhaps these are intentionally excluded since they are economic tools that are not typically used as part of a long-term solution. Rebates can and perhaps should have a place in long-term planning especially when purchasers do not consider more than one option much less the LCC. If the U.S. initiates a public policy that has at its goal, minimizing energy consumption, it may benefit from some form of rebates that encourage buying energy efficient appliances. Direct manufacturer rebates to companies who produce more efficient designs has been shown to be more economically efficient than consumer rebates. One such manufacturer rebate example is a program by the Southern California Edison utility (SCE) in 1992 with compact fluorescent light bulbs (CFL's) which reduced the final consumer cost while also reducing administrative costs with the same rebate dollars by working directly with manufacturers in lieu of the end consumers as shown in Table 24.

Table 24. Sample CFL costs for consumer and manufacturer rebates of $5 [SCE].

	Cost				
	Wholesale	Retail Markup (67%)	Retail	Final Consumer	Economic Efficiency
No Rebate	$10.00	$6.70	$16.70	$16.70	n/a
Consumer Rebate	$10.00	$6.70	$16.70	$11.70	$5/5 = 1.00
Manufacturer Rebate	$5.00	$3.35	$8.35	$8.35	$(5+3.35)/5 = 1.67

Residential appliances such as water heaters are typically used in service until failure where as appliances such as refrigerators are more often retired while still operational. For this reason, and due to quality of manufacturing differences, these appliances have varying life timelines. The Federal Energy Management Program (FEMP) estimates average lifetime for standard storage tank water heaters is 13 years while other sources report life expectancy ranges from 5 to 21 years depending on use patterns and the corrosiveness or harness of water. The Department of Energy (DOE) reports tankless water heaters to have a life expectancy of 20 years or more. Table 25 shows lifetime values for various water heater types by fuel type.

Table 25. Water heater fuel type lifetime (years).

	Minimum	Average	Maximum
Storage Tank			
Electric	6	14	21
Natural Gas	5	9	13
LPG and Oil	5	9	13
Tankless		20*	
Heat Pump		10*	20**
Solar		20*	

Source: DOE Technical Support Document, 2000 [DOE 2000a]
* High Efficiency Water Heaters [Energy Star 2006]
** [ACEEE 2007]

The Consortium for Energy Efficiency (CEE) reports standard refrigerators (refrigerator-freezers) have a life expectancy of 14 years and compact refrigerators have a 10-year life. Other sources report a 19 to 20-year life [FEMP 2006, Kreith 2007]. Reported life expectancies vary in part due to different definitions of life. The first ownership, that is for the purchaser and first user, of refrigerator-freezers is reported between 8 and 14 years but the service life is 20 years or more [E Source 2001].

2.4.1 Utility, Requirements, Comparison, & Selection of Features

The domestic water heater (DHW) is a good example of an appliance that delivers nearly pure utility to the user. That is, hot water is simply what is required and there are no other requirements such as form or ornamentation. As long as the DHW heater delivers hot water when it is needed most consumers don't pay much attention to it. Convenience, availability, or speed of installation followed by initial cost appears to be top criteria of DHW customers because there have typically been few features to choose from and they tend to be replaced urgently upon failure. Reliability and size is a concern as is noise of some newer technologies like heat pump water heaters, but as long as those are satisfied, the consumer makes the purchase decision based primarily on availability, secondarily on initial cost, and then on the life cycle cost (LCC). For an engineering study cost is an ideal metric of success since it is easily quantifiable. For an initial screening of technology, water heaters can then be accurately and easily evaluated by; 1) initial cost, 2) cost per daily use, and 3) LCC. Which cost is most important, depends on who is making the decision, the purchaser or the end user, and what the economic situation is at the time of purchase.

The residential refrigerator on the other hand requires a combination or a compromise of utility and ornamentation. Once the appliance deviates from the pure utility of refrigeration with the addition of some of these features, the cost and cost comparison can only be based on personal preference and what each consumer is willing to pay. It is a generally accepted notion that people will pay more for features that provide increased *value* to them. Except for a small sub-set of the population, energy-efficiency is not considered a feature in and of its self with *value*. The *value* of energy-efficiency increases when energy costs increase because consumers have a greater financial incentive to consume less energy. A recent quarterly report of architectural trends released by the American Institute of Architects (AIA) showed among its members, that energy efficient products are increasingly requested. Figure 22 from the second quarter of 2007 shows the architectural products that are increasing in popularity including “Energy Efficient” products [AIA 2007].

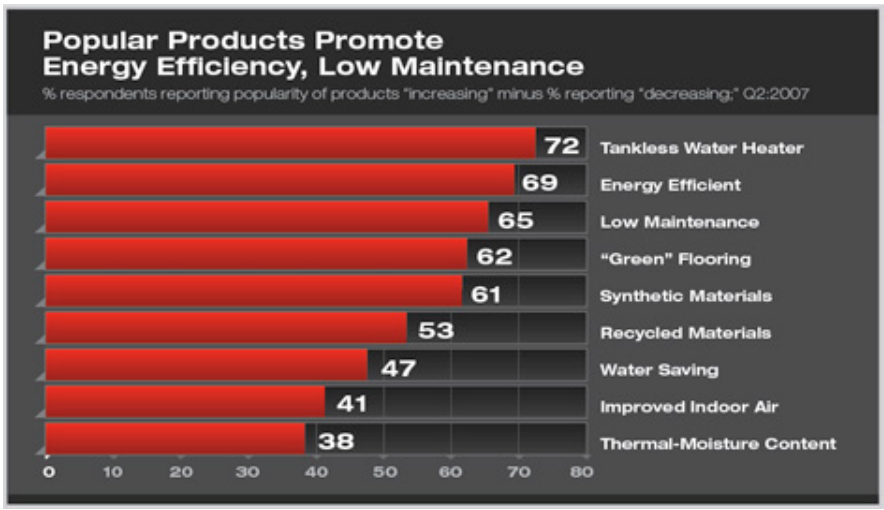


Figure 22. AIA Second Quarter of 2007, Reported Popularity of Products.

2.4.2 Evaluating Potential Benefit

As the potential of new technology depends heavily on the time and environment in which it is introduced, there are different types of potentials with varying levels of benefit. The International Energy Agency (IEA) defines five different types of potentials that we can use to frame discussion of energy efficiency, from the maximum theoretical potential, which is based on the limits of physics to the lower potential based on market trends.

1. Theoretical Potential
2. Technical Potential
3. Societal Potential
4. Economic Potential
5. Market Trend Potential

The greatest possible *known* potential is the *theoretical* potential, which in the case of the energy efficiency of a heat pump would be the thermodynamic ideal Carnot Cycle. It is reasonable to assume that yet undiscovered new technologies hold greater potential than the Carnot Cycle or that an entirely new concept would replace the heat pump all together. The addition of one or more restrictions is responsible for each step down in potential. Figure 23 shows the relationship of the different types of potentials of energy efficiency [WEA 2000].

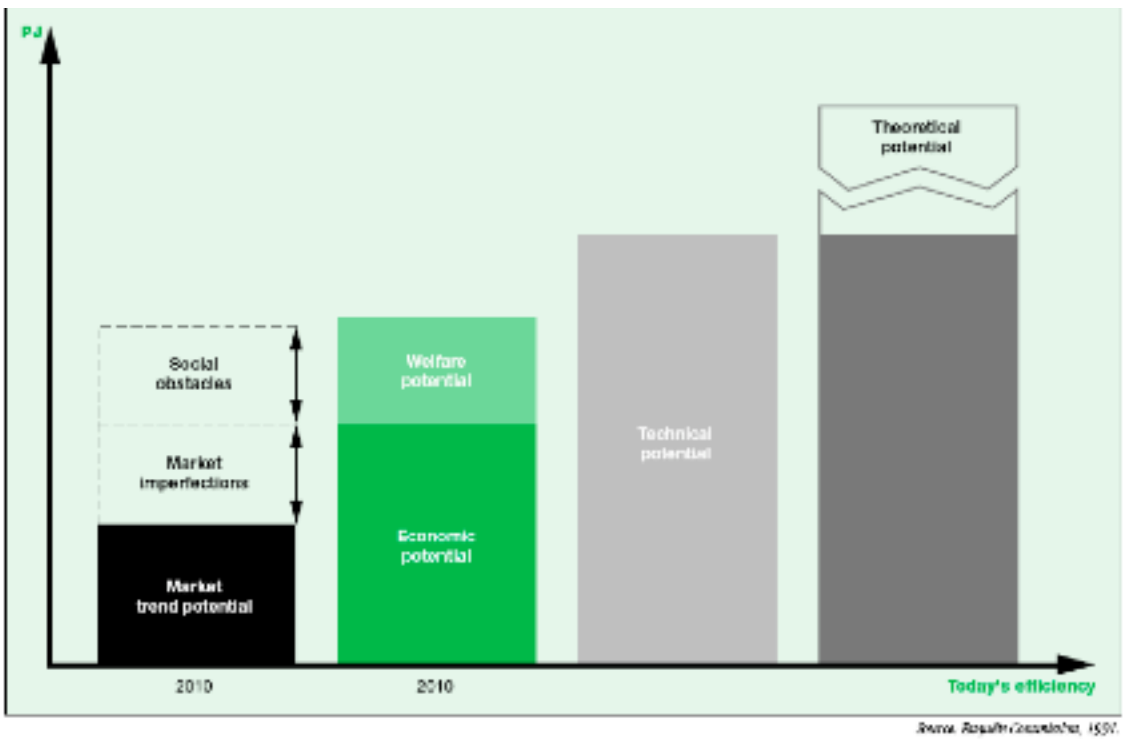


Figure 23. Theoretical, technical, economic, and market trend potentials of energy efficiency [WEA 2000].

Technical potential represents the thermodynamic and technical limitations, but with no regard to the practical feasibility or cost, such as the best laboratory prototype of a solar electric cell. The *economic* potential becomes the portion of the technical potential that is cost effective given the cost, benefit, and discount rate otherwise known as the economic environment. *Market trend* potential is literally what individuals will buy and represents the portion of the cost-effective potential with further restrictions known as market imperfections such as barriers to entry, taxes, subsidies, and resistance due to social norms. *Societal* or *welfare* potential is defined as the "cost effective" benefit when all externalities are taken into account such as carbon sequestration, health impact, and other ecological impacts for society.

Since the *theoretical* potential is the only type of potential defined by thermodynamic laws and represents the maximum attainable value, individual options should be compared against their theoretical potential to determine how much more improvement is possible. Two technologies that have comparable energy efficiency but operate at different percentages of their respective theoretical potential have different capabilities for improvement. Table 26 shows the capability for improvement for an electric resistance water heater compared to an electric heat pump water heater. The resistance water heater is within 8% of its theoretical limit, where as the heat pump has 63% more potential before hitting the theoretical limit. Clearly there are real barriers that stand in the way of the remaining potential including friction, finite size limits, and small amounts of heat transfer over long periods of time.

Table 26. Water Heater COP Potential.

	Theoretical	Economic*	% of Theoretical
Storage Tank – Electric Resistance	1	0.92	92%
Storage Tank – Heat Pump	8.3**	3	36%

* Based on commercially available models.
** COP Calculated from Carnot cycle efficiency with high temperature of 60deg C (333.15K) and low temperature of 20C (293.15K).

The potentials are a good tool for the preliminary screening of possible design options and technologies. The different definitions of potential cover the spectrum from concepts free of restrictions (theoretical), to feasible (technical), to possible humanitarian benefit (societal), to cost effective (economic), to ideas that can make an immediate impact (market trend). When problems require near term solutions, focus must be directed on *market trend* and *economic* potential for the screening of options. For long-term technology development, options should be screened based on their *technical* and *theoretical* potentials more than the current market trends.

Once the list of design or technology possibilities has been screened to the appropriate level, whether it is for Research and Development (R&D) or production improvements, more detailed analysis including lifecycle cost, payback period, sensitivity analysis, and tradeoff studies can be implemented. For energy efficiency improvements, cost of conserved energy (CCE) is a useful measurement. The CCE is defined as,

$$CCE = \frac{Cost}{Saved\ Energy}$$

where, Cost is the appropriate cost of the improvement such as the lifecycle cost (in $), and Saved Energy is the amount of energy saved in kilowatt hours or Btu for the respective timeframe of the cost. If the CCE is less than the market rate for energy than the energy efficiency improvement is cost effective. Table 27 is a sample of energy efficiency improvements with their respective cost, payback periods, and CCE.

Similar to the cost of conserved energy (CCE) is the incremental cost of implementing an energy efficiency measure. Incremental cost is the additional cost required to implement a new feature or technology, which can include amortized capital costs required to upgrade manufacturing equipment or facilities. The technology with the lowest incremental cost that meets the requirements is the best option. As long as multiple features can be independently integrated and do not interfere with each-others functionality, the incremental cost can be used to rank and select one or more options for implementation. Incremental cost works well for evaluating energy efficiency technology in residential refrigerators as well as DHW heaters because the incremental cost can be applied to all modern appliances at all price levels even though they have a wide difference in features, materials, and styles.

When optimizing a system, there are many possible goals including minimizing life-cycle cost (procurement to disposal), operational energy consumption, “cradle-to-grave” energy requirement, or exergy destruction. Since each goal has a different focus, each will result in different system architectures and have different associated costs.

Table 27. Estimated paybacks of energy conservation technologies [Kreith 2007].

Technology	Installed Cost ($)	Energy Savings (kWh/yr)	Cost Savings ($/yr)	Simple Payback (yr)	Life (yr)	Cost of Saved Energy (c/kWh)
Building Envelope						
Insulation, Wall	800	2831	226	1.2-4.6	50	1.9
Insulation, Ceiling	1140	4916	393	1-3.8	50	1.56
Insulation, Foundation	648	2880	230	2.1-8.4	50	1.51
Windows, double pane, Low-E, N	2.75/SF	6.3/SF	0.51/SF	5.4	20	2.93
Windows, double pane, Low-E, S	2.75/SF	4.5/SF	0.40/SF	6.9	20	3.42
Glass storm window	5/SF	9.5/SF	0.76/SF	4.6-10.6	20	3.54
Solar films	1.85/SF	4.75/SF	0.38/SF	4.8	3-15	3.26-13.78
Weatherstripping/ caulking	230	1852	148	1.6	2.5	5.23
HVAC/Motors						
Repair duct leaks	110	2300-4000	184-320	0.3-0.6	7	0.61
Duct insulation	0.85/LF	2-7.5/LF	0.16-0.58/LF	1.5-5.3	25	1.55
Heat pumps Air-source, hot climate	3920	3884	311	2.9	15	1.94
Destratification fans	415	2666	213	1.9	10	1.82
Efficient						
Air conditioners	300/ton	600/ton	48/ton	6.2	15	4.19
Heat exchangers	3760/kcfm	17000/kcfm	1190/kcfm	3.1	20	1.49
Direct						
Evaporative cooling	850/ton	1241/ton	87/ton	2.35-20	1.08-3.51	
EMCS	300	2790	195	1.5	10	1.26
Economizer	62.5	162-1785	11.5-125	0.5-5.5	15	2.38
Efficient motors (1-5 hp)	166	520	-	1.3	7	5.12
Optimum motor sizing	192	-	53	3.6	7	-
Appliances and Water Heating						
Heater wrap	21	273	19.11	1.1	10	0.9
Thermal traps	8	380	30	0.3	15	0.25
Pipe wrap	5	20	1.6	3.1	10	2.93
Low-flow shower head	9	275	22	0.4	10	0.38
Heat-pump Water heater	1350	2780	222	4.7	13	3.5
Heat-recovery Solar hot water heater	700	1100	88	7.9	13	5.98
Low flow system	42/SF	81/SF	6.50/SF	6.5	20	3.49
Refrigerators and freezers	731	590	47.2	1.3	20	0.68
Lighting						
Efficient incandescent	1.13	8	0.48	469 (hr)	750 (hr)	6.84
Compact fluorescent	13	57	5.67	2150 (hr)	10000 (hr)	2.44
High efficiency Fluorescent	18	153	9.3	0.9	20000 (hr)	2.42

Low-E, low emissivity coating on glass; SF, square foot; LF, lineal foot; kcfm, 1000 cubic feet per minute; EMCS, energy management and control systems; hr, hour; HVAC, heating, ventilation, and air conditioning.

2.4.3 Energy Cost

For U.S. residences, electricity and natural gas are the two most important fuel sources accounting for 41% and 43% respectively of the total domestic energy consumption [EIA 2007a]. Historical data on energy cost and consumption in the U.S. is available through the year 2005 with projections available through the year 2030 from the U.S. Department of Energy (DOE) and the Energy Information Agency (EIA) in the *Annual Energy Outlook 2007* report [EIA 2007a]. Current projections of primary energy sources in the U.S. show decreasing prices through approximately the year 2015 before slowly increasing through the year 2030. Figure 24 shows the price history and projection for these primary energy sources including petroleum, natural gas, coal, and electricity in 2005 dollars [EIA 2007a].

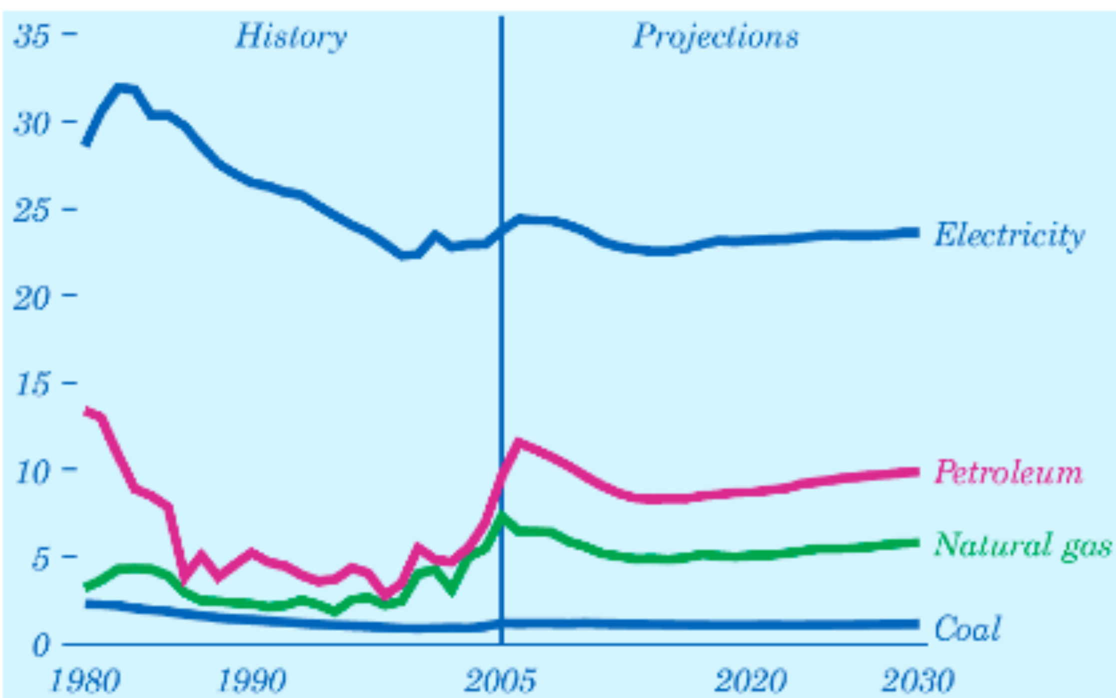


Figure 24. Energy prices, 1980-2030 (2005 dollars per million Btu) [EIA 2007a].

Petroleum or crude oil, which is the basis for heating oil used in some homes and gasoline used in automobiles, can provide an especially powerful lesson on instability and un-predictability of energy (fuels) cost. There have been several historically significant events in the history of U.S. oil generation and consumption including the creation of OPEC in 1960 and the oil embargo in 1973. With subsequent price increases and reduction of oil exports from the participating countries, the OPEC oil embargo forced oil prices to quadruple from $10 to $40 (1973 dollars) in only a few months and in the U.S., shortages resulted in rationing amongst the general population. Figure 25 shows the obvious price spikes of crude oil adjusted for inflation in 2006 dollars from the year 1947 to 2007.

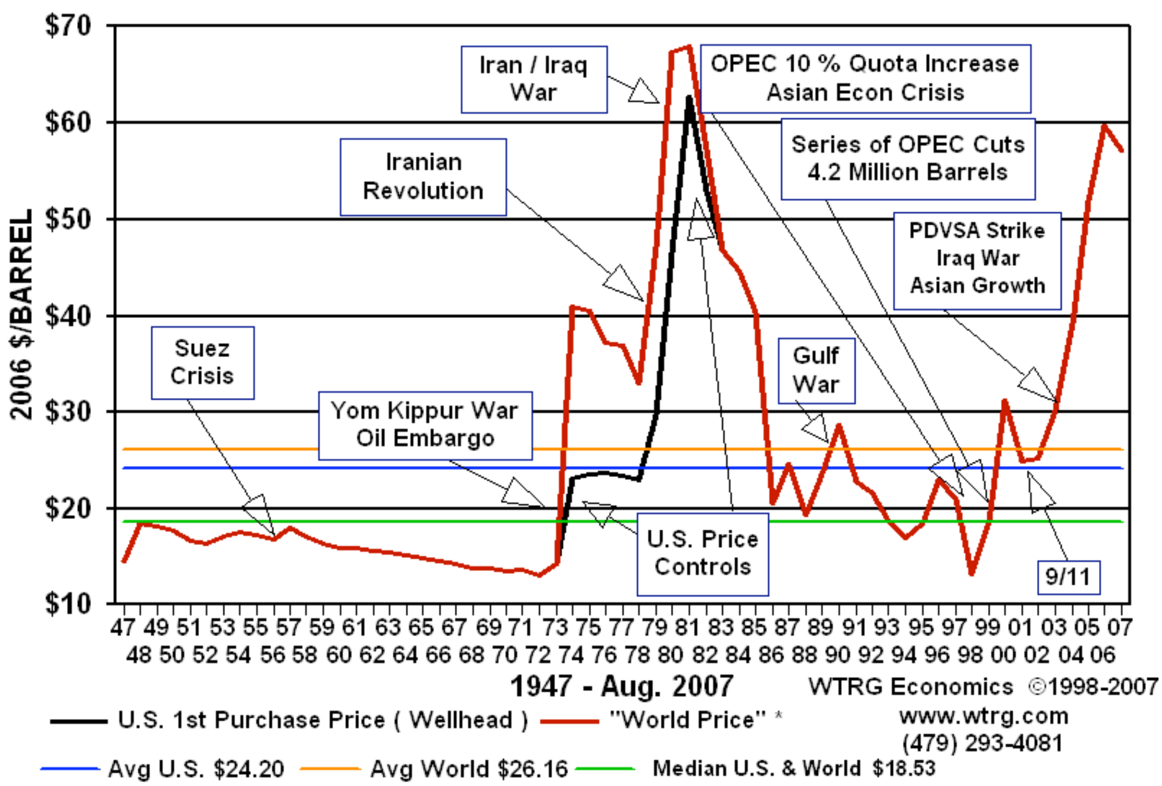


Figure 25. Crude oil prices in 2006 dollars [WTRG].

Comparing crude oil price estimates from 1997 and 2007 against current price. In the 1997 Annual Energy Outlook (AEO), the highest cost per barrel projection was from the *AEO High Price case* where the cost increased from $18.40 in 1995, to $25.53 in 2005, to $28.23 for the year 2015 (1995 dollars) [EIA 2007a]. In the 2007 AEO, the current oil price of $100 for a barrel (March 2008), which equates to $71-73 in 1995 dollars, is almost three times the highest projection in 1997. Table 28 summarizes the forecast and historical prices for each of the three price cases in the 1997 and 2007 AEO.

Table 28. Forecast and historical prices for crude oil.

				2006 dollars per barrel					
Forecast	1995	2000	2005	2008	2010	2015	2020	2025	2030
AEO 1997 reference	-	24.02	26.02	n/a	26.93	27.68	n/a	n/a	n/a
AEO 1997 high price	-	28.62	31.84	n/a	34.89	37.25	n/a	n/a	n/a
AEO 1997 low price	-	18.58	18.58	n/a	18.51	18.46	n/a	n/a	n/a
AEO 2007 reference	-	-	-	66.27	59.42	51.57	53.81	58.29	61.13
AEO 2007 high price	-	-	-	69.18	71.56	82.28	92.15	97.61	103.54
AEO 2007 low price	-	-	-	64.05	50.77	35.15	35.26	36.08	36.89
Historical*	22.78	32.47	58.69	94.26					
% Error reference case	-	26.0%	55.7%	29.7%					

* AEO 1997, EIA [67], AEO 2007, March 2008 market price of $100/barrel
** Inflation calculated from CPI from U.S. Bureau of Labor Statistics Data http://data.bls.gov/cgi-bin/cpicalc.pl
*** Future Inflation set at 2.20%

Although there is no single factor responsible for the crude oil price increase, generally it is attributed to increasing demand and decreasing supply that is amplified by uncertainty in political arenas and financial markets. In the long-tem view, supply and demand governs the cost of the fuel. Short-term prices have been affected by natural disasters (Hurricanes Rita and Katrina in the southern U.S.), conflicts abroad (Iraq War), and financial market distress. In the preface to the 2007 AEO, the Department of Energy describes the projections and provides a summary on the uncertainty involved in making projections,

> "Energy market projections are subject to much uncertainty. Many of the events that shape energy markets are random and cannot be anticipated, including severe weather, political disruptions, strikes, and technological breakthroughs. In addition, future developments in technologies, demographics, and resources cannot be foreseen with certainty."

The 2007 Annual Energy Outlook expresses crude oil prices in terms of the average price of imported light, low-sulfur crude oil to U.S. refiners. Crude oil prices are projected to decline gradually from their 2006 average level through 2015 before rising again as demand continues to grow [EIA 2007(a)]. EIA notes that although current oil prices are above the long-run equilibrium price the longer-term predictions remain valid, meaning that production is expected to catch up to, and make up for recent short-run disruptions. The predicted trends by EIA may turn out to be correct but the error in even recent predictions (10 years or less) shown in Table 28 are sizable enough to affect economic decisions regarding investment and energy use.

2.4.4 Sensitivity Study

Time permitting, energy costs should be viewed as a variable in engineering economics studies to explore the effects of rising or lowering prices, in what is known as a sensitivity study. Although this is not necessary when the analysis timeframe is short enough to limit the risk of energy price fluctuation, many residential, industrial, and commercial engineering economics studies span years if not decades. With the sensitivity study, probabilities of critical variables such as price fluctuation are not required, only the variation ranges that are of interest are required. Typically the levels of variation that are of interest are those values that are reasonable based on information at hand. If probabilities are known for the options, tools such *as decision tree* analysis or *Monte Carlo* simulations can be used to gain a better understanding of not only the possible outcomes but also the likely outcomes. Even though probabilities are sometimes referred to as being known, meaning they have been estimated from previous experience or calculated from statistics, the knowledge of the probabilities does not guarantee any single outcome. Decision tree analysis consists of branches of multiple choices expanding out from a single decision node. Each choice, if taken, has multiple possible outcomes with each outcome having a probability of occurrence and an expected value. The sum of the probabilities at each chance node is equal to one (100%) meaning that there are no other possible outcomes. Each chance node has an expected value equal to the sum of the possible outcome branches individual probabilities multiplied by their expected values. For each decision, the best choice is the one that leads to a chance node with the highest expected value. The decision tree can continue to grow with subsequent decisions expanding the branches. Figure 26 shows a sample decision tree for a decision to lease or build a facility with inferior choice branches marked with a double hash mark [Kreith 2007].

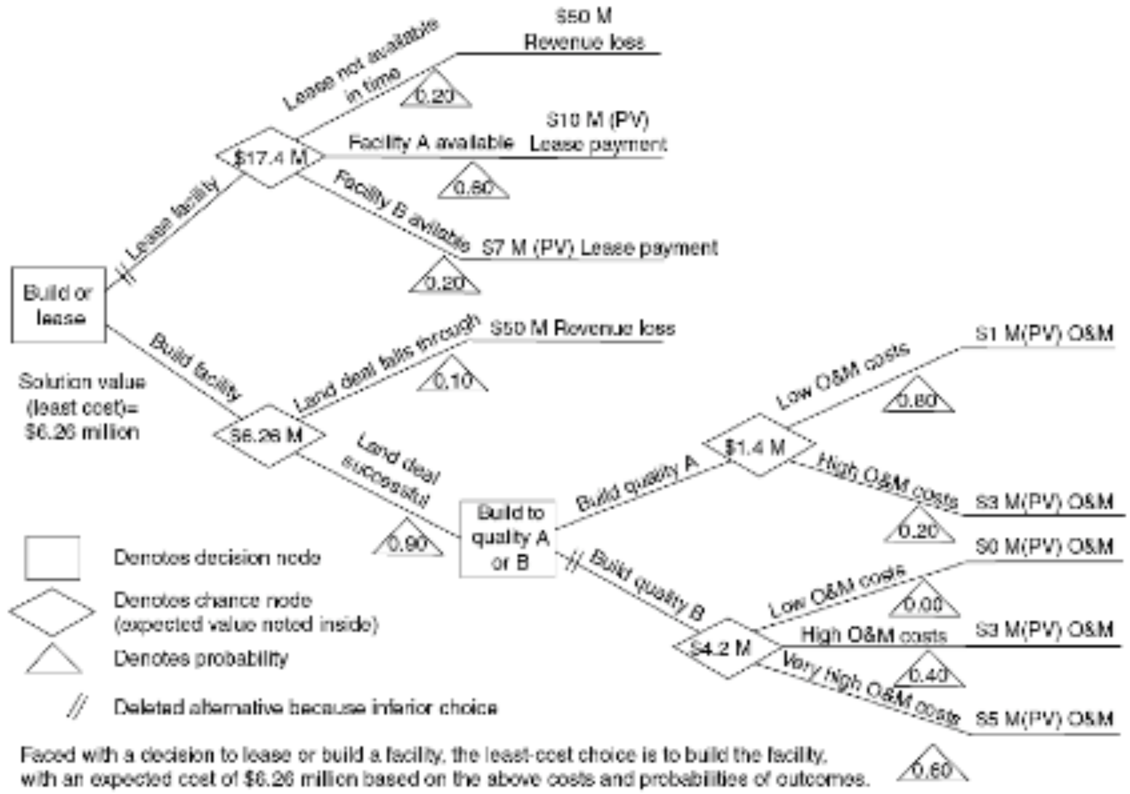


Figure 26. Decision tree to build or lease a facility.

Monte Carlo simulation on the other hand uses the probabilities in the form of cumulative distribution functions. A computer simulation program samples from the probability distributions and calculates the result, which can be the present worth, LCC, or another metric. The computer samples and calculates the result a large number of times to ensure the probabilities are random. Although the Monte Carlo method does not guarantee that the absolute maximum or minimum will be calculated, it does provide an array of possible results that can then be meaningfully analyzed other than just by looking at the maximum, mean, and minimum result. The benefit of running the simulation a large number of times is that a distribution of results is generated. For water heater LCC analysis, DOE sampled 10,000 times from a distribution on each variable to find out the percentage of population benefiting from the changes and the mean reduction in LCC [DOE 2000a]. Possible shapes for the distribution include a normal distribution, a distribution with multiple humps, or a distribution that is skewed significantly to one side. An example of result distributions that exhibit conclusive preference is shown in Figure 27, where Project B clearly dominates Project A. Not only does Project B have a greater maximum, mean, and minimum net present value than Project A, but also the net present value is more likely to occur. Figure 28 on the other-hand shows result distributions that are inconclusive as Project A has a higher average net present worth than Project B, but it is less likely to occur.

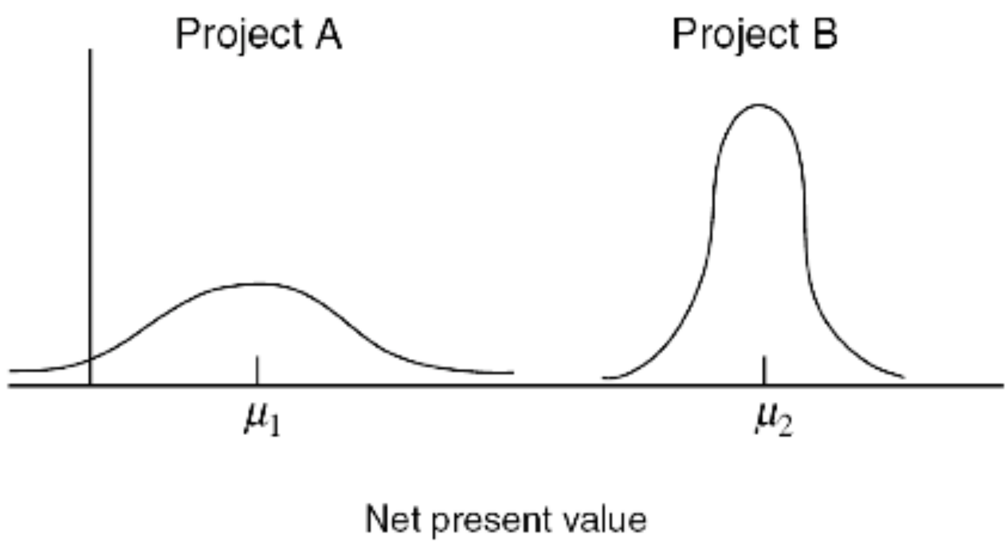


Figure 27. Result distributions from Monte Carlo simulations with conclusive preference.

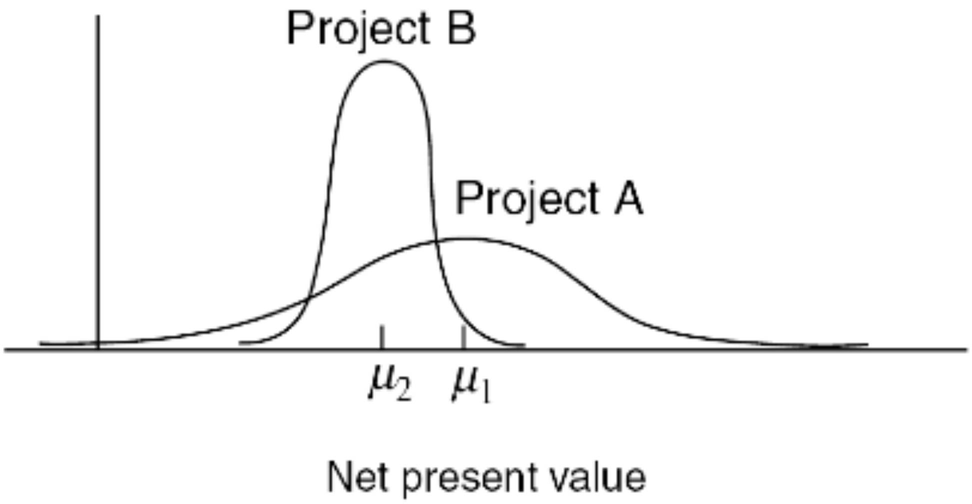


Figure 28. Result distributions from Monte Carlo simulations with inconclusive preference.

2.4.5 Energy Consumption

Total worldwide energy consumption energy consumption is projected to grow from 447 quadrillion Btu (quad) in 2004 to 702 quad in 2030, an increase of 57 percent over the 2004 to 2030 period despite high energy prices [EIA 2007b]. The United States is responsible for just over 22% of the worldwide energy consumption for the year 2004, consuming 100 quad of combined energy sources [EIA 2007a]. Clearly the U.S. is responsible for a significant portion of the current energy consumption, and as the developing nations continue to grow world wide, the U.S. will be responsible for consuming a smaller world percentage, albeit with an increase in total energy use. U.S. energy consumption is projected to increase at an average rate of 1.1% per year, compared to 2.2% worldwide. Figure 29 showing the energy use growth per capita and per dollar of GNP clearly shows that economic use of energy has increased in efficiency while as a nation we continue to use slightly increasing levels of energy.

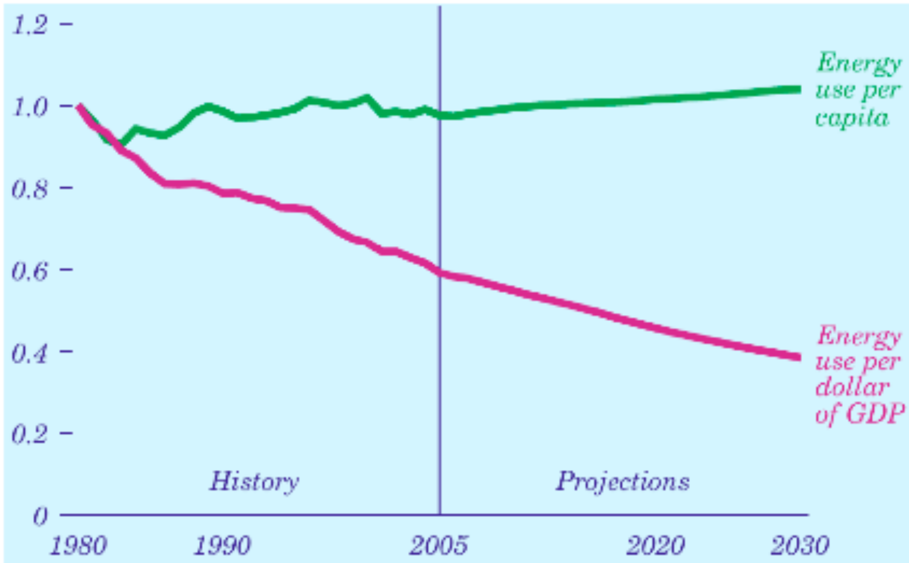


Figure 29. Energy use per capita and per dollar of gross domestic product [EIA 2007a].

Figure 30 shows the history and projections of U.S. energy use by fuel type with the trio of fossil fuels, petroleum, coal, and natural gas accounting for the majority of consumption.

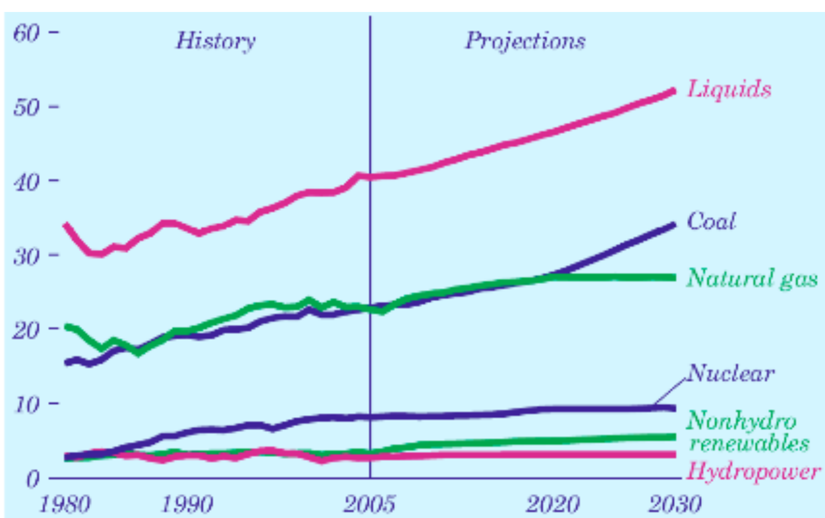


Figure 30. U.S. Energy consumption by fuel, 1980-2020 (quadrillion Btu) [EIA 2007a].

2.4.5.1 Residential energy consumption

Worldwide, residential energy use accounts for 11% the total energy consumption [EIA 2007b]. In comparison, residential energy use in the U.S. accounts for almost 22% of the total energy consumption [EIA 2007a]. *Total energy* consumption includes losses such as electrical transmission losses, where as *delivered energy* is literally the amount of energy received by the end user and would not include losses in electrical generation, transmission, and distribution. Figure 31 shows the U.S. total energy consumption by sector.

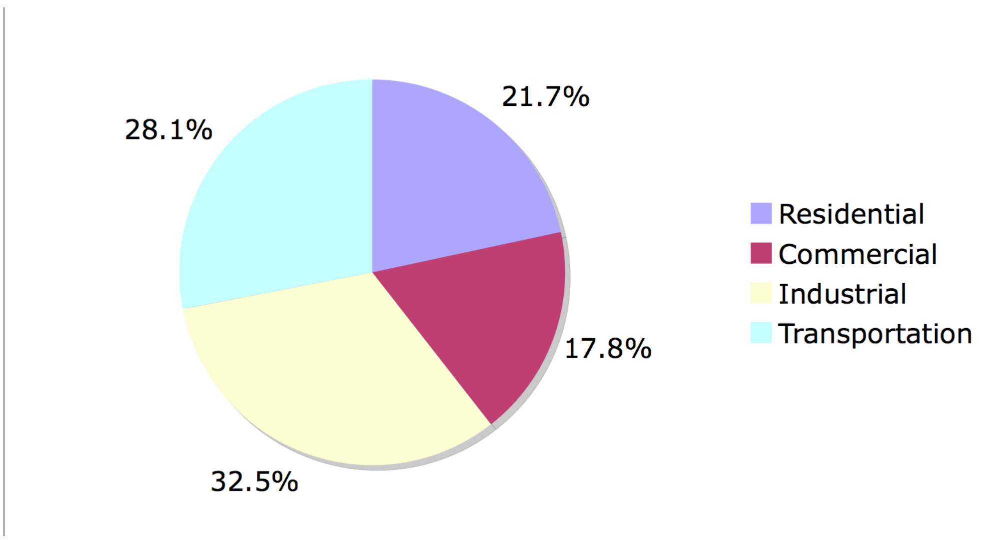


Figure 31. U.S. total energy consumption by sector [EIA 2004].

Figure 32 shows the *delivered energy* consumed in the U.S. by sector, thus some sectors such as the industrial sector which consumes significant amounts of electricity appears to consume less energy than transportation which uses very little electricity by comparison.

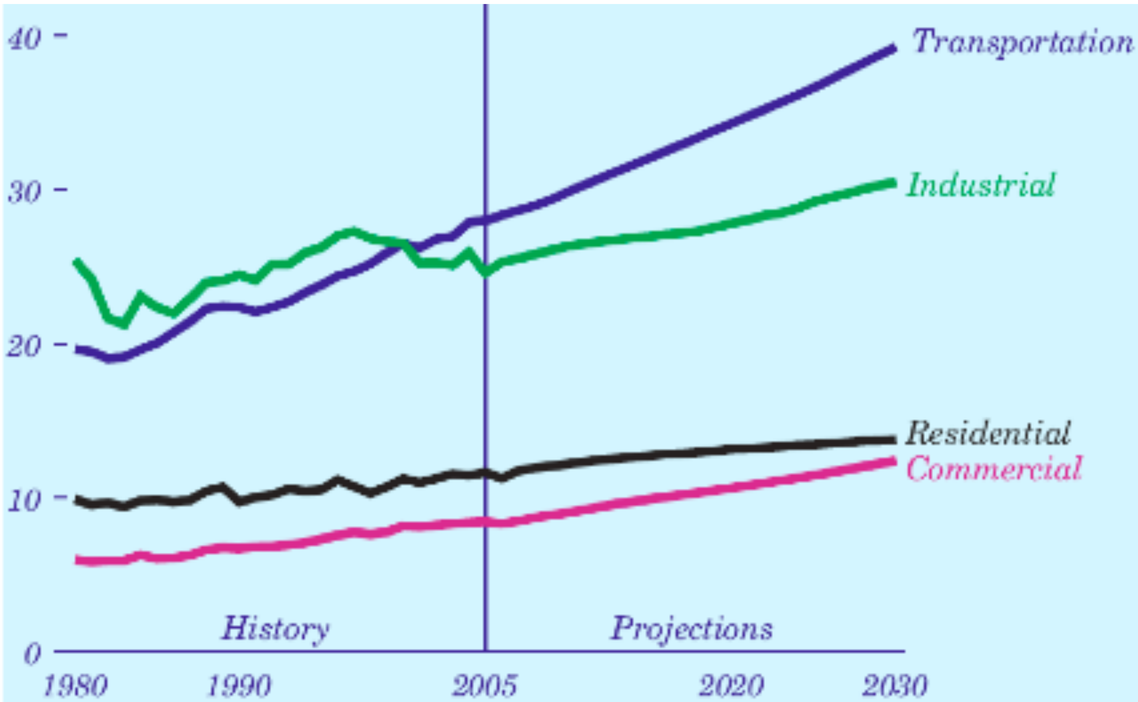


Figure 32. Delivered energy consumption by sector, 1980-2030 (quadrillion Btu) [EIA 2007a].

The greatest rate of growth in U.S. energy consumption is projected to be in the transportation sector while the residential sector will grow at the slowest rate. Per capita residential energy use is projected to decline slightly with factors such as greater energy efficiency and population shift to the warmer U.S. regions of the South and West outweighing the preference for larger homes, more and larger electronics, and new uses for energy. Although many consumer electronic devices have matured and have been refined to increase energy efficiency, many devices such as computers and TV's have increased in size, number in the household, and changed to more energy intensive technologies (plasma), all of which increase the total energy use [Kreith 2007, EIA 2007a].

The increase in U.S. home square footage since the 1950's places higher demands on space heating and cooling, more or higher wattage lighting, and longer runs of hot water, to name a few factors that increase energy consumption. For some who recognize the ecological and social impact of energy generation and subsequent consumption, design and build houses that balance energy production with energy consumption. For these balanced energy homes, called a zero-energy or alternately a net-zero-energy home, the primary goal is to balance energy production with energy consumption. Renewable energies are typically, but not exclusively, used to balance the energy consumption.

2.4.5.2 U.S. residential energy consumption by end use

Heating and cooling systems account for about 58% of the total energy expenditures in a typical residential home. The four heating and cooling sub-systems, space heating, hot water, space cooling, and food refrigeration, account for four of the top five residential energy end-use expenditures (lighting is number three). Figure 33 shows the distribution of residential energy consumption by end use. Although a reduction of energy consumption of any end use is a positive step, targeting the bigger end uses makes the most sense as small percentages can make a much larger impact on overall consumption.

Although refrigerators and freezers consume only 7.2% of total residential energy use, they are responsible for 11.9% of the electricity consumed in U.S. residences, just behind lighting (16.4%) and space cooling (14.7%). Space heating (8.9%) and water heating (8.4%) make up the next two largest consumers of residential electricity [EIA 2007a].

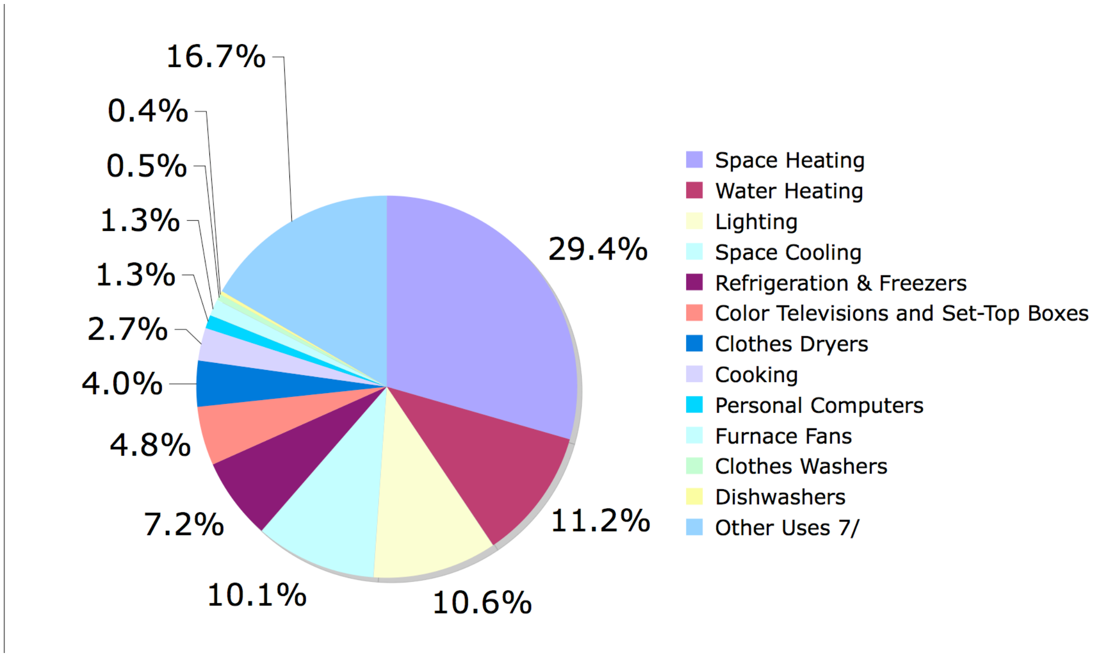


Figure 33. Residential energy consumption by end use [EIA 2004].

2.4.5.3 Electricity

In the United States, electricity consumption accounts for 17.2% of the total energy consumed and 40.6% of the residential energy consumed [EIA 2007a]. The average conversion of fuel to electricity is only 31.5% efficient in the U.S., meaning that over twice as much energy is lost as is consumed at the household. The other 68.5% is lost in generation, transmission, and distribution of electricity to households [EIA 2007a]. Many fuels can be converted into electricity but coal is the primary fuel used to generate electricity in the U.S. Coal accounts for 52.0% of the electricity generation in the U.S. with nuclear and natural gas responsible for 20.4% and 14.9% respectively [EIA 2007a].

According to the 2007 Annual Energy Outlook, the average electricity price in the U.S. remains relatively stable decreasing slightly from a current price of $0.083 (2005 dollars) to a low of $0.077 in 2013 before increasing to $0.081 in 2030, in dollars per kilowatt-hour [EIA 2007a]. Figure 34 shows the history and projection of price for electricity in 2005 dollars per kilowatt-hour.

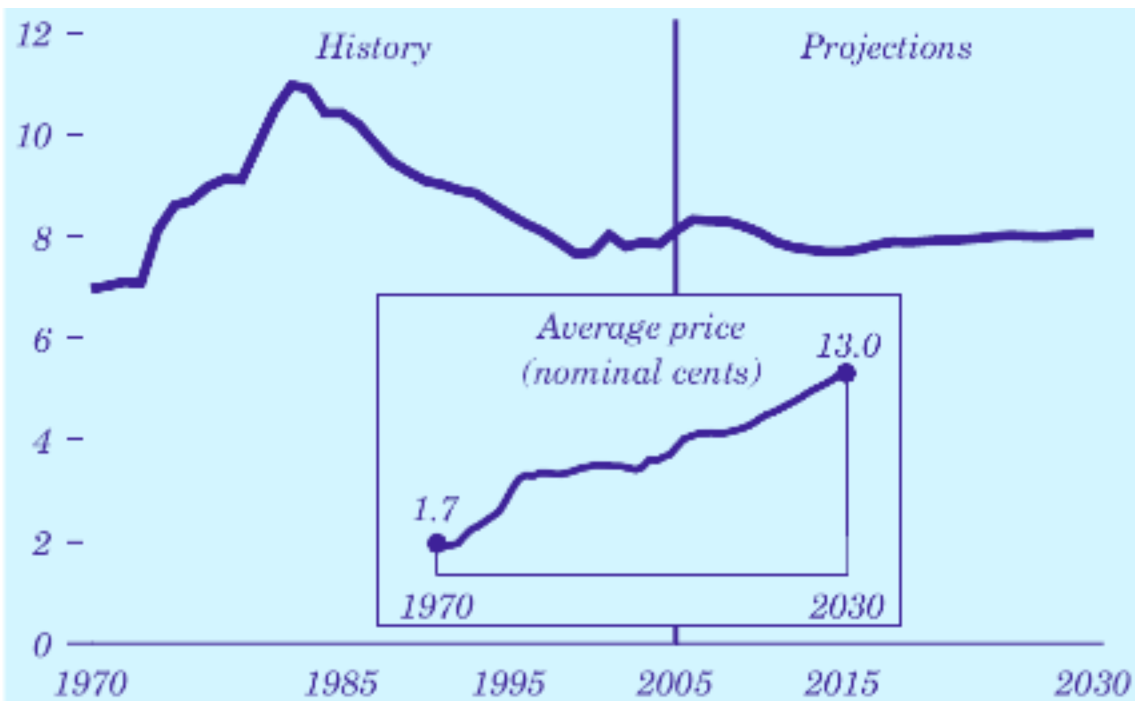


Figure 34. Average U.S. retail electricity prices, 1970-2030 (2005 cents per kilowatt hour) [EIA 2007a].

Across the U.S. electrical prices can vary significantly from the 2005 average of $0.0814 with prices as high as $0.1833 in Hawaii and as low as $0.0501 in Kentucky [EIA 2007c].

Figure 35 shows the price distribution in the U.S. for a kilowatt-hour of electricity for the year 2005.

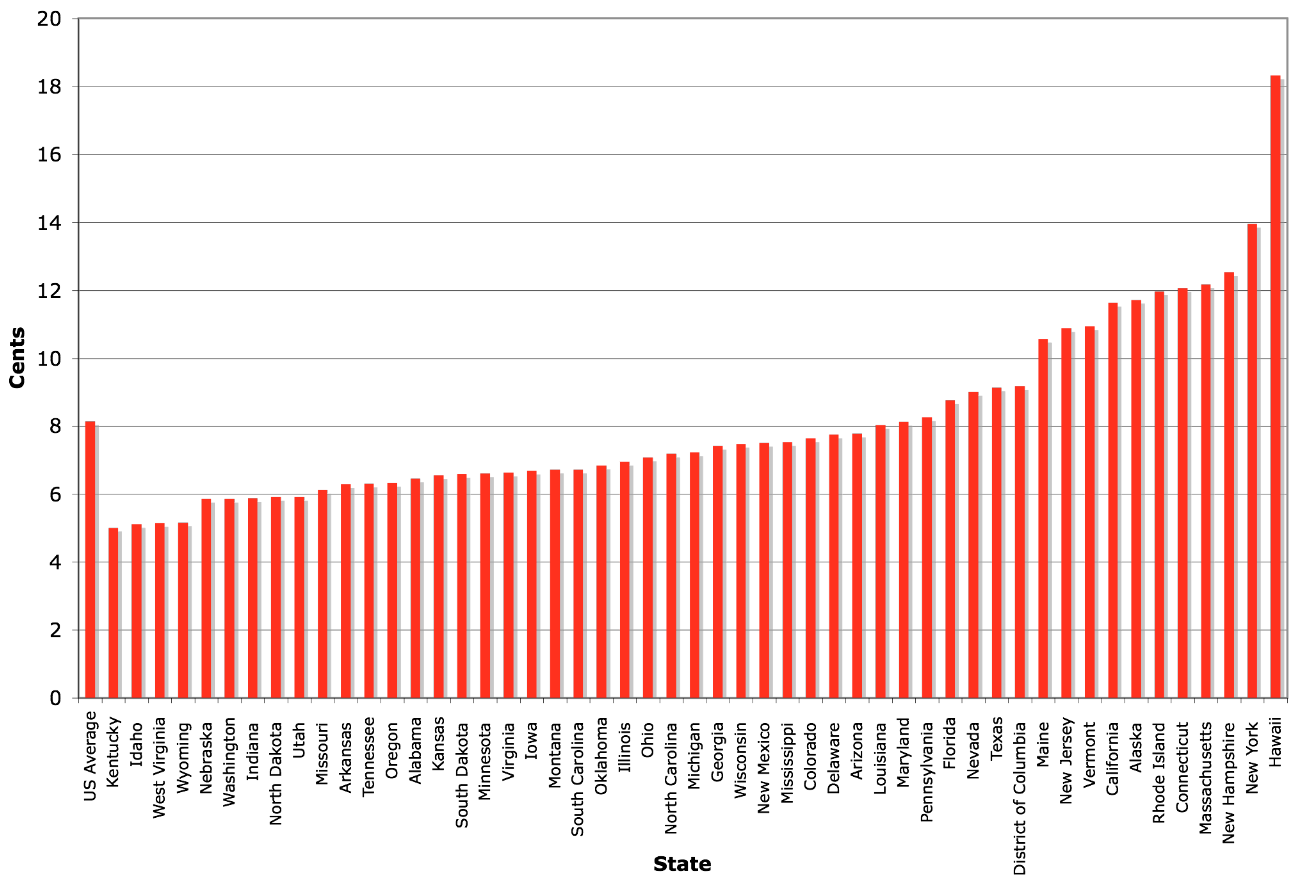


Figure 35. U.S. States average price for electricity (2005 cents per kilowatt hour) [EIA 2007c].

Short-term disruptions in price are rare but can have severe impact just as the market deregulation in California did. During this time electrical prices in California increased rapidly from about 11 cents ($0.11) per kilowatt-hour (nominal) in July of 1999 to approximately 16 cents ($0.16) per kilowatt-hour (nominal) only a year later [EIA 2005].

The increase in electrical rates was almost 69% in one year based on nominal rates or 67% once adjusted for inflation. Slowly the electrical price decreased as the market reached a new equilibrium and in 2005 California had the eighth highest average electrical price in the U.S. states with a price of $0.1163 (nominal) per kilowatt-hour [EIA 2007c].

2.4.5.4 Natural Gas

For residences, natural gas makes up 42.6% of the total energy consumption and is used for space heating, water heating, and cooking. In industry, natural gas is also used for electricity generation as well as heating. EIA projections maintain increasing long-term natural gas consumption for uses other than electrical generation as shown in Figure 36.

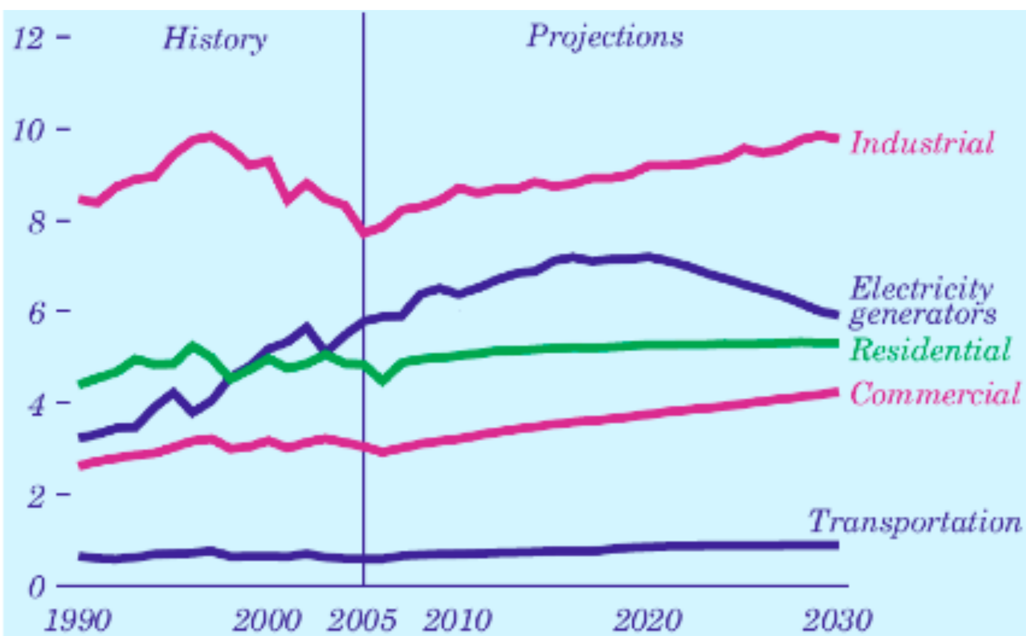


Figure 36. Natural gas consumption by sector, 1990-2030 (trillion cubic feet) [EIA 2007a].

Just as with crude oil, natural gas prices are expected to decline from the current peak price of $7.29 in dollars per million Btu to $4.84 in 2015 (2005 dollars) before gradually increasing for the foreseeable future. Natural gas price can be reported by either volume or energy content, where one thousand cubic feet (Mcf) is equal to about 1.031 million Btu (MMBtu) [EIA 2007a]. The exact specific energy can change from time-to-time or from location to location. Another energy content measurement used with natural gas is the therm, which is defined as being equal to one hundred cubic feet (ccf) of natural gas with a heat factor (HF) equal to one at standard temperature and pressure.

$$therm \equiv ccf_{HF=1}$$

where, the HF is a measure of the quality or energy density of the natural gas.

CHAPTER 3. COMPUTER MODEL & SIMULATION

3.1 Introduction

A program is written with Matlab software to simulate a refrigerator, hot water heater, and the combined system. The refrigerator and hot water heater are developed individually, compared to known results, then combined for system simulation. Combined program flowchart is shown in Figure 37.

3.2 Refrigerator Simulation Model

Many names are used to describe a cold food storage appliance such as refrigerator, freezer, or refrigerator/freezer but for the purpose of this study, the term *cabinet* is used to identify just the insulated enclosure while *refrigerator* is used to identify the entire refrigeration system. The schematic of a simple refrigerator is shown in Figure 38.

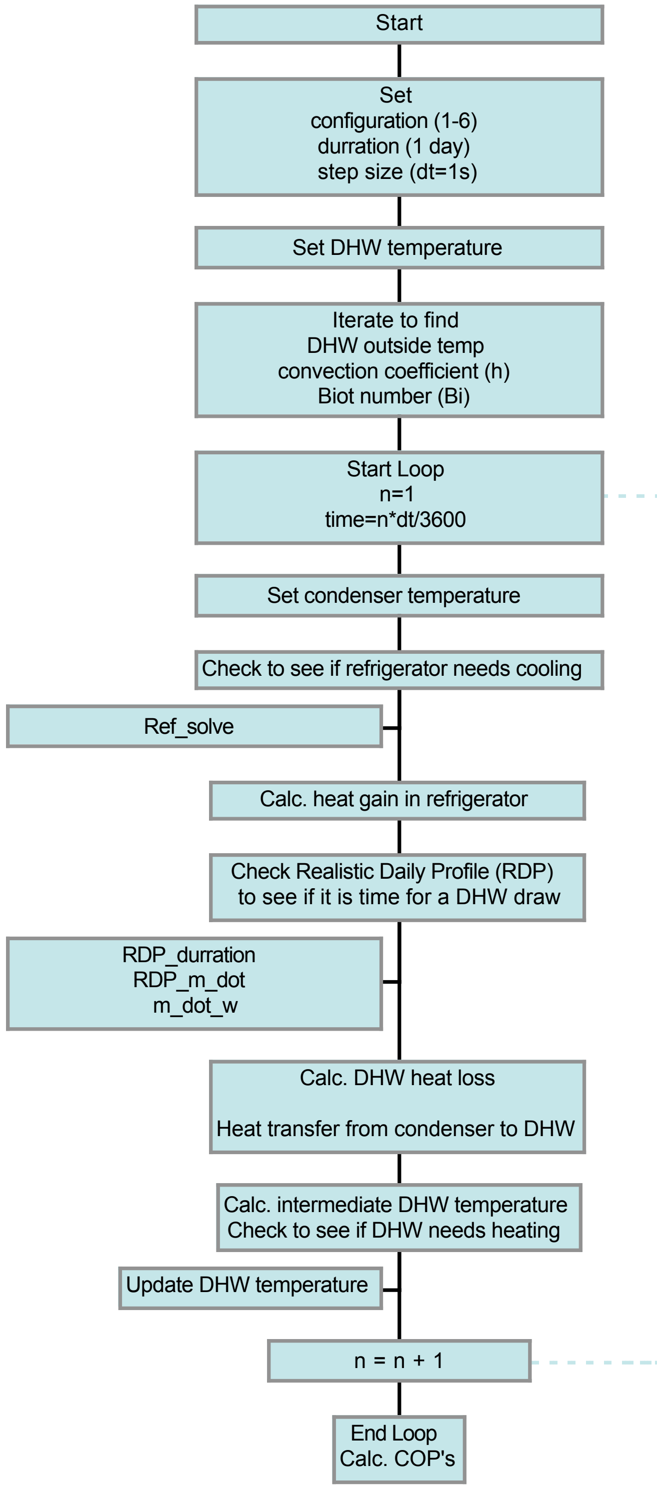


Figure 37. Simulation flowchart.

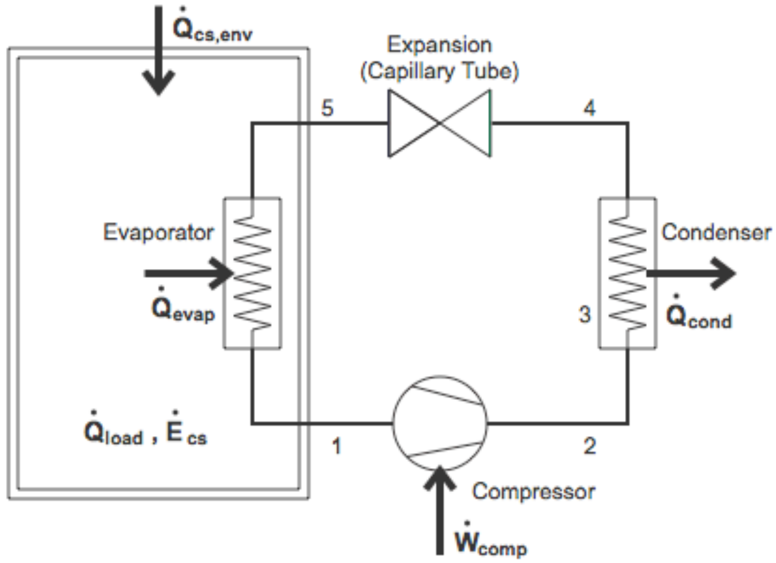


Figure 38. Schematic of refrigerator.

Assumptions for refrigerator model:

1. Neglect auxiliary loads (light, cabinet fan, defrost)
2. Refrigerant cycle is steady-state
    1. No energy storage in refrigerant cycle
    2. Refrigerant is saturated vapor at evaporator outlet / compressor inlet
    3. Compression is adiabatic and uses a constant isentropic compression efficiency value
    4. Refrigerant is super-heated vapor at compressor outlet condenser inlet
    5. No pressure drop in condenser
    6. Refrigerant is subcooled liquid at condenser outlet.
    7. Refrigerant continues to reject heat to the environment until the condenser outlet temperature is 5°C above the ambient temperature
    8. Expansion in capillary tube is constant enthalpy and adiabatic
    9. Two-phase refrigerant at the evaporator inlet
    10. No pressure drop in the evaporator
3. Overall heat transfer coefficient (UA) for condenser and evaporator is constant based in values from literature
4. Condenser environment is sufficiently large such that condenser ambient temperature is constant

5. Compressor and condenser heat do not enter the cabinet. Heat transfer to the cabinet is only from the environment.

The simplified refrigerant thermodynamic cycle is shown in Figure 39 over a p-h diagram. Compressor inlet (1) is on the saturated vapor curve, compressor outlet (2) is in the super heat region, condenser outlet (4) includes subcooling, and the evaporator inlet (5) is in the 2-phase region under the dome. Isentropic compression state (2s) is also shown on the thermodynamic cycle diagram, as is the extent of the condenser de-superheater (2-3) and the extent of the condenser subcooling (4a-4).

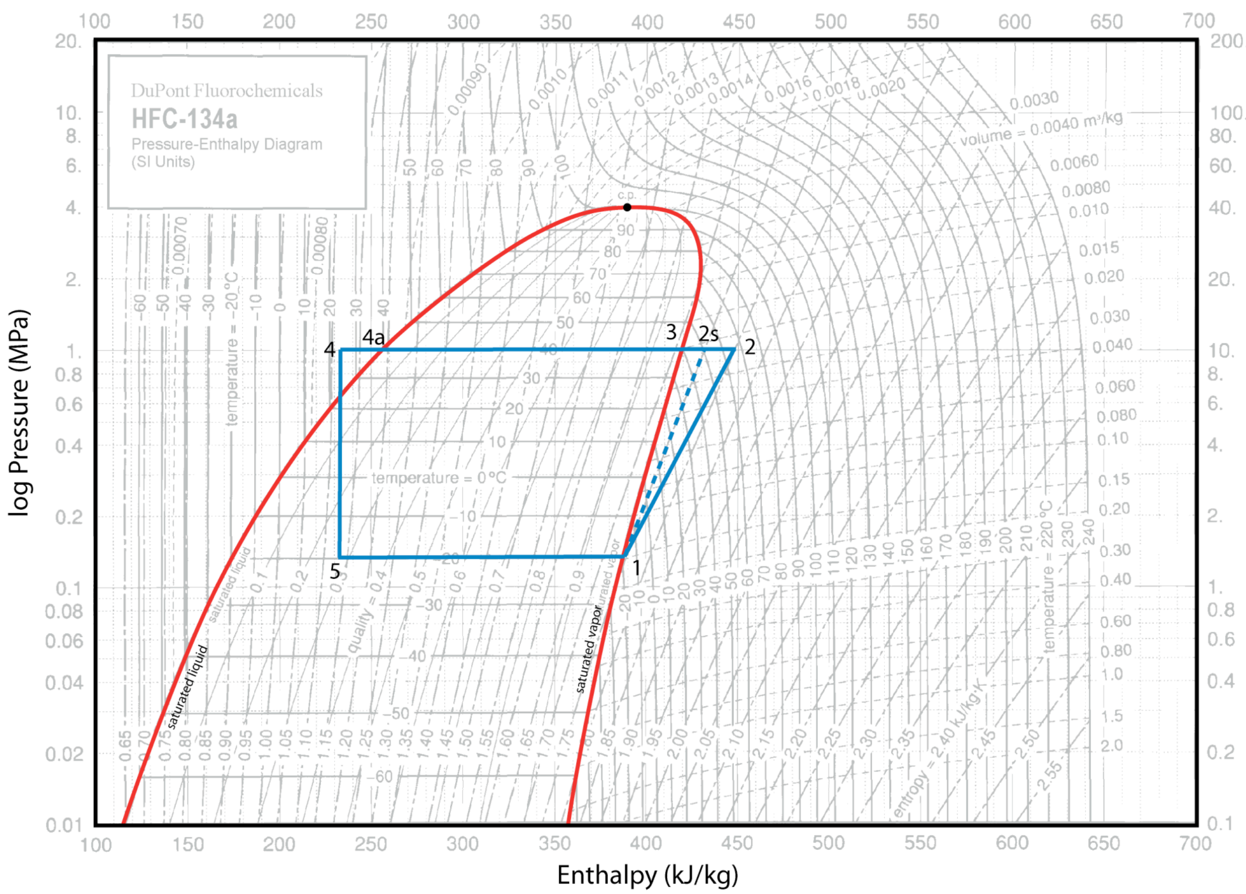


Figure 39. Refrigerant thermodynamic cycle simplified.

Energy rate balance for the cold storage,

$$\dot{Q}_{cab,env} + \dot{Q}_{load} - \dot{Q}_{evap} = \dot{E}_{cab}$$

where $\dot{Q}_{cab,env}$ is the heat gain from the environment through the cabinet walls and $\dot{Q}_{load}$ is the heat gain due to warm food placed in the cabinet. Although they are neglected here, the auxiliary loads such as the automatic defrost, cabinet fan, and cabinet light would be an added burden on the cold storage increasing the required heat extraction by the evaporator and could be grouped in the $\dot{Q}_{load}$ term.

Energy rate balance for the refrigerant cycle,

$$\dot{W}_{comp} + \dot{Q}_{evap} - \dot{Q}_{cond} = \dot{E}_{r}$$

Summing the equations,

$$\dot{Q}_{cab,env} + \dot{Q}_{load} + \dot{W}_{comp} - \dot{Q}_{cond} = \dot{E}_{cab} + \dot{E}_{r}$$

The refrigerant cycle modeled as steady-state and steady-flow so there is no net change in energy storage in the refrigerant and so the refrigerant energy storage term drops out,

$$\dot{Q}_{cab,env} + \dot{Q}_{load} + \dot{W}_{comp} - \dot{Q}_{cond} = \dot{E}_{cab}$$

For DOE testing, refrigerator doors remain closed so no additional warm food is placed in the cabinet so the load term drops out,

$$\dot{Q}_{cab,env} + \dot{W}_{comp} - \dot{Q}_{cond} = \dot{E}_{cab}$$

Coefficient of performance for the refrigerator is defined as the desired work, divided by the required work to achieve it.

$$COP_{ref} = \frac{\dot{Q}_{cold}}{\left|\dot{W}_{net}\right|} = \frac{\dot{Q}_{evap}}{\dot{W}_{comp,elec} + \dot{W}_{fan,evap} + \dot{W}_{fan,cond}}$$

where $\dot{W}_{comp,elec}$ is the electrical power in to the compressor.

3.2.1 Compressor

The compressor is responsible for increasing the pressure of the refrigerant in the refrigeration cycle. Only in an *ideal* thermodynamic cycle can isentropic compression occur. In a *real* thermodynamic cycle the pressure and entropy increase as the temperature is increased to a greater extent than in the isentropic compression. Because the presence of liquid can damage most compressor types, the residential refrigeration cycle is designed to super-heat the refrigerant slightly in what is known as *Dry Compression*. The requirement of *dry compression* further increases the refrigerant temperature at the compressor inlet and subsequently at the compressor outlet. Although super-heating of the refrigerant is required for most residential refrigeration systems, it should be minimized as it reduces the efficiency of the refrigeration cycle [Radermacher 2005].

There are four basic models presented in literature: Efficiency Model, Map-based Model, Lumped Parameter Model, & Distributed Parameter Model. The simplest efficiency model, an isentropic analysis model, requires only the isentropic and volumetric efficiency as inputs and is suitable for a parametric design study. Efficiency values can be derived from experimentation or varied as a design parameter. The most complex model is also the most accurate, the distributed parameter model solves a complete set of energy, momentum, and continuity equations and is suitable for detailed analysis of the compressor. What the efficiency model may lack in accuracy it makes up for in flexibility and reduced computational effort. For this study the efficiency model has been chosen because the focus is on the system configuration and overall feasibility. The three compressor efficiencies that are required for the basic model are listed in Table 29.

Table 29. Compressor efficiencies [Domanski 2005].

Type	Symbol	Efficiency
Electric motor efficiency	$\eta_{comp,motor}$	0.85
Compressor isentropic efficiency	$\eta_{comp,isen}$	0.65
Compressor volumetric efficiency	$\eta_{comp,vol}$	0.7

The model for the compressor is simplified by assuming that the compressor is operating in steady-state adiabatic compression. For a given electrical input to the compressor the work done on the refrigerant is,

$$\dot{W}_{comp} = \dot{W}_{elec}\eta_{comp,motor}$$

Refrigerant mass flow rate depends on a number of factors including the cooling load and the heat exchanger temperatures. The mass flow rate for an ideal gas can be found from the compressor inlet temperature and pressure using the equation [Ueno 2003],

$$\dot{m} = f_c \eta_{comp,vol} \frac{P_1 V_s}{R T_1}$$

where, f_c is the compressor rotational frequency in Hz, $\eta_{comp,vol}$ is the volumetric efficiency of the compressor, V_s is the swept volume of the compressor, P_1 and T_1 are the pressure and temperature at the compressor inlet respectively, and R is the ideal gas constant. The same equation for compressor mass flow rate using density in place of the ideal gas law is,

$$\dot{m} = f_c \eta_{comp,vol} V_s \rho$$

where the refrigerant density is found in a lookup table using temperature and pressure of the refrigerant at the compressor inlet. The compressor used in literature is a piston compressor (ZEM HQY75AA) with a cooling capacity 118 W and 1.49 COP at ASHRAE conditions (55/-23.3°C) for R600a refrigerant [Bjo¨rk 2006]. Compressor product literature provides data for input power in Watts and COP as a function of the refrigerant evaporating temperature in degrees Celsius, this data is used in a lookup table in the model. The known compressor characteristics are shown in Table 30.

Table 30. Known compressor characteristics.

Variable	**Symbol**	**Value**
Compressor rotational frequency	f_c	50 (Hz)
Swept volume of compressor	V_s	7.39 cm^3
Evaporating Temperature (°C)	**Input Power (W)**	**COP**
-10	118	1.89
-15	105	1.71
-20	93	1.52
-23.3	85	1.40
-25	81	1.34
-30	69	1.16

Refrigerant mass flow rates reported in literature for domestic refrigeration range from 0.0009 kg/s [Hepbasli 2007] to 0.0015 kg/s [O'Brien 1998].

3.2.2 Heat Exchanger Models

A detailed heat exchanger model is not required since the overall heat transfer coefficient (UA) of the evaporator and condenser are documented in existing literature. A number of models can be found in literature that generally increase in complexity with increased accuracy.

1. Single-Node Model (Lumped Parameter Model - LPM)
    a. With Log Mean Temperature Difference (LMTD) is simple but not good for phase change.
2. Multi-Node Model (Distributed Parameter Model)
    a. Control Volumes set with parameters lumped in each.

3. Zone Model
    a. Several zones are set up and parameters in each are lumped.
    b. The accuracy is better than the lumped parameter and faster than the distributed or finite model.
    c. Typically 3 zones for a condenser (super-heated gas, two-phase, & sub-cooled).
    d. Typically 2 zones for an evaporator (two-phase & super-heated).
4. Finite Model
    a. Highest accuracy of results but computationally expensive.

A selection of calculation method is not required since the refrigerant is primarily 2-phases in both heat exchangers, although the condenser does de-super heat the refrigerant. In the simplified refrigerant cycle used, the evaporator heat exchanger inlet and outlet temperature are equal. The condenser heat exchanger has a slightly higher inlet temperature, with super-heated refrigerant entering, than outlet temperature but the temperature difference is small and can be considered constant at the condensation temperature.

If the model grows in scope and complexity, to increase accuracy the LMTD method would be preferred as the size and type of heat exchanger will be known and only the heat exchanger(s) inlet and outlet temperatures will be required. Since the simulation will be a steady-state system, convergence must occur, and the evaporator outlet temperature is the same as the compressor inlet temperature.

1. Log-Mean-Temperature-Difference (LMTD)
    a. Design Problem
    b. Easy when inlet and outlet temperatures are known or specified.
    c. When only inlet temperature is known the solution must be found through an iterative procedure.

$$q = UA\Delta T_{lm}$$

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)}$$

2. Effectiveness-Number of Transfer Units (ε-NTU)
    a. Performance Calculation
    b. Suitable when only the inlet temperatures are known along with the effectiveness of the heat exchanger.
    c. Requires tables/figures to correlate effectiveness (ε) and NTU.

$$q = \varepsilon C_{\min}\left(T_{h,i} - T_{c,i}\right)$$

$$NTU \equiv \frac{UA}{C_{\min}}$$

3.2.2.1 Condenser

The condenser has a fixed overall heat transfer coefficient (UA = 7.7 W/K) with an overall size of 1.33x0.51x0.008 m^3 based on literature [Bjo¨rk 2006]. It is comprised of 53 steel vertical wires on each side of the tubing, each of diameter 1.5 mm. The tubing has an internal and external tube diameter of 3.5 mm and 5.0 mm respectively. Heat transfer on the exterior of the condenser is free convection with the refrigerant flow horizontally downward.

To simplify the model, potential, kinetic, and chemical changes in the refrigerant are neglected, as is axial heat transfer. Refrigerant enters the condenser super-heated and since the pressure drop is neglected, the refrigerant remains at a constant pressure throughout the condenser. The refrigerant temperature decreases along iso-baric lines while super-heated until the refrigerant is saturated vapor. While the refrigerant is in the 2-phase region the temperature then remains constant then continues decreasing along iso-baric lines while subcooled where it exits the condenser. The environment that the condenser is exposed to is assumed to be a significantly large heat sink so that the environmental temperature T_{env} is a constant temperature of 25°C or 32°C based on ISO test standard.

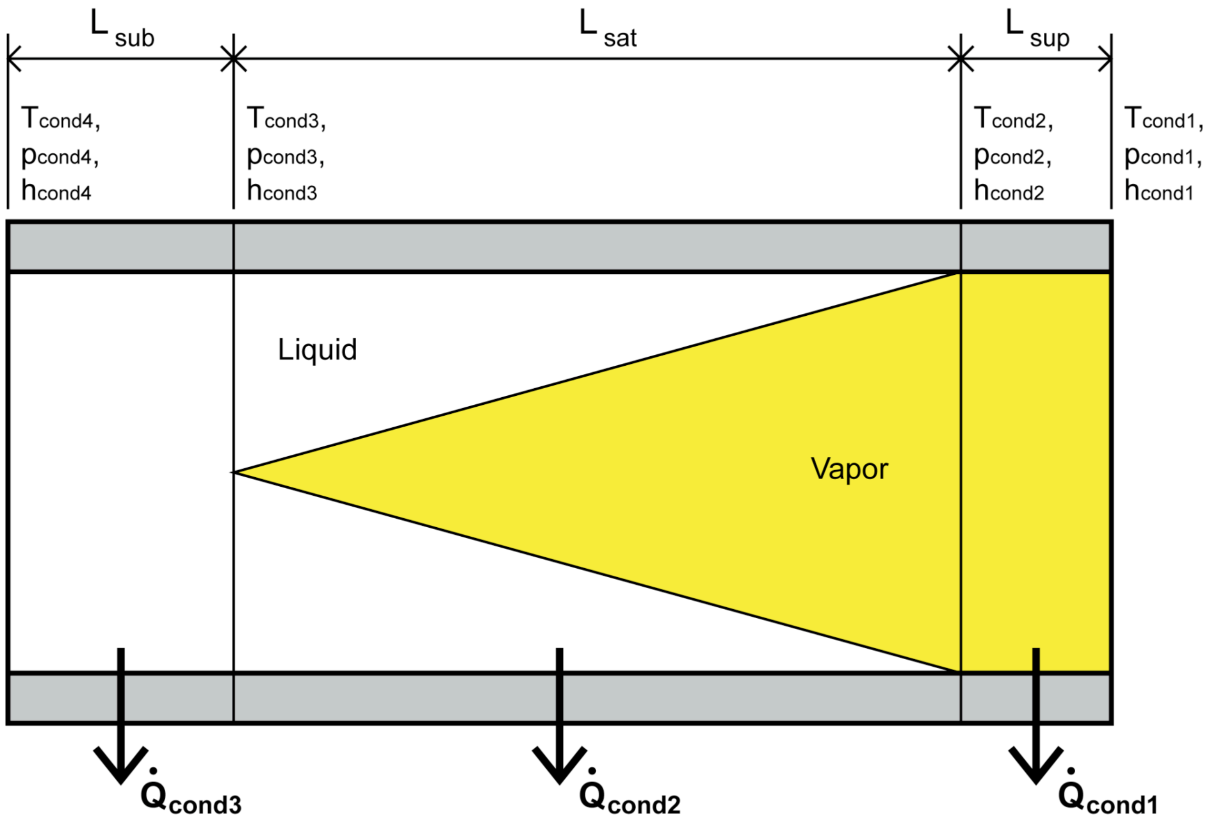


Figure 40. Condenser control volume.

A basic model has one control-volume for the super-heat vapor, one for the two-phase region, and another for the subcooled liquid as shown in Figure 40. The heat flux of each control-volume is summed for the total condenser heat flux.

$$\dot{Q}_{cond} = \dot{Q}_{cond1} + \dot{Q}_{cond2} + \dot{Q}_{cond3}$$

Expressions of heat flux for each control-volume,

$$\dot{Q}_{cond1} = \dot{m}_r\left(h_{cond1} - h_{cond2}\right)$$
$$\dot{Q}_{cond2} = \dot{m}_r\left(h_{cond2} - h_{cond3}\right)$$
$$\dot{Q}_{cond3} = \dot{m}_r\left(h_{cond3} - h_{cond4}\right)$$

When each control-volume heat flux is combined, the total heat flux shows only the input and outlet state is required.

$$\dot{Q}_{cond} = \dot{m}_r\left(h_{cond1} - h_{cond4}\right)$$

Relating back to the system states the condenser heat flux becomes,

$$\dot{Q}_{cond} = \dot{m}_r\left(h_2 - h_4\right)$$

By neglecting energy storage in the refrigerant, the condenser heat flux must be equal from the refrigerant side and from the heat exchanger side. The equation for the condenser heat flux from the heat exchanger side is,

$$\dot{Q}_{cond} = UA_{cond}\left(T_{cond} - T_{env}\right)$$

These two equations for the condenser heat flux must be solved simultaneously, or as in the Matlab model, solved iteratively to find the condenser temperature T_{cond} and the condenser heat flux $\dot{Q}_{cond}$.

3.2.2.2 Evaporator

The evaporator has a fixed overall heat transfer coefficient (UA = 3.7 W/K) with an overall size of 0.66x0.49x0.0014 m^3 based on literature [Bjo¨rk 2006]. Aluminum, plate type, with internal hydraulic diameter 3.2 mm.

To simplify the model, potential, kinetic, and chemical changes in the refrigerant are neglected, as is axial heat transfer. Refrigerant enters the evaporator as a variable quality 2-phase. Since the pressure drop is neglected, refrigerant remains at a constant pressure and temperature throughout the evaporator to exit as saturated vapor at evaporator outlet. The environment that the evaporator is exposed to is the refrigerator cabinet temperature, T_{cab} which is maintained at 5±1°C per the ISO test standard.

Enthalpy and temperature at the evaporator inlet are used to find the quality,

$$h_5 = x_5 h_{5v} + (1 - x_5) h_{5l}$$

$$x_5^2 (h_{5v} - h_{5l}) + x_5 (h_{5l} - h_5) = 0$$

$$x_5 = \frac{-(h_{5l} - h_5) \pm \sqrt{(h_{5l} - h_5)^2 - 0}}{2(h_{5v} - h_{5l})}$$

$$x_5 = \frac{h_{5l} - h_5}{h_{5l} - h_{5v}}$$

Entropy found from quality and temperature,

$$s_5 = x_5 s_{5v} + (1 - x_5) s_{5l}$$

Relating back to the system states the evaporator heat flux becomes,

$$\dot{Q}_{evap} = \dot{m}_r \left(h_1 - h_5 \right)$$

As with the condenser, the evaporator heat flux must be equal from the refrigerant side and from the heat exchanger side. The equation for the evaporator heat flux from the heat exchanger side is,

$$\dot{Q}_{evap} = UA_{evap} \left(T_{cab} - T_{evap} \right)$$

These two equations for the evaporator heat flux must be solved simultaneously, or as in the Matlab model, solved iteratively to find the evaporator temperature T_{evap} and the evaporator heat flux $\dot{Q}_{evap}$.

3.2.3 Expansion (Capillary Tube)

The expansion device in most refrigerators is a small diameter tube known as a capillary tube. The model assumes that there is constant enthalpy, adiabatic expansion in the capillary tube. This means that there is no heat lost from the resulting pressure and temperature decrease. There is typically such a small amount of energy lost in the real expansion of the refrigerant that it is reasonable to neglect it.

3.2.4 Envelope Model "Cabinet"

The refrigerator envelope, or cabinet, is based on Electrolux model number ER8893C domestic refrigerator unit with a single door and single, all-refrigerator compartment with a declared energy consumption of 0.68 kW h/24 h [Bjo¨rk 2006]. Exterior height, width, and depth dimensions are 1.75m, 0.6m, and 0.6m respectively. Internal volume is 350 liters and since there is no freezer, the adjusted volume is the same.

Wall insulation has an average insulation value of R-7.5$\left(\frac{ft^2\ {}^\circ F\ hr}{Btu}\right)$ [EERE 2008e]. Insulation thicknesses 3.8 cm (1.5 inch) for the doors, 5.5 cm (2.2 inch) average for the freezer walls, and 4.3 cm (1.7 inch) average for the fresh food walls [DOE 1995b]. Freezer wall thickness is not used since the refrigerator has no freezer unit. Declared overall heat transfer coefficient (UA) for the cabinet is 2.3 W/K [Bjo¨rk 2006]. A simple calculation of the overall heat transfer coefficient (UA) for the cabinet using just the conduction through the wall insulation results in a value of 1.8 W/K. It is expected that the value of the simple calculation would be less than the actual due to the neglect of convection, radiation, increased heat transfer at the seams (edge effects), gaskets, holes in the envelope for refrigeration and electrical components, and air infiltration. It is assumed that no heat from condenser re-enters the refrigerator cabinet and that the environment T_{env} is sufficiently large and has constant temperature.

The cabinet is modeled as a finite lumped capacitance volume. Although spatial temperature distributions inside the cabinet are neglected, the cabinet temperature T_{cab} is transient to capture the heating and cooling cycles. The relative humidity of the air in the cabinet is assumed to be constant at 25% with the mass fraction held constant based on 25%

relative humidity at 5°C. Cabinet thermal mass includes cabinet components such as metal racks, glass shelves, plastic bins, and plastic liner. Cabinet thermal mass excludes all refrigeration components and refrigerant.

An attempt has been made to capture a realistic thermal mass inside a refrigerator cabinet by documenting "typical" contents. Items with small mass and low specific heat have been ignored while foods high in water content have been modeled simply as water. Thermal load inside cabinet is modeled as,

1. 2x Pyrex 1-1/2cup/390ml glass container (310g), plastic lid (31g), & water (293ml).
2. 3x Pyrex 1 quart/946ml glass container (621g), plastic lid (g), & water (710ml).
3. 1x gallon of milk
4. 1x 2-liter bottle of soda
5. 6x glass bottle beverages 558g total for each bottle with 355ml liquid (355g liquid, 203g glass)
6. 16x 400ml condiments (water)
7. 10x 16oz cheese, cream cheese, tofu, etc. (16floz = 473ml)

Accessory loads from the light, defrost and fans have been ignored. The light load has been ignored since the door remains closed for both the ANSI and ISO test procedures. Defrost cycle is also ignored even though most domestic units have automatic cycles as this would likely remain unchanged as the system configuration changes. Fans in the cabinet, and on the heat exchangers have been ignored. Since the heat exchanger UA values are fixed and based on free convection, the performance is not dependent on fan operation, so the fan loads are ignored. Furthermore, without detailed heat exchanger heat transfer calculations, the fan operation would only add a small and fixed percentage to the refrigerator electrical consumption. In this system configuration study, the difference in energy consumption is more important than the value of energy consumption for any one scenario. Condenser and evaporator fan motors would typically run while the compressor is on and cooing is occurring. Sample power consumption values for the evaporator fan motor and the condenser fan motor are 8.0W and 11.6W respectively [DOE 2005].

3.2.5 Refrigerant Properties

Although many refrigerants can be used in a vapor-compression refrigeration cycle, only a few are commercially viable due to a combination of constraints such as cost, regulatory issues, and overall system efficiency. The two that are used in the simulation are 1,1,1,2-tetrafluoroethane (R134a) and isobutane (R600a). Although the refrigerant R134a is the most common refrigerant used in residential refrigeration application in the United States, isobutane (R600a) is used as the primarily refrigerant in the simulation as it can most accurately be verified against literature [Bjo¨rk 2006]. General characteristics of R600a, R134a, and the refrigerant they replaced, Freon-12 (R12), are listed in Table 31.

Table 31. Refrigerant properties.

Refrigerant	R 600a	R 134a	R 12
Name	Isobutane	1,1,1,2-Tetra-flouro-ethane	Dichloro-di-fluoro-methane
Formula	$(CH_3)_3CH$	CF_3-CH_2F	CF_2Cl_2
Critical temperature in °C	135	101	112
Molecular weight in kg/kmol	58.1	102	120.9
Normal boiling point in °C	-11.6	-26.5	-29.8
Pressure -25 °C in bar (absolute)	0.58	1.07	1.24
Liquid density at -25 °C in kg/l	0.6	1.37	1.47
Vapor density at t_o -25/+32 °C in kg/m3	1.3	4.4	6
Volumetric capacity at -25/55/32 °C in kJ/m3	373	658	727
Enthalpy of vaporization at -25 °C in kJ/kg	376	216	163
Pressure at +20 °C in bar (absolute)	3	5.7	5.7

Refrigerant properties of state are calculated in a steady-state refrigerant cycle sub-routine from tabulated values from the National Institute of Standards and Technology (NIST) on-line Webbook [NIST]. Values of two known parameters of the refrigerant cycle are sent to the sub-routine, where the two unknown refrigerant cycle variables are found through the process of iteration.

Table 32. Parameters and variables of refrigerant sub-routine.

Known parameters	**Unknown variables**
Temperature, cabinet (T_{cab})	Temperature, Evaporator outlet/Compressor inlet (T_{evap}, T1c)
Temperature, condenser environment (T_{env}, T_{cond_env})	Temperature, Condenser inlet (T_{cond}, T3c)
	Mass flow rate of refrigerant ($\dot{m}$)

3.3 Domestic Water Heater (DHW) Storage Tank Simulation Model

The purpose of the storage tank model is to first develop a baseline for energy consumption in a typical domestic hot water heater configuration and then to perform as either a heat sink or a combined heat sink and water heater in a combined system state. The basic assumptions for the DHW storage model are:

1. Lumped capacitance water model
2. Transient temperature model with energy storage
3. Energy generation by an electric resistance element
4. Water thermodynamic properties taken at a constant temperature and pressure
5. Water is drawn from the storage tank based on flow rate and flow duration from a realistic daily profile (RDP)
6. Drawn water is replaced by constant temperature make-up water
7. Heating element cut-in and cut-out temperature is 54°C and 56°C respectively
8. Environment is sufficiently large such that ambient temperature is constant

Temperature set-point of the storage tank when in the DHW heating mode is 55±1°C. The environment is assumed to be large and hold a constant temperature of 25°C or 32°C in a combined configuration based on ISO test points for refrigerators. Thermodynamic properties for water are constant, evaluated at 50psig (0.3448 MPa) and 315°K [NIST] while air properties are taken at 300°K [Incropera 2002].

3.3.1 Energy Balance

In the general energy equation for the DHW, the sum of energy in and energy generated equals the sum of energy out and energy stored.

$$E_{in} + E_{gen} = E_{out} + E_{st}$$

The same energy equation based on the rate of change is,

$$\dot{E}_{in} + \dot{E}_{gen} - \dot{E}_{out} = \dot{E}_{st}$$

The rate of energy inflow is due to water mass flow of the mains make-up and heat exchanger (condenser) heat transfer to the DHW,

$$\dot{E}_{in} = \dot{m}_w c_{p,w} T_{w,in} + \dot{Q}_{hx}$$

If the DHW is at higher temperature than the condenser the heat transfer would be negative in value and the DHW would loose heat to the refrigerant cycle.

With the control volume drawn around the DHW tank, the mass balance is,

$$\dot{m}_{w,out} = \dot{m}_{w,in} = \dot{m}_w$$

where the mass flow of water leaving the DHW (draw-off) is equal to the mass flow of water entering the DHW (make-up).

Energy generation in the DHW is due to an electric resistance heater, the value is the product of the electrical power of the heater element and the recovery efficiency of the DHW. The heating elements have 4.5 kW heating power and 96.8% recovery efficiency, consistent with the DOE characteristic values.

$$\dot{E}_{gen} = \dot{W}_{elec}\, \eta_{recovery}$$

The rate of energy outflow is due to the mass flow of water from the DHW and the heat loss to the environment. Here heat loss to the environment is defined as negative, if the mains water is colder than the environment, energy out could be positive.

$$\dot{E}_{out} = \dot{m}_w c_{p,w} T_{w,out} + \dot{Q}_{dhw,loss}$$

Energy storage term is equal to the rate of change in the internal thermal energy of the water,

$$\dot{E}_{st} = \frac{dU_{st}}{dt} = \frac{d}{dt}\left(\rho V c T\right)$$

Taking the density (ρ), volume (V), and specific heat (c) as constants the energy storage term can be expressed as,

$$\dot{E}_{st} = \rho V c \frac{dT}{dt}$$

where specific heat c, is equal to c_v and for a liquid c_p is equal to c_v.

The transient temperature model of the storage tank requires that the water temperature be calculated repeatedly on the time interval dt. The water temperature at the next time step, T_{t+dt}, is dependent only on the current water temperature T_t and the rate of change of energy storage $\dot{E}_{st}$ as the density, volume, and specific heat remains constant.

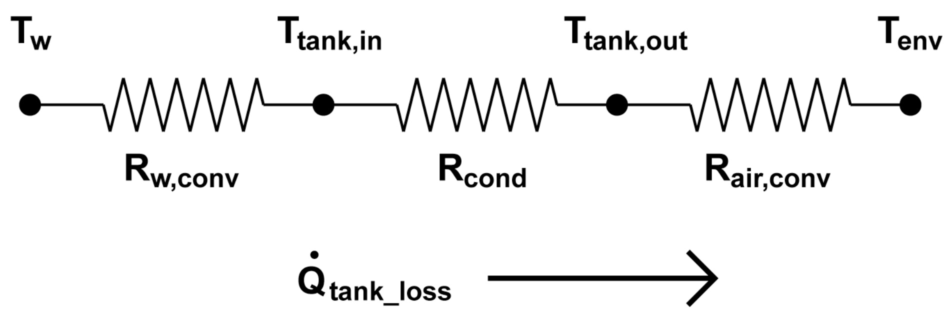
$$T_{t+dt} = T_t + \frac{\dot{E}_{st}}{\rho V c}$$

3.3.2 Lumped Capacitance

The thermodynamic model of the storage tank utilizes lumped capacitance to keep the model simple. Logically, we know that there is some temperature gradient within the tank and we have seen earlier in the literature review that stratification, the vertical temperature distribution, should be encouraged to increase energy efficiency.

3.3.3 Heat Transfer

Heat loss from the water in the DHW to the environment includes convective heat transfer from water to the inside tank wall, conduction through the multiple layers of materials in the tank wall, and convection of the exterior of the DHW. Radiation to the environment also occurs in real systems but is likely to be small since the temperature difference is small between the environment (air) and exterior surface of the storage tank.



Overall heat transfer coefficient, UA, is interpolated from DOE characteristic values so that the detailed calculation of conduction, convection, and radiation is not required. A sample tank volume of 40 gallons results in an interpolated UA value of 1.79 W/K. A simple calculation of the overall heat transfer coefficient using only conduction through only insulation is performed to verify the DOE value. In this simple calculation the multiple layers in the tank wall are ignored as the thermal conductivity (k) of the insulation is many times less than steel or glass, which comprise most domestic water heater tanks.

$k_{insulation}$ = 0.0219 W/mK

k_{steel} = 63.9 W/mK (AISI 1010 Steel)

k_{glass} = 1.4 W/mK

The equation for heat transfer for one-dimensional conduction is,

$$\dot{Q} = \frac{k\ A}{t} \Delta T = UA\ \Delta T$$

where A is equal to the area of the surface where heat is transferring and t is the insulation thickness. The resulting equation for the overall heat transfer coefficient is,

$$UA = \frac{k\ A}{t}$$

The overall heat transfer coefficient for a simplified forty-gallon tank with one, two, and three inches of insulation is 1.46, 0.73, and 0.49 W/K respectively. While all UA values are less than the interpolated DOE value of 1.79 W/K, this is to be expected as heat conduction paths that offer less resistance such as heating elements and steel pipe fittings have been neglected.

3.3.4 Water Temperature In

The temperature of water mains (make-up) is an average annual value of 12.5°C based on a minimum daily average temperature in colder climates of 3°C and maximum daily average temperature of 23°C [Biaoua 2007].

A more detailed simulation could use a sine wave with a minimum of 2°C and a maximum of 23°C with a period of 365 days and a phase shift of 46.25 days so that the minimum temperature coincides with February 14th and the maximum August 14th. This sine wave temperature profile is shown in Figure 41.

$$T_{w,in} = 12.5 - 10.5\left(\sin\frac{(d + 46.25)2\pi}{365}\right)$$

$$\begin{aligned}
&Amplitude = \frac{23° - 2°}{2} = 10.5° \\
&Period = 365\,days \\
&Phase\ Shift = 46.25\,days \\
&Offset = 2° + 10.5° = 12.5°
\end{aligned}$$

where d is the day of the year with day one on January 1st.

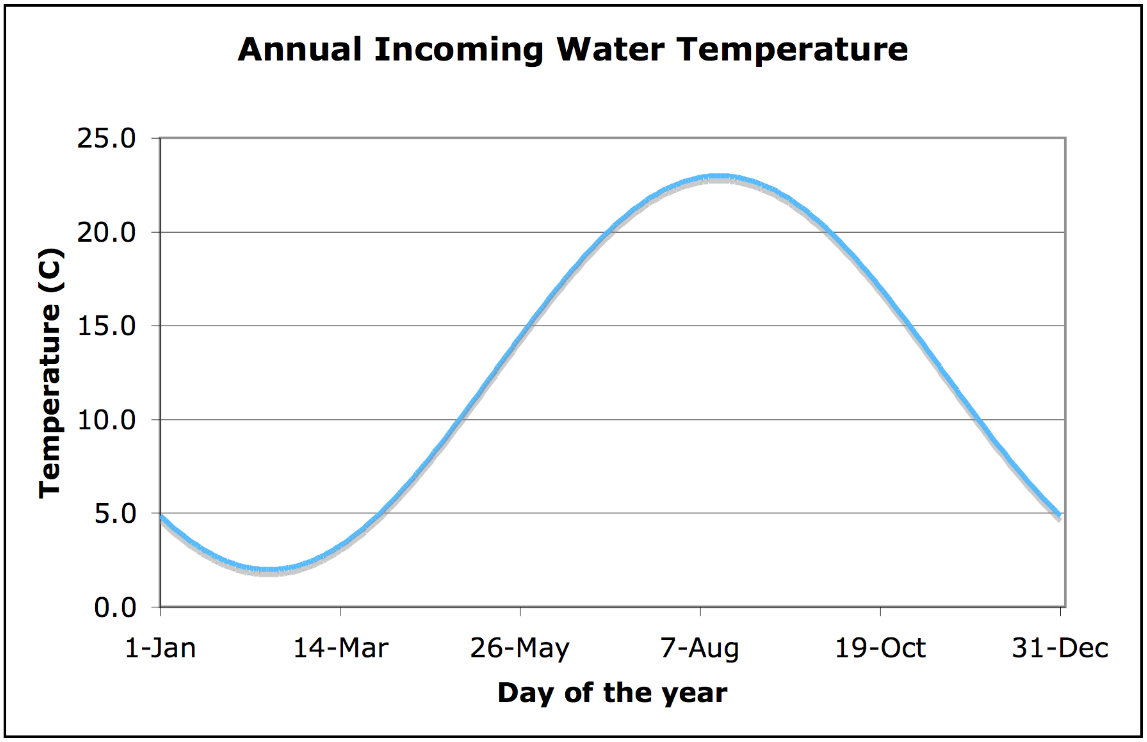


Figure 41. Annual incoming water temperature.

In the combined system, warmer incoming water and warmer storage tank water temperatures will increase the performance of the DHW system but reduce the performance of the refrigerator condenser (heat exchanger) by forcing heat rejection to a higher temperature environment. Conversely, the refrigerator generally performs best with lower condenser temperatures as long as frost does not accumulate as to impede heat transfer to the environment.

3.4 Verification of model

3.4.1 Refrigerator Verification

The Matlab refrigerator model is verified by representing a residential refrigerator unit that is well documented in size, heat transfer characteristics, refrigeration cycle components, refrigerant, and temperature conditions [Bjo¨rk 2006]. The unit is an Electrolux ER8893C, single door and single compartment domestic refrigerator with the following data:

- Cabinet: (1.75x0.6x0.6 m^3), 350 l internal volume, UA value 2.3 W/K.
- Evaporator: free convection, UA value 3.7 W/K.
- Condenser: free convection, UA value 7.7 W/K.
- Piston compressor (ZEM HQY75AA) with low pressure oil sump and 265 ml mineral oil charge. Cooling capacity 118 W and COP 1.49 at ASHRAE conditions (55/-23.3°C).
- Refrigerant: Isobutane (R600a).
- Capacity control by intermittent run (on/off cycling) with self-defrosting in every off-period. Declared energy consumption 0.68 kW h/24 h.

Assuming that there is no change in cabinet contents and the door closed, the minimum energy required to maintain steady state conditions is equal to the heat gain through the walls of the cabinet. Based on given values for the overall heat transfer coefficient and ISO test points for the cabinet and ambient temperature, the rate of heat transfer from the environment to the refrigerator cabinet is,

$$\dot{Q}_{cab} = UA_{cab}\left(T_{env} - T_{cab}\right)$$

$$\dot{Q}_{cab} = 2.3(25 - 5) = 46\ W$$

For a 24 hour period the heat transfer from the environment to the refrigerator cabinet is,

$$Q_{cab} = \dot{Q}_{cab} * 24h$$

$$Q_{cab} = 46 * 24 = 1104\ Wh$$

Based on the declared daily energy consumption, the coefficient of performance of the refrigerator must be,

$$COP = \frac{Q}{W} = \frac{1104}{680} = 1.62$$

Similar low-back pressure compressors have coefficients of performance of 1.40 and 1.48 for refrigerant R600a and 1.336 and 1.44 for refrigerant R134a at ASHRAE conditions and an evaporating temperature of -23.3°C. The ASHRAE compressor test conditions for are shown in Table 33.

Table 33. Compressor test conditions.

Testing cycle conditions	CECOMAF		ASHRAE	
	LBP(A)	HMBP(C)	LBP(B)	HMBP(D)
Condensing temperature	55		55	55
Liquid temperature	55		32	46
Superheating temperature	32		32	35
Suction temperature	32		32	35
Ambient temperature	32		32	35

The Matlab model results are shown in Table 34 compared against experimental test data from literature [Bjo¨rk 2006]. Also shown in Table 34 is declared manufacturer performance for the refrigerator and compressor. Matlab results are generated for one day with a time step of ten seconds.

Table 34. Refrigerator verification results.

	Temperature (°C)				Qevap (kWh)	Qcab (kWh)	Wref (kWh)	COP
	Ambient	Condenser	Cabinet	Evaporator				
Refrigerator Declared	25		5			1.104	0.68	1.62
Experimental	25	44.7	5	-10.9			0.61	
Matlab	25	46.4	5	-24.6	1.097	1.103	0.808	1.36
Matlab Error		1.7 °C		-13.7 °C			32%	16%
Compressor Declared	32	55		-23.3				1.48
Experimental	31	53.2	5	-11.8			1.01	
Matlab	31	52.9	5	-23.9	1.418	1.435	1.096	1.29
Matlab Error		-0.3 °C		-12.1 °C			9%	13%

The Matlab model matches the cabinet heat gain and the condenser temperature very well, but represents the evaporator temperature and the energy use with significant error. The time step duration of ten seconds appears to be a reasonable compromise as the average temperature rise or fall of the cabinet that takes place in that duration is only 0.005°C. With a ten second time step a single 24 hour run of the Matlab model takes between two and four hours to complete.

3.4.2 Domestic Hot Water Heater Verification

Energy consumption of the Matlab DHW model is compared to DOE average values to validate the assumptions in the Matlab model. The DOE model, WHAM, is also evaluated for another reference point for comparing the Matlab model since the volume and thermal characteristics of the DOE average data is not available. As shown previously in Table 14, the average energy consumption for electric resistance water heaters is 3460 kWh per year for an average water draw of 171.5 liters per day, or 9.479 kWh per day for the same water draw.

For the electric water heater, checking the daily energy consumption values using WHAM, average US hot water use, characteristic values for a 40 gallon tank and representative temperatures,

Vol = 171.5 (l/day)
$T_{\tan k}$ = 55 (°C)
T_{in} = 12.5 (°C)
T_{∞} = 25 (°C)
UA = 1.79 (W/°K)
η_{RE} = 0.968
P_{on} = 4.5 (kW)
for water at 315°K
ρ = 991.6 (kg/m^3)
c_p = 4.179 (kJ/kg*°K)

From WHAM the daily energy requirement compares favorably with the DOE US average value of 9.479 kWh/day,

$$Q_{in} = 9.853\frac{kW \cdot hr}{day}$$

The Matlab model was configured alternately with a continuous water draw and two realistic daily profiles that bracket the average draw volume to generate energy consumption values for the average US daily water draw of 171.5 liters. To eliminate error due to the heating and cooling cycle in the dynamic model, the average of four consecutive day's energy consumption values are used. Results of the

Matlab and WHAM models are compared to the DOE average value in Table 35.

Table 35. DHW Daily total energy consumption during operation.

		Model - Light RDP	Model - Medium RDP	Model - US average (Interpolate from RDP)	Model - US average (Continuous Draw)	WHAM - US average	DOE - US average
Draw volume	liters	100.85	181.52	171.5	171.5	171.5	171.5
Energy	kWh/day	6.2165	10.084	9.603	10.037	9.853	9.479
Error from DOE	%			1	6	4	0

* Model and WHAM based on a 40 gallon tank, electrical resistance heat, and thermal properties based on DOE characteristic values.

With daily energy consumption values correlating within ten percent to DOE values, the DHW Matlab model can be used with confidence for further study.

3.5 Combined System Simulation Model

Assumptions for the combined model include assumptions for the independent refrigerator and DHW models in addition to the following:

1. Combined configuration
    a. Refrigerator condenser is placed in the DHW storage tank.
    b. Overall heat transfer coefficient (UA) of the condenser heat exchanger is adjusted for the change in environment. In a refrigerator only system the condenser rejects heat to air, in the combined system the condenser rejects heat to water. In an optimized combined system the condenser would likely

change from a heat exchanger of steel wires on tubes to a copper or stainless steel coil.

2. Combined pre-heat configuration
    a. Refrigerator condenser is placed in a heat sink, a smaller pre-heat tank
    b. As with the combined configuration, the overall heat transfer coefficient of the condenser is adjusted for rejection of heat to water.
    c. Heat sink water is fully mixed
    d. DHW make-up water temperature is equal to heat sink

3.5.1 Home Energy Centre (HEC)

Based on the Home Energy Centre (HEC) evaluated in the International Journal of Energy Research [O'Brien 1998] and the recommendations from the same paper, I propose a different configuration of the integrated refrigerator and hot water heater. Changes based on the evaluation of the HEC:

1. Remove the heat sink from the hot water tank to reduce thermal losses from the hot water tank.
2. Use the heat sink as a pre-heat reservoir for the hot water tank. Utilize stratification in the heat sink.
3. Keep the heat sink at a lower temperature so, more refrigerator waste heat can be absorbed by the domestic water heater supply (DHW) and the refrigerator operates more efficiently because the condenser, where the refrigerator is rejecting heat to, is at a lower temperature.

Similar to the HEC, the system recovers heat from the refrigerator condenser. Instead of rejecting heat to the high-temperature heat sink or DHW storage tank, the heat from the refrigerator condenser is rejected to a lower-temperature heat sink responsible for pre-heating water for the DHW storage tank shown in Figure 42.

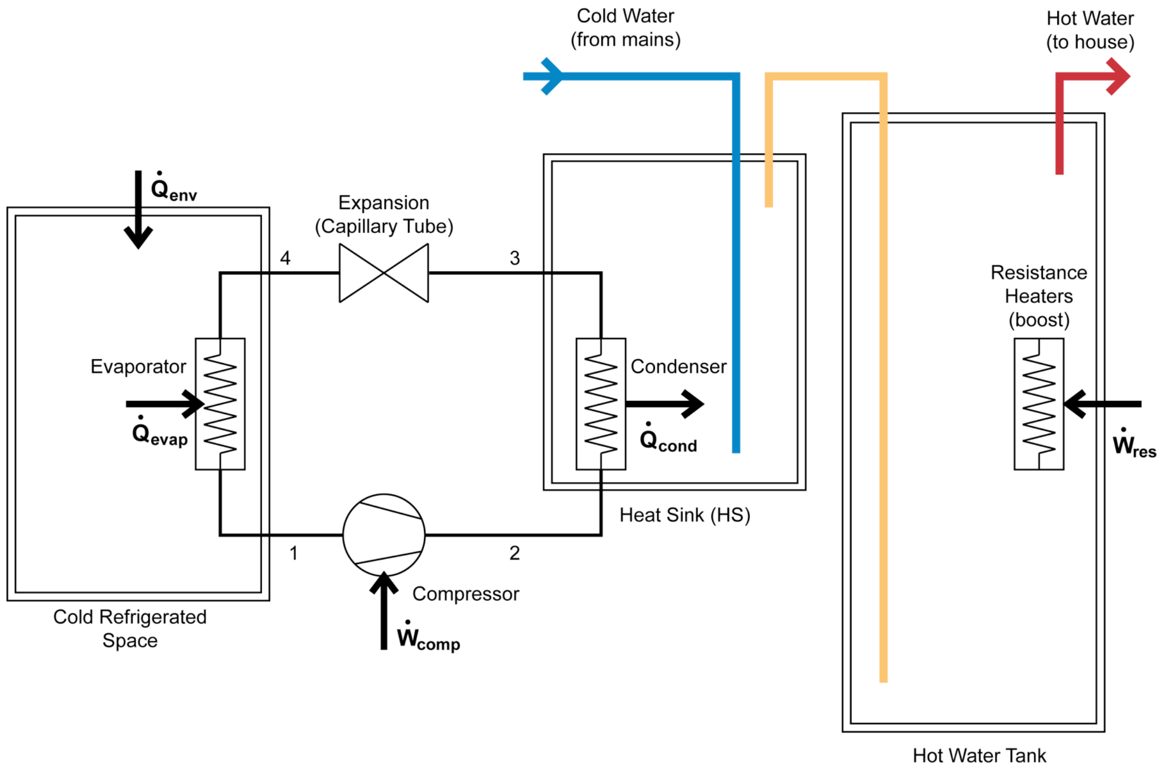


Figure 42. Revised HEC configuration, preheat for DHW.

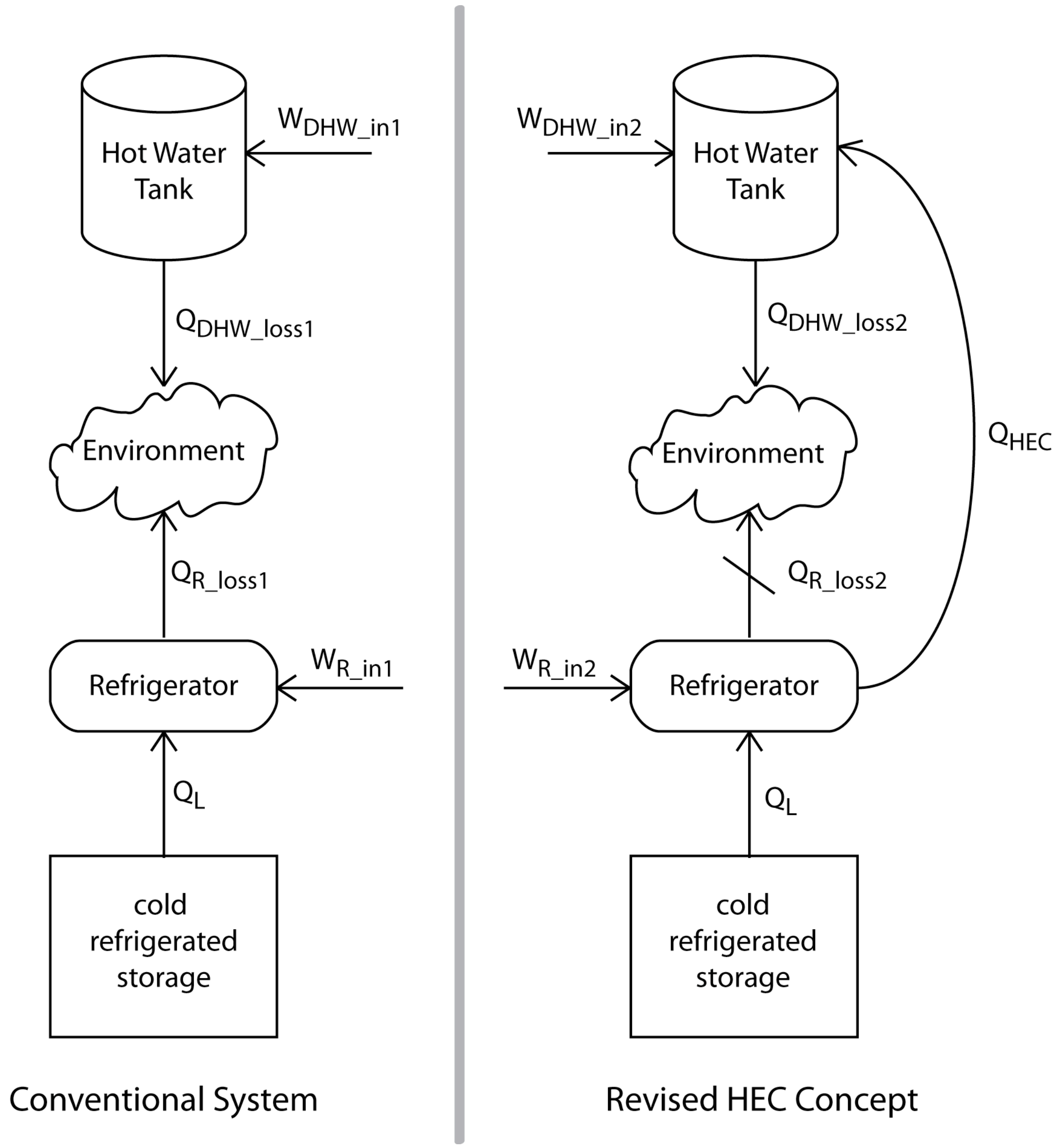


Figure 43. Revised HEC concept objectives.

The primary goal is to reduce the energy into the system ($W_{R\_in} + W_{DHW\_in}$). To accomplish this we will first determine the conditions required to capture as much waste heat (Q_{HEC}) from the refrigerator that would otherwise be lost to the environment as shown in Figure 43. That is to maximize Q_{HEC} and minimize Q_{R_loss2} in order to reduce W_{DHW_in2}.

The second step is to reduce the input to the refrigerator (W_{R_in2}) again by determining the appropriate conditions.

Coefficient of performance of the HEC is defined as,

$$COP_{HEC} = \frac{Q_{evap} + Q_{w\_out}}{W_{comp} + W_{DHW}}$$

The efficiency comparison is introduced by O'Brien et all as,

$$\phi_{ref} = \frac{W_{HEC,ref}}{W_{ref}}$$

$$\phi_{DHW} = \frac{W_{HEC,DHW}}{W_{DHW}}$$

where $W_{HEC,ref}$ and $W_{HEC,DHW}$ are the energy consumed by the refrigerator and the DHW components respectively in a HEC system, W_{ref} and W_{DHW} are the energy consumed when the refrigerator and DHW respectively are operated independently, and a ϕ value of less than one indicates reduced energy consumption by the HEC.

3.5.2 Correction for Condenser Overall Heat Transfer Coefficient (UA)

To correct the overall heat transfer coefficient (UA) for the condenser when submerged in water it is assumed that the area of the heat exchanger remains the same, only the effectiveness changes based on the ratio of heat transfer resistances for air and water.

The overall heat transfer coefficient is equal to the inverse of the effective resistance R_{eff} to the heat transfer,

$$UA = \frac{1}{R_{eff}}$$

As the thermal resistances are in series, the effective resistance is the sum of individual resistances,

$$R_{eff} = R_{conv,i} + R_{cond,tube} + R_{conv,o}$$

where $R_{cond,tube}$ is the thermal resistance due to conduction through the radial wall of the tube and $R_{conv,i}$ and $R_{conv,o}$ are the thermal resistance values due to convection on the inside and outside of the tube wall respectively. The general equation for conduction in a one-dimensional radial system is,

$$R_{cond} = \frac{\ln\left({r_o}/{r_i} \right)}{2\pi\, k\, L}$$

where r_o and r_i are the external and internal radius of the cylinder respectively, k is the thermal conductivity of the tube wall, and L is the length of the tube. The general equation for convection in a one-dimensional radial system,

$$R_{conv} = \frac{1}{2\pi\, r\, h\, L}$$

where r is either the inside or the outside tube radius and h is the corresponding convection coefficient. With the overall heat transfer coefficient given and the wall conduction and outside convection resistance in air easily calculated, the inside convection resistance can be solved for.

3.5.2.1 Convection Coefficients

The average convection coefficient, $\bar{h}$, for the outside surface of the condenser in air is based the Nusselt number,

$$\overline{Nu}_D = \frac{\bar{h}D}{k}$$

where D is the characteristic diameter of the geometry, k is the thermal conductivity of the air. The Nusselt number can be found from the following correlation for a long cylinder in natural convection [Incropera 2002],

$$\overline{Nu}_D = \left\{ 0.60 + \frac{0.387 Ra_D^{\ 1/6}}{\left[1 + (0.559/\Pr)^{9/16}\right]^{8/27}} \right\}^2 \qquad Ra_L \le 10^{12}$$

where the Rayleigh number, the product of the Grashof number and the Prandtl number,

$$Ra_D = Gr_D \Pr = \frac{g\beta(T_s - T_\infty)D^3}{\nu\,\alpha}$$

is based on the gravitational constant (g), kinematic viscosity (ν), thermal coefficient of expansion (α), and the expansion coefficient (β).

For ideal gasses the expansion coefficient is,

$$\beta \equiv \frac{1}{T_\infty}$$

The wires on the condenser tubes act as fins to increase the surface area of the heat exchanger to reduce the resistance to and increase the rate of heat transfer to the environment. To account for the increased heat exchanger surface area contributed by the wires, the overall surface efficiency parameter (η_o) has been added to the convection resistance term,

$$R_{conv} = \frac{1}{\eta_o 2\pi\, r\, h\, L}$$

The overall surface efficiency parameter is defined by,

$$\eta_o = 1 - \frac{A_f}{A}\left(1 - \eta_f\right)$$

where A_f is the fin surface area, A is the total surface area, and η_f is the efficiency of a single fin. The efficiency of a single fin that is assumed to be straight with an adiabatic tip is,

$$\eta_f = \frac{\tanh(mL)}{mL}$$

where L is the length of the fin and m can be found by the equation,

$$m^2 \equiv \frac{hP}{kA_c}$$

where h is the convection coefficient, P is the perimeter, *k* is the thermal conductivity, and A_c the cross-sectional area of a single fin.

3.5.2.1 Correlation Results

With typical condenser conditions shown in Table 36, the overall heat transfer rate for the condenser increases to 13.8 W/K in water from 7.7 W/K in air. Intermediate and final results in the correlation calculations are shown in Table 37.

Table 36. Typical condenser conditions for air/water correlation.

Parameter	Symbol	Value
Condensing temperature ($^\circ C$)	T3c	55
Environmental temperature ($^\circ C$)	T_{env}	25
Tube diameter (mm)	D_{tube}	5.0
Wire diameter (mm)	D_{wire}	1.5
Area, tubes (m^2)	A_{tubes}	0.2353
Area, wires (m^2)	A_f	0.6644
Area, overall (m^2)	A	0.8997

Table 37. Intermediate and final results for air/water condenser correlation.

	Air	Water
Overall heat transfer coefficient (W/K)	7.7	13.8
Convective heat transfer coefficient (W/m^2K)	20.36	2405
Heat transfer resistance (K/W)		
Convection, outside	0.0580	6.054e-4
Conduction	5.930e-5	5.930e-5
Convection, refrigerant side	0.0718	0.0718
Overall	0.1299	0.0725

3.5.3 Configuration Tests

The independent refrigerator and DHW configurations (test # 1, 2, & 3) are run to serve as baseline results for combined system comparison. Combined system is simulated in configurations both of high-temperature heat rejection (test # 4 & 5) and of low-temperature heat rejection as a DHW pre-heat (test # 6 & 7). Test configuration descriptions are shown in Table 38.

Table 38. Test configurations.

Test #	Configuration	Refrigerator Operating	DHW Operating	DHW Heating Element On	DHW Water Draw	Couple Systems
1	Refrigerator standing test	Y				
2	DHW standing loss test		Y	Y		
3	DHW operating test		Y	Y	Y	
4	Combined standing loss test	Y	Y	Y		Y
5	Combined operating test	Y	Y	Y	Y	Y
6	Combined pre-heat standing test	Y	Y	Y		Y
7	Combined pre-heat operating test	Y	Y	Y	Y	Y

3.5.4 Desired Results

Results collected for each test configuration include electrical energy consumption (kWh), coefficient of performance (COP), and efficiency comparison ϕ for performance parameters shown in Table 39.

Table 39. Results collected for each configuration.

Result	Symbol	Units
Consumed energy, refrigerator	E_{ref}	kWh
Consumed energy, DHW	E_{DHW}	kWh
Consumed energy, HEC	E_{sys}	kWh
Coefficient of performance, refrigerator	COP_{ref}	
Efficiency, DHW	η_{DHW}	
Coefficient of performance, HEC	COP_{sys}	
Efficiency comparison, refrigerator	ϕ_{ref}	
Efficiency comparison, DHW	ϕ_{DHW}	

The most promising test configuration of combined operating (configuration five) and combined pre-heat operating (configuration seven) will be subjected to a sensitivity study including the following parameters with low and high level values shown in Table 40. The sensitivity study is a full factorial experiment with three factors and two levels (2^3) resulting in 8 total experiments.

Table 40. Sensitivity study parameters with high and low level values.

Parameter	Low (-)	Baseline	High (+)
UA Condenser in air (W/K)	4	7.7	16
Temperature of environment (°C)	18	25	32
Volume of storage tank (gallons)	30	40	60

CHAPTER 4. RESULTS

4.1 Introduction

Simulations for each of the seven configurations are performed using a Matlab program for a simulated duration of one day and a time step of ten seconds. The first three configurations are used to validate the Matlab model and to establish baseline temperature profiles and energy consumption values.

4.1.1 Configuration 1: Refrigerator only

Used for validating the refrigerator model, the first configuration operates the refrigerator alone rejecting condenser heat to the ambient environment, which in this case is air. Refrigerant properties for configuration #1 are shown in Table 41.

Table 41. Refrigerant properties of state for refrigerator only configuration.

Configuration 1: Refrigerator only

State Number	Description	T (°C)	p (MPa)
1	Compressor inlet	-25.2	0.058
2	Compressor outlet	66.3	0.631
3	Condenser saturated vapor	46.7	0.631
4	Condenser outlet	30.0	0.631
5	Evaporator inlet	-25.2	0.058
5	Refrigerant quality	0.34	
$\dot{m}_r$	Refrigerant mass flow rate	0.00043	

Refrigerator cabinet temperature profiles for ambient temperatures of 25°C and 31°C are shown in Figure 44.

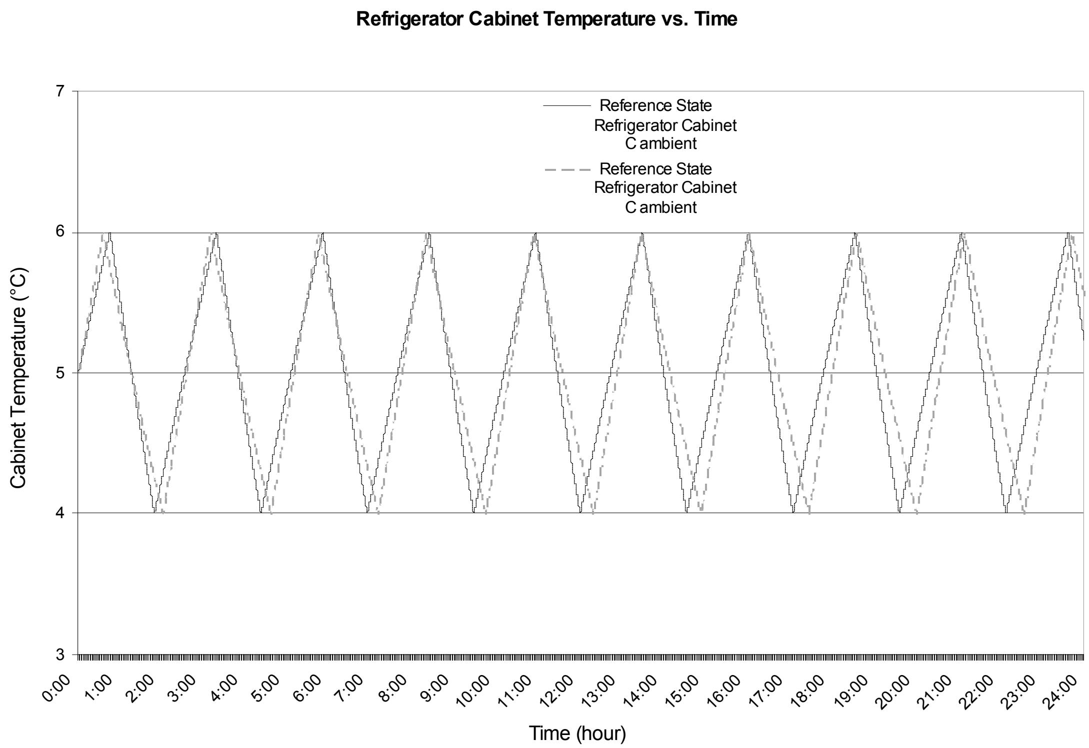


Figure 44. Refrigerator cabinet temperature profile in refrigerator model verification for 25°C and 31°C ambient temperatures.

4.1.2 Configuration 2: DHW standing

For verification of the hot water storage tank, the second configuration simulates the standing, or stand-by energy requirements. There is no water draw as the storage tank functions in a 25°C ambient air. Storage tank water temperature profile with no water draw for configuration two is shown in Figure 45.

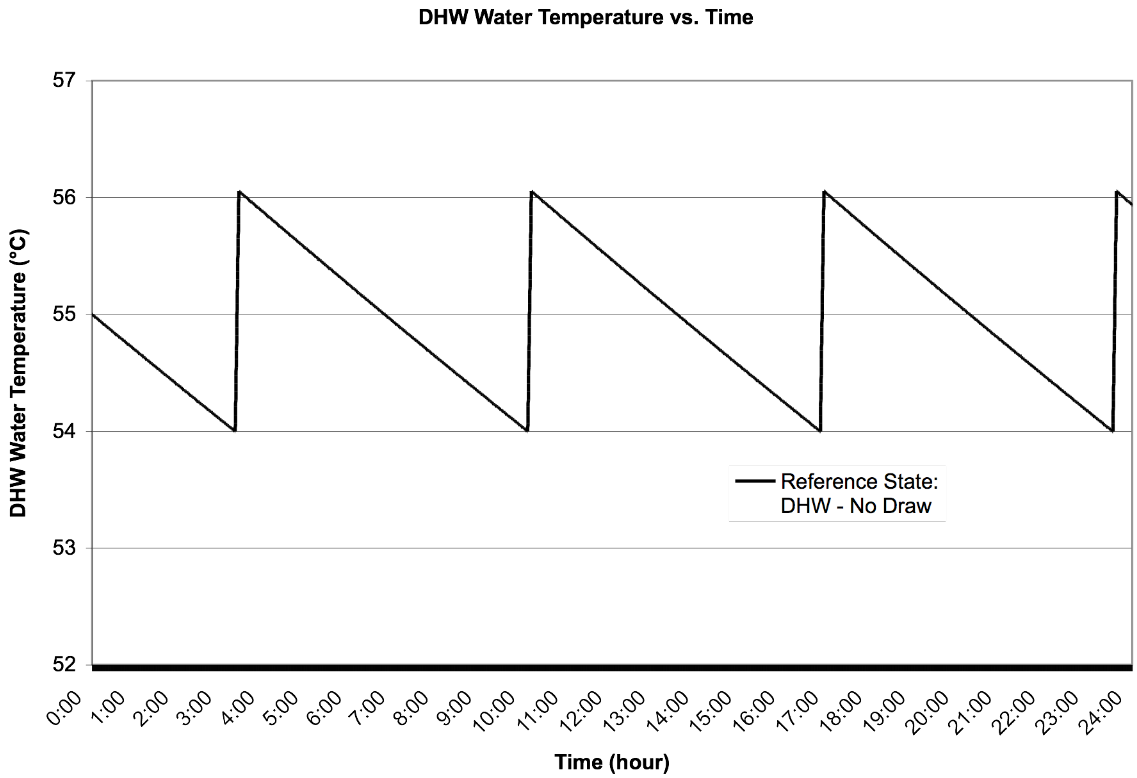


Figure 45. Water temperature profile in standing DHW model verification.

4.1.3 Configuration 3: DHW operating

As with the second configuration, only the DHW is considered, this time with water draw. Water temperature profiles for no water draw and light and medium realistic daily profiles (RDP) are shown in Figure 46.

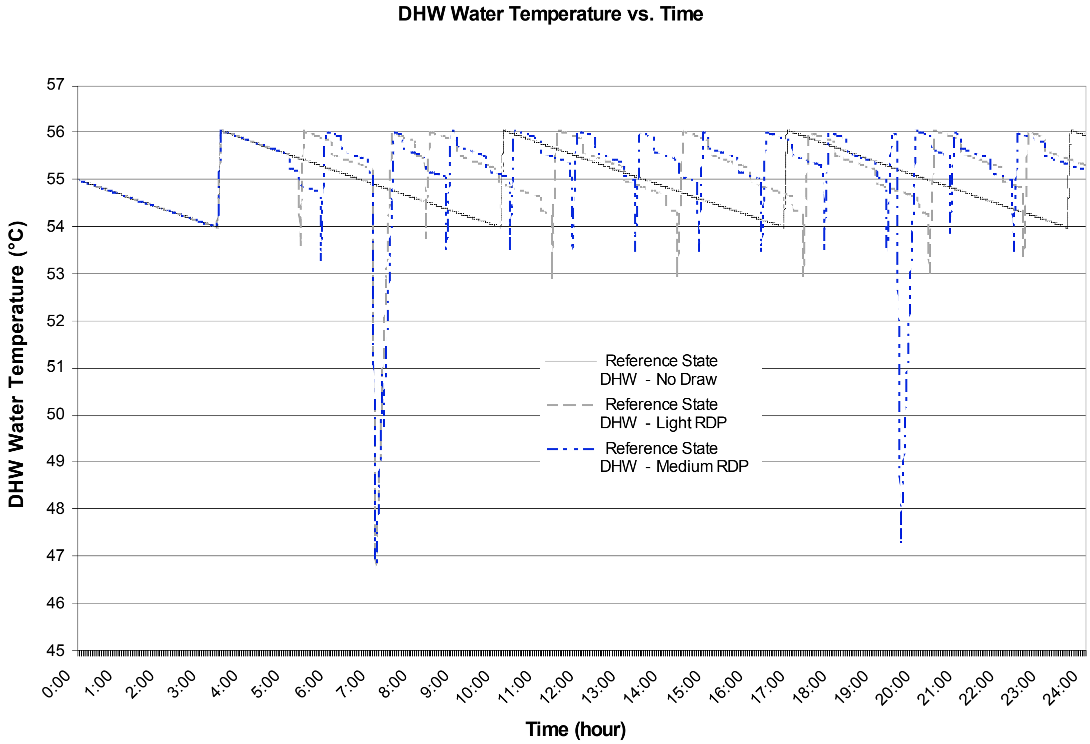


Figure 46. Water temperature profiles in DHW model verification for various draw profiles.

4.1.4 Configuration 4: Coupled system, DHW standing

The first of the combined systems is a simulation of the refrigerator operating and rejecting the condenser heat to the hot water heater storage tank. There is no water draw from the hot water heater storage tank and both the refrigerator cabinet and the storage tank are exposed to 25°C ambient air. The temperature profile of the storage tank water can clearly be seen increasing in response to the refrigerator heat rejection in Figure 47.

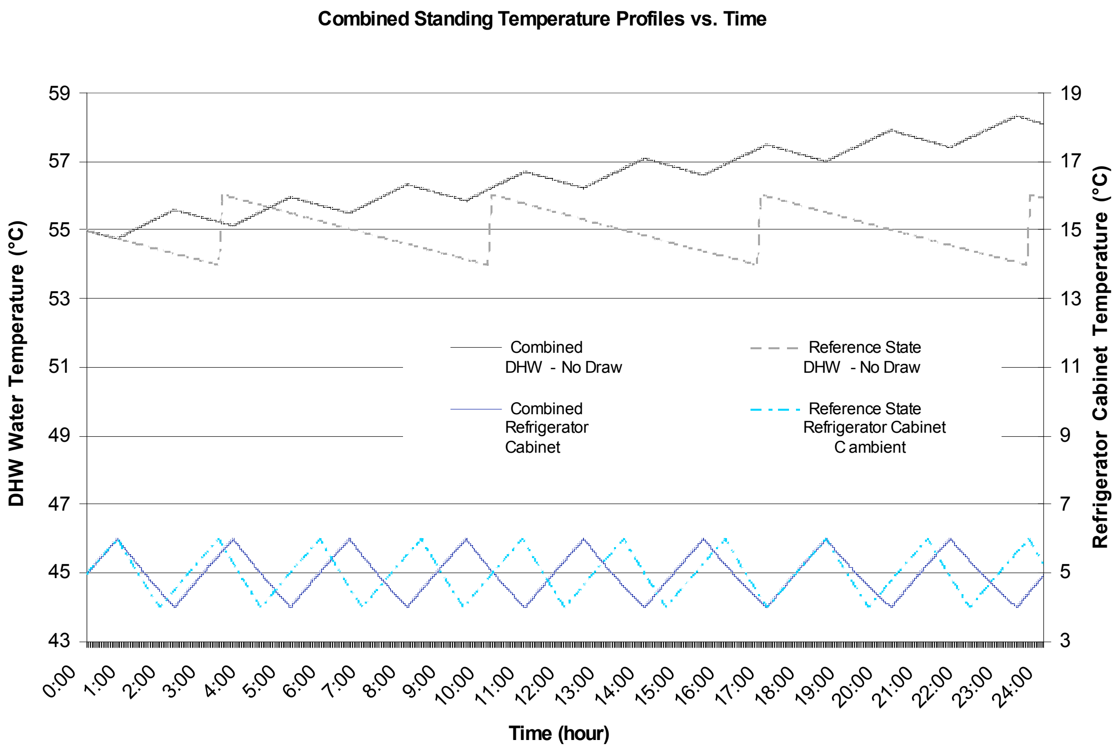


Figure 47. Directly coupled system standing, refrigerator and DHW temperature over reference state.

For this particular configuration and conditions, the heat rejection from the refrigerator condenser provides more energy than is required to maintain the stand-by operation of the hot water heater resulting in no energy consumption by the DHW. Refrigerant properties for configuration #4 are shown in Table 42.

Table 42. Refrigerant properties of state for directly coupled, standing configuration.

Configuration 4: Directly coupled, standing

State Number	Description	T (°C)	p (MPa)
1	Compressor inlet	-20.6	0.071
2	Compressor outlet	84.6	1.043
3	Condenser saturated vapor	68.0	1.043
4	Condenser outlet	60.6	1.043
5	Evaporator inlet	-20.6	0.071
5	Refrigerant quality	0.53	
$\dot{m}_r$	Refrigerant mass flow rate	0.00052	

4.1.5 Configuration 5: Coupled system, DHW operating

In the directly coupled configuration under medium realistic daily water draw, the refrigerator rejects condenser heat directly to the DHW storage tank. The reduction in energy consumed by the DHW can be seen in Figure 48 as the DHW temperature always remains above the reference state temperature. Refrigerator cooling cycles are prolonged when compared to the reference state, presumably due to the less effective condenser heat rejection in the DHW storage tank. The prolonged cooling cycles result in increased energy consumption by the refrigerator.

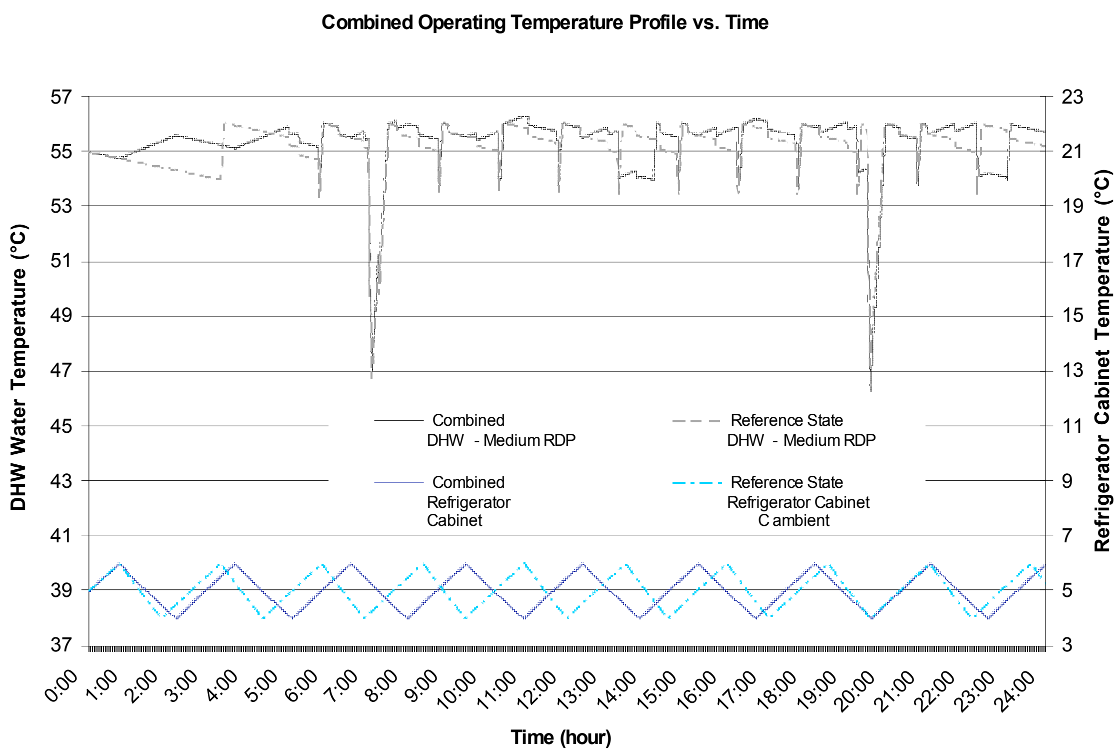


Figure 48. Directly coupled system operating with medium draw, refrigerator and DHW temperature over reference states.

Refrigerant properties for configuration #5 are shown in Table 43.

Table 43. Refrigerant properties of state for directly coupled, operational configuration.

Configuration 5: Directly coupled, operating

State Number	Description	T (°C)	p (MPa)
1	Compressor inlet	-20.6	0.071
2	Compressor outlet	84.6	1.043
3	Condenser saturated vapor	68.0	1.043
4	Condenser outlet	60.6	1.043
5	Evaporator inlet	-20.6	0.071
5	Refrigerant quality	0.53	
$\dot{m}_r$	Refrigerant mass flow rate	0.00052	

4.1.6 Configuration 6: Combined pre-heat, DHW standing

Configured as a pre-heat to the DHW storage tank, the heat sink begins at the temperature of the make-up water and increases as the refrigerator rejects heat to it. Temperature profiles are shown in Figure 49. As there is no water draw in the standing test, there is no interaction between the heat sink and the DHW storage tank and the performance of the DHW is unchanged.

Due to the low ambient temperature of the make-up water in the heat sink, the refrigerator requires less energy to operate when compared to the reference state. Refrigerant properties for configuration #6 are shown in Table 44.

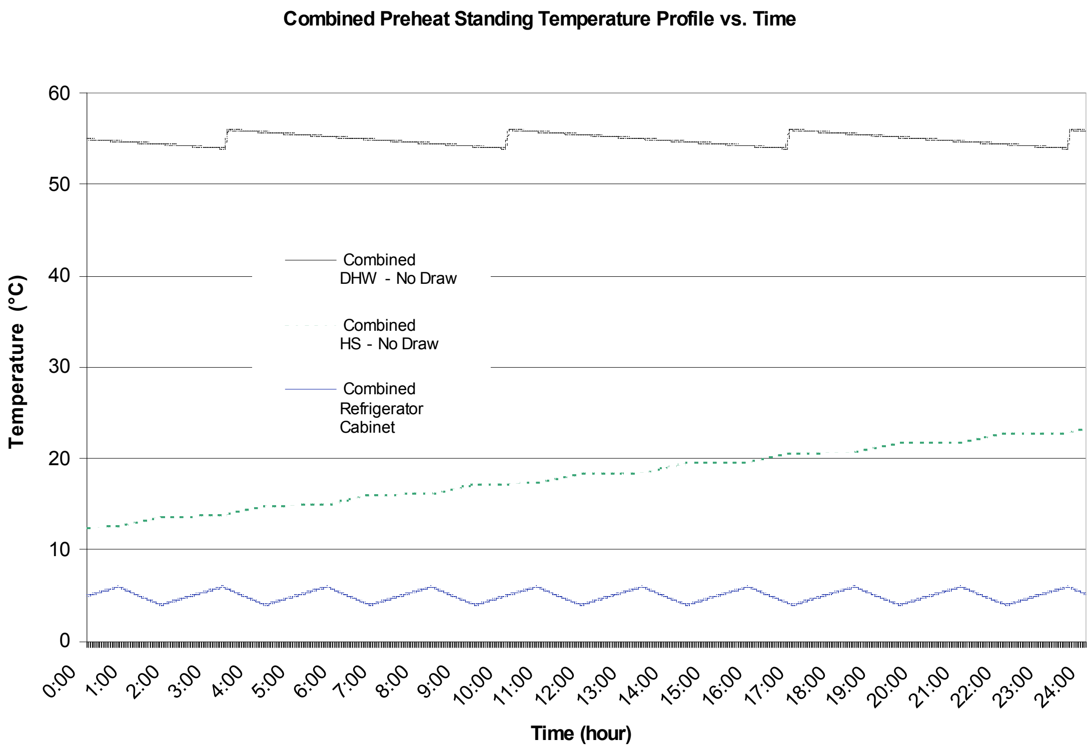


Figure 49. Combined system in preheat, standing configuration, refrigerator, heat sink, and DHW temperatures shown.

Table 44. Refrigerant properties of state for preheat, standing configuration.

Configuration 6: Preheat, standing

State Number	Description	T (°C)	p (MPa)
1	Compressor inlet	-24.3	0.060
2	Compressor outlet	56.4	0.498
3	Condenser saturated vapor	37.5	0.498
4	Condenser outlet	31.0	0.498
5	Evaporator inlet	-24.3	0.060
5	Refrigerant quality	0.38	
$\dot{m}_r$	Refrigerant mass flow rate	0.00045	

4.1.7 Configuration 7: Combined preheat, DHW operational

In the final combined configuration, the system is subjected to a medium realistic daily water draw. The heat sink starts at the make-up water temperature (12.5°C) and warms as the refrigerator heat is rejected to it, but does not reach steady-state operation. The temperature of the DHW is higher during water draws than in the reference state due to the pre-heated water from the heat sink. The temperature of the heat sink can be seen to decrease in response to the water draw from the DHW, and the colder make-up water filling.

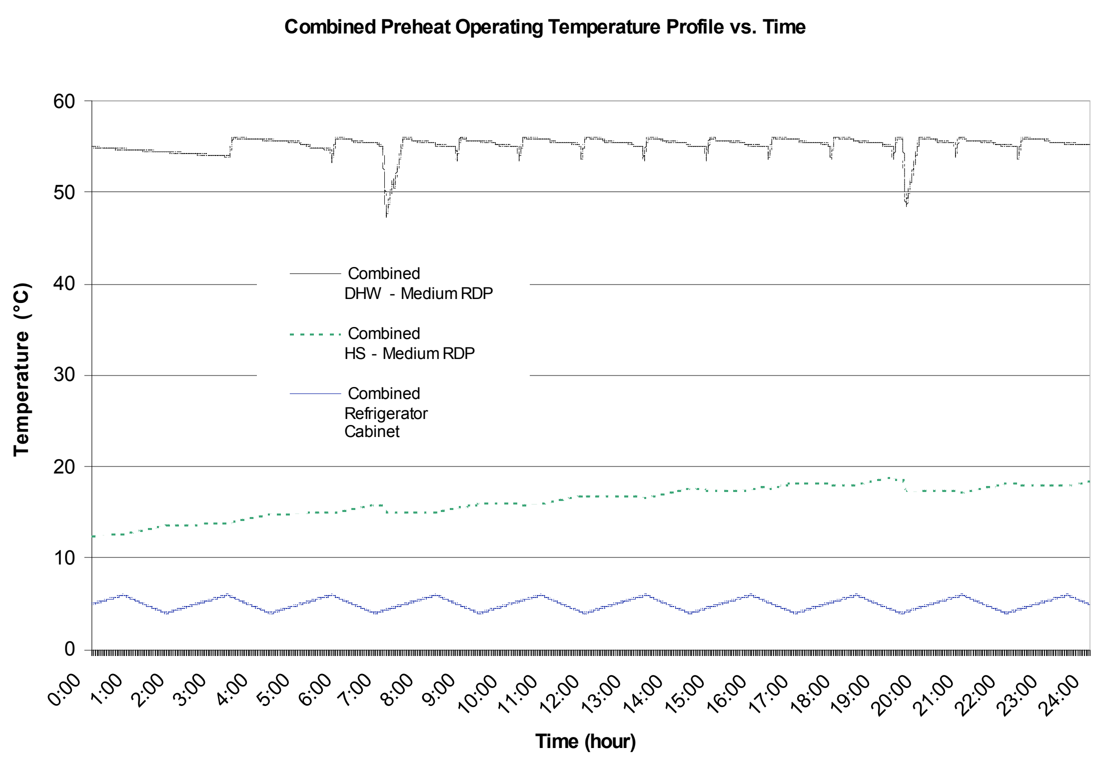


Figure 50. Preheat operating configuration, showing refrigerator, heat sink, and DHW temperatures.

To determine the heat sink steady state operation the initial heat sink temperature was set equal to the ambient temperature (25°C) and the simulation duration increased to three days. Figure 51 shows the heat sink temperature generally decreasing as it approaches 20°C.

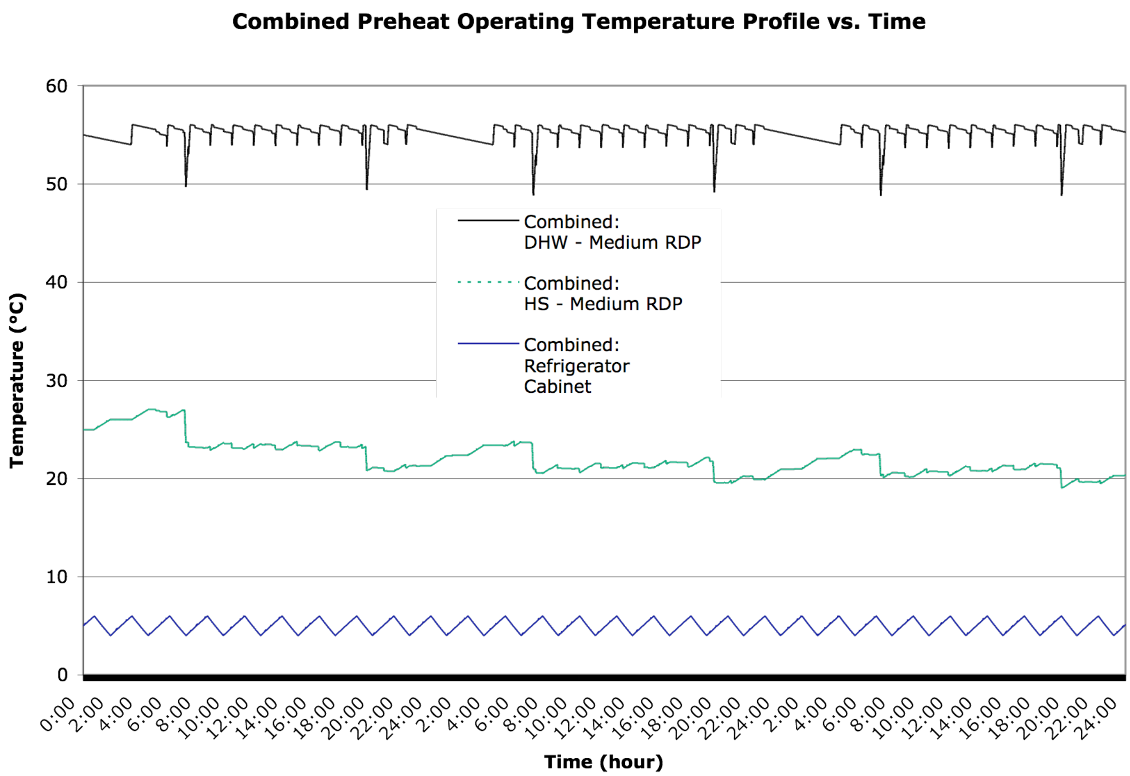


Figure 51. Preheat operating configuration, duration of three days.

The refrigerator again has a reduced energy requirement due to the more effective heat rejection in the colder heat sink than in either the DHW or the reference state. Refrigerant properties for configuration #7 are shown in Table 45.

Table 45. Refrigerant properties of state for preheat, operating configuration.

Configuration 7: Preheat, operating

State Number	Description	T (°C)	p (MPa)
1	Compressor inlet	-24.3	0.060
2	Compressor outlet	56.4	0.498
3	Condenser saturated vapor	37.5	0.498
4	Condenser outlet	31.0	0.498
5	Evaporator inlet	-24.3	0.060
5	Refrigerant quality	0.38	
$\dot{m}_r$	Refrigerant mass flow rate	0.00045	

4.1.8 Summary of Results

Table 46 shows the performance parameters of the seven configurations simulated with the Matlab model including energy consumed and efficiencies. While the energy consumption of the DHW and the home energy center (HEC) decreased for both the directly coupled and the preheat configurations, only in the preheat configuration did the refrigerator realize a reduction in energy consumption. The reduction in energy consumption by the refrigerator was negligible, only 2W and 7W for the standing and operating conditions respectively.

Heat transfer results for the seven configurations are shown in Table 47. The heat transfer by the evaporator remained virtually constant for the baseline, directly coupled, and preheat configurations. The condenser heat transfer increased significantly for the directly coupled configuration while decreasing only slightly for the preheat configuration. Slightly more energy is delivered by the DHW in the water for both the directly coupled and preheat configurations than in the baseline

Table 46. Energy consumption results for configuration simulations.

		Configuration	Refrigerator	DHW Standing	DHW Operating	Baseline		Coupled		Preheat	
						Standing	Operating	Standing	Operating	Standing	Operating
Result	**Symbol**	**Units**	1	2	3	1&2	1&3	4	5	6	7
Consumed energy, refrigerator	W_{ref}	kWh	0.808			0.808	0.808	1.146	1.051	0.806	0.801
Consumed energy, DHW	W_{DHW}	kWh		1.500	10.125	1.500	10.125	0	8.388	1.500	9.263
Consumed energy, HEC	W_{HEC}	kWh				2.308	10.933	1.146	9.439	2.306	10.064
COP, refrigerator	COP_{ref}		1.358					0.967	1.019	1.362	1.380
Efficiency, DHW	η_{DHW}			-	0.836			-	1.010	-	0.921
COP, HEC	COP_{HEC}					0.475	0.875	0.967	1.011	0.476	0.958
Efficiency comparison, refrigerator	ϕ_{ref}							1.418	1.301	0.998	0.991
Efficiency comparison, DHW	ϕ_{DHW}							0	0.828	1	0.915

Table 47. Thermal energy results for configuration simulations.

			Refrigerator	DHW	DHW	Baseline		Coupled		Preheat	
				Standing	Operating	Standing	Operating	Standing	Operating	Standing	Operating
Result	**Symbol**	**Units**	1	2	3	1&2	1&3	4	5	6	7
Removed energy, refrigerator (evaporator)	Q_{evap}	kWh	1.097			1.097	1.097	1.107	1.071	1.098	1.105
Rejected energy, refrigerator (condenser)	Q_{cond}	kWh	1.687			1.687	1.687	2.109	1.984	1.578	1.579
Delivered energy, DHW	$Q_{w,out}$	kWh		0	8.468	0	8.468	0	8.475	0	8.532

4.2 Sensitivity Study

As configuration 5, the directly coupled HEC, has the lowest energy requirement of the operating configurations, it provides the basis for the sensitivity study. Initially introduced in Table 40, the factors used in the full factorial sensitivity study are shown in Table 48 in standard order.

Table 48. Sensitivity study test matrix.

Sensitivity Study	Condenser UA	T_{env}	DHW tank volume	UA (W/K)	T_{env} (°C)	Vol (gallons)
1	-	-	-	4	18	30
2	+	-	-	16	18	30
3	-	+	-	4	32	30
4	+	+	-	16	32	30
5	-	-	+	4	18	60
6	+	-	+	16	18	60
7	-	+	+	4	32	60
8	+	+	+	16	32	60

4.2.1 Sensitivity study results

Energy consumption results of each component, refrigerator and DHW, as well as the overall system (HEC) are collected for each Matlab simulation run. The results of the eight full factorial study are shown in Table 49 along with the repeated test matrix for reference.

Table 49. Sensitivity study results.

Factor Assignment								Values			Response Table (Data)		
								A	B	C			
Run#	A	B	C	D=AB	E=AC	F=BC	G=ABC	UA (W/K)	T_{env} (°C)	Vol (gal)	W_{HEC}	W_{ref}	W_{DHW}
1	-	-	-	+	+	+	-	4	18	30	9.240	0.690	8.550
2	+	-	-	-	-	+	+	16	18	30	9.385	0.648	8.738
3	-	+	-	-	+	-	+	4	32	30	8.133	1.458	6.675
4	+	+	-	+	-	-	-	16	32	30	8.577	1.427	7.150
5	-	-	+	+	-	-	+	4	18	60	10.106	0.681	9.425
6	+	-	+	-	+	-	-	16	18	60	10.293	0.643	9.650
7	-	+	+	-	-	+	-	4	32	60	8.855	1.455	7.400
8	+	+	+	+	+	+	+	16	32	60	9.226	1.426	7.800

4.2.2 Significance of main effects

Significance of the main effects are calculated to make it easer to identify patterns and determine which factors have the greatest effect on energy consumption. As can be seen in Table 50, the factor with the greatest impact on overall system performance is factor B, the temperature of the environment. The factor with the least impact on the overall system performance is factor A, the overall heat transfer coefficient (UA). As with the overall system performance, the change in environmental temperature is the most significant factor for both the DHW and the refrigerator. The least significant factor for the refrigerator and DHW is the DHW tank size and overall heat transfer coefficient (UA) respectively.

Table 50. Significance of main effects.

			W_{HEC}	W_{ref}	W_{DHW}
Factor A	Runs Where A is + :	Average =	9.371	1.036	8.334
UA	Runs Where A is - :	Average =	9.084	1.071	8.013
		Total Effect = Difference =	0.287	-0.035	0.322
Factor B	Runs Where B is + :	Average =	8.698	1.441	7.256
T_{env}	Runs Where B is - :	Average =	9.756	0.666	9.091
		Total Effect = Difference =	-1.059	0.776	-1.834
Factor C	Runs Where C is + :	Average =	9.620	1.051	8.569
Vol	Runs Where C is - :	Average =	8.834	1.056	7.778
		Total Effect = Difference =	0.786	-0.004	0.791

From the significance of the main effects shown in Table 50, it can be seen that the energy consumption of the refrigerator and DHW have an opposite relationship with respect to the environmental temperature. While the refrigerator energy consumption increases with the environmental temperature, the DHW energy consumption decreases at more than twice the rate of the refrigerator energy consumption. The result is that a higher environmental temperature (factor B) results in lower overall system energy consumption.

Run number three (Table 49) resulted in the lowest overall energy consumption, a 14% reduction in energy consumption from configuration five, and a 26% reduction in energy consumption from the baseline. This run represents the smaller overall heat transfer coefficient, warmer environmental temperature, and smaller DHW storage tank.

CHAPTER 5. SUMMARY AND DISCUSSION

The topic of this study is a simulated combined refrigerator and domestic hot water (DHW) system modeled in Matlab. The combined refrigeration and DHW system, also referred to as the home energy center (HEC), is comprised of a refrigerator only cold food storage and a storage tank type DHW where the condenser from the refrigerator is located in the water storage tank instead of being exposed to ambient air so the heat extracted from the cold food storage cabinet can be rejected to the hot water storage tank. The refrigerant cycle is assumed to operate steady-state while the water storage tank and the refrigerator cabinet are simulated as transient thermal masses. The system configuration with the lowest daily energy requirement of the four simulated was subjected to various conditions, three factors with two levels each for a total of eight simulation runs. This section will discuss the performance of the overall system, changes in performance of the individual components, economic savings, and recommendations for further study.

5.1 Overall System Performance

As shown in Table 46, the HEC configuration with the lowest overall energy requirement is with the refrigerator directly coupled to the DHW storage tank. This configuration has the lowest overall energy requirement for both standing and operating conditions with almost 1.5kWh less daily while operating than the baseline configuration. The efficiency improvement in this configuration of the DHW is greater than the drop in refrigerator coefficient of performance (COP) resulting in the lower overall system energy consumption. In the directly coupled configuration (#5) all of the heat extracted from the refrigerator cabinet is rejected to the water storage tank, thus reducing both the DHW energy consumption and the fraction of system energy consumption. The fraction of DHW energy consumption of the whole system drops to 88.9% in the directly coupled system from 92.6% in the baseline.

5.2 Refrigerator Performance

In the experimental testing performed by O'Brien et al [O'Brien 1998], the refrigerator performance was not affected by the HEC configuration because the refrigerator had a secondary condenser. This secondary condenser allows excess heat to be rejected to the environment (air) in consistent conditions regardless of the effectiveness of the primary condenser to extract heat. In all of the combined configurations simulated, all condenser heat is forced to be rejected to the water in the heat sink or the DHW. In the directly coupled configurations (4-5) the refrigerator is required to reject condenser heat to a water storage tank maintained at 55°C. Water temperatures simulated in the DHW ranged from $55 \pm 1\ °C$ with no loads to $55^{+1}_{-9}\ °C$ while operating under a medium realistic daily profile (RDP). The preheat configurations reduced the refrigerator energy consumption to the lowest values in the study by reducing the condenser ambient temperature from 25°C to the make-up water temperature of 12.5°C. When the heat sink temperature was arbitrarily set equal to the make-up water temperature of 12.5°C, the heat sink increases in temperature not only in response to the refrigerator heat rejection but also in response to the warmer ambient air surrounding it. The heat sink appeared to reach steady state temperature of approximately 20°C in both simulations when the heat sink was set initially equal to the make up water temperature and the ambient air temperature of 12.5°C and 25°C respectively.

The simulation results showed that the refrigerator performance is negatively influenced by the HEC configuration with the lowest energy consumption. This is acceptable because as a whole, the HEC system consumes less energy than the baseline configuration.

5.3 DHW Performance

For the directly coupled DHW standing test, there was no stand-by energy consumed by the DHW storage tank. The heat rejected by the refrigerator condenser was more than enough to maintain the water set-point of 55°C. In fact, water temperature in the DHW storage tank continued to generally increase for the entire duration of the simulation exceeding 58°C at the end of one day. This is similarly noted in experimental testing by O'Brien et al [O'Brien 1998], where low water draws (6 liters) could be taken without incurring energy consumption in the DHW. The preheat configuration required energy consumption of the DHW to maintain the set point just as the independent DHW in the standing tests because the DHW was isolated from the refrigerator and the DHW. During simulations of the directly coupled system operating, the DHW required significantly less energy to maintain the temperature set point than the baseline configuration. The directly coupled DHW requires almost 2kWh or 17% less energy than the baseline DHW.

The only adverse affect that the HEC has on the water storage tank is that during prolonged periods of no water draw, the DHW temperature continues to rise in response to continued heat rejection from the refrigerator. In the short term, increasing the storage tank water temperature results in more energy consumption by the refrigerator to keep the cabinet cold. As the water temperature continues to rise it is likely to reach a temperature at which the condenser cannot easily reject heat and damage is done to the refrigerant cycle components. Greater levels of insulation on the storage tank would increase the likelihood of refrigerant damage during periods of little or no water draw making thermal protection necessary on either the refrigerator or DHW or both.

The results of the operational simulations show that the DHW performance is generally improved by the HEC configuration.

5.4 Economics

Assuming that the maximum payback period acceptable to a household is five years and a discount rate of 1.6%, the marginal cost of implementing the HEC based on the baseline conditions would have to be less than $253 as shown in Table 51.

Table 51. Calculation of acceptable marginal cost for coupled system.

		Baseline		Coupled	
Configuration		Standing 1&2	Operating 1&3	Standing 4	Operating 5
Consumed energy, HEC	kWh	2.308	10.933	1.146	9.439
Reduction	kWh			1.163	1.495
	%			50%	14%
Annual savings	kWh			424.3	545.5
	$			$41.29	$53.08
Discount rate	%			1.6%	
Payback period	yr			5	
Marginal cost	$			$196.88	$253.10

Energy cost assumed $0.973/kWh.

Natural gas water heaters typically cost less to operate but are less efficient as not all of the fuel combustion heat is transferred to the water. Referring to the annual DHW energy costs in Table 16, the estimated savings for a natural gas storage water tank drops to less than $27 compared to the nearly $62 saved annually with a conventional electrical resistance heat water storage tank. The refrigerator requires $8.63 more energy annually no matter what fuel is used to heat the DHW, resulting in annual savings of $53.08 and $18.16 for electrical resistance and natural gas heat respectively. This results in only $86.60 in savings over five years for a natural gas water heater.

In its most basic form, the directly coupled HEC could be delivered as one unit with the water storage tank directly attached to the refrigerator cabinet. This would reduce or eliminate the need for expensive on-site connections and charging the refrigerant lines. As this single unit design may be impractical due to the large size and weight or aesthetically undesirable in most residential settings, the two devices would most likely be delivered separately and the location of the DHW be hidden from sight. To maintain the integrity of the refrigerant lines, an external refrigerant to water heat exchanger would be likely required. The Matlab model by comparison, simulates an internal heat exchanger that is immersed in the water storage tank. The cost of the additional components to the manufacturer would have to be a fraction of the amount saved over five years to account for price mark-up by a distributor and retailer. One of the most significant barriers to implementing a coupled system like this would likely be gaining the cooperation installers, specifically the plumbing, heating ventilation and air conditioning (HVAC), and electrician trades people.

From these limited cost savings, it can be surmised that only in a nearly optimized system that requires no skilled labor to install could the directly coupled HEC be viewed as cost effective with current energy costs.

5.5 Further Study

Although the directly coupled system should be able to provide adequate cold food storage and hot water, further study is required on the Home Energy Center (HEC) before its benefits could be realized in a residential or commercial setting. Some of the possible problems with the directly coupled HEC are;

Regulatory
- Local building codes may require double wall heat exchangers for potable water, how will this affect system performance?

Practical
- How many days of no water draw would it take for the increase in refrigerator energy consumption to exceed the baseline energy consumption value?

Technical
- The simulation duration should be extended to reduce the effect of measuring over an uneven number of heating and cooling cycles.
- Improvement of the Matlab model, most notably the inclusion of thermal stratification in the water storage tank.
- Choice of refrigerant? R600a was used as it provided the easiest verification against test literature. This is not as popular as R134a is in the United States but it may be advantageous in this type of system due to its higher critical temperature of 135°C compared to R134a of 101°C. This may provide a greater safety margin if the DHW stands for an extended period of time with no water draws and the refrigerator continues to reject heat to it.

Economic
- How much will the HEC configuration add to the manufacturing price once a double wall heat exchanger is incorporated?
- What is the optimal configuration with respect to money, energy, and exergy?

5.5.1 Variability in use

Establish a realistic annual profile (RAP) based on the published realistic daily profiles (RDP). This profile would include sequential days of no water draw to simulate homeowners away from the house on vacation. The remaining days would be assigned a randomly generated RDP based on their probability of occurrence. The work of developing daily, weekly, annual, and holiday probabilities for water draw has most recently been carried out by Jordan and Vajen in 2001 as part of solar combisystem investigation [Jordan 2001a, Jordan 2001b]. Their model considers the probabilities of draw occurrences to create an annual hot water draw profile for daily draw volumes of 100, 200, 400, 800 liters in time intervals down to each minute.

Since application environment and user behavior can vary, Monte Carlo simulation of the HEC should be performed. Uncontrollable factors affecting performance and safety such as ambient temperature, make-up water temperature, draw volume and draw rate must be included to make sure that the system operates in a controlled and reliable manner. Factors that can be controlled such as the storage tank size, overall heat transfer coefficient of the heat exchangers, insulation levels, etc. could be included to optimize the system configuration.

5.5.2 Thermal stratification

Since thermal performance of a heat exchanger declines as the temperature difference is reduced, maximizing temperature gradients is important to maximizing efficiency in thermal systems. Heat can be more easily rejected to a cold sink (the environment) than it can to a hot sink. Since the temperature is spatially uniform inside the simple tank model, the simulation cannot take advantage of thermal stratification that typically increases energy efficiency. Proposed conditions and assumptions for incorporating stratification in the DHW storage tank are:

1. DHW tank is a stratified storage tank with a moderate temperature difference from the top of the tank to the bottom of the tank of 30°C.
2. Refrigerator heat exchanger (condenser) heat rejected at the bottom, cooler location of the DHW
3. Mains make-up water added at the bottom of the DHW
4. Hot water removed from the top of the DHW (draw-off)
5. Temperature varies only with height in storage tank, horizontal cross-section held constant.

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