TECHNICAL ARTICLE

Optimised Liquid Flooded Gas Cycle for Heat Pump and External Heat Engine Applications

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Gas cycles, in their ideal form, promise the highest efficiency heat pump and external heat engine systems. The Ericsson and Stirling cycle promise Carnot efficiency, yet their seeming simplicity present significant engineering challenges in implementation. The primary challenge of both cycles is isothermal compression and expansion where finite time and the mechanical system's insufficient surface area prevent ideal performance. Liquid flooding of the Ericsson cycle has been explored previously. Flooding with a high heat capacity liquid in effect increases the surface area for heat exchange, yet the experimental mechanical system uncounted significant losses, preventing suitable performance. This paper models a novel gas cycle system that combines aspects of the reverse Brayton air cycle and the Liquid Flooded Ericsson cycle. The enthalpy model in EES (Engineering Equation Solver) allows optimization of possible cycle arrangements and yields an optimal arrangement that constitutes a new gas cycle utilizing liquid flooding to achieve superior performance in a simpler cycle that previously envisioned. Possible applications of the new cycle include heat pump and external heat engines.

Introduction

With the rapidly increasing threat posed by global warming and climate change, there is a growing interest in clean and efficient energy sources such as solar and wind power generation. While these will become major sources of power, efficient use of this power is paramount. Buildings account for 36% of global energy use and 39% of carbon emissions, of which space heating accounts for 30%. (UN Report 2017). While electric heating allows use of solely green sourced electrical energy, a much more efficient method to create heat is through the use of heat pumps, where the heat of even low temperature ambient air or other source is upgraded, creating 2–4 times the amount of heat with the same electricity consumed. While heat pumps are highly efficient, the majority use working fluids of HFCs with 2000–4000 the global warming potential as CO2. By treaty these fluids must be eliminated, yet few satisfactory alternatives exist. The immediate need of high efficiency, HFC free heat pumps system is paramount.

Liquid Flooded Ericsson Cycle
Ericsson Cycle
The Ericsson external heat engine is a closed cycle system that, in the ideal form, achieves Carnot efficiency, the highest possible between two heat sinks. The cycle

consists of the following segments: Isothermal compression, isobaric heat addition, isothermal expansion, and isobaric heat rejection. The working fluid of the cycles can be Helium, Hydrogen, Nitrogen, or even air; it does not require HFCs. The isothermal compression and expansion segments require the working fluid to take in or expel heat through the volumetric chamber walls. The isobaric heat addition and rejection is normally in the form of a counter flow heat exchanger, allowing the cycle to recuperate the heat energy of the cycle. Importantly, the volume of a well-insulated counter flow heat exchanger or 'recuperator' does not affect the cycle performance, allowing a large recuperator, which is a significant advantage, unlike the Stirling cycle, which is commonly associated with the Ericsson cycle. While the Ericsson heat engine cycle creates a shaft power output, in the cycle can run in the reverse thermodynamic direction, and shaft power can drive the system to create a heat pump and mechanically move heat in the opposite direction to the natural flow of heat. This Ericsson cycle heat pump also, in the ideal form, achieves ideal, Carnot efficiency for a heat pump. The Ericsson cycle heat engine is shown in **Figure 1**.

As with all gas cycles, the mechanical implementation of the ideal cycle presents significant engineering challenges. While the recuperator is a counter-flow heat exchanger with commercial versions achieving greater than 95% effectiveness, isothermal compression and expansion of the working fluid is not possible in mechanical systems due to the limited surface area for conduction and limited
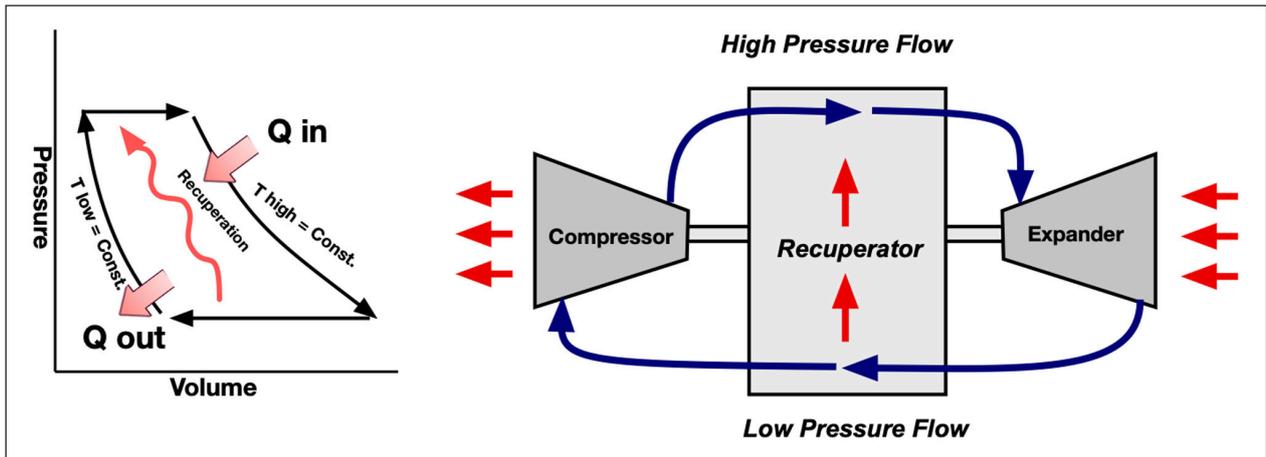
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Figure 1: Ericsson cycle heat engine.

time of the process. Additionally, as with all gas cycles, the Ericsson cycle is extremely sensitive to the efficiencies of the compressor and expander (Lemort, 2008).

Liquid Flooded Ericsson Cycle
Compression and expansion segments of the ideal Ericsson cycle are isothermal yet in practice are more adiabatic, forming a quasi-isothermal segment at best. This is due to non-infinite time of the segment and insufficient heat exchange of the working fluid to the chamber wall. Many expander and compressor designs have been explored in attempts to create chambers with sufficient conduction surface, yet with increase in surface area, there is the corresponding increase in seal length, and thus friction (Igobo, 2014). The efficiency gains of approaching isothermal segments is countered by the increase in seal friction, making the concept of very large surface areas at best very challenging and cost prohibitive for any commercially viable system.

An alternative method to approach isothermal compression and expansion through the use of liquid flooding of the volumetric chambers was proposed by Hugenroth of Purdue University (Hugenroth, 2006). Liquid flooding of gas compressors and expanders (1–5% by mass) has been used in industry to increase effective sealing of the system. It is also known in industry that by increasing the flooding ratio of liquid to gas mass, compressor work can be minimized as the compression is cooled and becomes more isothermal (Igobo, 2014).

In order to approach isothermal compression and expansion, the flooding liquid's heat capacity needs to be significantly higher than the gas. During the segments, depending on the liquid-gas ratio, the gas approaches isothermal compression or expansion as the heat energy is exchanged with the incompressible liquid. The liquid is chosen to remain liquid throughout the cycle process. In effect, liquid flooding increases the heat exchange surface area to the gas, allowing isothermal segments and large system efficiency gains.

The concept of the Ericsson cycle liquid flooding was explored by Hugenroth to overcome the substantial difficulties of achieving isothermal segments. While the development of a lab scale evaluation system concentrated on a small heat pump system using paired scroll compressor and expanders as a small refrigeration unit, the theoretical work of the Liquid Flooded Ericsson Cycle (LFEC) developed applies to Ericsson heat pumps with other compressor and expander types, as well as heat engines.

Liquid Flooded Ericsson Cycle Modelling
Theoretical LFEC Modelling
An analytic model was developed, building on the original work of Hugenroth, for the compression and expansion volume with gas-liquid fluid mixture within. Standard enthalpy thermodynamic relations are used to determine the resultant power and temperature changes of the fluid during compression and expansion. The compression chamber is considered a rigid, sealed container with heat added or removed to the fluid in accordance with the first law of thermodynamics. Nitrogen, the working fluid of the modelling presented, is modelled as an ideal gas in the derivation below. The equations below apply to both compression and expansion with negative work for compression and positive work for expansion, consistent with the convention adopted here that all energy flows into a control volume are positive and all energy flows out are negative.

Starting with the energy of the gas and liquid in the volumetric chamber, Eq. 1. By definition, Eq. 2 follows, and thus, Eq. 3.

Eq. 1 $$\delta Q = dU = du_g m_g + du_l m_l$$

Eq. 2 $$du = dT c_v$$

Eq. 3 $$dU = m_g c_{v,g} dT + m_l c_l dT$$

Where:
U *internal energy*
m *mass (g: gas, l: liquid)*
T *temperature*
c_v *constant volume heat capacity*
c_l *heat capacity of liquid (treated as incompressible)*

Rearranging gives Eq. 4, where $c_{v,f}$ is the effective constant volume specific heat of the fluid (gas and liquid combined) and is the average specific heat of the fluid mix weighted by the respective masses. The same approach is used to give $c_{p,f}$, the effective constant pressure specific heat, Eq. 5. These can be combined as the ratio of specific heat of an ideal gas, to give the isentropic exponent of the combined liquid and gas in the volumetric chamber, Eq. 6., that is Eq. 5 divided by Eq. 4. From this, it can be seen that an ideal gas with a constant specific heat ratio of k that is undergoing a compression or expansion process in thermal equilibrium with an incompressible liquid having a constant specific heat behaves exactly like an ideal gas having a constant specific heat equal to k_f.

$$\text{Eq. 4} \qquad c_{v,f} = \frac{m_g c_{v,g} + c_l m_l}{m_g}$$

$$\text{Eq. 5} \qquad c_{p,f} = \frac{m_g c_{p,g} + c_l m_l}{m_g}$$

$$\text{Eq. 6} \qquad k_f = \frac{m_g c_{p,g} + c_l m_l}{m_g c_{v,g} + c_l m_l}$$

With the theoretical development above, the LFEC can be modelled in the same method as a cycle with ideal gas. To observe the effect of liquid, the dimensionless term, Capacitance Rate Ratio is introduced as Eq. 7. In this, the ratio of liquid to gas capacitance is weighted by the mass flow of each. This equation allows the liquid-gas ration to be quantified qualitatively where C_p of the gas is used arbitrarily as opposed to C_v. Eq. 8 is the relative ratio of fluid temperatures during compression.

$$\text{Eq. 7} \qquad \gamma_{CR} = \text{Capacitance Rate Ratio} \equiv \frac{\dot{m}_l c_l}{\dot{m}_g c_{p,g}}$$

$$\text{Eq. 8} \qquad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{k_f - 1}{k_f}}$$

Where:
T_1 *inlet temperature*
T_2 *outlet temperature*
P_1 *start pressure*
P_2 *end pressure*

Cycle Gas and Flooding Liquid

For modelling and the forthcoming heat engine and heat pump lap scale systems, Nitrogen was chosen for the working gas of the cycle. Being an inert gas, Nitrogen avoids any explosive hazard of under pressure present in air with lubricants. Duratherm LT was chosen as the liquid flooding fluid for both heat pump and lower temperature heat engine applications. Fluid properties are detailed in **Table 1**. The Duratherm LT fluid model was developed from the manufacture's tabular data by linear regression, creating polynomials for use in the EES models. Duratherm LT is suitable for applications from –29C to 315C. Other liquids may be used as research continues. While numerous other heat transfer fluids can be used in the system, Duratherm LT was chosen due to it's high heat capacity and vapour pressure.

It is envisioned for heat pump and heat engine development prototypes, a flooding liquid that can serve not only as the high heat capacitance flooding liquid, but also assist in sealing of the compressor and expander, such as common with screw and scroll compressors. Volumetric systems under development that rely on clearance seals would be ideal for this application. Additionally, the fluid may also be selected to provide lubrication to the compressor and expander bearings. In such a system, the flooding liquid could serve all of the purposes, greatly simplifying the system design and engineering.

Figure 2 shows the mass of the respective gas and liquid (left) with the Capacitance Rate Ratio (right) as a function of mass ratio. This example is of an Ericsson heat engine model to accentuate the difference of Capacitance Rate Ratios of the compressor (5C) and expander (300C) which is due to the heat capacity of both the liquid and gas and volume of the gas being a function of pressure and temperature.

The effect of liquid flooding of a compressor and expander is shown in **Figure 3**, where the capacitance ratio (Eq. 7) is increased and the relative temperature between fluid entry and fluid exit is observed (Eq. 8) with different compression ratios (CR). It can be seen with increasing flooding ratio, the compression and expansion processes approach isothermal. The effect is decreased with increased compression and expansion ratios (ER). With a capacitance ratio of $g_{CR=}$ 15, the outlet temperature is within 0.5% of the inlet temperature, essentially isothermal. It is shown later that values of $g_{CR=}$ 7 are optimal for the applications of the proposed heat pump system.

The effect of liquid flooding and the work required for compression can be seen in **Figure 4**. In this, the work with respect to adiabatic work of the compressor is minimized with g_{CR} = 4 as the compression approaches isothermal (Igobo, 2014). With increasing flooding, work increases, where the compressor essentially become partially a

Table 1: Gas and Liquid Properties.

Fluid	Density [kg/m³]	Heat Capacity [J/kgK]
Nitrogen	1.16	1040 C_p
Duratherm LT	811.18	2139

Temperature: 300K, Pressure: 1 bar.

liquid pump and associated work increasing with liquid mass flow. This is well known in industry where limited flooding is used to minimize work of compression. The work required for compression can be optimized, but it will be seen, cycle optimization results in higher flooding ratios than of the compressor. **Figure 5** shows the effects of compression ratio on optimum flooding ratio to minimize compression power.

Purdue Liquid Flooded Ericsson System

The Liquid Flooded Ericsson Cycle (LFEC) system arrangement developed by Hugenroth is shown in **Figure 6**. This was the configuration first developed as a heat pump cooler. The same arrangement was later used in an Ericsson heat engine lab scale development of a micro-Concentrated Solar Power (mCSP) system (Nelson, 2014).
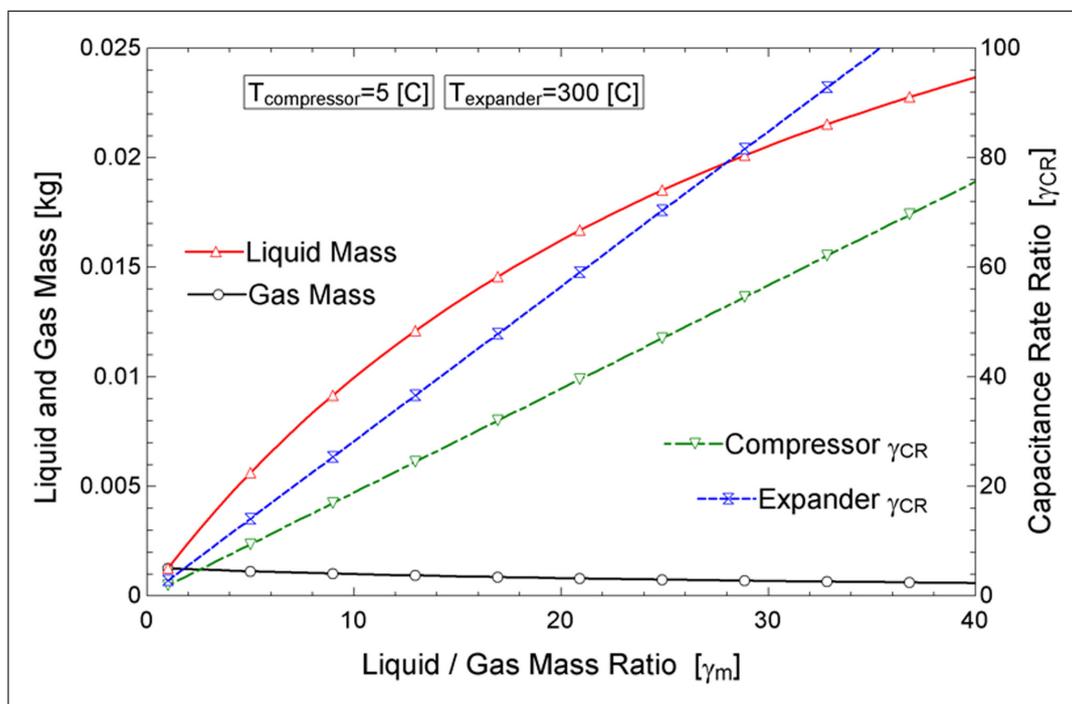
Figure 2: Liquid and Gas Relationship.
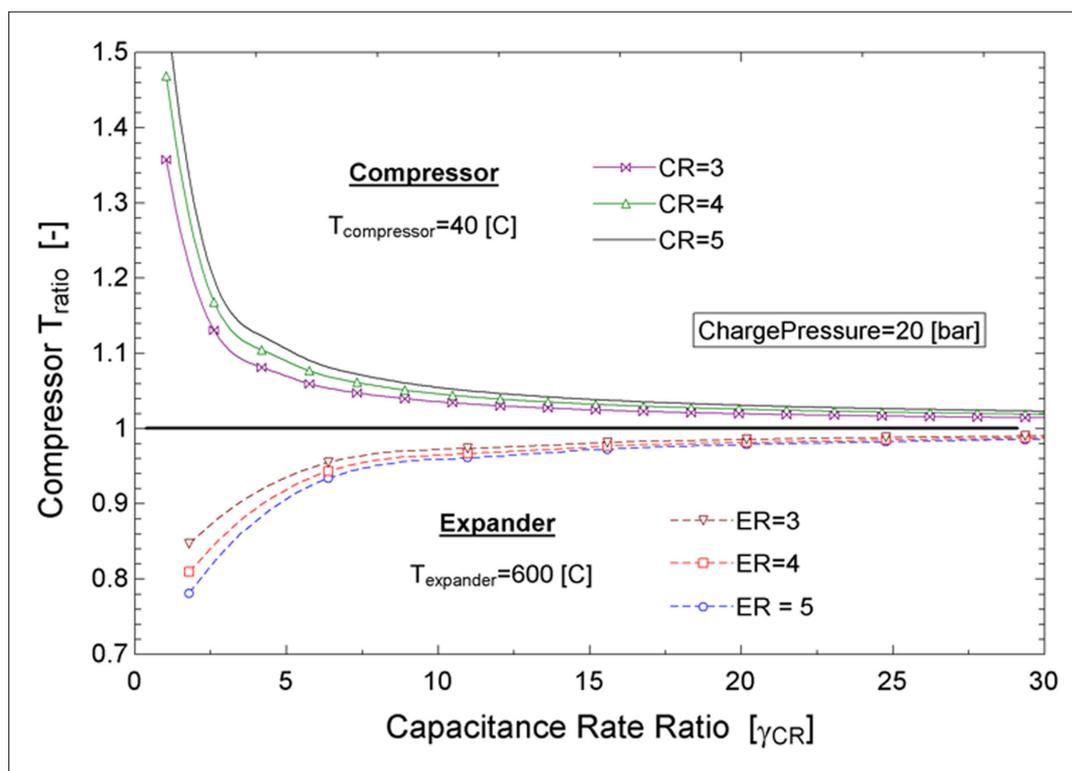
Figure 3: Liquid Flooding and Isothermalities.
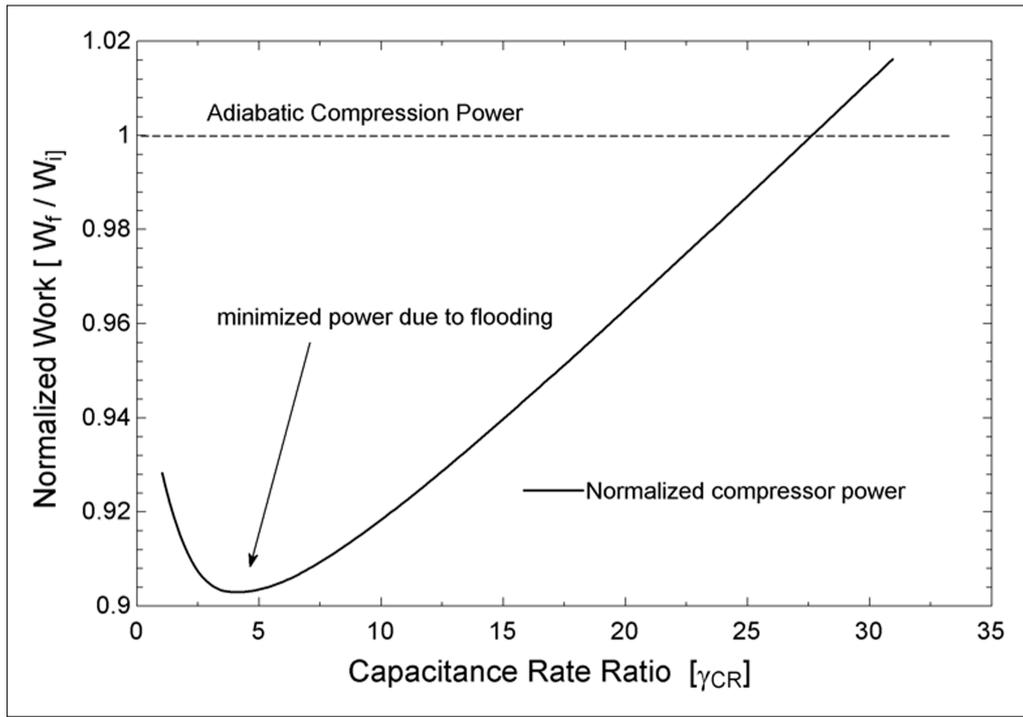
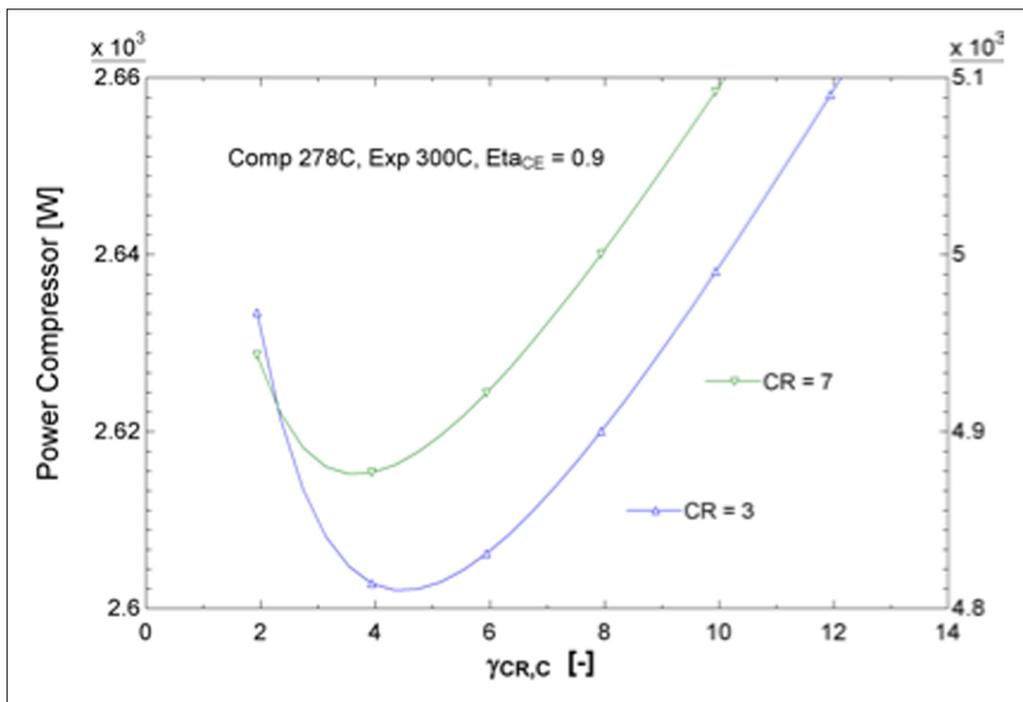**Figure 4:** Compression Work with Flooding.

Figure 5: Compression Ratio and Work.

In this there are three fluid loops; the working gas loop of the Ericsson cycle (1–8), the compressor liquid flooding loop (1–2, 9–11) and the expander fluid loop (5–6, 12–14). The separate liquid and working gas fluid loops were used since a suitable recuperator that can tolerate the gas and liquid counter flow were not available. While an additive manufacturing recuperator would allow such flows, the cost is prohibitive. For the system, commercial, off the shelf plate heat exchangers, pumps, hydraulic motors were used. Bespoke fluid separators were

fabricated. Scroll compressors, which can be modified to perform as expanders are known to tolerate high ratios of liquid, where used in this system.

Eq. 9 $Second\ Order\ Efficiency = \mathrm{eta}_{Carnot} = \dfrac{COP_{System}}{COP_{Carnot}}$

In order to quantify system performance, Second Order Efficiency is used, defined in Eq. 9. The LFEC system,
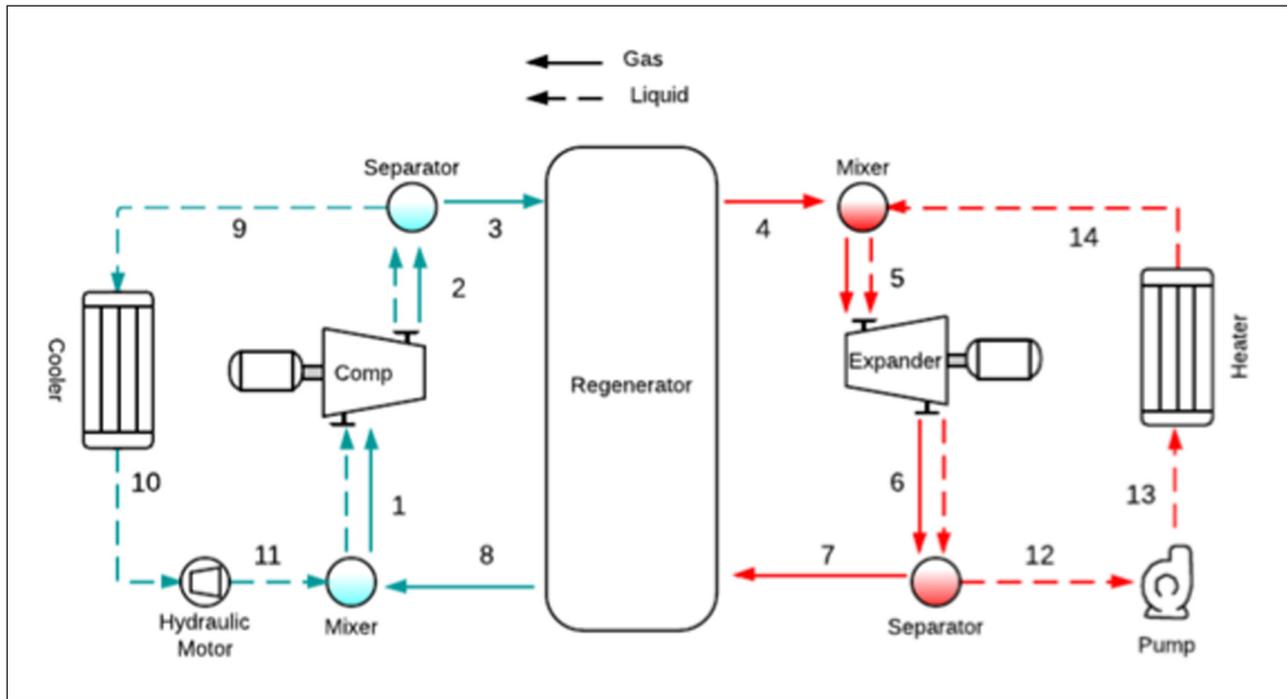
Figure 6: LFEC System Arrangement.

modelled with ideal components, achieves Second Order Efficiency greater than 0.9, yet with actual component efficiencies of the scroll compressors, pressure and heat losses, the actual system showed fundamental issues with the concept and implementation. The primary issue was extreme sensitivity of closed cycle systems to the limited isentropic efficiency of the scroll systems, less than 0.7. This was compounded by the fluid routing system, pumps, separators and associated routing pressure losses. The compressor fluid loop (1–2, 9–11) included a hydraulic motor which allowed fluid, pumped by the compressor, to return to the low-pressure side, incurring losses due to efficiencies. The expander fluid loop (5–6, 12–14) includes a pump, required to pump liquid from low pressure to high, incurring significant work ($w = v_l (P_f - P_i)$). Additionally, in the LFEC test system, fluid lines up to 1.2m in length were used between elements, incurring pressure drops. All of this resulted in systems with second order efficiencies less than 0.15 and in the case of the cooler, suited only for very limited temperature ranges.

Hybrid Liquid Flooded Gas Cycle Model
In order to explore the optimum liquid flooded Ericsson cycle, a detailed model was developed in EES as shown in **Figure 7**. This model employed an enthalpy model using the liquid-gas energy equations developed above in a similar model to Hugenroth. Yet in the developed EES model, the gas and liquid are modelled as real fluids. In this, there are the compressor and expander with the two counterflow heat exchangers, one each for the gas and liquid flooding loops. Fluid separators and mixers are shown as blue dots. A total of eight heat exchangers are show, four for the gas loop, four for the liquid loop. The recuperators and heat exchangers use effectiveness models. For

heat pump or heat engine applications all of these would not be used and in the model can be bypassed by setting the effectiveness to zero. The model, with all the heat exchangers, allows all possible arrangements of both heat engine and heat pumps, to be explored to seek optimum arrangements for specific applications.

The cycle sequences of **Figure 7** is as following: Fluid (gas and liquid) is compressed (12 → 1), fluid is separated to gas and liquid (1 → 2, 13), in parallel, gas enters Hx3, then gas recuperator, then Hx4 while the fluid enters Hx7, the liquid recuperator, and Hx8. The liquid and gas are then combined (5, 16 → 6), then expanded (6 → 7). The fluid is then separated (7 → 8, 17) and the gas follows Hx1, gas recuperator, and Hx2 while the liquid follows Hx5, liquid recuperator, Hx6. The gas and liquid are then combined (11, 20 → 12) and then compressed.

A similar arrangement, yet with only heat exchangers Hx6 and Hx8, was proposed by Hugenroth as an alternative to the developed model (**Figure 6**) in their patent filling. It is not known if the arrangement was developed into a lab scale version.

A significant advantage of this arrangement over that of **Figure 6** is that it avoids the multiple liquid loops described above and the required pump pumping across low to high pressure with significant power requirements. The fluid separation and mixing in both models (**Figures 6** and **7**) were modelled as adiabatic. In the proposed model, the fluid separation and mixing is accomplished with a novel, simple system to be detailed in future work.

Cycle Optimization
The Engineering Equation Solver (EES) (Klein, 2009) model using effectiveness models for the heat exchangers allows optimization of the cycle. In this, the effectiveness
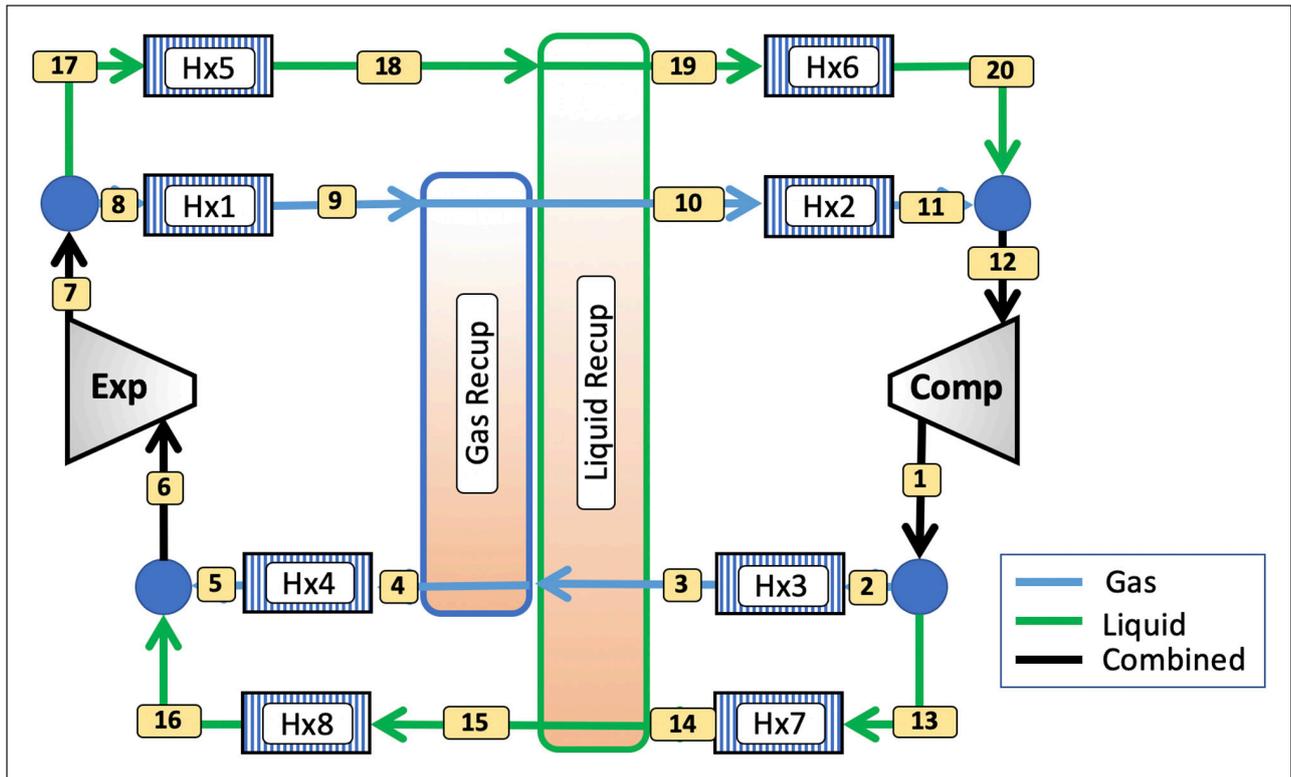
Figure 7: Universal Cycle Model.

of the gas and liquid recuperator is set to 90%, which is reasonable for commercially available brazed plate heat exchangers. The isentropic efficiency for the compressor and expander was modelled at 85%, commensurate with systems in development.

Optimization was conducted using the EES Generic Optimization method to determine the optimum effectiveness of each of the heat exchangers (Hx1 through Hx8). The Genetic method is the most robust method in the sense that is able to find a global optimum even if there are local optima, but it is also the slowest.

Heat exchanger effectiveness was constrained between 0 to 90%, reflecting readily available heat exchangers. This optimization is intended to yield the optimum effectiveness values as well as the optimum heat flow through each heat exchanger which will allow selection of suitable heat exchangers for a lab scale system. This method was applied successfully to both heat pump and heat engine models.

New liquid flooding gas cycle
While there are 512 possible permutations of heat exchanger use in the universal model, only about 10 represent physical viable systems. These were then optimized with and without the use of the gas recuperator. When modelled without a gas recuperator, the gas path was routed directly (between points 3–4 and 9–10). The results were counter intuitive for optimization of a gas cycle. **Figure 8** shows the same heat exchanger configurations in column 1–10 and 11–20, yet the first with gas recuperation and the second without. In every examined case of heat pump application and limited evaluation of a heat engine, the case of a system without gas recupera-

tion yielded higher 2nd order efficiency, as shown in column 13. This shows for a gas cycle with liquid flooding, gas recuperation is not desired in most cases. The resultant cycle is in fact a new gas cycle. While the Ericsson cycle requires gas recuperation, optimization show a gas cycle with only liquid flooding recuperation is superior, which by definition would be a liquid flooded reverse Brayton cycle.

The liquid flooded gas cycle without gas recuperation also has advantages in application to real systems. The requirement for complete separation of gas and liquid before the gas recuperator is avoided. The cycle does not require a robust gas-liquid separation system, but a more simple, lower cost and lower volume system as the liquid recuperator can tolerate some gas. The resultant optimized cycle (**Figure 9**) allows the liquid flooding liquid to be routed for the application. For example, as a domestic heat pump, the Hx8 and Hx5 can be located in the exterior, exposed to ambient air, while Hx6 and Hx7 can be routed for an interior heat exchanger. Such a system would not require an addition heat transfer fluid system.

Cycle Applications
The new cycle has applications as a heat pump and external heat engine. As an example of the new cycle applied to a domestic hot water heat pump is developed below. In such and application, heat energy is drawn from exterior air-to-liquid heat exchangers by the heat pump and then heats water for domestic use. The modelled cycle is that of **Figure 10**. Liquid recuperator effectiveness is set to 90% and isentropic efficiency of the compressor and expanders are set to 85%.
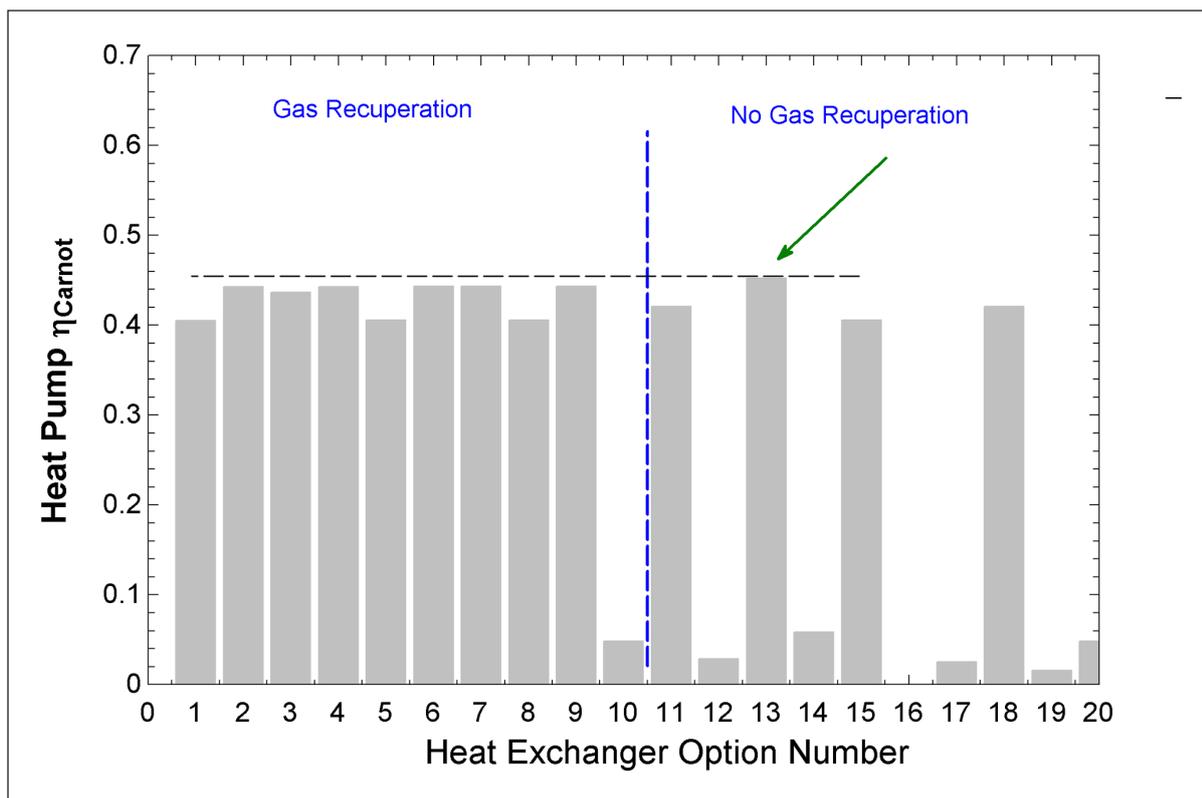
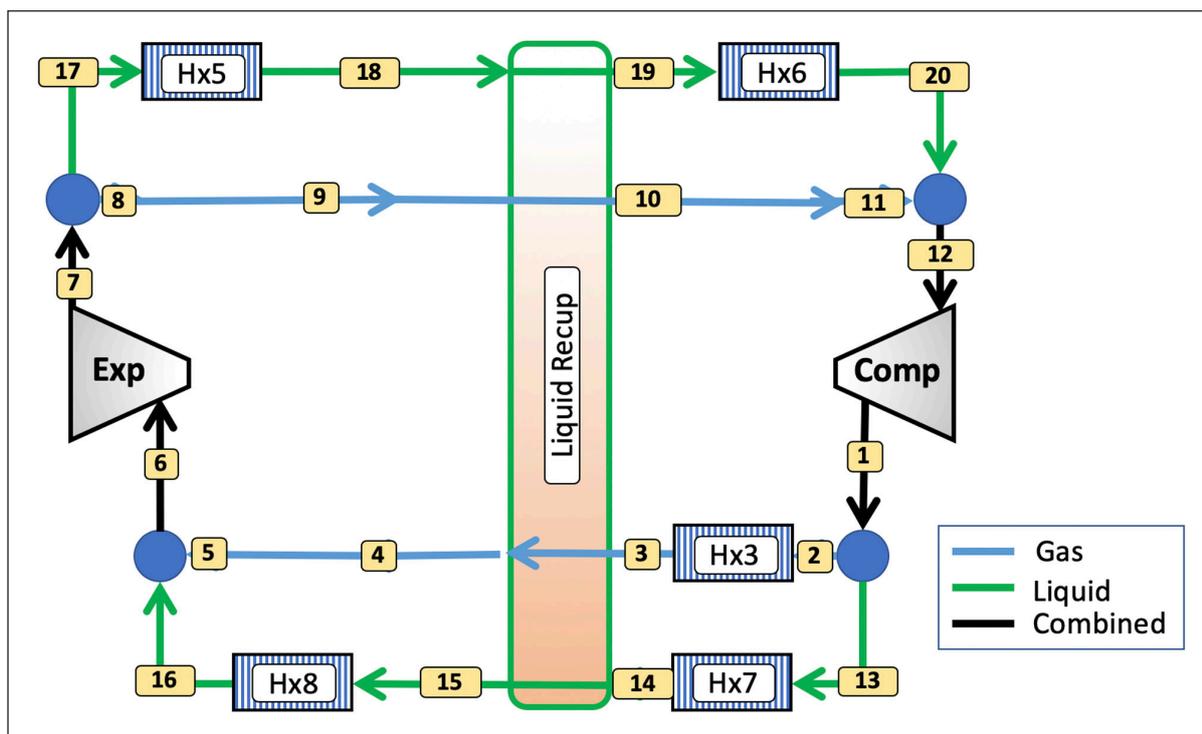**Figure 8:** Heat Exchanger Optimization.

Figure 9: Optimum Heat Pump Arrangement.

Figure 11 shows the performance of the modelled heat pump with varying temperatures of the evaporators (exterior heat exchangers, Hx5, Hx8) from –20C to 20C exterior temperature. The performance is on par with vapour compression (VC) systems and above trans-critical CO_2 (tcCO2) heat pumps. A unique advantage over VC and tcCO2 systems is the ability to generate hot over a variety of temperatures as shown. The proposed system ability to generator hot water in a low cost, low pressure system in excess of 90C is unique. The same system performance with respect to COP is shown in **Figure 11**.
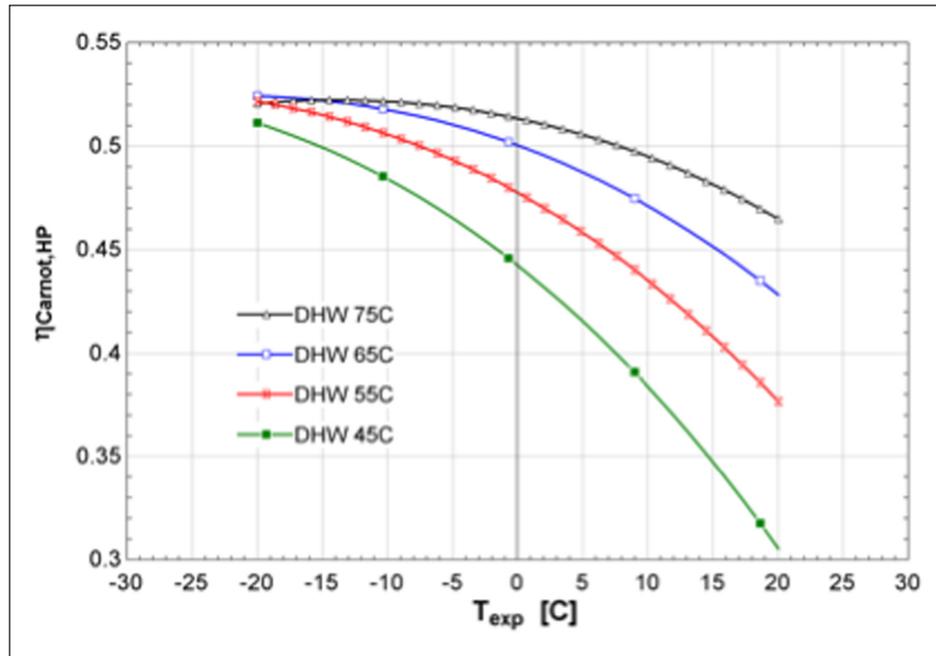
Figure 10: DHW Heat Pump 2nd Order Efficiency.
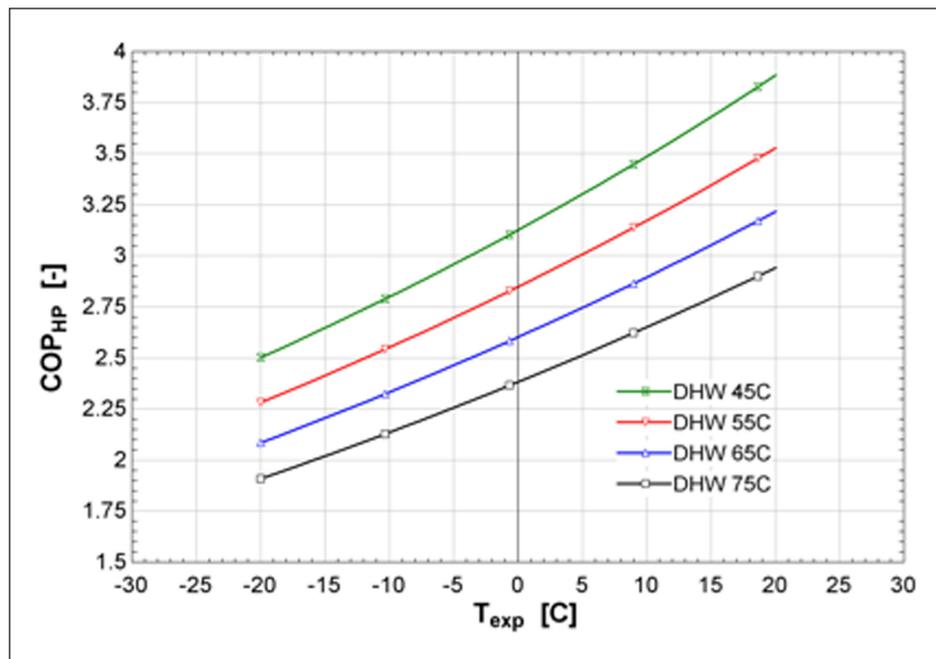
Figure 11: DHW Heat Pump COP.

Conclusion

Gas cycles, such as Stirling and Ericsson cycles, in their ideal form, can achieve Carnot efficiency, the maximum possible both as a heat pump and heat engine. Yet mechanical implementations of the cycles suffer from significant inefficiencies, primarily due to difficulties achieving isothermal compression and expansion. One method to approach isothermal compression and expansion is the use of liquid flooding in order to effectively increase the heat exchange surface area. Yet present methods and cycles applying liquid flooding to the Ericsson cycle suffer from efficiencies, nullifying the advantage of liquid flooding. The work presented here showed that an optimised for gas cycle with liquid flooding can achieve significantly better Carnot efficiencies than known Ericsson systems. The developed cycle is in fact a new cycle using liquid flooding recuperation and no gas recuperation and can be defined as a liquid flooded reverse Brayton cycle. The new cycle shows promise as a heat pump and heat engine. Modelling will continue and a lab scale heat engine and heat pump system are planned.

Competing Interests

The author has no competing interests to declare.

References

DuraTherm, LT. Fluid Data, DuraTherm Heat Transfer Fluids Corp. Lewiston, NY, USA. https://duratherm-fluids.com/products/duratherm-lt/.

Hugenroth, J. 2006. PhD Thesis. Liquid Flooded Ericsson Cycle Cooler. Purdue University. https://docs.lib.purdue.edu/dissertations/AAI3287416/.

Hugenroth, J, et al. 2006. Liquid-flooded Ericsson cycle cooler Part 1 – thermodynamic analysis. *International Refrigeration and Air Conditioning Conference*, Purdue. https://docs.lib.purdue.edu/iracc/823/.

Igobo, O and **Davies, PA.** 2014. Review of low-temperature vapour power cycle engines with quasi-isothermal expansion. *Energy*, 70: 22–34. DOI: https://doi.org/10.1016/j.energy.2014.03.123

Klein, SA. EES – Engineering Equation Solver, Version 10.668. 28 July 2019. *F-Chart Software*. http://fchart.com.

Lemort, V, et al. 2008. Analysis of Liquid-Flooded Expansion Using a Scroll Expander. *International Compressor Engineering Conference*. https://pdfs.semanticscholar.org/ab33/16f83c1a1e80d240b0c85275717771589903.pdf.

Nelson, A. 2014. High temperature flooded expansion for solar thermal power generation. MSc Thesis. Purdue. https://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=1484&context=open_access_theses.

UN Global Status Report. 2017. Towards a zero-emission, efficient, and resilient buildings and construction sector. https://www.worldgbc.org/sites/default/files/UNEP%20188_GABC_en%20%28web%29.pdf.