Utilization of Atmospheric Heat Exchangers in LNG Vaporization Processes:

A comparison of systems and methods

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Abstract

Recent developments in LNG vaporization systems have relied upon atmospheric heat exchange to capture heat for the vaporization process. While atmospheric heat exchange for cooling purposes is universally understood, so far there are not many published quantitative comparisons of the various available techniques and the equipment used for atmospheric heat exchange for large scale heating applications. The LNG industry is generally slow in accepting the new technologies due to the large investment cost and the risk involved.

This paper examines many factors associated with the use of atmospheric vaporization systems, including capital and operating cost and risk. The basis for comparison in this paper is a 1.5 billion cubic feet per day (BCF/day) LNG vaporization terminal in the U.S Gulf Coast. Vaporization system options evaluated include; Submerged Combustion Vaporizer (SCV) only, two-loop system with direct contact heat exchangers, single loop system with indirect contact heat exchangers, and hybrid systems combining SCVs, Open Rack Vaporizers (ORVs) and direct contact heat exchangers.

The paper also discusses the variations and weather patterns and how these affect plant design and heat exchanger operations, including sensitivity analysis for multiple site variables.

Basics of LNG Vaporization

The increasing demand of LNG along with the ever-increasing cost of crude oil is making the investment in LNG chain very attractive. . However the cost of first liquefying the LNG and then vaporizing it is very high. Typically about 8 to 9 % of the plant input is used as fuel in LNG liquefaction and about 1.5% of plant throughput is used as fuel in LNG regasification. Moreover, the long-term viability of the LNG transportation chain is challenged by emerging GTL technologies and newer technologies like gas by wire, CNG transportation and others. The cost dependence of the various types of fuels is extremely complex and it is important to reduce the cost of vaporization.

Process heating remains the single largest cost of the vaporization operation. The average 1.0 billion standard cubic feet per day (bscfd) LNG vaporization terminal requires approximately 5,500 billion btu per year of heat for LNG vaporization. At a cost of US$ 10.0 per MMbtu of natural gas, the energy cost is about US$ 55 million per year. Ideally, operators would like to use “free” sources of heat such as from seawater via open rack vaporizers (ORV) or from the waste heat generated by a co-located facility such as a power plant, refinery or petrochemical plant. In situations where seawater or waste heat cannot be used, the traditional alternative is the use of submerged combustion vaporizers (SCV) or Shell and tube vaporizers (STV) with an intermediate heating fluid. Utilization of ambient air will not only save expensive fuel and the operating cost but will help in reducing the emissions significantly.

Current innovations are focused on developing systems that can capitalize on another source of “free” heat – specifically – energy from the ambient air. Two such terminals have been built on a large scale; one is in operation with a second scheduled for commissioning in 2008. At least one other atmospheric heating system is under construction. Each of these three terminals utilizes a different system approach, and additional innovations in vaporization system design will likely lead to future terminals having lower cost, more efficient configurations than these early installations.

Design Basis for Analysis

The vaporization of liquefied natural gas (LNG) from a liquid to vapor state involves latent and sensible heat transfer through a series of heat exchangers. For the purpose of this study, the inlet LNG is assumed to be saturated at a temperature of -256 degree F and a pressure 2.0 psig. The LNG is heated to a superheated condition of 40 degrees F at a pressure of 1250 psig. In all cases, this vaporization is performed in a continuous flow process, with an exit volumetric flow rate of 1.5 billion cubic feet (BCF) per day. For shell and tube vaporizers, an ethylene glycol solution is used as an intermediate fluid for this evaluation.

Existing vaporization systems

Several existing vaporization systems were modeled as well as theoretical systems utilizing standard components. As a basis for comparison, all costs and footprints were compared to a standard SCV installation.

Comparative analysis

The design optimization of an air based LNG vaporization facility requires consideration of a large number of variables and assumptions, making the possible optimized solutions almost limitless. The performance of air based technologies are sensitive to weather conditions like wind speed, wind direction, wet bulb temperature etc. There is clearly no “best” system for all locations, load profiles and economic conditions. For the purpose of this paper, we selected a location in the United States Gulf Coast based on average weather conditions. This study assumes no constraints that prohibit any particular system such as permitting restrictions, emissions restrictions or space limitations. Obviously, such constraints may favor use of one technology over the other We did not attempt to evaluate and compare these constrained scenarios.

For each system, the following results were compared: footprint, total installed cost and annual operating cost.

Assumptions for analysis

For the purpose of this study, we have selected a 1.5 BCF per day vaporization facility on the United States Gulf Coast. Plant throughput was modeled as base-loaded with a 99% capacity factor. Operating costs are driven by energy prices, both for natural gas and electricity. Due to the cyclic and volatile nature of these commodity price curves, analysis was performed based on MMBTU of gas consumed and MWhr of electricity consumed per year. Typical price curves for power and gas prices were later applied to generate a “base case” cost of operation. A sensitivity analysis was performed to evaluate the volatility of this operating cost compared to changes in these commodity price curves.

Submerged Combustion Vaporization (SCV)

Footprint

For each evaluated system, the footprint of the LNG unloading docks, pumping stations, storage tanks and ancillary systems was assumed to be constant. As such, the only equipment evaluated was that required for actual vaporization of LNG. For the SCV base case, the system boundary starts at the LNG inlet flange to the SCV and ends at the natural gas outlet. The overall footprint for the SCV system, utilizing 8 vaporizers is 42,500 square feet.

Costs

System installation cost was compared based on 2007 price and cost estimates and averaged across various projects. SCV operating cost includes 8.2 E6 MMBtu /year of gas consumed as well as 61,477 MWhr for blower operations. Total LNG vaporization system power with LNG pumps and some miscellaneous power consumes a total of 162,655 MWhrs. Additionally, $0.07 million (based on caustic consumption) a year wastewater treatment cost is expected.

Environmental Impact

Operation of SCVs is expected to generate 185 tons/year of NOx and 220 tons/year CO emissions, which will require appropriate permitting. Additionally, 66 million gallons of wastewater will have to be treated for carbonic acid removal and neutralization.

Technology

Future advances in SCV technology include higher efficiency and lower emissions burners, which could potentially reduce the operating costs of these units.

Atmospheric heat exchange systems

Basics of atmospheric heaters

This paper evaluates two common and well-established heat exchanger designs. Direct contact heat exchangers operate on the same general principles as a traditional cooling tower, comprised of a water distribution system, a “fill” section that facilitates direct contact between the working fluid and air, an induced draft fan to create air flow and a basin to recover water and condensate. Indirect contact heat exchangers consist of a coil section that contains the working fluid. Coils are outfitted with a series of fins to increase the available surface area. Forced draft fans are used to blow ambient air down around the tubes, and a collection basin below the unit is employed to recover condensate falling from the tubes.

Direct vs. indirect heat exchange

There are two different types of atmospheric heat exchangers being evaluated and compared in this study, direct and indirect heat exchangers. The direct heat exchangers utilize direct contact between the atmosphere and the working fluid, while indirect heat exchangers maintain a boundary between the working fluid and the atmosphere through which heat must be transferred. While most engineers are familiar with the design and operating differences in direct versus indirect atmospheric *cooling* systems, (i.e. wet cooling towers vs. air cooled heat exchangers), the physics are quite different in a heating application and the traditional rules of thumb can’t be applied. For example, unlike a cooling application, where only the wet system takes advantage of a latent heat transfer with the atmosphere – both direct and indirect heating systems will condense atmospheric moisture and take advantage of available latent heat transfer. Hence, both units benefit from “wet” heat transfer.

Two key differences in the process mechanics affect unit design and footprint. Firstly, indirect system utilizes heat transfer only while a direct system benefits from heat and *mass* transfer, increasing the amount of atmospheric energy that can be captured in a given heat transfer area. Secondly, the direct system passes heat directly from the atmosphere to the working fluid whereas the indirect system requires heat transfer across a series of coils and fins, to provide the necessary surface area for heat transfer.

Beyond the mass and energy transfer mechanics within the heat exchange surface area; the design of these units is drastically affected by wind and weather conditions. Unlike cooling systems, where the majority of the effluent air is more buoyant than the ambient air, and rises into the atmosphere away from the unit, the air leaving these heat exchangers is colder and less buoyant than the ambient air and remains at grade level, potentially mixing with influent air and reducing the inlet air temperature and enthalpy. As a result, under certain atmospheric conditions, the heat transfer driving force (differential temperature and enthalpy) can be greatly reduced or stalled altogether. This performance issue can be practically resolved either through increasing the size of the unit or modifying the effluent air handling system to reduce or prevent recirculation. . For the purpose of this study, we have assumed that state-of-the-art heat exchangers are in use, employing anti-recirculation technology.

Theoretically, the ideal solution for ambient heat vaporization would be the utilization an indirect heat exchanger as the vaporizer, utilizing no intermediate working fluid. These ambient air vaporizers (AAV) are widely used in small scale cryogenic gas vaporization applications, the large required footprint as well as dramatic effluent air recirculation potential have prevented this technology from gaining commercial acceptance in the 1BCF/day size range. For this reason, AAVs have not been included in this particular study. Future experience and advances in AAV technology should be evaluated, however, when considering system options in future LNG terminals.

Mass and energy balance

Figure 1 illustrates a simple mass and energy balance across indirect and direct heat exchangers. Performance of the heat exchangers requires modeling multiple heat transfer mechanisms within the unit. For an indirect heat exchanger, the following mechanisms are modeled:

1. Sensible Heat Transfer: from ambient air → fins → tube wall → to fluid
2. Latent Heat Transfer: from condensing vapor → film → fins → tube wall → to fluid
3. Sensible Heat Transfer: from ambient air → film → fins → tube wall → to fluid

Note that the sensible heat transfer is modeled EITHER in direct contact in the non-wetted surface area of the heat exchange surface, OR in contact with the condensate film on the heat exchanger surface.

For the direct heat exchangers, the following heat transfer mechanisms are modeled:

1. Sensible Heat Transfer: from ambient air → to fluid
2. Latent Heat Transfer: from condensing vapor → to fluid
3. Mass & Energy Transfer: from condensate mixing with fluid

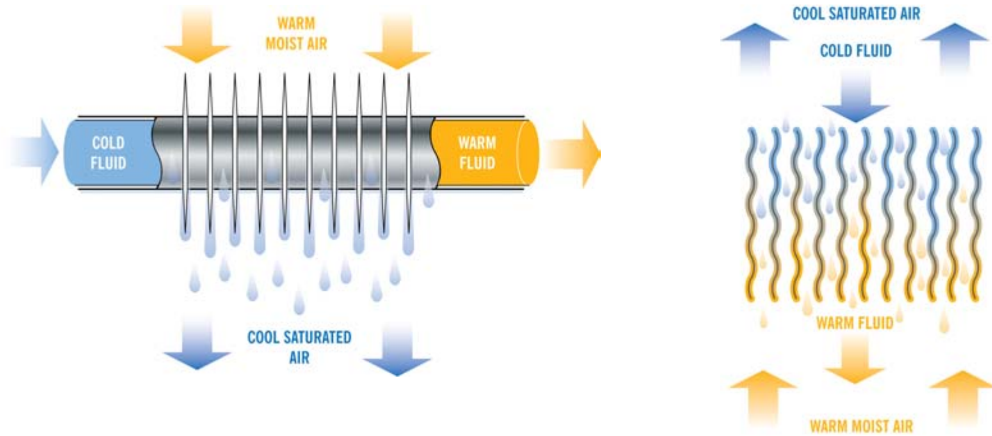

Figure 1 – Mass and energy balance, indirect and direct heat exchangers

From a system perspective, the condensate formed in the direct heat exchanger must eventually be removed from the system, practically, this is done upstream of the heat exchanger, after the available heat from the condensate has been transferred into the vaporizer or intermediate heat exchanger. Alternatively, the condensate may be discharged just downstream of the atmospheric heat exchanger, but only after equalizing with working fluid temperatures, thereby transferring more available heat than the indirect heat exchanger.

Qualitatively, for a given heating duty and atmosphere, it follows that the direct contact heat exchanger would require less heat transfer surface area for a given air rate, or a lower air flow rate (and associated fan power) for a given footprint.

Performance vs. weather

Both direct and indirect atmospheric heat exchangers have a performance curve that is dependent upon the inlet air wet bulb condition and approach temperature. It is important here to point out that the “inlet” air conditions and “ambient” air conditions are not always identical. In a zero-recirculation case, these conditions will be the same, but any amount of recirculation of effluent air into the inlet will reduce the inlet dry bulb and wet bulb temperature below the ambient conditions – making heat transfer more difficult.

For a given air rate and inlet cold fluid temperature, the outlet fluid temperature is a function of inlet wet bulb. As wet bulb drops, so does outlet temperature. At a sufficiently low inlet condition, the unit no longer heats the fluid, as the inlet temperature continues to drop, the heat transfer will reverse directions and the unit will begin to cool the working fluid. Figure 2 is shows the resultant operating

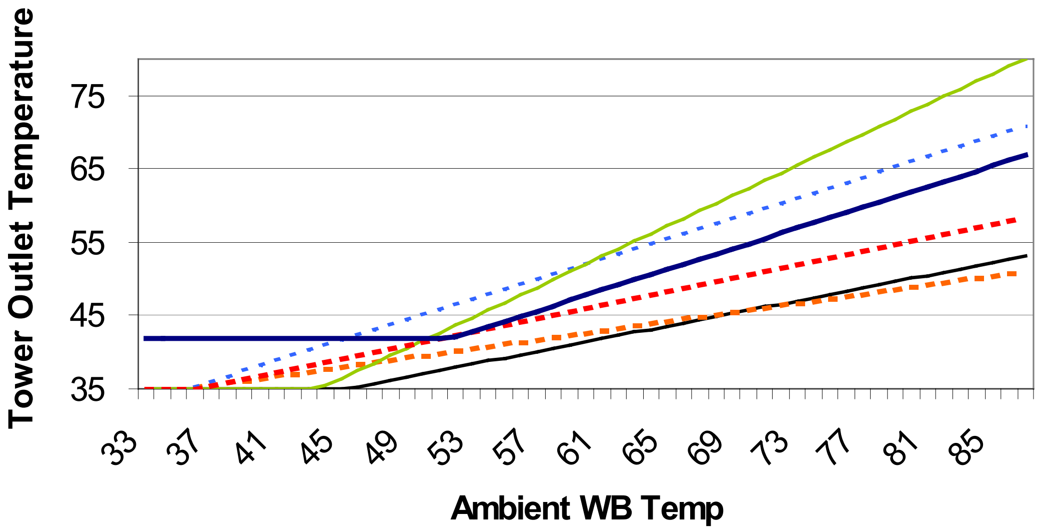

Figure 2: Example operating curves for multiple direct and indirect heat exchanger selections

curves for six different atmospheric heat exchangers that may be evaluated for an LNG installation.

Indirect systems allow the use of low freezing point fluids and can potentially operate at inlet temperatures below 32F, (0C). However, the condensation on the exterior of the heat exchangers will freeze in these conditions, risking coil damage and reducing heat transfer efficiency. For this reason, sub-freezing unit operations are not assumed or modeled.

Another important operational consideration is recirculation. Modern computational fluid dynamics (CFD) software suites can be utilized to predict the behavior of the effluent air stream and resulting reduction of inlet air temperatures. Readers should be cautioned that saturated air CFD modeling is a very complex exercise, and most off-the-shelf CFD packages are only equipped to model dry air or liquid state fluids. For this exercise, CFD analysis was performed to predict heat exchanger recirculation behavior under a variety of wind conditions. The orientation of the unit was chosen to minimize the negative impact of adverse wind conditions during winter months.
Figure 3 shows a CFD model of an indirect heat exchanger experiencing significant recirculation under moderate wind conditions.

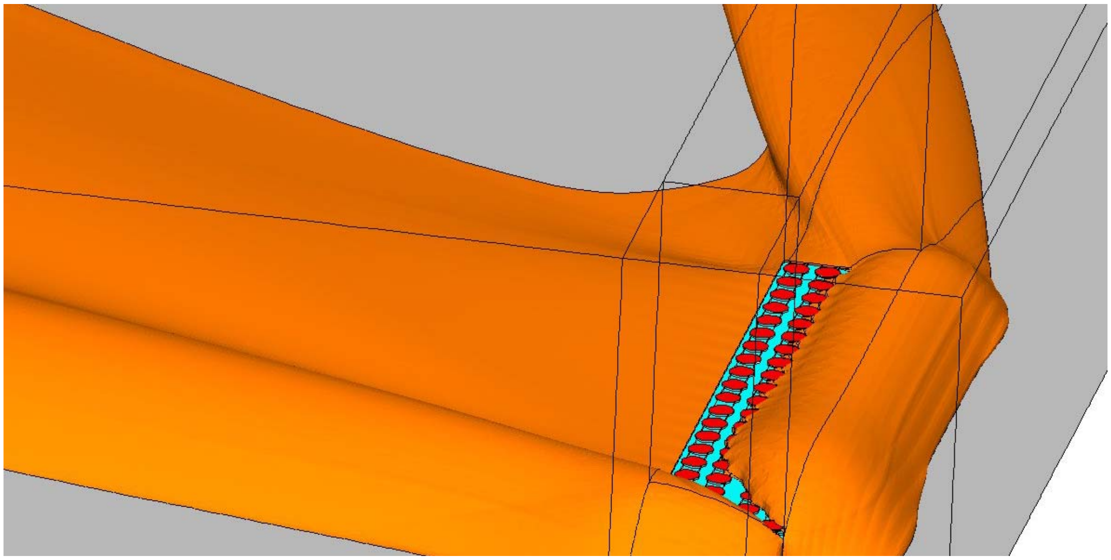

Figure 3: CFD model of 1.5 BCF/d indirect heat exchanger, demonstrating recirculation condition.

Sizing, selection and optimization

System design temperatures and flow rates acutely affect heat exchanger performance capabilities and sizing. A lower inlet fluid temperature results in a smaller heat exchanger and a wider range of available operating conditions. Each system evaluated has a different design fluid inlet temperature, which contribute significantly to the performance difference between different configurations.

Sizing and selection of atmospheric heat exchangers is a multi-step iterative process. The initial size was determined using a design point for each system and a design wet bulb of 65 degrees F.

With differences in heat exchange surface area and fan horsepower, there are a large number of possible configurations that can meet this design point. The selection is optimized based on lowest total operating cost, and a set of unit operating curves is generated. For the purpose of this study, further optimization was not performed. During a detailed system design, multiple design points are evaluated on a full year simulation model of the facility to optimize the selection based on a typical year's operating cycle. This simulation model calculates hourly operating parameters and is called an "8760 model", referring to 8760 hours in a year. While we generated 8760 models for each plant configuration that was evaluated, we did not perform multiple design point optimizations based on the output of the 8760 model.

Single Loop system with indirect heat exchanger

The simplest STV system evaluated was a single loop system comprised of a shell and tube vaporizer and an indirect atmospheric heat exchanger operating in a closed loop. Supplemental heat is supplied via a gas fired fluid heater. This system had the most favorable design condition in terms of flow rate and cold fluid inlet temperature to the atmospheric heat exchanger. Fluid circulating rate in this system was 74,000 gpm with a heat exchanger inlet temperature of 30.15 degree F. The operating model for this system did not assume any loss in heat exchanger efficiency due to icing, but it did assume the use of a non-standard heat exchanger designed to mitigate recirculation losses. Use of a standard heat exchanger will produce significantly reduced performance due to cold air recirculation.

Footprint

The total plan footprint of this system is 160,000 square feet, which is 375 % of the base SCV configuration. Figure 4 shows a simplified diagram of this system.

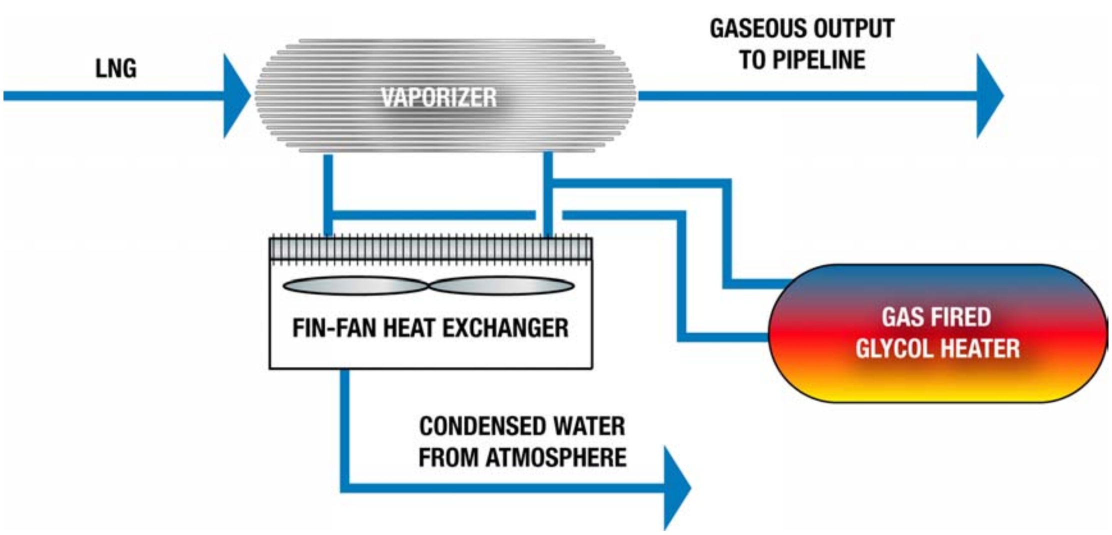

Figure 4: STV indirect system

Costs

The total installed cost for this configuration was 141% of the based SCV installation. Total operating cost included a 81.9% reduction in gas consumption due to atmospheric heat exchange, but an annual gas consumption increase of 120,000 MMBTU/year due to the use of lower efficiency heaters. Pump and fan operation, with operation of blowers for the backup fired heaters results in total power consumption of 239,539 MWhr/year, resulting in a total consumption difference of 76,884 MWhr/yr.

Environmental Impact

Operation of the single loop STV system in the Gulf Coast environment will generate 242 million gallons of condensate at an average temperature of 60 degrees F. This water can either be collected and treated for industrial use or discharged to the environment.

While large amounts of cold air will be discharged to the atmosphere, significant fogging is not expected.

Technology – differing working fluids

One variable that may allow for reduced system cost is altering the working fluid used in the system. Use of a phase change material like propane will reduce the required heat exchanger size by utilizing latent heat transfer within the tubes, thereby reducing the necessary heat transfer surface area of the unit. Other working fluids will have varying degrees of specific heat capacity, with an associated reduction or increase in required heat transfer surface area and pumping power. Regardless of the freezing point of the working fluid, operating constraints are in place to minimize freezing on the outside of the heat exchanger tubes.

Two loop system with direct contact heat exchanger

In order to take advantage of the higher heat transfer surface area of a direct contact heat exchanger, the two loop system replaces the indirect heat exchanger with an intermediate plate and frame heat exchanger with a secondary fresh water loop comprised of a pump and direct contact heating tower. Supplemental heat is supplied via a gas fired fluid heater on the primary working fluid. The direct contact atmospheric heat exchanger has a design flow rate of 84,000 gpm with a design inlet temperature of 35 degree F. Operation of the system at reduced inlet temperature will have a positive impact on system performance.

Footprint

The total plan footprint of this system is 90,000 square feet, about 210% larger than the SCV base system. Figure 5 shows a simplified diagram of this system.

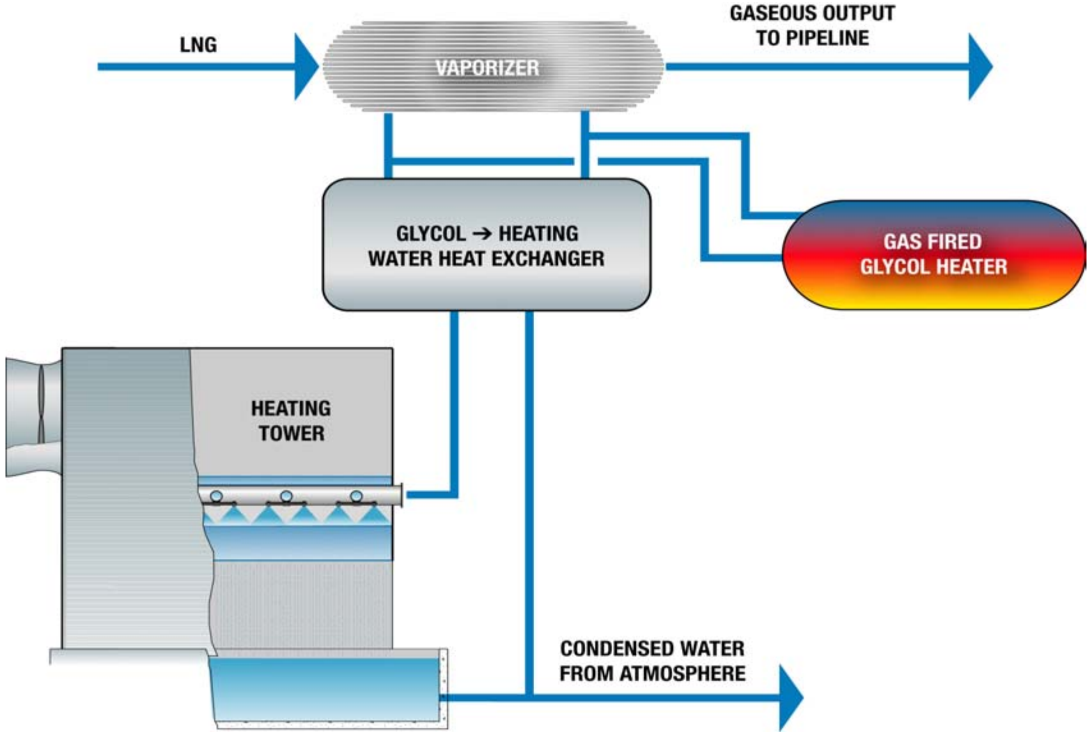

Figure 5: STV direct system

Costs

The total installed cost for this configuration was 131 % of the based SCV installation. Total operating cost included a 73.9 % reduction in gas consumption due to atmospheric heat exchange, but an annual gas consumption increase of 173,000 MMBTU due to the use of lower efficiency heaters. Pump and fan operation with backup fired heater blower power increases power consumption for a total consumption of 208,234 MWhr/year, resulting in a total consumption difference of 45,580 MWhr/yr.

Environmental Impact

Operation of the two loop STV system in the Gulf Coast environment will generate 347 million gallons of condensate at an average temperature of 55 degrees F. Warmer discharge temperatures can be obtained by locating the blowdown line upstream of the plate and frame heat exchanger. This water can either be collected and treated for industrial use or discharged to the environment.

While large amounts of cold air will be discharged to the atmosphere, significant fogging is not expected.

Technology

One variable that may allow for reduced system cost is altering the primary working fluid used in the system. Use of a phase change material like propane will reduce the required plate and frame and shell and tube heat exchanger size by utilizing latent heat transfer on the primary fluid side of these exchangers, thereby reducing the necessary heat transfer surface area of the unit. Other working fluids will have varying degrees of specific heat capacity, with an associated reduction or increase in required heat transfer surface area and pumping power.

Other technologies to evaluate are cross-flow heating towers vs. the counterflow design in this study. Cross flow towers have a wider footprint yet lower height, which could provide a reduction in total installed cost, yet no change in operating costs.

SCV – hybrid with direct contact heat exchanger

In order to take advantage of high efficiency burner technology, a direct contact heat exchanger may be operated with an SCV, circulating and heating water directly from the SCV. Compared to an SCV-only system, this configuration adds a circulating water pump and a heating tower. For the purpose of this study, we assumed no space constraints and selected a heating tower based on maximum gas savings. Practical applications can utilize smaller heating towers that deliver less annual process heating, still saving a significant amount of fuel. Additionally, this study assumed no changes in the operating temperature of the SCV water bath, resulting in a heat exchanger inlet temperature of 55 degrees F and a flow rate of 327,000 gpm. Extremely significant improvements can be achieved in this system by adjusting SCV operating conditions.

Footprint

The total plan footprint of this system is 300,000 square feet. Figure 6 is a simplified diagram of this arrangement.

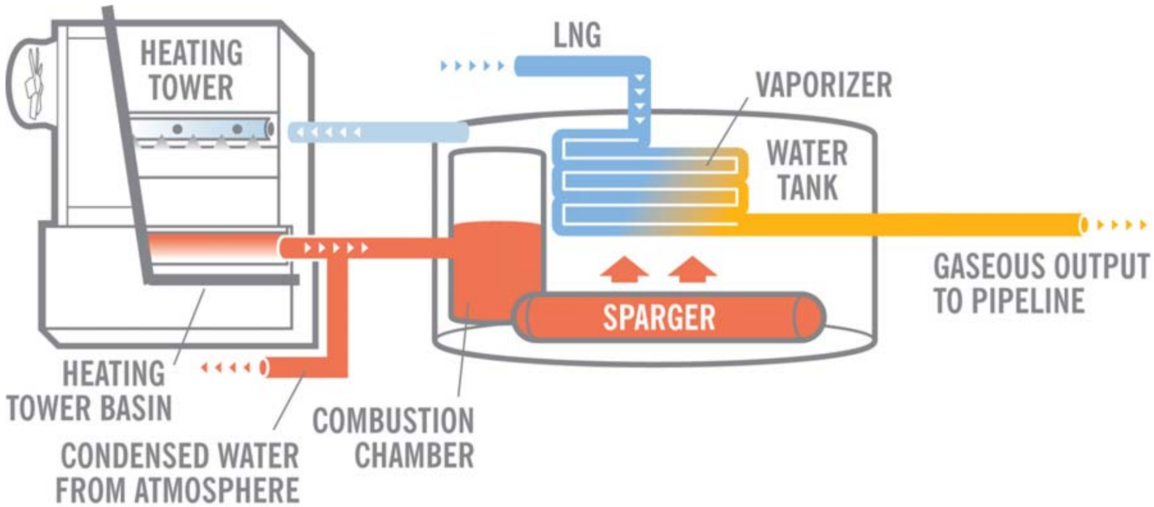

Figure 6: SCV hybrid with direct heat exchanger

Costs

The total installed cost for this configuration was 138 % of the based SCV installation. Total operating cost included a 66 % reduction in gas consumption due to atmospheric heat exchange. Pump and fan operation also increases power consumption to a total of 278,906 MWhr/year. SCV blowers operate at the same rate as standard SCV operation, so no deduct is taken for reducing blower operations, for a total difference of 116,251 MWhrs.

Environmental Impact

Operation of the system in the Gulf Coast environment will generate 313 million gallons of condensate at an average temperature of 61 degrees F. Warmer discharge temperatures can be obtained by locating the blowdown line upstream heating tower. This water can either be collected and treated for industrial use or discharged to the environment. During certain operating conditions, this water will include carbonic acid, a by product of submerged combustion, yet at a lower concentration that SCV-only operations. The system will need to employ the same discharge water treatment strategy as standard SCV-only operations, but with a significantly reduced treatment chemical cost.

While large amounts of cold air will be discharged to the atmosphere, significant fogging is not expected.

Technology

The proposed system assumes no changes to current SCV designs. There may be future development opportunities that reduce the overall system operating cost by performing certain modifications to the SCV to allow a lower energy consumption. Another system configuration could employ indirect contact heat exchangers. This configuration will have roughly the same gas savings, but with a significantly larger footprint, installed cost and operating cost.

ORV – hybrid with direct contact heat exchanger

An ORV can potentially be operated in a closed loop circulating water to a heating tower. For the system evaluated, an SCV is retained to supplemental vaporization heating during unfavorable atmospheric conditions. The evaluated system has a heat exchanger design inlet temperature of 40 degrees F with a flow rate of 112,000. Further optimization of the system should allow reduction of these numbers and significant overall system performance.

Footprint

The total plan footprint of this system is 95,000 square feet. Figure 7 is a simplified diagram of this arrangement.

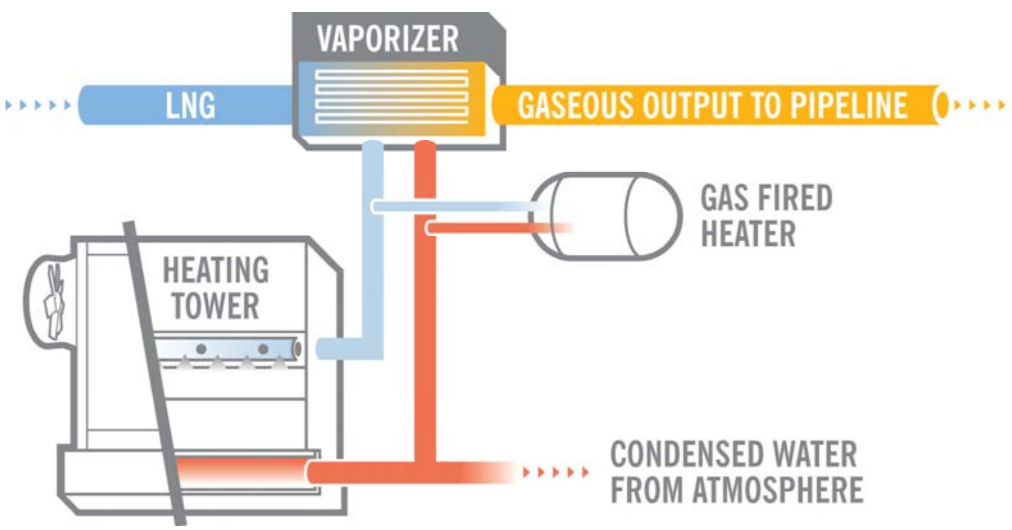

Figure 7: ORV hybrid direct system

Costs

The total installed cost for this configuration was 135 % of the base SCV installation. Total operating cost included a 70 % reduction in gas consumption due to atmospheric heat exchange. Pump and fan operation with backup fired blower power yields total power consumption of 164,758 MWhr/year, resulting in a total consumption difference of 2,104 MWhr/yr. The use of lower efficiency fired water heaters increases gas consumption by 199,000 MMBtu per year.

Environmental Impact

Operation of the system in the Gulf Coast environment will generate 328 million gallons of condensate at an average temperature of 55 degrees F. Warmer discharge temperatures can be obtained by locating the blowdown line upstream of the heating tower. This water can either be collected and treated for industrial use or discharged to the environment.

While large amounts of cold air will be discharged to the atmosphere, significant fogging is not expected.

Technology

Further operational cost savings may be obtained by utilizing seawater as the working fluid. This would involve a higher water discharge rate and a seawater intake system. The flow rate of seawater intake would be at least two orders of magnitude lower than a comparable ORV system, potentially clearing regulatory hurdles. As a result, by utilizing the inherent freeze-protection technology of ORVs, total system operating costs will be reduced significantly.

System Summary Table

System	SCV	STV-Indirect	STV-Direct	SCV-hybrid	ORV-hybrid
Footprint	1.0	3.8	2.1	7.1	2.2
Total Installed Cost	1.0	1.4	1.3	1.4	1.4
Annual Operating Cost					
Annual net MMBTU	1.0	0.28	0.40	0.34	0.30
Annual MW-hrs	1.0	1.47	1.28	1.72	1.01

Financial evaluation factors

Ranking and comparing these system options requires several economic evaluation factors to be established. More importantly, the expected operating cycle has the strongest impact on the financial assessment. Since most of the heat captured from the atmosphere occurs in warmer months, the an operating cycle that includes less than 100% throughput from April through September will have a significantly different gas savings profile, heavily favoring the two systems that utilize high efficiency submerged combustion. Financial risk assessment models should include evaluation of multiple plant operating models.

Care must be taken in performing calculations with the above referenced “Net MMBTU” figures. Net MMBTUs include MMBTUs saved from atmospheric heat exchange operation less additional MMBTUs burned through the use of lower efficiency burners. In most cases, the market price of the additional gas burned will be higher than the market price of the gas saved, so multiplying the net gas savings by a single value is inappropriate.

To compare the economics of the various systems, the following economic constants:

Land cost = $70/sq.ft
Min. gas price = $7/MMBtu
Winter gas price = $12/MMBtu
Power cost = $32/MWhr

Since this is a comparative analysis, comparing alternatives to SCV systems, we calculated the return on the incremental investment above the investment into an SCV-only system. In this manner, we accurately retain the integrity of the study as a comparative exercise. For each option, we calculated the total incremental operating savings based on the financial constants above. For simplicity, we calculated a simple payback and an investment return rate for each system. The simple payback period ranged from two years to just under 1.5 years. Achievement of these rates of return is completely dependent on continuous operation of the plant, especially in warmer hours of the year.

System	**STV-Indirect**	**STV-Direct**	**SCV-hybrid**	**ORV-hybrid**
Simple payback (yrs)	**1.71**	**1.36**	**1.73**	**1.57**
Annual return (%)	**59%**	**74%**	**58%**	**64%**

Because of the relatively high price of gas compared to the system installation cost, each system generates a high return on investment. The assessment for a particular plant, therefore, should include physical constraints associated with the site, as well as risk analyses that evaluate volatility in gas and electricity pricing, LNG supply and environmental issues.

Conclusions

While the ranking of systems will change based on individual project constraints and economics, for the site and financial factors evaluated, the following conclusions were reached:

1. The STV with direct heat exchanger is the most compact system, offering a good choice for space-constrained sites. This system also requires the lowest investment, making it the least risky in terms of potential stranded costs.
2. The STV with direct contact heat exchanger also represents the highest financial payback in the scenario analyzed.
3. The STV with indirect heat exchanger requires the highest initial investment and represents the highest risk in terms of potential stranded costs. However, for the operating temperatures that were modeled, this system has a slightly higher annual savings potential.
4. The SCV-hybrid requires the greatest amount of real estate, but can be improved with design changes that reduce the cold water inlet temperature to the atmospheric heat exchanger.
5. The operating temperatures chosen for the system primarily drove the differences in performance for all three direct-contact heat exchangers. Further reduction in cold-water inlet temperatures is possible, potentially improving the annual gas savings of these systems.