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THE ECONOMIC VIABILITY OF A MICROTURBINE COGENERATION SYSTEM

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ABSTRACT

Currently, electrical resistance heaters are used to produce most of the hot water in South Africa. Increasing electricity tariffs make these devices very expensive. This paper investigates the economic savings potential of using a cogeneration system made of microturbines, heat pumps and heat exchangers. Specifically the heating of water for large residential units is investigated. Different economic parameters are used to compare microturbine heat pump systems with electrical resistance heaters, natural gas boilers and heat pumps. For different main centres in South Africa, the amount of hot water is determined where a cogeneration system is economically more viable than other types of water heaters. It has been concluded that the most important influence factor is the electricity tariff. The higher the electricity tariff in a city, the smaller the number of domestic consumers where a cogeneration system becomes viable.

INTRODUCTION

The largest part of the energy needs in South Africa is supplied by non-recovery fossil fuels, especially coal. During the past 100 years approximately half of the world's known fossil fuel reserves have been used. Therefore, if there is no cutback in the consumption of fuel, serious shortages may arise in the near future.

Two strategies can be followed to decrease the consumption of fossil fuels [1]. Firstly, alternative energy sources may be developed, i.e. nuclear fission, nuclear fusion, solar, hydro, biomass, wood or photovoltaic cells. The second strategy is to use energy more efficiently. Among the methods are steps to prevent the wastage of energy, the recovery of waste heat, the improvement of process efficiency, including manufacturing and production, as well as the use of cogeneration.

The savings potential of the above-mentioned steps is very large. Furthermore, many of the steps are feasible in practice in the short term, and the implementation thereof has immediate economical advantages. Although large amounts of money have to be spent on the development of alternative energy sources, it is expected that savings will enjoy more attention in the future.

The cogeneration system used in this study (figure 1) consists of a microturbine with natural gas as fuel, a heat pump driven by the electricity produced by the microturbine, and a heat exchanger which uses part of the remaining heat in the turbine exhaust gases to heat water. Both the heat pump and heat exchanger produce hot water of 60°C.

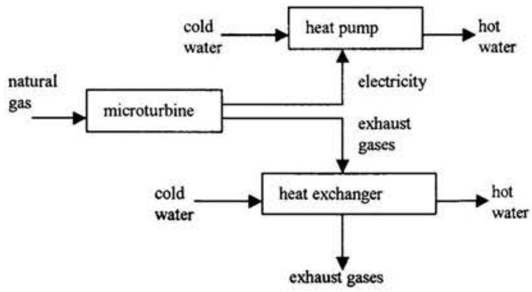

Figure 1: Schematic representation of cogeneration system

The aim of this paper is to compare the economic viability of this cogeneration system, also called *microturbine heat pump system,* with that of an electrical resistance heater, a natural gas boiler, and a heat pump water heater. Four economic parameters will be used: the payback period, the net present value, the internal rate of return and the life cycle cost. For different main centres in South Africa, the volume of hot water will be determined above which the cogeneration system is more viable than other types of water heaters.

NOMENCLATURE

a_0, a_1, a_2, a_3, a_4, a_5	Constants
AP	Payback period (year)

b_0, b_1, b_2, b_3	Constants
c_0, c_1, c_2, c_3	Constants
C	Initial capital layout required to purchase and install water heater (rand)
C_e	Initial capital layout required to purchase and install electrical resistance heater (rand)
C_g	Initial capital layout required to purchase and install gas-fired boiler (rand)
C_h	Initial capital layout required to purchase and install heat pump (rand)
C_p	Heating capacity of water at constant pressure (J/kg·K)
C_s	Initial capital layout required to purchase and install microturbine heat pump system (rand)
COP_m	Average heat pump coefficient of performance for month m
COP_{mr}	Real long-term coefficient of performance for month m
d	Number of days in the relevant month
$d_0, d_1, d_2, d_3, d_4, d_5$	Constants
E	Monthly electricity consumption (kWh)
E_{gas}	Monthly energy input of microturbine heat pump system (kWh)
E_w	Energy needed to heat water for one month (kWh)
E_l	Tariff validity (kWh)
f_0, f_1, f_2, f_3	Constants
F	Monthly energy cost (rand)
F_e	Monthly energy cost of electrical resistance heater (rand)
F_g	Monthly energy cost of natural gas boiler (rand)
F_h	Monthly energy cost of heat pump (rand)
F_j	Annual energy cost (rand)
F_s	Monthly energy cost of microturbine heat pump system (rand)
H	Height above sea level (metres)
LCC	Life cycle cost (rand)
n	Duration of project in months
NPV	Net present value (rand)
Q	Monthly average heat output of microturbine heat pump system (kW)
Q_{eg}	Monthly average heat output of exhaust gases of microturbine (kW)
Q_{hp}	Monthly average heat output of heat pump (kW)
Q_{ne}	Nominal heat output of electrical resistance heater (kW)
Q_{neg}	Nominal heat output of exhaust gases of microturbine (kW)
Q_{ng}	Nominal heat output of gas-fired boiler (kW)
Q_{nhp}	Nominal heat output of heat pump (kW)
Q_{ns}	Nominal heat output of microturbine heat pump system (kW)
r	Annual interest rate (fraction)
S_j	Annual saving on energy (rand)
S_{me}	Monthly saving on energy by using a microturbine heat pump system instead of an electrical resistance heater (rand)
S_{mg}	Monthly saving on energy by using a microturbine heat pump system instead of a natural gas boiler (rand)
S_{mh}	Monthly saving on energy by using a microturbine heat pump system instead of a heat pump (rand)
t_{ka}	Average temperature of cold feed-water during month concerned (°C)
t_{km}	Monthly minimum cold feed-water temperature (°C)
t_{wa}	Average temperature of hot water (°C)
T	Number of hours for heating (h)
T_b	Source or average ambient hourly wet bulb temperature (°C)
T_d	Average ambient hourly dry bulb temperature (°C)
T_n	Cost of natural gas (rand/kWh)
T_p	Sink or average hot water temperature in storage tank (°C)
T_1, T_2	Unit cost tariffs (rand/kWh)
W	Hot water consumption per day (kg)
η_{el}	Electrical efficiency of microturbine
η_g	Efficiency of gas-fired boiler
η_{tot}	Total efficiency of microturbine

CALCULATIONS

Energy requirements

The energy requirements of large residential units for the production of hot water have to be determined. The large residential units considered are hotels, hospitals, jails, boarding houses, holiday resorts and blocks of flats housing more than 100 residents. With a feed-water temperature mostly varying between 15°C and 25°C [2], the hot-water consumption per person per day at 60°C will vary between 75 and 140 litres [3, 4]. The purpose of this study is to determine what amount of water has to be heated for the microturbine heat pump system to be cheaper than the other types of water heating systems.

The amount of energy needed to heat water for a period of one month is:

$$E_w = W\, C_p\, (t_{wa} - t_{ka})\, d/(3600 \cdot 1000) \qquad (1)$$

For large hot-water installations, the heat loss from the storage tank is negligible in comparison with the energy needed to heat the water. Therefore, the heat loss is not taken into consideration, and equation (1), therefore, represents the total energy needed to heat a certain quantity of water for a period of one month [1].

Energy cost

The costs of energy to heat water with an electrical resistance heater, a gas-fired boiler, a heat pump and a microturbine heat pump system, have to be determined before the monthly saving in energy cost can be calculated [4, 5].

The electrical resistance heater and the heat pump use electricity as *fuel*. The electricity cost functions of municipalities for residential units usually assume the form:

$$F = T_1 E, E \le E_1$$
$$F = T_1 E_1 + T_2 (E - E_1), E > E_1 \quad (2)$$

The heat output of an electrical resistance heater is about the same as the electricity input, so the electricity cost can be written as:

$$F_e = T_1 E_w, E_w \le E_1$$
$$F_e = T_1 E_1 + T_2 (E_w - E_1), E_w > E_1 \quad (3)$$

The heat output of a heat pump equals the electricity input multiplied by the coefficient of performance. Therefore, the monthly cost in electricity for a heat pump installation is given by:

$$F_h = T_1 E_w/COP_{mr}, E_w/COP_{mr} \le E_1$$
$$F_h = T_1 E_1 + T_2 (E_w/COP_{mr} - E_1), E_w/COP_{mr} > E_1 \quad (4)$$

where:

$$COP_{mr} = 0.8\ COP_m \quad (5)$$
$$COP_m = a_0 + a_1 T_b + a_2 T_b^2 + a_3 T_p + a_4 T_b T_p + a_5 T_p^2 \quad (6)$$
$$T_p = 0.75\ t_{wa} + 0.25\ t_{ka} \quad (7)$$

$a_0 = 5.21333$, $a_1 = 1.03693\times10^{-1}$, $a_2 = 2.23706\times10^{-4}$,
$a_3 = -6.80730\times10^{-2}$, $a_4 = -1.04650\times10^{-3}$,
$a_5 = 3.10204\times10^{-4}$

The gas-fired boiler and the microturbine heat pump system use natural gas as a fuel. The weighted average price of natural gas, for project evaluation purposes in South Africa, is \$2/Gigajoule[1]. This means 15.33 rand/GJ or 0.0552 rand/kWh.

The efficiency η_g of a gas-fired boiler is approximately 80% and is independent of ambient temperature or height above sea level. Therefore, the energy cost for a gas-fired boiler is:

$$F_g = T_n E_w/\eta_g \quad (8)$$

The fuel of the cogeneration system is natural gas. The energy cost for a microturbine heat pump system is given by:

$$F_s = T_n E_{gas} = T_n E_w/(\eta_{tot} + \eta_{el} (COP_{mr} - 1)) \quad (9)$$

where:

$$\eta_{el} = b_0 + b_1 T_d + b_2 H + b_3 T_d H \quad (10)$$
$$\eta_{tot} = c_0 + c_1 T_d + c_2 H + c_3 T_d H \quad (11)$$

$b_0 = 3.18\times10^{-1}$, $b_1 = -1.20\times10^{-3}$, $b_2 = -3.18\times10^{-6}$,
$b_3 = 1.20\times10^{-8}$, $c_0 = 7.76\times10^{-1}$, $c_1 = 1.60\times10^{-3}$,
$c_2 = -5.18\times10^{-6}$, $c_3 = -1.07\times10^{-8}$

The monthly saving in energy cost is given by the difference between the running costs of two heating systems.

$$S_{me} = F_e - F_s \quad (12)$$
$$S_{mh} = F_h - F_s \quad (13)$$
$$S_{mg} = F_g - F_s \quad (14)$$

Capital cost

The economic attraction of different heating methods depends not only on the energy cost, but also on the capital cost of the apparatus. The capital cost is a function of the capacity of the apparatus. Estimates have been done to determine the capital costs within the capacity range 50 – 500 kW heat output [4]. Quotations for the cost of electrical resistance heaters, gas-fired boilers and heat pumps were obtained from South African companies which manufacture water heaters in multi-model series. After the prices had been adjusted to provide for installation and General Sales Tax (i.e. 14%), the total cost was plotted against the nominal output of the water heater. A linear regression of the results gives rise to the following equations, which describe the total cost:

Electrical resistance heater: $C_e = 200\ Q_{ne}$ (15)

Gas-fired boiler: $C_g = 94\ Q_{ng} + 7\,960$ (16)

where:

$$Q_{ne} = W C_p (t_{wa} - t_{km})/(1000\cdot3600\cdot T) \quad (17)$$
$$Q_{ng} = W C_p (t_{wa} - t_{km})/(1000\cdot3600\cdot T) \quad (18)$$

The cost of a heat pump is determined by the size and design of the unit. In order to keep costs down, the unit has to be as small as possible. Therefore, the nominal size of a heat pump is determined as

$$Q_{nhp} = W C_p [(Q_{hp}/Q_{nhp})_{min}]^{-1} (t_{wa} - t_{km})/(1000\cdot3600\cdot T) \quad (19)$$

The nominal capacity of the heat pump is the heating capacity at a water outlet temperature of 55°C and an ambient wet bulb temperature of 10°C. The minimum ratio $(Q_{hp}/Q_{nhp})_{min}$ is the minimum value of all the monthly averages, calculated by:

$$Q_{hp}/Q_{nhp} = d_0 + d_1 T_b + d_2 T_b^2 + d_3 T_p + d_4 T_b T_p + d_5 T_p^2 \quad (20)$$

where:

$d_0 = 9.94564\times10^{-1}$, $d_1 = 3.70009\times10^{-2}$, $d_2 = 3.45315\times10^{-4}$,
$d_3 = -3.14830\times10^{-3}$, $d_4 = -1.59120\times10^{-4}$, $d_5 = -4.69420\times10^{-5}$

Again, the total cost of different heat pumps was plotted against the nominal output. A linear regression gives rise to the following equation:

$$C_h = 1\,780\ Q_{nhp} + 24\,700 \quad (21)$$

The capital cost of the microturbine heat pump system can be calculated as follows:

$$C_s = 2\,285\ Q_{ns} + 242\,400 \quad (22)$$

where:

$$Q_{ns} = W C_p [(Q/Q_{ns})_{min}]^{-1} (t_{wa} - t_{km})/(1000\cdot3600\cdot T) \quad (23)$$

The minimum ratio $(Q/Q_{ns})_{min}$ is the minimum value of all the monthly averages, given by:

$$Q/Q_{ns} = 0.583\ Q_{hp}/Q_{nhp} + 0.417\ Q_{eg}/Q_{neg} \quad (24)$$

The ratio Q_{hp}/Q_{nhp} can be calculated with equation (20), the ratio Q_{eg}/Q_{neg} can be calculated by the following equation:

$$Q_{eg}/Q_{neg} = f_0 + f_1 T_d + f_2 H + f_3 T_d H \quad (25)$$

where

$f_0 = 1.02$, $f_1 = -1.80\times10^{-3}$, $f_2 = -1.23\times10^{-4}$, $f_3 = 2.16\times10^{-7}$

Equation (25) can only be used when the dry bulb ambient temperature is higher than 0°C, so it can generally be used in South Africa.

Economic parameters

The following parameters are used in determining the economic advantages of microturbine heat pump systems:

The payback period is given by:

[1] The exchange rate is \$1 = 7.665 rand (as on 20/11/2000).

$$AP = C_s/S_j \quad (26)$$

where the initial capital layout includes the purchase price as well as the cost of freight and installation of the system. The annual saving on energy is the sum of the monthly savings, given by equations (12), (13) and (14) for the different types of water heaters.

The net present value is given by:

$$NPV = (S_j/r)\,(1 - (1 + r/12)^{-n}) - C_s \quad (27)$$

where an annual interest rate of 20% is used (i.e. $r = 0.2$). The actual interest rate in South Africa is 14.5%. With $r = 0.2$, a rather conservative approach is used. It is also assumed that a microturbine heat pump system will last 10 years. In other words, the duration of the project is 120 months. It is further assumed that the microturbine heat pump system will have no resale value at the end of the project.

The internal rate of return is that value of the annual interest rate which reduces the net present value to zero.

The life cycle cost is defined as the capital cost plus the present value of all the energy costs across the lifespan of the apparatus:

$$LCC = C + (F_j/r)\,(1 - (1 + r/12)^{-n}) \quad (28)$$

RESULTS

Calculations were executed in an Excel spreadsheet to determine the payback period, the net present value, and the internal rate of return for 27 main centres in South Africa. Calculations were done for storage tank capacities of 10 000, 20 000, 30 000, 40 000, 50 000, 60 000, 70 000, 80 000, 90 000 and 100 000 l/day. Four different heating times were considered for each storage tank capacity: 12, 16, 20 and 24 h/day respectively. The system is considered to be operational throughout the year. The life cycle cost of the different types of water heaters was calculated for the same storage tank capacities as above. Calculations were done for the heating apparatus operational for 12 months/year and respectively 16 and 20 h/day. In this paper, only the results for a few representative cities will be discussed. More detailed figures and results for other main centres in South Africa can be found in [4].

The payback period is a measure of the approximate number of years that are required for the total value of savings to equal the total required investment. The lower the payback period, the more viable the investment.

When the microturbine heat pump system is compared with an electrical resistance heater, it can be deduced from figure 2 that the payback period varies between 2.5 and 21.0 years for a heating time of 12 h/day; between 2.0 and 19.2 years for a heating time of 16 h/day; between 1.7 and 18.2 years for a heating time of 20 h/day and between 1.5 and 17.5 years for a heating time of 24 h/day during the coldest month of the year. The increase in the heating period from 12 to 24 hours results in the installation of a smaller system, therefore, a decrease in payback period. Systems with a payback period longer than 10 years are not viable at all, because a microturbine heat pump system only lasts 10 years. A payback period shorter than five years is considered to indicate a viable system.

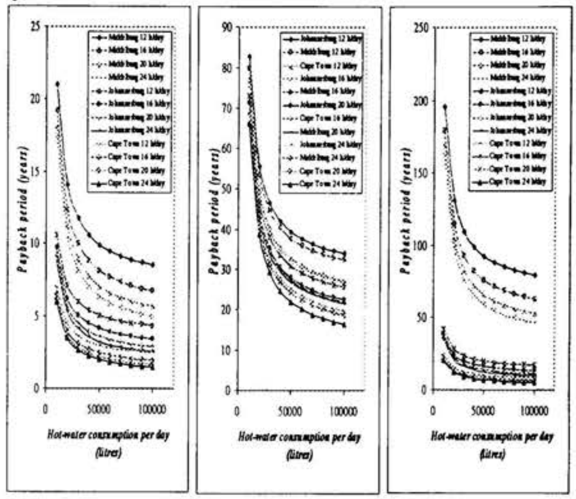

Figure 2: Payback period of microturbine heat pump system compared with electrical resistance heater (left), natural gas boiler (middle) and heat pump (right)

The hot-water volumes where the payback period is exactly five years for a heating time of 16 h/day for the centres Johannesburg, Pretoria, Durban, Cape Town and Port Elizabeth are shown in table 1. This means that for hot-water volumes higher than those shown in table 1, the cogeneration system is considered more viable than the electrical resistance heater.

Centre	Hot-water volume [l/day] where payback period = five years	net present value = 0 rand	life cycle cost break-even point
Johannesburg	31052	44301	39179
Pretoria	24834	31817	29332
Durban	19082	23000	21790
Cape Town	15951	17684	16894
Port Elizabeth	18704	22587	21217

Table 1: Hot-water volume above which microturbine cogeneration system is more viable than electrical resistance heater (heating time of 16 h/day) [l/day]

Comparison between a microturbine heat pump system and a natural gas boiler (figure 2) leads to average payback periods varying between 18.7 years (100 000 l/day, heating time of 24 h/day) and 82.5 years (10 000 l/day, heating time of 12 h/day). These high values indicate that natural gas boilers are more viable than a combination of microturbines and heat pumps for any volume of hot water.

When the microturbine heat pump system is compared with heat pumps directly connected to the electrical grid, big variations in payback periods can be seen (figure 2). The most important reason is the variation in electricity tariff. In Middelburg the electricity is very cheap (0.14 rand/kWh), which results in very long payback periods. In Johannesburg and Cape Town electricity is much more expensive (respectively 0.24 and 0.33 rand/kWh), which results in much shorter payback periods. In general, the payback periods are very long, which indicates that the microturbine heat pump system is not economically viable, compared with a heat pump water heater.

A second parameter used to describe the economic viability of a microturbine heat pump system is the net present value. This value is more efficient than the payback period because the changing value of money along the lifespan of the system is brought into account. When the net present value is positive, the project is considered viable.

From figure 3 (comparison with electrical resistance heater), it can be deduced at what amount of hot water the net present value becomes positive. The longer the heating time, the higher the net present value. For most centres, the higher the volume of hot water, the higher the net present value. Two centres, however, Middelburg and Pietermaritzburg, do not follow this trend for the shorter heating times. The reason is the cheap electricity tariff in these two centres. The second term in equation (27), which represents the capital cost of the microturbine heat pump system, becomes too dominant, and causes the net present value to decrease when the hot-water production increases.

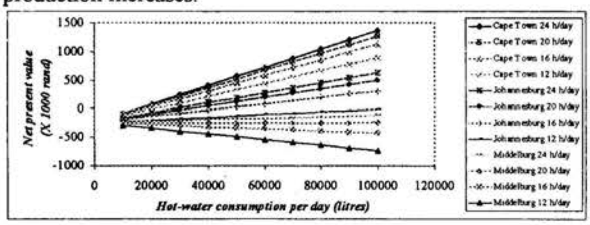

Figure 3: Net present value of microturbine heat pump system compared with electrical resistance heater

Comparison with natural gas boiler and heat pump only show negative net present values. The higher the amount of hot water, the more negative the net present value, which indicates that a microturbine heat pump system is never economically viable when compared with a natural gas boiler. The reason is that the savings are not high enough to equal the investment. At this moment, the capital cost of a microturbine heat pump system is very high. The expectation is, however, that this cost will drop as soon as more manufacturers enter the market. The result will be an increase in the net present value, which will even become positive for the higher volumes of hot water, when the microturbine heat pump system is compared with a heat pump.

In table 1 the volumes of hot water where the net present value becomes positive, are shown for five main centres in South Africa. For hot-water volumes higher than the ones shown in table 1, the microturbine cogeneration system is considered more viable than the electrical resistance heater.

The internal rate of return is defined as the interest rate that will result in a net present value of zero. The higher the internal rate of return, the more viable the system.

It can be deduced from figure 4 that the shorter the heating time, the lower the internal rate of return. It is, therefore, important that a microturbine heat pump system must not be unnecessarily large. Again, the values of the internal rate of return are negative or very low when the system is compared with a natural gas boiler or a heat pump (figure 4), which shows the lower viability of the microturbine heat pump system.

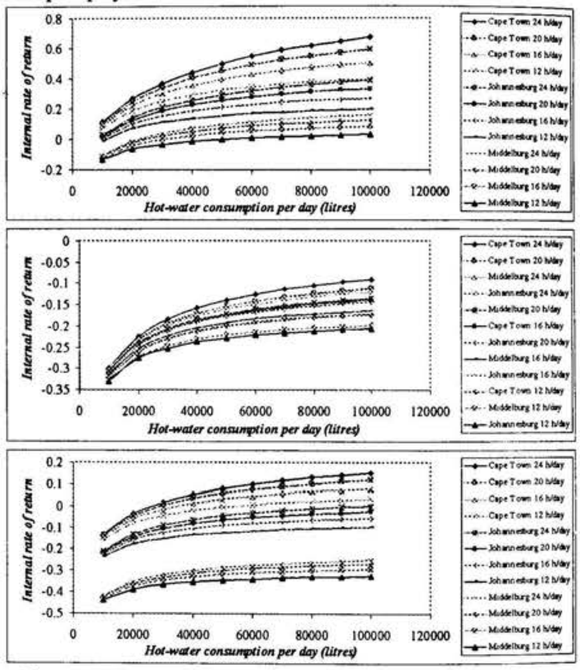

Figure 4: Internal rate of return of microturbine heat pump system compared with electrical resistance heater (top), natural gas boiler (middle) and heat pump (bottom)

The life cycle cost represents the sum of the capital cost and the present value of all the energy costs across the lifespan of the water heater. The system with the lowest life cycle cost is the most viable. The life cycle cost for the different types of water heaters is shown in figure 5 for the centres Johannesburg, Middelburg and Cape Town, for a heating time of 16 h/day and 20 h/day. Figure 5 shows that the life cycle cost of the electrical resistance heater and the natural gas boiler does not change much with a varying heating time. The life cycle cost of the heat pump and the microturbine heat pump system is lower for a longer heating time.

Table 1 shows at what hot-water volume the life cycle cost of the microturbine heat pump system equals the life cycle cost of the electrical resistance heater. Above this volume, the microturbine cogeneration system is more viable.

The life cycle cost is the only economic parameter to show a volume of hot water below 100 000 l/day where the microturbine heat pump system is more viable than a heat pump water heater. The reason is the capital cost of the water heater. The life cycle cost is the only parameter that brings this cost into account (see equation (28)) and is therefore the most accurate parameter to base the decision on.

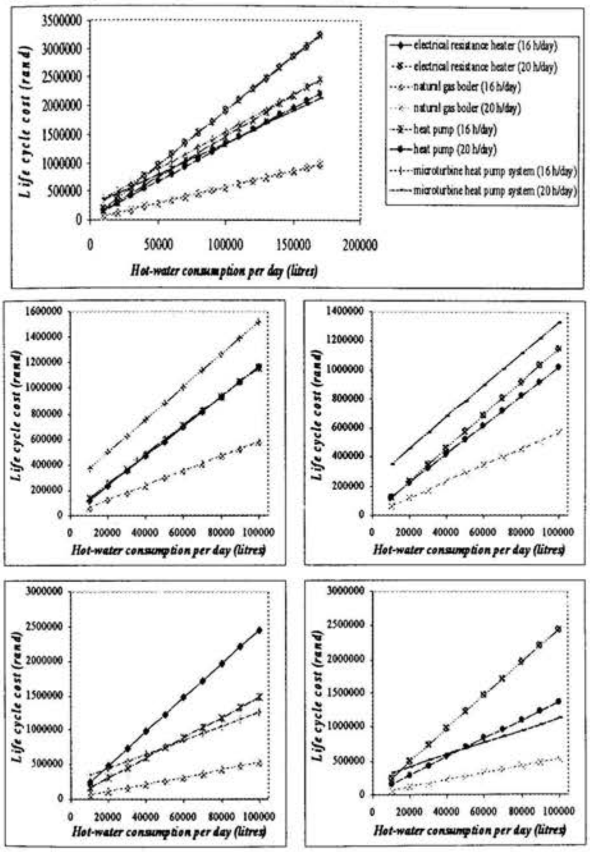

Figure 5: Life cycle cost for different types of water heaters in Johannesburg (top), Middelburg (middle) and Cape Town (bottom)
(heating time of 16h/day (left) and 20 h/day (right))

One person uses approximately 100 litres of hot water per day. According to table 1, a microturbine heat pump system working 16 h/day, is thus more viable than an electrical resistance heater in a building of 170 residents in Cape Town, 220 residents in Durban and Port Elizabeth, 295 residents in Pretoria and 390 residents in Johannesburg.

The influence of different parameters on the economic viability of a microturbine heat pump system was investigated. The life cycle cost decreases when the heating time per day increases. This makes a cogeneration system more viable when it works more hours per day. The most important influence on the life cycle cost of a system is the electricity tariff. It is more viable to install a microturbine heat pump system in a centre with high electricity prices. Factors such as the ambient temperature, the feed-water temperature and the height above sea level have no visible influence. These parameters tend to cancel each other.

Presently, the capital cost of microturbines is still very high. As more microturbine manufacturers enter the market, the prices will drop and the use of microturbines combined with heat pumps will become viable for lower amounts of hot water.

CONCLUSION

To study the economic savings potential of a cogeneration system, four different economic parameters were used: the payback period, the net present value, the internal rate of return and the life cycle cost. The cogeneration system made of microturbines, heat pumps and heat exchangers was compared with electrical resistance heaters, natural gas boilers and heat pumps. For different main centres in South Africa, the amount of hot water where the cogeneration system is more viable than other water heaters was determined. The most important influence factor is the electricity tariff: the higher the electricity tariff, the smaller the number of domestic consumers where the cogeneration system becomes viable.

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TEMPERATURE CONTROL IN REFRIGERATED TRANSPORT WITH A SNOW BAG

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ABSTRACT

Snow shooting is a recently patented process that is used in refrigerated transport. Pressurised liquid carbon dioxide is injected via a "snow" lance into a permeable snow bag mounted near the ceiling of the insulated container. The decrease in pressure causes the liquid carbon dioxide to convert to "snow" and vapour inside the snow bag. It is the purpose of the paper to develop a model that can be used to predict the heat transfer and more specifically the heat transfer coefficient of a snow bag. This model could then be used to predict the amount of snow and/or size of the snow bag for a given size of body and heat load. This is done with experimental and theoretical energy balances over the container walls and the snow bag.

INTRODUCTION

Countries with a warm climate and high ambient temperatures require careful temperature control of all perishable products from producer to consumer. Should the cold chain be broken, spoilage of food products will occur mainly through microbial growth such as bacteria, yeast and mould. Wide temperature fluctuations will also affect the quality of the frozen and chilled foods during transportation. The demands placed on the refrigerated transport industry over the last decade has increased as more is understood about the importance of cargo temperature maintenance.

Refrigerated transport equipment [1], can be broadly classified by the type of refrigeration system used. These types are ventilation, product subcooling, water ice, dry ice (carbon dioxide), liquid nitrogen or liquid carbon dioxide spray, eutectic plates for holdover, mechanical refrigeration and snow shooting. Snow shooting will now be discussed.

Snow shooting

Snow shooting should not be confused with liquid carbon dioxide spray. It is a new technology on which no literature has been published as far as could be determined. It has been recently patented in South Africa [2]. After loading the cargo into the insulated body, the doors are closed and the pressurised liquid carbon dioxide (R-744) at a pressure of 2 MPa (gauge) is injected via a "snow" lance into a permeable snow bag mounted near the ceiling of the insulated container (Figure 1). As is shown in Figure 2 the decrease in pressure causes the liquid carbon dioxide to convert to "snow" and vapour inside the snow bag. At an atmospheric pressure of 101.325 kPa one kilogram of liquid carbon dioxide expands to form 0.54 kg of gas at a temperature of approximately -78.5°C and an enthalpy of 47 kJ/kg as well as 0.46 kg of solid snow, also at a temperature of -78.5°C, but with an enthalpy value of 640 kJ/kg. This results in 8% of the cooling capacity in a gas phase and 92% in a solid phase.

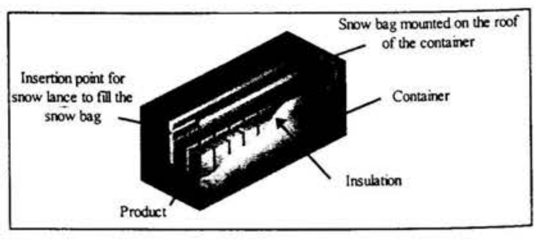

Figure 1: Container with snow bag.

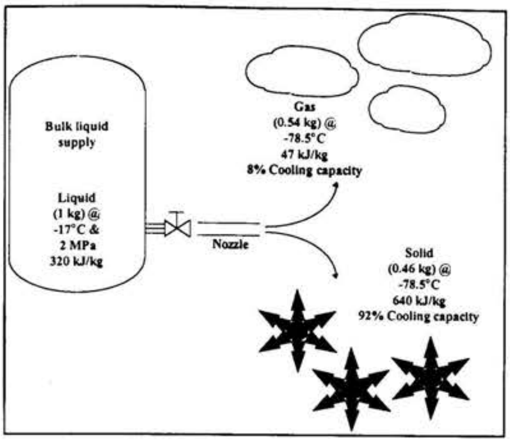

Figure 2: Carbon dioxide snow formation.

The snow bag is designed so that only the snow particles are retained, thus allowing the cold vapour to escape and pre-cool the container walls and product. The permeable snow bag acts as a phase separator, separating the carbon dioxide snow from the gas. The snow in the snow bag sublimes to cold carbon dioxide gas, flows down the cargo keeping it cold. The snow is not in direct contact with the product or the container walls, therefore this method can be used for most chilled as well as frozen products. Semi-compaction of the solid carbon dioxide snow phase occurs during charging, which reduces the sublimation rate of the snow, resulting in a more controlled release of the refrigeration capacity.

Snow shooting compared to direct injection into the snow bag results in limited thermal shock on the transport container or vehicle walls because of the containment of the snow within the snow bag. Therefore, a more uniform time versus temperature profile within the transport container or vehicle and cargo is obtained because of the relatively low rate of "snow" sublimation and separation of the snow from the cargo.

A further advantage of using snow shooting is that the inert carbon dioxide atmosphere reduces all bacterial activity in the cargo and hence snow shooting is intrinsically hygienic. In addition, the cargo is reliably kept at or below the temperature at which it is loaded into the truck. Another advantage is that the cold carbon dioxide is twice as dense as normal air and this dense gas quickly sinks and a natural thermal convection pattern is set up. There is therefore no need for a forced draft that assures a negligible weight loss due to dehydration of fresh produce. It compares very favourably to the typical two percent per 24 hour weight loss that occurs with mechanical refrigeration units that blow dry air past the fresh produce and pick up its moisture.

Snow shooting releases carbon dioxide to the atmosphere that contributes to global warming. However, the carbon dioxide used in South Africa where the process was developed is a by-product from chemical plants that releases it to the atmosphere. The process of liquifying and storing the carbon dioxide before it is used for snow shooting among others therefore does not directly increase global warming but delays it.

The amount of snow injected into the snow bag determines the length of time that the cargo may be maintained at its loaded temperature. This is established from experience with similarly insulated and sized vehicles.

Some of the advantages of snow shooting against mechanical refrigeration are: A small capital investment is needed as no refrigeration unit is required. The simpler design results in almost no maintenance costs; less dehydration of product occurs due to the absence of forced air circulation; the equipment is easier to install; no downtime is needed due to defrost cycles; faster cool down of internal structure; easier cleaning and maintenance; shorter delivery and installation time of equipment; it is environmentally friendly, no CFC's or HCFC's are released and no noise pollution occurs; less wear-and-tear on vehicle itself because the bag is much lighter than a mechanical refrigeration unit and therefore results in a higher payload; no personnel discomfort during off-loading due to the absence of forced air circulation and a higher payload since this method is lighter than a mechanical refrigeration unit.

Some of the disadvantages of snow shooting are: The single, most important health hazard that carbon dioxide presents is the risk of suffocation: by excluding oxygen from the lungs, the necessary precautions must therefore be taken when the doors of the delivery vehicle are opened.

Carbon dioxide is normally stored as a cold liquid at 2 MPa (gauge) pressure and at -17°C. To maintain these conditions and prevent loss through evaporation, special vessels are needed to store the very cold liquid. These are designed to limit the heat exchange between air and the cold liquid. They are double skinned and filled with an expanded insolent. The small losses due to the heat in-leak of the vessel are taken care of by a small integral mechanical refrigeration unit. The refrigeration unit recondenses the evaporated carbon dioxide back into a liquid, thus reducing the pressure in the vessel. The capital cost and running cost of this "plant" is therefore a disadvantage.

It is important to ensure that the correct amount of snow is added for the duration of the journey (too much will increase the operating cost and too little will prevent the refrigeration effect) - this factor can be complicated by non-constant journey times and vehicle breakdowns. Additional leftover snow at the end of the journey is of little practical use. The COP of snow shooting is lower than that of mechanical refrigeration and therefore results in higher operating costs. Snow-shooting is not a cost-effective means of lowering the temperature of the product. It is best used to maintain product temperatures. Products that need to respire during transit such as fruit or vegetables should be transported with caution, as prolonged exposure to high concentrations of carbon dioxide could cause spoilage.

The conversion rate efficiency using the snow bag is very close to the theoretical value of 50% (compared to the measured value of 46%), i.e., one kilogram carbon dioxide theoretically converts to 0.5 kg carbon dioxide snow. There is a slight (±0.3 to ±0.7%) decrease in percentage solid mass formed at 30 kPa atmospheric pressures compared to 101.325 kPa depending on the starting liquid pressure.

Presently the size of the snow bag and length of filling for a particular journey is determined through practical experience with similarly insulated and sized vehicles. This is not cost effective and sufficient to answer to the needs of different customers. It is the purpose of this paper to develop a model that can be used to predict the heat transfer and more specifically the heat transfer coefficient of a snow bag. This model could then be used to predict the amount of snow and/or size of the snow bag for a given size of body and heat load.

The layout of the paper is as follows. Firstly the overall heat transfer coefficient of the container will be calculated after which the heat transfer coefficient of the snow will be calculated. Both named coefficients are calculated with experimental work. Thereafter the heat transfer coefficient of the snow is calculated with theoretical equations and compared to the previous calculated heat transfer coefficient.

EXPERIMENTAL WORK

The container used is a thermal insulated container used for transport. The snow bag construction is mounted to the roof of the container with the shade netting material used as the phase separator (snow bag). K-type thermocouples are positioned on the inner and outer face of each wall to determine the temperature on the inside and outside respectively. The thermocouples used are calibrated to an accuracy of $\pm 0.5^{0}$C.

For heating the inside of the container six electric heating elements of 1 kW each are placed at even intervals inside the container. In order to measure the heat load inserted into the container via the heating elements a kilowatt-hour meter is connected on the power supply line of the heating elements. Four fans are placed inside the container to circulate the air if a uniform temperature inside the container is required.

The snow bag construction is suspended with two s-type load cells from the roof to measure the weight of the snow inside the snow bag. Three depth gauges are positioned at the top of the container to measure the height of the snow in the snow bag at different positions during the experiment without opening the doors of the container.

The container and snow bag dimensions and the position of the thermocouples, elements, fans, depth gauges and load cells are shown in Figure 3.

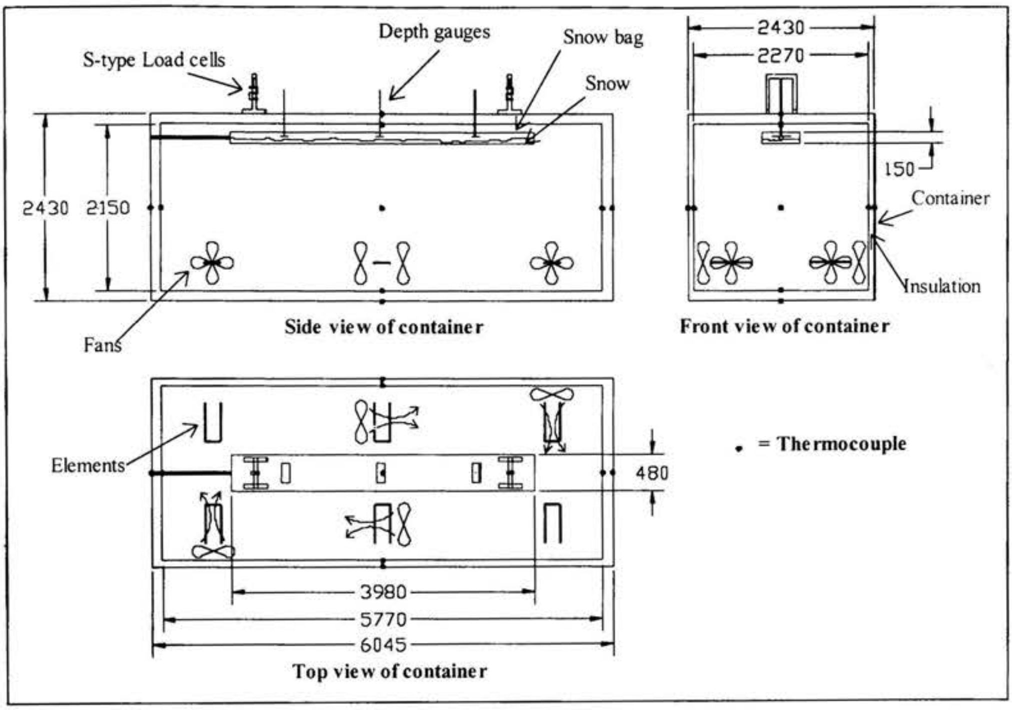

Figure 3: Diagrammatic view of the container.

The overall heat transfer coefficient of the container is determined by heating the inside of the container until a temperature difference of approximately 50°C between inside and outside is reached. With a constant temperature difference an energy balance (Equation 1 and 2) is used to calculate the overall heat transfer coefficient of the container walls.

$$Q_l = Q_t + mC_p\Delta T/t$$
$$\therefore Q_l = UA(T_{ia} - T_a) + mC_p(T_{i2} - T_{i1})/t \quad (1)$$
$$\text{where } T_{ia} = (T_{i2} + T_{i1})/2 \quad (2)$$

The heat transfer coefficient of the snow in the snow bag can thereafter be calculated with Equation 3 and 4. The area of the snow is determined with the height of the snow measured with the depth gauges on the roof. The heat transferred rate of the snow calculated in Equation 3 is checked by using the sublimation rate of the snow determined from the load cell readings according to Equation 5.

$$Q_l + Q_t = Q_s + mC_p\Delta T/t$$
$$\therefore Q_l + UA(T_a - T_{ia}) = h_s A_s (T_{ia} - T_s) + mC_p(T_{i2} - T_{i1})/t \quad (3)$$
$$\text{where } T_{ia} = (T_{i2} + T_{i1})/2 \quad (4)$$

$$Q_{lc} = \Delta m_t (h_g - h_i)/t \quad (5)$$

THEORETICAL HEAT TRANSFER COEFFICIENT

The equations used to find the theoretical heat transfer coefficient of the snow are obtained from reference [3]. In the equations all the properties are calculated at the film temperature defined in Equation 6. The snow bag is separated into three different sections according to the orientation of the surface and the way the free convection gas moves over the surface. For the sides and points the characteristic length, Rayleigh number, Nusselt number and heat transfer coefficient are calculated with Equation 7, 9, 10 and 13 respectively. For the top the characteristic length, Rayleigh number, Nusselt number and heat transfer coefficient are calculated with Equation 8, 9, 11 and 13 respectively. For the bottom the characteristic length, Rayleigh number, Nusselt number and heat transfer coefficient are calculated with Equation 8, 9, 12 and 13 respectively.

$$T_f = (T_{ia} + T_s)/2 \quad (6)$$
$$L_c = h_{av} \quad (7)$$
$$L_c = \frac{A_s}{P} \quad (8)$$

$$Ra_L = \frac{g\beta(T_s - T_{ia})L_c^3}{\upsilon\alpha} \quad (9)$$

$$Nu_L = 0.68 + \frac{0.670 Ra_L^{1/4}}{\left[1 + (0.492/\Pr)^{9/16}\right]^{4/9}} \qquad Ra_L \leq 10^9 \quad (10)$$

$$Nu_L = 0.27 Ra_L^{1/4} \qquad 10^5 \leq Ra_L \leq 10^{10} \quad (11)$$

$$Nu_L = 0.15 Ra_L^{1/3} \qquad 10^7 \leq Ra_L \leq 10^{11} \quad (12)$$

$$h_s = \frac{k}{L_c} Nu_L \quad (13)$$

RESULTS

The overall heat transfer coefficient of the container is determined with the temperature inside the container stabilising with approximately a 50°C temperature difference between the inside and outside of the container. Figure 4 shows the temperature curves of the inside and ambient temperatures. The product of *UA* is determined to be 45 *W/K*.

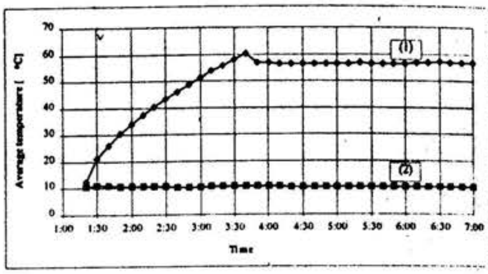

Figure 4: Average inner (1) and ambient (2) temperature versus time with four elements and fans switched off at 3:50 am.

The heat transfer coefficient of the snow in the snow bag is determined experimentally and theoretically and shown in Figure 5. The corresponding temperatures inside and outside the container are also shown on Figure 5.

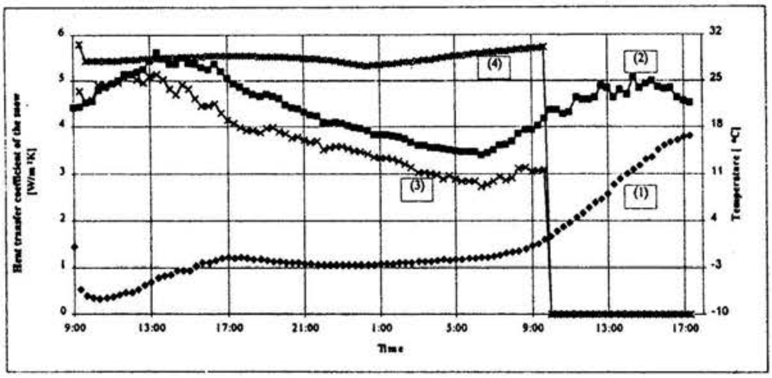

Figure 5: Average inner (1), ambient (2) temperature, experimental (3) and theoretical (4) heat transfer coefficient of the snow versus time.

DISCUSSION OF RESULTS

The product of the overall heat transfer coefficient and the area of the container (*UA*) determined to be 45 *W/K* can unfortunately not be verified theoretically because of deterioration of the container insulation and damage to the container itself. Fortunately the verification of the energy balance with the load cells also verifies this result. The verification of the heat transferred rate of the snow with the load cells result in an error of 12%, which is accurate for an energy balance.

The average difference between the experimental and the theoretical calculated heat transfer coefficient of the snow is 19%. The reason for this can be explained by the fact that the equations used to calculate the theoretical heat transfer coefficient describes a flat surface with free natural convection. In the snow sublimation this is not 100% the case since:

1. The snow bag creates a boundary between the free stream and the snow and therefore cause unnatural flow patterns.
2. The bag is close to the roof, therefore at the top there is not a free stream flow.
3. A critical reason is that a layer of ice forms on the snow bag from the moist air. This creates an insulation layer that prevents the snow from sublimating.
4. The free stream temperature around the snow bag was assumed to be constant. Actually there is a temperature gradient (warmer at the top side than the bottom side since the cold CO_2 moves down).
5. 100% CO_2 was assumed around the snow bag while it was actually a mixture of CO_2 and air.

The first four reasons will give a higher theoretical than experimental value since they all create restriction for natural convection to take place. Through calculations it is seen that the fifth reason actually gives a slight increase in theoretical heat transfer coefficient of the snow.

CONCLUSION

With an average difference of 19% between the theoretical and experimental heat transfer coefficient it can be concluded that the theoretical equations can be used as a first order estimation to calculate the amount of snow needed for a specific journey. However more experiments have to be done, perhaps in conjunction with finite element analysis to achieve more accurate results. Finite element analysis can be used to take into consideration the insulation layer created by the ice and the snow bag as well as the temperature gradient inside the container. A further advantage is that actual products can be simulated to increase the value of the results obtained.

NOMENCLATURE

A area of the container used in experiments $[m^2]$
A_s area of the snow in the snow bag $[m^2]$
α thermal diffusivity [m^2/s]
β volumetric thermal expansion coefficient $[K^{-1}]$
C_p specific heat at constant pressure $[J/kg.K]$
g gravitational acceleration [m/s^2]
h_{av} average height of the snow in the snow bag $[m]$
h_g enthalpy of saturated vapour $[kJ/kg]$
h_i enthalpy of saturated solid $[kJ/kg]$
h_s heat transfer coefficient of the snow in the snow bag $[W/m^2.K]$
k thermal conductivity $[W/m.K]$
L_c characteristic length $[m]$
m mass of carbon dioxide gas in container $[kg]$
m_t total mass of snow bag construction and snow $[kg]$
Nu_L Nusselt number based on the characteristic length
Pr Prandtl number
P perimeter of surface area $[m]$
ρ density of carbon dioxide $[kg/m^3]$
Q_l product heat load $[W]$
Q_{lc} heat transferred rate of snow calculated with load cells $[W]$
Q_s heat transferred rate of snow $[W]$
Q_t heat transmitted through container walls $[W]$
Ra_L Rayleigh number based on the characteristic length
T_a ambient temperature outside the container $[^0C]$
T_f film temperature $[^0C]$
T_{i1} temperature inside the container at a given instant $[^0C]$
T_{i2} temperature inside the container after a certain period $[^0C]$
T_{ia} average temperature inside the container $[^0C]$
T_s temperature of snow in the snow bag $[^0C]$
t time in which temperature change $[s]$
U overall heat transfer coefficient of the container walls $[W/m^2.K]$
v kinematic viscosity $[m^2/s]$

REFERENCES

[1] ASHRAE *1994 ASHRAE Handbook Fundamentals SI edn* pp. 27.3-27.5 (1994).

[2] Patent number 91/5027 hold by *African Oxygen Limited*, South Africa.

[3] Incropera FP & DeWitt DP, *Introduction to Heat Transfer*, John Wiley & Sons Inc. (1996)/