Programme Area: Smart Systems and Heat

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Title: Review of gas heat supply to heat pumps

Abstract:

This report investigates heat network operating temperatures and the effect current decisions regarding the design and operation of networks will have on the long term operational efficiencies. The report demonstrates how heat networks can be future proofed for changes to operating temperatures. This is applied to a heat network in Bury to highlight and quantify impacts.

Context:

The Spatial Energy Plan for Greater Manchester Combined Authority project was commissioned as part of the Energy Technologies Institute (ETI) Smart Systems and Heat Programme and undertaken through collaboration between the Greater Manchester Combined Authority and the Energy Systems Catapult. The study has consolidated the significant data and existing evidence relating to the local energy system to provide a platform for future energy planning in the region and the development of suitable policies within the emerging spatial planning framework for Greater Manchester.

Energy Systems Catapult

Support for EnergyPath Networks

Task 017: Review of gas heat supply to heat pumps

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Contents

1 Executive Summary

This report investigates heat network operating temperatures and the effect current decisions regarding the design and operation of networks will have on the long term operational efficiencies. The report demonstrates how heat networks can be future proofed for changes to operating temperatures. This is applied to a heat network in Bury to highlight and quantify impacts.

The report shows the viable flow temperature limits to network operation lie between 80 °C and 50 °C due to required domestic hot water delivery temperatures. Maximum network return temperatures lie between 60 °C and 40 °C, dependent on the level of alteration to existing building space heating systems. The minimum network return temperature is 25 °C, as the theoretical limit from a space heating system, and a typical return temperature from a domestic hot water system.

The report demonstrates that heat network pipes can be optimally sized, determined through a combination of minimising the heat loss and minimising the pumping energy. This is dependent on the design operating temperatures, and as such changing the operating temperature from the design can lead to less efficient performance.

This report finds that a network designed and operated at 50 °C flow and 40 °C return gives the lowest annual energy consumption. These operating temperatures minimise losses in the system, and enable better Coefficient of Performance of generation from heat pumps (CoP of 4.4). To operate at these return temperatures, extensive retrofitting to the building heating systems will be required. This is expected to be a significant cost and may outweigh the benefits associated with the lower temperatures. Operating this network at higher temperatures increases the energy loss, although this increase is minimal (+0.5% /annum) when using flow temperatures of 80 °C to 50 °C variable to outside air temperature and 60 °C return at design conditions, dropping to 25 °C at minimal demand. This could be used as a means of delaying any extensive retrofitting programmes.

This report finds that a network designed and operated at 80 °C to 50 °C variable flow and 60 °C return gives comparatively low energy losses across the system. Using a variable flow temperature improves the heat pump Coefficient of Performance (CoP 3.7) when compared to a constant 80 °C flow (CoP 3.0). To operate at these temperatures, only minimal retrofitting to the building heating systems will be required. Additionally, operating this network at a lower temperature (50 °C flow and 40 °C return) will reduce the energy losses across the network. This provides lower lifetime energy losses if the network operates at 50 °C flow and 40 °C return for less than 2/3 of its lifetime.

As the capital costs for both 80 °C to 50 °C variable flow, and 50 °C flow will be similar. The decision on which network has the lowest lifetime cost should be based on if the costs of the lower lifetime energy consumption from the 50 °C flow network outweigh the costs of the extensive retrofitting needed to achieve these temperatures in the existing building stock.

For the selected network in Bury, the total losses as a percentage of total heat demand (2,521 MWh/annum) are shown below.

	Designed for 50/40	**Designed for 80v/60**
Operated at 50/40	1.65%	1.71%
Operated at 80v/60	2.13%	2.04%

2 Introduction

It is anticipated that the dominant heat source for heat networks built in the next few years will be natural gas fired (CHP and boilers). However, to achieve low carbon emissions in the future it is likely that natural gas fired heating technologies will need to be phased out and retrofitted with electric systems using low carbon electricity.

Arup has been appointed to carry out a desktop review of the implications of the switch from a hot water network (>70 °C) powered using gas to a warm water network (50 °C-55 °C) powered using electric heat pumps.

This report determines viable flow and return conditions available to warm water (50 °C-55 °C) and hot water (>70 °C) networks. These are compared and the implications of switching between them once a network is installed is shown. This allows informed decision making of the design of a network to sufficiently future proof its operation over the scheme lifetime.

2.1 Scope of Review

The scope of this review excludes assessment of the financial viability of the proposed heat network connections within the worked examples. The following justification has been provided from EnergyPath Networks Request for Quotation documentation:

The local area energy strategy is designed to meet an agreed carbon constraint for the study area at the lowest cost to society. It is expected that an additional cost over business as usual will be incurred and that many of the proposed measures will not be viable without some form of incentive or carbon price.

3 Building considerations

This report considers heating systems which primarily aim to maintain comfort for heat consumers. Heating systems thereby provide a service where consumers are kept warm whilst hot water is provided on demand to meet bathing, showering and washing needs. To maintain warmth, the space heating system should be capable of heating rooms to at least 20 °C. To provide a hygienic supply, the hot water system should also provide hot water at 44 °C. These requirements should be fulfilled regardless of whether a warm water (50 °C-55 °C) or hot water (70 °C) network is selected.

3.1 Domestic Hot Water

Domestic hot water (DHW) refers to the hot water used in sinks, showers and baths in any type of building (not just domestic dwellings). For the purposes of this report, DHW does not refer to hot water used in industrial processes. DHW demand is weather independent, as external temperatures do not influence the demand for DHW.

3.1.1 Storage

Where DHW is stored, whether in a cylinder with a coil or a calorifier with external plate heat exchanger, there is a risk of legionella proliferation. These bacteria cause diseases including Legionnaire’s disease, a potentially fatal form of pneumonia. To prevent legionella growth, DHW storage must not:

- Be between 20 °C and 45 °C.
- Allow water droplets to be produced.
- Have deposits that can support bacterial growth such as rust, scale, sludge, biofilms and organic matter (Health and Safety Executive, 2013).

The Department of Health require for stored hot water that ‘the flow temperature of hot water out of the calorifier should be a minimum of 60 °C’.

To ensure a flow temperature of 60 °C out of the calorifier, the approach temperature to the calorifier must be higher, i.e. the temperature from the network must be above 60 °C. This means that DHW storage is not suitable for warm water networks where the flow temperature is less than 55 °C, unless some form of heating is used to raise the temperature of the water in the supply pipework.

3.1.2 Instantaneous

Instantaneous DHW can be used in place of water storage. In instantaneous hot water systems connected to a heat network, a heat exchanger is utilised to heat a cold-water feed to the required temperatures instantaneously.

The Department of Health provide the following maximum recommended set delivery temperatures for various outlets (Department of Health, 2016):

Table 1 Maximum recommended set delivery temperatures

Activity	Maximum recommended set delivery temperatures
Showers and hair wash facilities	41 °C
Unassisted baths	44 °C
Bidets	38 °C

An instantaneous DHW system should therefore be designed to provide a minimum set delivery temperature of 44 °C. The supply temperature to the DHW should be hotter than the set point. Therefore, a supply temperature of 50 °C from a warm water network should meet all the building's DHWS demand. For the purposes of this report, warm water networks can supply all the instantaneous DHW demand.

Therefore, the lowest temperature possible to supply DHW demands is a 50 °C flow temperature. As stated previously, due to Legionella, a 50 °C flow temperature requires all DHW to be instantaneous.

3.2 Space Heating

Space heating systems heat the air in a closed space to maintain the comfort of occupants. Space heating systems are typically designed for a room temperature of ~20 °C with capacities based on a nationally defined design weather event. Space heating demands are subject to ambient air temperatures and exhibit weather dependency.

Existing buildings typically use fan coil units, air handling units, and/or steel or cast-iron radiators to fulfil their space heating requirements. In North America, forced-air heating with fan coils is the most commonly used method for heating, which is to be expected in light of a long tradition of using air conditioning. In Europe, where air conditioning is mainly used in commercial buildings rather than in private homes, systems of this kind are mostly confined to commercial buildings. Both radiators and fan coils work at high temperatures (>70 °C) and are suitable for less well insulated buildings with high peak loads and annual heating demands. Emitter sizes have traditionally been chosen for high design temperatures which can easily be supplied by fossil fuels that have dominated heat supply in most countries.

3.2.1 Impact of reducing flow temperatures

Most newly installed radiators operate at 80 °C flow, 60 °C return. Older systems typically operate at 82 °C flow, 71 °C return. Table 2 shows the impact of adjusting flow temperatures on radiator sizing for a room requiring 2,750 W of heat to maintain an internal temperature of 21 °C. The table shows that dropping the flow temperature from 80 °C to 50 °C whilst maintaining the same flow rate requires a radiator size 3.5 times the original size to provide the same capacity given the same room conditions. Therefore, reducing the flow temperature to 50 °C under similar flow conditions would require new, larger radiators throughout the building, which also reduces the area available in the building.

Table 2: Radiator Size to meet 2,750 W room heating requirement at 21 °C. Sizing approach follows BG30 methodology (BSRIA, 2007).

Flow	Return	Multiple of 80/60 Capacity
80	60	1
70	50	1.4
60	40	2.0
50	30	3.5

Radiators are typically oversized to allow the space to be heated up quickly; i.e. to minimise the period of time required to preheat the space to the desired temperature. The speed that the building is heated and the building's inertia is used to calculate an oversize factor compared to the steady state for the design condition (extreme weather event). Assuming a radiator is designed to operate at 80 °C flow, 60 °C return and is deliberately oversized by 20%, then the flow temperature from such a radiator could drop by 6 °C to 74 °C to maintain thermal comfort in the room at the same flow rate (i.e. provide 2,750 W of heating). The same deliberate oversizing of 20% allows an 82 °C flow, 71 °C return radiator to be operated as an 80 °C flow, 60 °C return radiator. The longer pre-heat period would need to be managed.

This highlights that the without major retrofits, flow temperatures can be reduced by small amounts without affecting the comfort of the building occupants.

3.2.2 Variable temperatures

Most modern radiator systems are capable of being operated at variable temperatures. Peak heating demands only occur during a few short periods of the year when the ambient temperature is coldest. At warmer ambient temperatures, a control box can be used to turn down heating system temperatures whilst still meeting thermal comfort requirements. Therefore, provided connected buildings can operate at variable temperatures, the network can also operate under variable temperatures.

3.2.3 Retrofit measures

To achieve a consistently low flow temperature in buildings designed to operate at higher temperatures, retrofit measures would be required. These could include longer preheat times, improved insulation levels, larger or more radiators or installing alternative heating emitters. Installing larger or more radiators is unlikely to be practical given space constraints in most buildings. Within offices, air handling units and fan coil units are more common than radiators and redesign may be required to operate at lower flow temperatures. Terminal unit upgrades would therefore be required for building heating systems to be compatible with a 50 °C flow temperature.

3.3 Heat Exchange (Network to Buildings)

To transfer heat from a heat network to building heating systems, some form of heat exchanger is typically installed. Although direct connections are feasible whereby the same water flowing through the network also flows through the heating emitters; the risk of contamination and system leaks is reduced when hydraulic separation (i.e. a heat exchanger) is installed. For individual dwellings, heat interface units (HIUs) are typically used whilst for larger buildings, heat substations are preferred.

This report assumes that indirect connections are used throughout the network, as this is considered lower risk. Including a heat exchanger introduces temperature changes across the exchanger. The heat network code of practice stipulates that the temperature difference across a plate heat exchanger – shall not exceed 5 °C. Generally, approach temperatures of 1 °C -2 °C, are feasible with plate heat exchangers.

3.3.1 Flow temperatures

The two limits to provide connected buildings with suitable flow temperatures, as discussed in the previous subsection are:

1. 50 °C peak flow temperature, which will require major refurbishment and retrofits. This limit is determined by the DHW requirement and includes allowance for the temperature difference across the heat exchanger.
2. 80 °C peak flow temperature, which will require relatively minor retrofits (controls adjustment and system rebalancing), much of which would be required when connecting a building to a heat network, independent of the supply temperature. This limit is determined by historic space heating system temperatures.

Note that new build buildings may be supplied via either temperature.

3.3.2 Return temperatures

Network return temperatures are a result of the temperature levels used in internal piping systems in buildings connected to the network and of the design and control of substations. Maintaining low return temperatures is an important consideration when designing heat substations and heat interface units. Low return temperatures offer benefits to the performance of the heat network. The limiting return temperature to a network is made up of a combination of the return temperature from the space heating systems (which is influenced by room temperatures) and the cold-water inlet to the domestic hot water system.

Return temperatures will be seasonal and will be dependent on the space heating demand. To enable heat transfer, there must be a temperature differential between the room and the radiator. Given a room temperature of ~20 °C this limits the space heating return aspect of the district heating return to a theoretical minimum of 25 °C.

Following minimal retrofit (80 °C peak flow temperature) it is assumed that the space heating system in an existing building will be rebalanced so that the maximum return temperature to the network is 60 °C. For warm water networks (50 °C peak flow temperature) the required extensive refit of existing building heating systems should allow return temperatures to be lowered. For the purposes of this analysis, it has been assumed that a 40 °C maximum is achievable in the heat network. Further reduction of the maximum return is anticipated to require unfeasible alterations to the building.

The return temperature from the DHW system, will be dependent on the cold-water supply temperature. It is assumed that 25 °C is a typical return temperature from a DHW system.

To determine an average return temperature over the course of the year, an estimated split of DHW and space heating usage is required. In the UK in 2013, across domestic, services and industry sectors, the ratio of space heating demand to domestic hot water demand was 82:18 (gov.uk, 2014).

Actual return temperatures tend to be much higher than theoretical return temperatures. The difference between theoretical and actual return temperatures can be explained by faults which occur in heat exchangers, control chains and system designs. Faults could be due to mistakes made in the design phase, commissioning or operation or due to malfunctioning components or set point errors. Faults are more prevalent in warm water networks than hot water networks because of more rigorous operating conditions.

Return temperatures have been calculated for the viable flow temperature operating conditions over a year supplying 2,000 MWh/annum to a typical demand profile. This analysis has used real weather data to capture demand variations. The maximum and minimum return temperatures calculated over the year for an 80 °C flow temperature was 58 °C, and for a 50 °C flow was 38 °C.

4 Heat Distribution

Heat network pipes use water to distribute heat from the energy centre to the building heat exchangers. Typically heat network pipes are located below ground. Heat network pipes are inefficient in that they lose heat and require pumping. Heat is lost as the pipes are hotter than the surrounding ground. Pumping is required to overcome the friction across the internal surfaces of the pipes and fittings to allow water to circulate and distribute heat.

4.1 Temperature

The viable network temperatures as determined in the last section are: flow temperatures between 80 °C max and 50 °C min, and return temperatures of between 60 °C (minimal refit) or 40 °C (major refit) max. These are laid out as three viable options in the table below. In addition to these set flow temperatures, it is possible to vary the flow temperature based on the demand. This may lead to more efficient operation of the network. This leads to a further option in which the flow varies between the two limits.

Table 3 Viable network temperature limits

Name	Maximum Flow Temperature	Minimum Flow Temperature	Maximum Return Temperature	Minimum Return Temperature
80/60	80 °C	80 °C	60 °C	25 °C
80/40	80 °C	80 °C	40 °C	25 °C
50/40	50 °C	50 °C	40 °C	25 °C
80v/60	80 °C	50 °C	60 °C	25 °C

For a given supply requirement determined by the building demand, and at a set supply temperature (and corresponding mass flow rate), there will be an optimum pipe diameter to supply the heat. This optimum point will be determined at the point where the combined heat loss and pumping requirement are lowest.

4.2 Heat Loss

Heat loss is a function of the ground conditions (soil type, wetness, depth, temperature), pipe diameter, flow and return temperatures and insulation type and thickness. This report assumes constant ground conditions for all networks. Larger pipe diameters increase the surface area of the pipe, causing larger heat losses. All pipe diameters selected throughout the report are industry standard sizes, as defined in the European Standard for district heating pipes (EN253).

Reducing flow and return temperatures helps to reduce losses as the temperature difference, the driving force for heat loss, is lowered. Heat loss is directly proportional to pipework water temperature.

With more insulation, heat losses are lower. The level of insulation on heat network pipes is defined (in EN253) in three levels; Series 1, Series 2 and Series 3. Series 1 is the thinnest and Series 3 is the thickest. The thickness of the insulation within each Series is linked to the pipe diameter (although due to the manufacturing of the pipes does not increase linearly with pipe diameter). As such, the choice of insulation is between the Series and an assessment of the preferred insulation Series is an economic assessment based on initial cost compared to long term heat loss reductions. This is not required to compare the effect of operating temperatures. To provide comparable results this report assumes Series 1 (EN253 standard insulation) is used throughout.

4.3 Pumping Energy

The internal surface of a pipe is not frictionless, therefore to circulate water around a heat network a pump needs energy to overcome the friction of the water flowing across the unsmooth pipe.

$$energy\ rate\ of\ pumping = volume\ flow\ rate * pressure\ difference$$

Frictional losses are proportional to the bulk velocity squared. This means that the pressure difference is also proportional to the volume flow rate squared. Therefore, the pumping energy required is proportional to the volume flow rate cubed.

4.4 Pipe Sizing

For a heat network, thick insulation and small pipe diameters are required to minimise heat losses. This should be balanced with a large pipe diameter to minimise pumping energy/ frictional losses. Practically, in operation, there is a balance to be achieved between heat loss and frictional loss. There is also a capital cost implication when considering trench size and pump size.

The following analysis displays the energy required to supply 2,000 MWh/annum of heat over 1 km, under previously determined flow and return temperatures. This is simulated over a year using weather data to more accurately capture variations in demand.

It should be noted that this analysis assumes a circulating pump efficiency of 2/3. The heat pump CoP is calculated using a source temperature of 10 °C, supply temperatures corresponding to each option (i.e. 80 °C, 50 °C, and variable between 80 °C and 50 °C), and an efficiency of 2/3 to the theoretical CoP. The CoP achieved is between 3.0 and 4.4. The return temperature stated indicates the maximum return of the heat network.

4.4.1 Light building retrofit

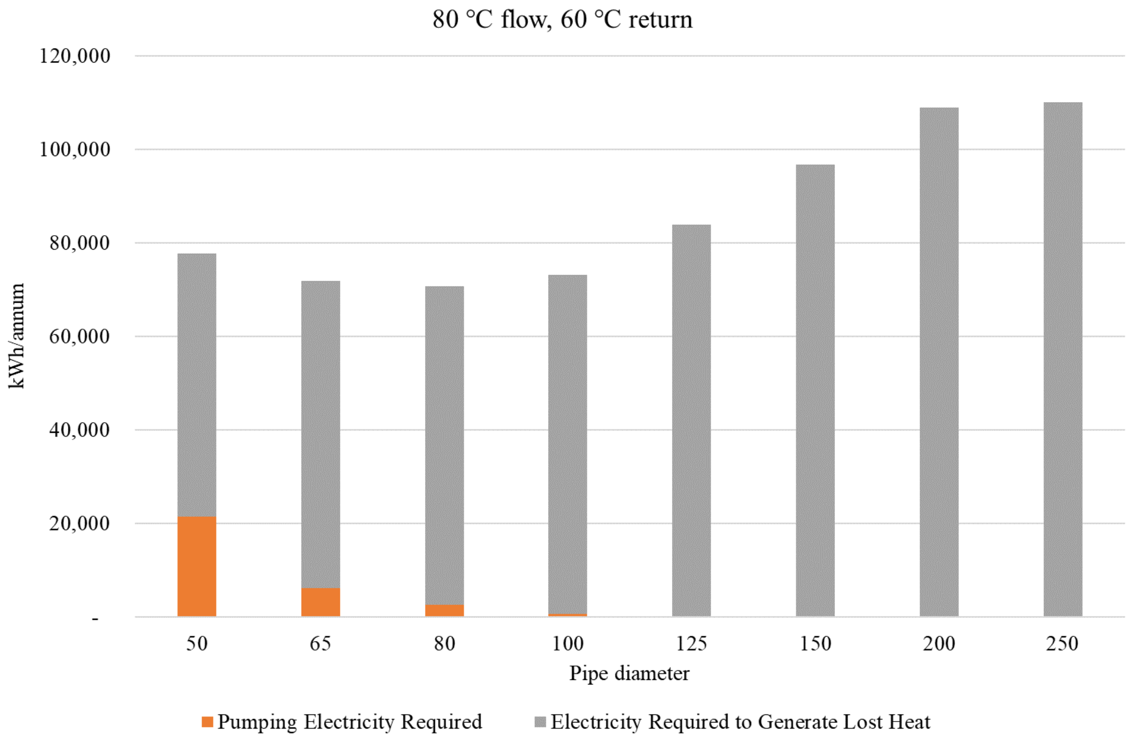


Figure 1 Energy losses for 80 - 60 network

4.4.2 Full building retrofit

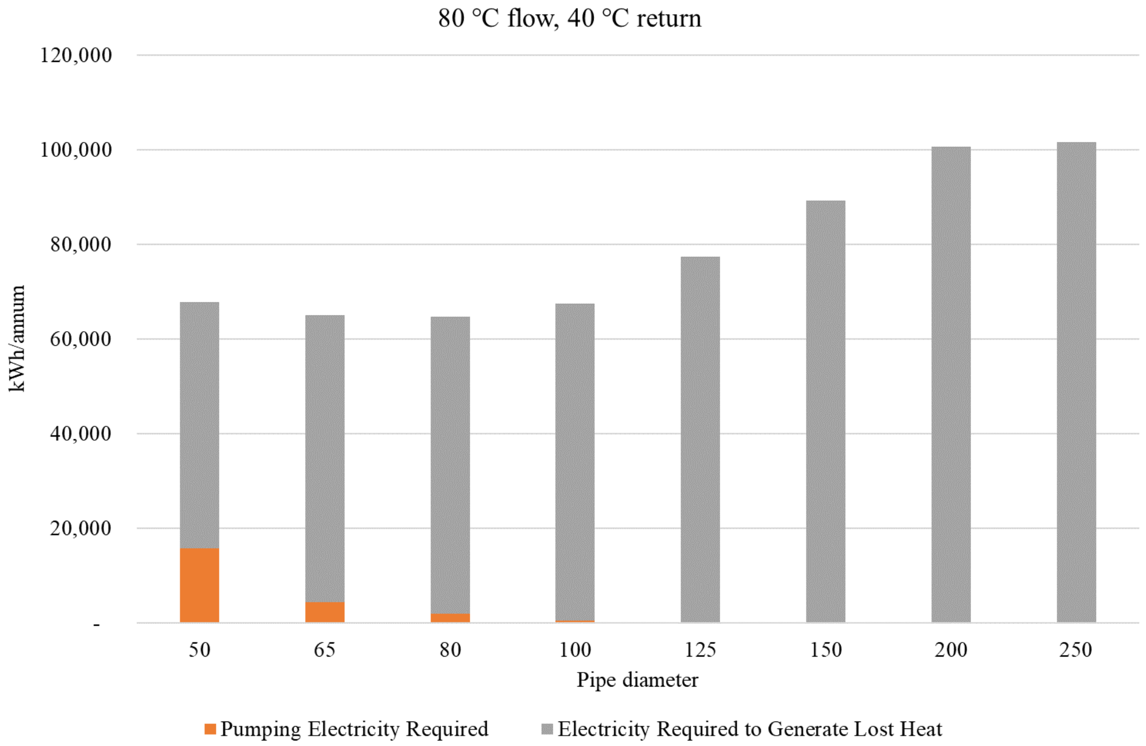


Figure 2 Energy losses for 80 - 40 network

4.4.3 Full building retrofit

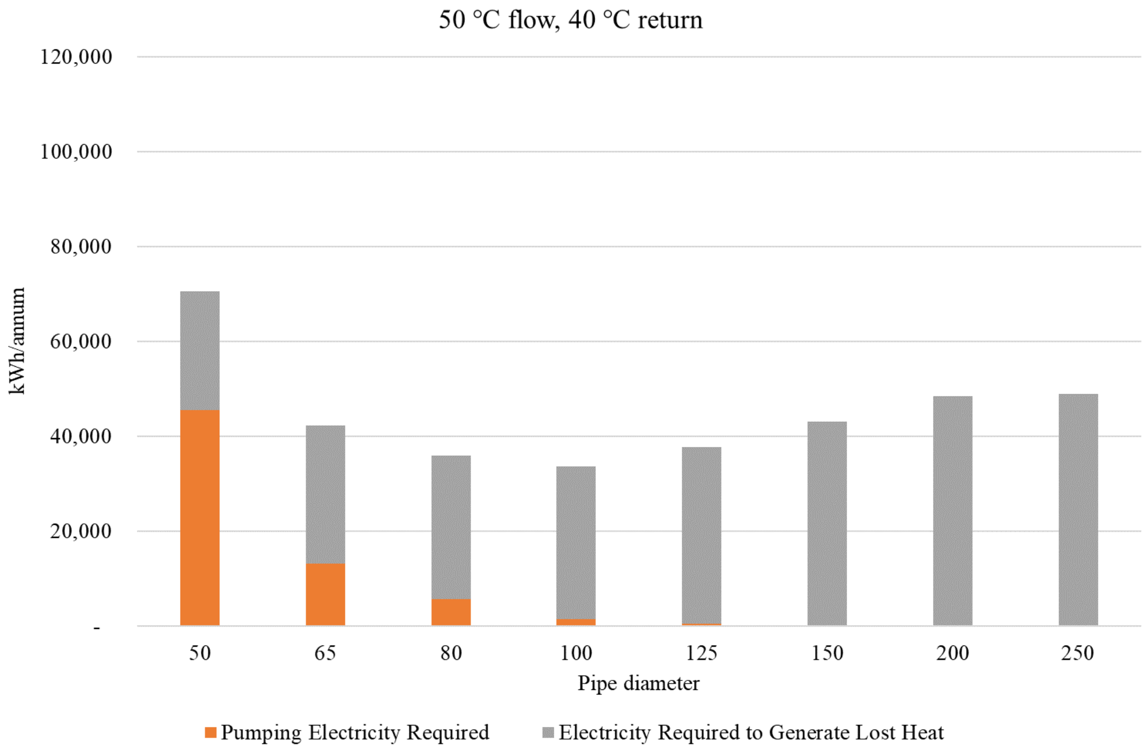


Figure 3 Energy losses for 50 - 40 network

4.4.4 Light building retrofit (variable)

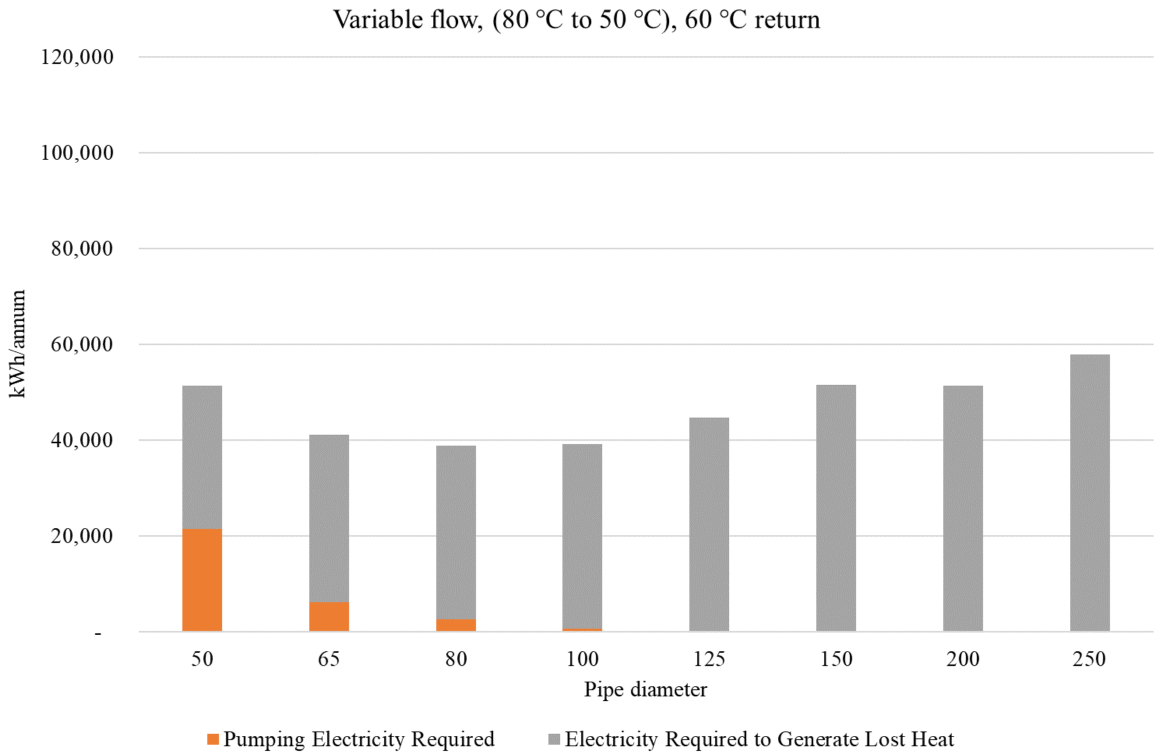


Figure 4 Energy losses for 80 (to 50 variable) - 60 network

These results show that for a given pipe length and demand, there is an optimum pipe diameter which provides the lowest total energy loss. This optimum, and energy loss, is not the same for all flow and return temperatures, shown in the below table.

For different annual demands, although the optimum pipe diameter will change, the relationship between the efficiencies of the optimum and pipes larger and smaller will remain similar. As a result, it is important to note the impact of increasing or decreasing the pipe size away from the optimum. This is shown in the table below as a percentage increase in total energy loss when increasing or decreasing the pipe diameter one size away from the optimum. The impact of going further than one pipe size can been seen visually in the previous charts.

Table 4 Optimum pipe sizing and corresponding total energy loss

Flow & return temperatures	**Optimum pipe diameter**	**Energy loss at optimum as percentage of annual demand**	**Energy loss at higher pipe size as percentage of annual demand**	**Energy loss at lower pipe size as percentage of annual demand**
80 - 60	DN80	3.5% (70.7 MWh/annum)	3.7% (73.2 MWh/annum)	3.6% (71.7 MWh/annum)
80 - 40	DN80	3.2% (64.7 MWh/annum)	3.4% (67.4 MWh/annum)	3.2% (64.9 MWh/annum)
50 - 40	DN100	1.7% (33.7 MWh/annum)	1.9% (37.7 MWh/annum)	1.8% (36.0 MWh/annum)
80/50variable - 60	DN80	1.9% (38.9 MWh/annum)	2.0% (39.3 MWh/annum)	2.1% (41.1 MWh/annum)

4.4.5 Summary

From the table, a 50 °C flow and 40 °C return network gives the lowest total annual energy loss. The sizing shows that a larger pipe diameter is more suitable for a lower temperature network.

A pipe network serving completely retrofitted buildings (50 °C flow, 40 °C return) provides only minimally greater energy savings, 5.2 MWh/a, than a pipe network serving minimally retrofitted buildings (80 °C to 50 °C variable flow, 60 °C return).

The analysis shows there is an increase in annual energy loss if the pipes are not sized optimally. The optimal pipe size will, over a wider network with a variety of pipe sizes supplying different loads, vary based on the temperature used to design the pipes. Therefore, designing for one condition, and then operating under different conditions result in sub-optimal pipe sizes being used. This will in turn lead to an increase in annual energy loss over what is possible at those operating temperatures, as shown above.

4.5 Future proofed pipe sizing

It is assumed that once heat network pipes are installed they will not be replaced to optimise sizing based on updated flow and return temperatures. Thus, a decision must be made on the appropriate means to size the pipes initially to future proof the pipes, reducing the energy losses over the project lifetime. This is dependent on the proportion of time spent in each temperature state, and the additional annual energy losses resultant from operating the network at a different temperature to which it was designed.

It should be noted that the impact on capital costs for installation as a result of different operating temperatures is negligible. The effect on pipe sizing is not significant enough to impact on materials or trench sizes.

The highest energy losses for all pipe diameter are under the 80 °C flow, 60 °C return operating temperatures. The lowest energy losses for all pipe diameters (except for DN50) are under the 50 °C flow, 40 °C return operating temperatures. Therefore, these have been selected to demonstrate the following analysis.

From analysis of the optimum pipe sizes for the temperature scenarios:

- If a network designed based on 80 °C flow, 60 °C return operating temperatures is operated under 50 °C flow, 40 °C return operating temperatures, the pipes will be undersized.
- If a network designed based on 50 °C flow, 40 °C return operating temperatures is operated under 80 °C flow, 60 °C return operating temperatures, the pipes will be oversized.

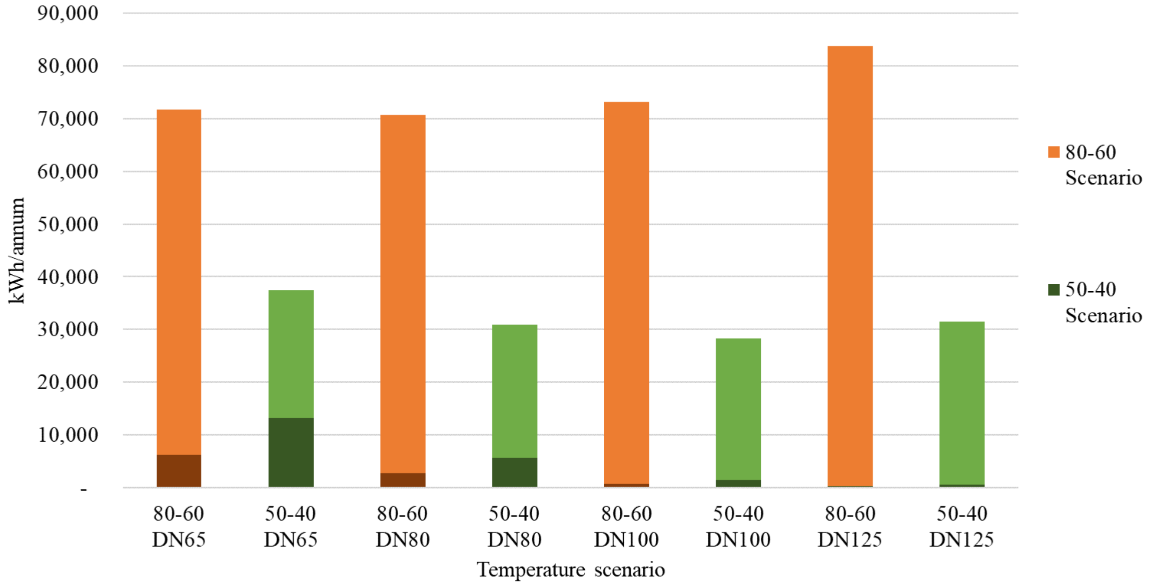


Figure 5 Energy loss comparison between 50-40 and 80-60 networks

The chart shows the increase in total annual energy loss compared to the optimum. The darker colour variation shows the pumping energy; the lighter colour shows the heat loss. For this analysis of 2,000 MWh/annum over 1 km of pipe, operating at the 50 °C flow, 40 °C return the optimum pipe size is DN100. Using undersized pipes will increase the total energy loss by 6.8% (+2.3 MWh/annum) with DN80 pipes, to 25.6% (+8.6 MWh/annum) with DN65 pipes. Operating the 80 °C flow, 60 °C return the optimum pipe size is DN80. Using oversized pipes will increase the total energy loss by 3.5% (+2.5 MWh/annum) with DN100 pipes, to 18.5% (+13.1 MWh/annum) with DN125 pipes.

If the time spent in each state differs, this needs to be included in the analysis as it will impact the preferred option.

5 Heat Generation

5.1 Technology overview

5.1.1 Natural Gas Generation

Heat generation from natural gas fuelled technologies can be split into gas CHP technologies and gas boilers. The gas CHP considered in this study are reciprocating engines.

Gas boilers operate through burning gas in a controlled environment, heating a heat exchanger, which heats up water. The efficiency of a gas boiler varies due to manufacturer, size and if the boiler is condensing. At flow temperatures below 55 °C, there is the potential to operate gas boilers in condensing mode. Condensing boilers achieve higher efficiencies by condensing the water vapour in the exhaust gases and so recovering the latent heat of vapourisation which is otherwise lost at higher temperatures. The condensed vapour leaves the system in liquid form, via a drain. This report assumes gas boiler efficiencies of 85% which is considered industry standard for non-condensing boilers. To simplify the analysis this report does not consider condensing boilers, as the increased complication in modelling the efficiency gains detract from the narrative of this analysis. Gas boilers are efficient at modulating to meet part-load demand, and do not suffer efficiency reductions at part load, hence they are modelled in this report as a flat 85% efficiency independent of demand and supply conditions. The efficiency is not expected to increase significantly in the future as gas boilers are a mature technology.

In a reciprocating engine CHP, useful heat can be extracted from three main sources, the intercooler, the engine and the exhaust gases. The intercooler is an intake air cooling device commonly used on turbocharged and supercharged engines to cool the air compressed by the turbo/supercharger, reducing its temperature and increasing the density of the air supplied to the engine. This improves combustion and increases the power output of the engine. Intercooler heat is typically at the lowest temperature (~45 °C) of the three waste heat sources; it is usually not commercially attractive nor technically feasible to utilise this heat in hot water networks. However, with warm water networks, this heat can be mixed with engine waste heat to increase the temperature of the intercooler heat so that it can be utilised on the network. This report assumes CHP heat is extracted from the engine block and exhaust, but not the intercooler. It assumes a constant efficiency.

5.1.2 Heat Pumps

Heat pumps utilise low grade waste heat occurring both naturally (in the air, ground or water bodies) and from man-made processes including industrial processes and cooling. A heat pump captures this waste heat using a refrigerant, a fluid which evaporates at low temperatures. The temperature of the waste heat can be increased using a reverse Carnot cycle, making the heat useable in heat networks.

The required energy input to raise the temperature to the desired value depends on:

1. The input temperature of the waste sink heat source;
2. The output temperature, in this case a heat network;
3. The refrigerant used as the heat transfer medium;
4. The non-perfect performance of physical plant and equipment.

As a result, the ratio of useful heat output to fuel (electricity) input, known as the Coefficient of Performance (CoP), is highly sensitive to these items. The higher the Coefficient of Performance, the more efficient the heat pump.

The temperature of the waste heat source has been assumed to be 10 °C. This is based on an approximation of year-round stable ground temperatures in the UK. This represents an average borehole for a ground source heat pump.

The temperature differential across the heat pump heat exchangers, required for heat transfer from the source to refrigerant and from refrigerant to the heat network, has been assumed to be 5 °C.

The Coefficient of Performance can be calculated using the following formulae:

$$COP = \eta * \frac{T_{output}}{T_{output} - T_{input}}$$

Where:

$\eta = system\ efficiency$

$T_{output} = T_{supply\ (heat\ network)} + dT_{heat\ exchanger}$

The temperature of the refrigerant at the output side of the heat pump, i.e. the flow temperature for a heat network with a temperature differential across the heat pump heat exchanger.

(Note the units must be in Kelvin, [K] = [°C] + 273.15).

$$T_{input} = T_{source} - dT_{heat\ exchanger}$$

The temperature of the refrigerant at the source side of the heat pump, i.e. the temperature of the source with a temperature differential across the heat pump heat exchanger.

(Note the units must be in Kelvin, [K] = [°C] + 273.15).

The maximum theoretical efficiency limit can be found when η = 1. In practice, this limit is not achievable due to the working efficiencies of the heat pump such as heat transfer through the heat exchangers and compressor efficiency.

Currently achieved system efficiencies are between 50% and 70%. As heat pumps are a mature technology, major breakthroughs are not expected and they are close to a cost-effective peak. As such, the expected efficiency for heat pumps installed in the next 20 years has been assumed to be 2/3.

Table 5: Heat Pump CoPs at reference values, with 5 °C temperature difference over the internal heat exchangers.

Source temperature	**Supply temperature**	**System efficiency**	**Achieved CoP**
10 °C	80 °C	2/3	2.98
10 °C	50 °C	2/3	4.37

As a worked example of the heat pump CoP for an 80 °C supply temperature:

$$COP = \frac{2}{3} * \frac{(80 + 273) + 5}{\big((80 + 273) + 5\big) - \big((10 + 273) - 5\big)}$$

$$COP = 2.98$$

The CoP can also be improved by decreasing the supply (flow) temperature.

The below figure shows that the CoP is not linearly related to the flow temperature. Greater gain in CoP are found when the small decreases are made to the flow temperature at lower temperatures in comparison to at higher temperatures. This means that minimal retrofits to building heating systems to reduce the flow temperatures from 82 °C to 80 °C or 70 °C will have minimal impact on the heat pump CoP compare to major retrofits which reduce the temperatures to 50 °C.

In summer months, a reduction in supply temperature may be feasible for consumers. In winter months, the heat pump could supply a lower base temperature, and this could be ‘topped up’ to the required supply temperatures by a gas fuelled boiler or CHP.

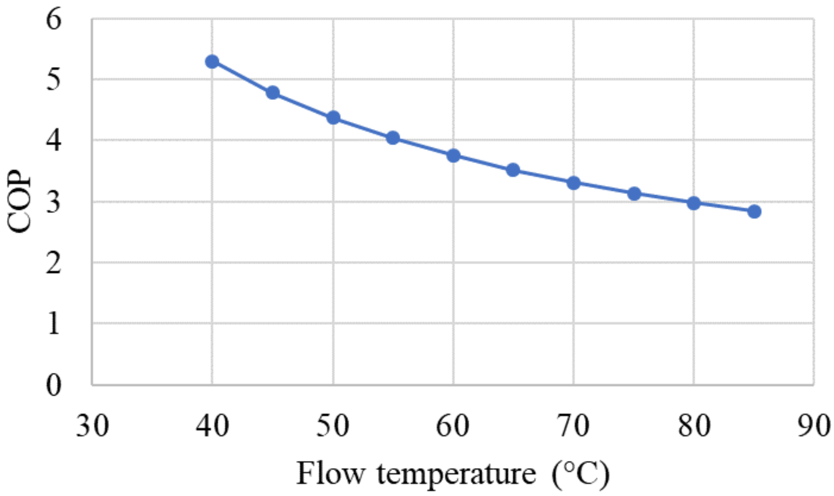


Figure 6: Theoretical Heat Pump CoPs for 10 °C source temperature, at 2/3 efficiency, with 5 °C temperature difference over the internal heat exchangers.

5.1.3 Combined heat pump and gas generation

This report assumes that the gas generation is used as a heat top up to a heat pump system delivering the base load. This is expected to be the preferred operating strategy for an energy centre utilising both technologies as it allows the energy centre operator greater flexibility to meet key performance targets. This assumes that installed capacity is not a limiting factor.

Under this operational strategy, the energy centre can run at full capacity using heat pumps alone if required. However, if the CoP to reach the higher temperatures is not preferred, the operator is able to output lower temperature (and increase the CoP) using the heat pumps and use gas generation to achieve the higher temperatures when required.

To operate under this strategy, the heat network return should first go through the heat pumps, the output of which should then go through the boiler. The boiler can be bypassed should higher temperatures not be required.

5.2 Optimum generation supply

Future optimisation of generation will depend heavily on the total cost of the fuel. This includes economic, social and environmental costs. Predicting the future relationship of gas to electricity is outside the scope of this report. The following section analyses a range of scenarios and the impact on generation at key points to allow for informed decision making given appropriate forecasting of the aforementioned costs.

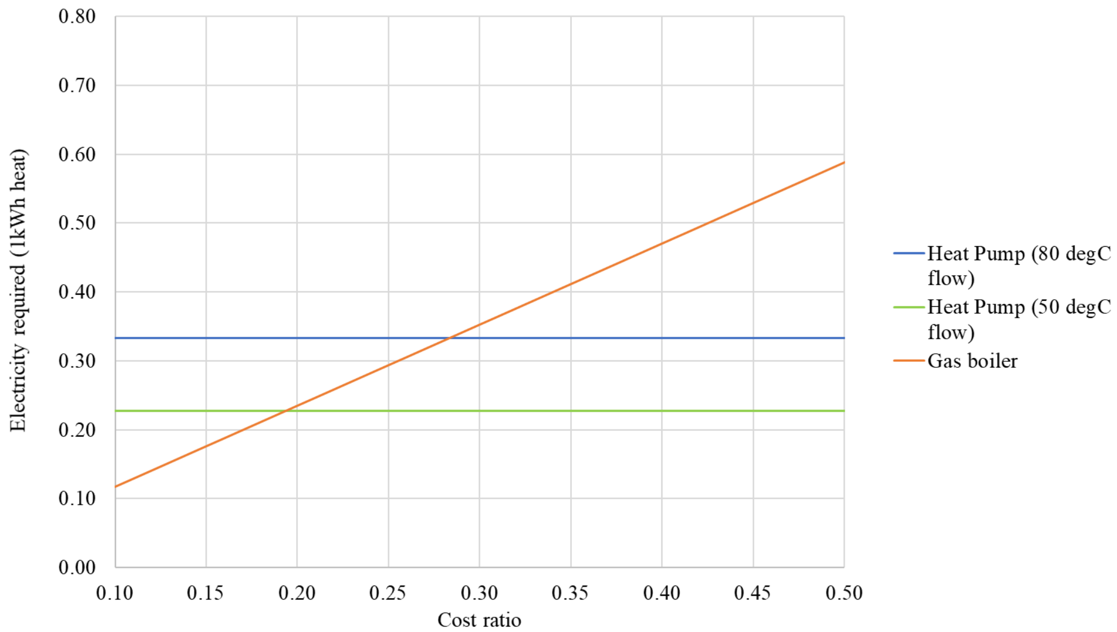
5.2.1 Preferred fuel switch point

The following analysis compares the electricity required to generate 1 kWh of heat using:

- A heat pump with 2/3 efficiency (from a 10 °C source with 5 °C temperature difference over the internal heat exchangers)
- Gas boiler 85% conversion efficiency

The cost ratio is defined as:

$$Cost\ ratio = \frac{Gas}{Electricity}$$



The lowest electricity required line shows the preferential heat supply technology for a cost ratio. Less than a 0.19 ratio, a gas boiler is preferential over either heat pump supply temperature. Above 0.19 a 50 °C flow heat pump is the preferred technology. Above a 0.28 ratio, either heat pump is preferred over a gas boiler.

For a variable supply temperature, the switch point between gas boiler and heat pump would be dependent on the supply temperature, but would vary between the upper limit shown above (0.28 at 80 °C flow), and the lower limit of the return temperature (0.12 at 25 °C flow, not shown on graph).

This shows that in the future, gas will need to be less than 1/5 the total cost of electricity to be a cost-effective fuel against heat pumps under all scenarios.

The analysis can be extended to consider CCGT, in which gas is used to produce electricity. The analysis can therefore be used to determine the minimum efficiency of a CCGT used with these heat pumps to produce heat at a lower cost of gas than an 85% efficient gas boiler. For 80 °C flow heat pump, a 29% efficient CCGT is the minimum required. For a 50 °C flow heat pump, a 19% efficient CCGT is required. Current EU guidelines give CCGT efficiencies of 47.7% (gross). This means that it is always preferable to use CCGT electricity with heat pumps rather than gas boilers.

5.2.2 Impact on generation temperatures

Analysis has been performed on the heat pump flow temperature upper limit to show its impact on the annual heat output from each technology. Because of the gas generation temperature top-up operational strategy described previously, the gas generation will provide additional energy to top up the heat from the heat pump flow output to the required network flow temperature. By altering the heat pump output temperature, this affects both the heat pump CoP, described previously, and the amount of gas required to supply the network. This analysis can be used to indicate if annual proportions of supply to the network from each technology are valid, given expected cost ratios.

1. Four temperature scenarios as in the pipe sizing section.
2. 2,000 MWh/annum supplied over 1 km using the optimum sized pipes for each temperature scenario.
3. Weather data to capture varying demand.

The temperature of the heat pump output which gives 0%, 50% and 100% gas generation has been determined. This can be equated to a minimum cost ratio which would make this economically preferential for the heat pump. For example, the impact of this is that if the network requires 100% heat pump generation, for 80 °C flow, the cost of electricity to gas must be a 1 to 0.28 ratio. Whereas if only 50% of generation via heat pump is required, a 51 °C setpoint will achieve this, for which a 0.19 cost ratio is viable.

This is shown visually for the 80 °C /50 °C variable flow, 60 °C return scenario in the subsequent Figure.

Table 6 Cost ratio switch point for given gas generation

	0% gas generation		**50% gas generation**		**100% gas generation**	
Temperature scenario	**Max output temperature of heat pump**	**Cost ratio**	**Max output temperature of heat pump**	**Cost ratio**	**Max output temperature of heat pump**	**Cost ratio**
80 °C /50 °C variable flow, 60 °C return	80 °C	0.28	51 °C	0.19	< 25 °C	0.10
80 °C flow, 60 °C return	80 °C	0.28	60 °C	0.23	< 25 °C	0.10
80 °C flow, 40 °C return	80 °C	0.28	55 °C	0.21	< 25 °C	0.10
50 °C flow, 40 °C return	50 °C	0.19	41 °C	0.16	< 25 °C	0.10

Note that as 25 °C is the lowest return temperature from the network.

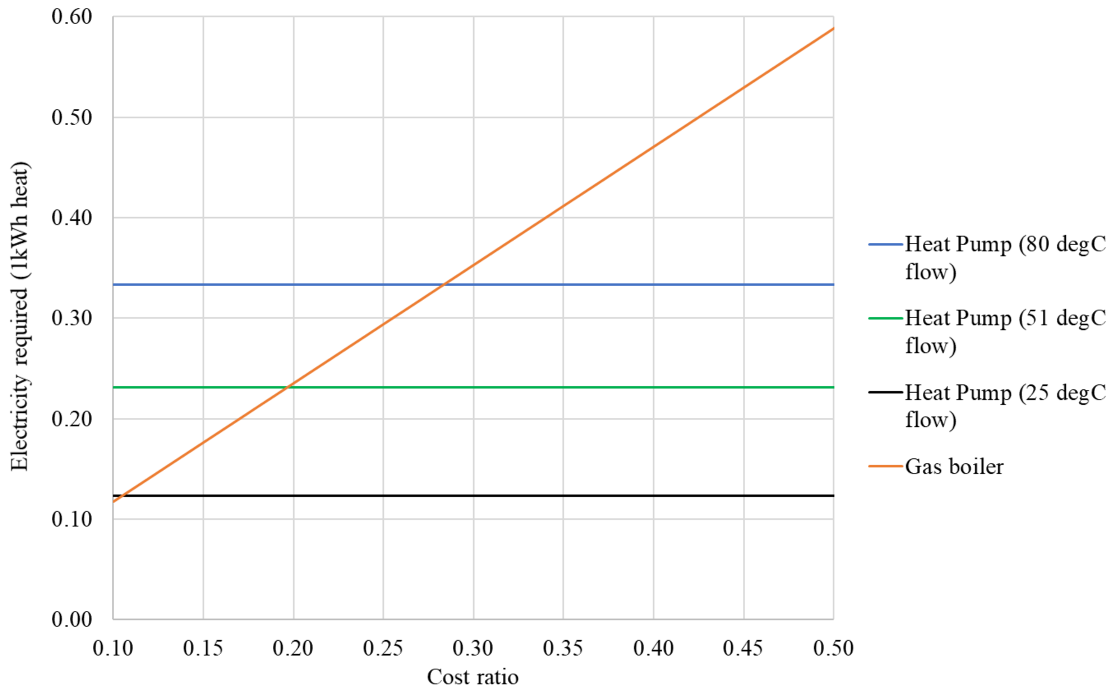


Figure 7 Visualisation of cost ratio for given gas generation (80v/40)

6 Bury

6.1 Heat network

The data provided covered an area across Bury including more than 18,000 buildings identified as having potential for some form of heat network connection. Developing such a large network is unfeasible given the number of connections, and has been determined as unrequired to provide a study the impact of switching from a gas-powered heat network to an electricity-powered heat network.

Network routing, optimum pipe sizing for energy loss analysis has been performed for a smaller subset of connections which are considered representative of the bigger scheme. This has been done focusing on a smaller network cluster, with a high linear heat density to highlighted the effect of the changing pipes while using only a small number of buildings. This approach has been assessed to be the most time-efficient approach to provide a solution. The heat network has been assumed to be representative of the impact of switching from gas to electricity given the magnitude of the heat demand.

GIS has been used to filter buildings with the highest heat demand to determine a cluster around Bury town centre. 12 buildings each with an annual heat demand above 150 MWh/year have been selected and are shown in black in the Bury map below.

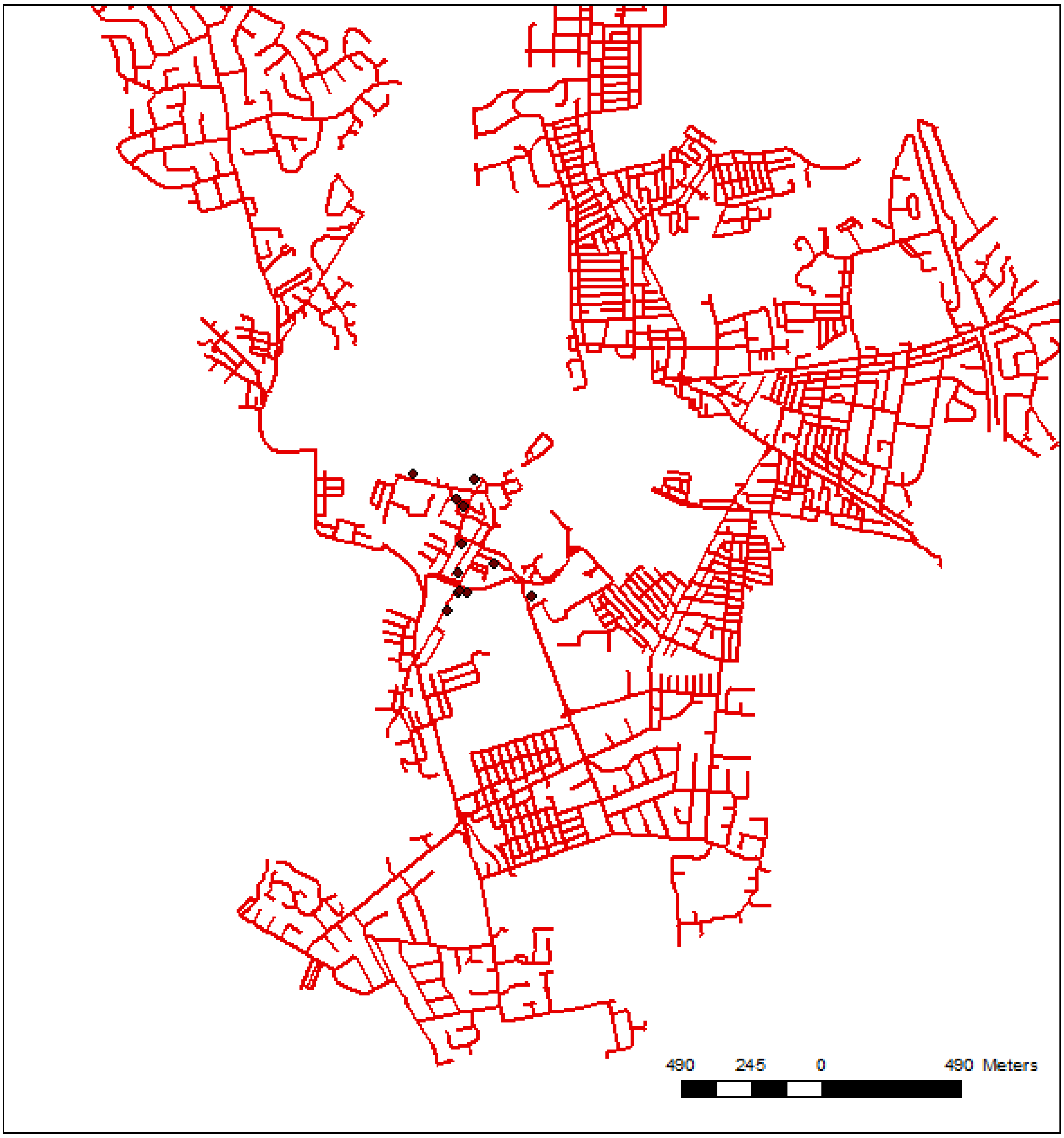


Figure 8 Selected Buildings

An arbitrary energy centre location has been selected for modelling purposes only, in land close to the buildings to be connected. NetSim GSS has been used to draw the following heat network that has been considered for pipe sizing and heat loss analysis.

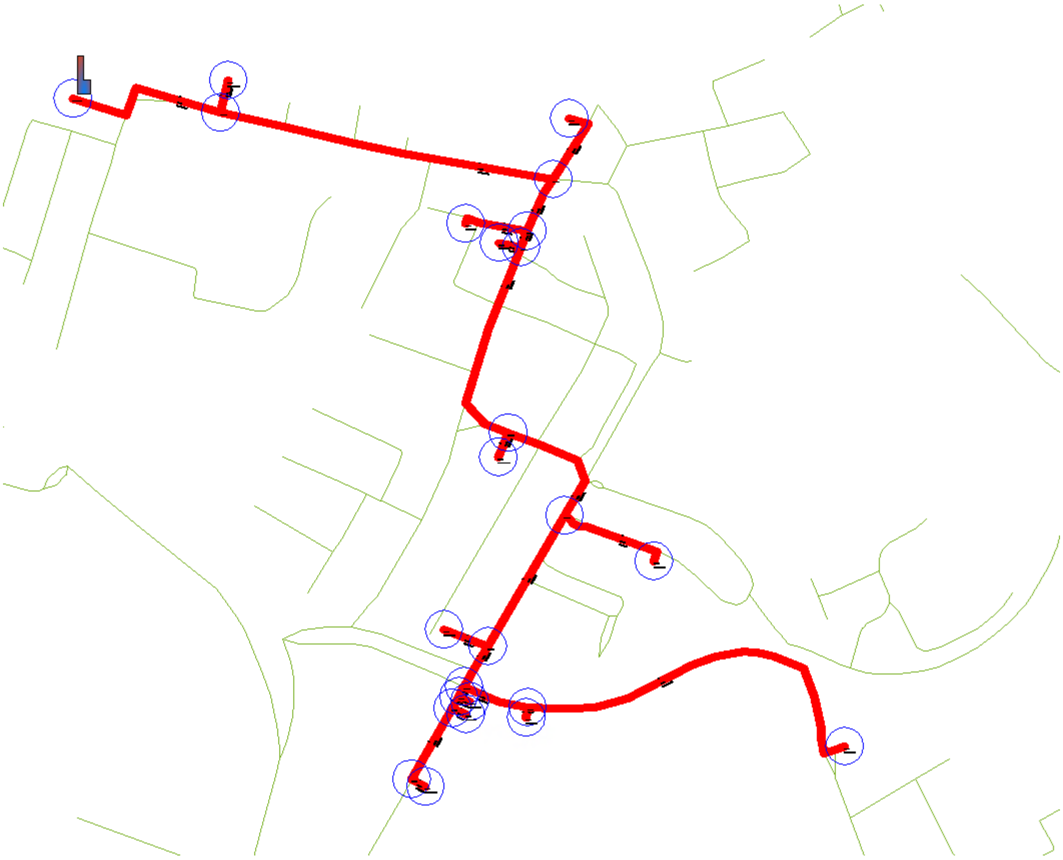


Figure 9 Selected Heat Network

6.1.1 Pipe sizing method

To determine the optimum pipe sizes across a network, a pressure drop (Pa/m) sizing criteria is used. By gradually increasing the maximum allowable pressure drop in the network this allows higher mass flow rates within a given pipe diameter. This allows smaller pipes to be used to deliver the same energy.

As discussed previously, the optimum occurs at the point where the heat losses and pumping energy are at a combined minimum. Hence for a given energy supply requirement, increasing the pressure drop reduces the size of pipe required which reduces the heat loss but increases the pumping required.

This method has been used for the Bury networks for each of the four viable temperature scenarios. From this, an optimum pipe size is found for each section of pipe which can be compared to the other temperature scenarios. This is then used to quantify the impact of operating at non-design temperatures.

6.1.2 Pipe sizing results

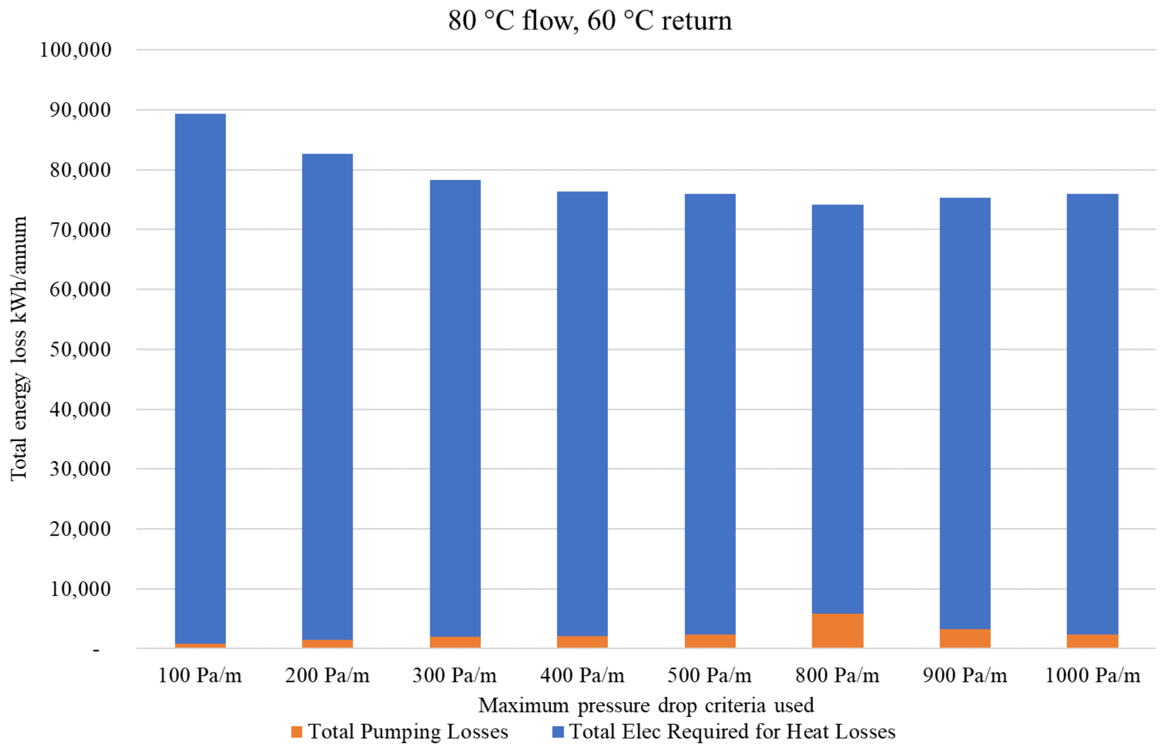


Figure 10 80/60 commercial pipe sizing results

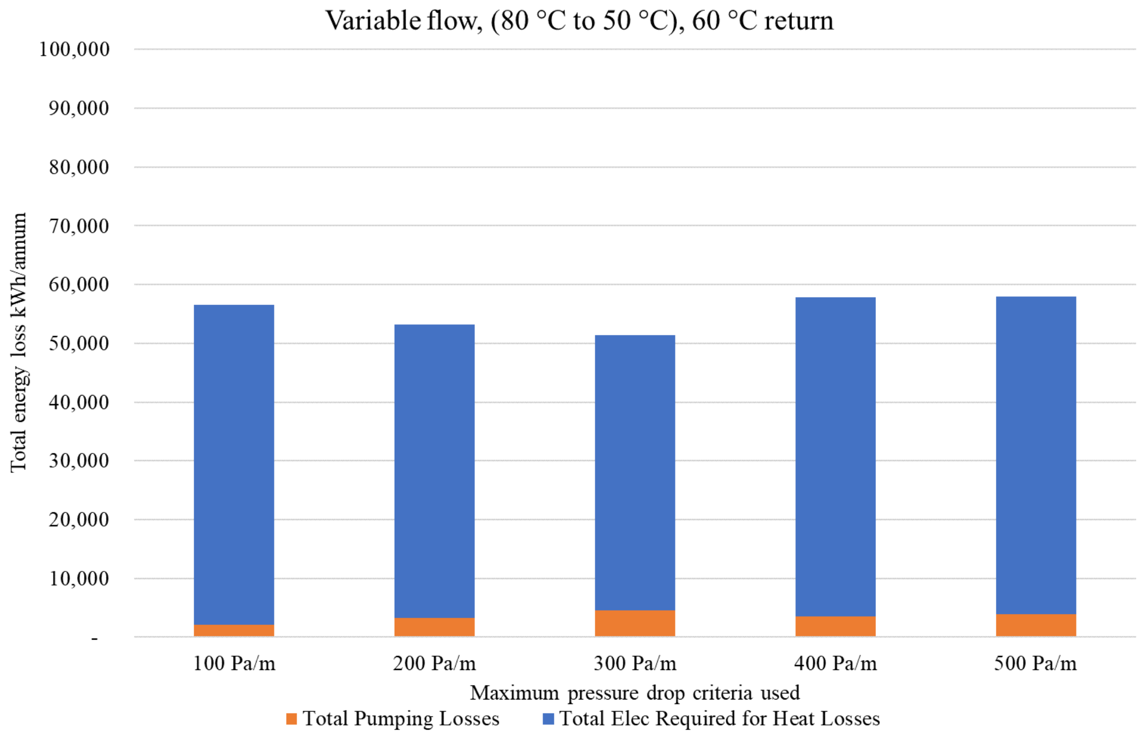


Figure 11 80v/60 commercial pipe sizing results

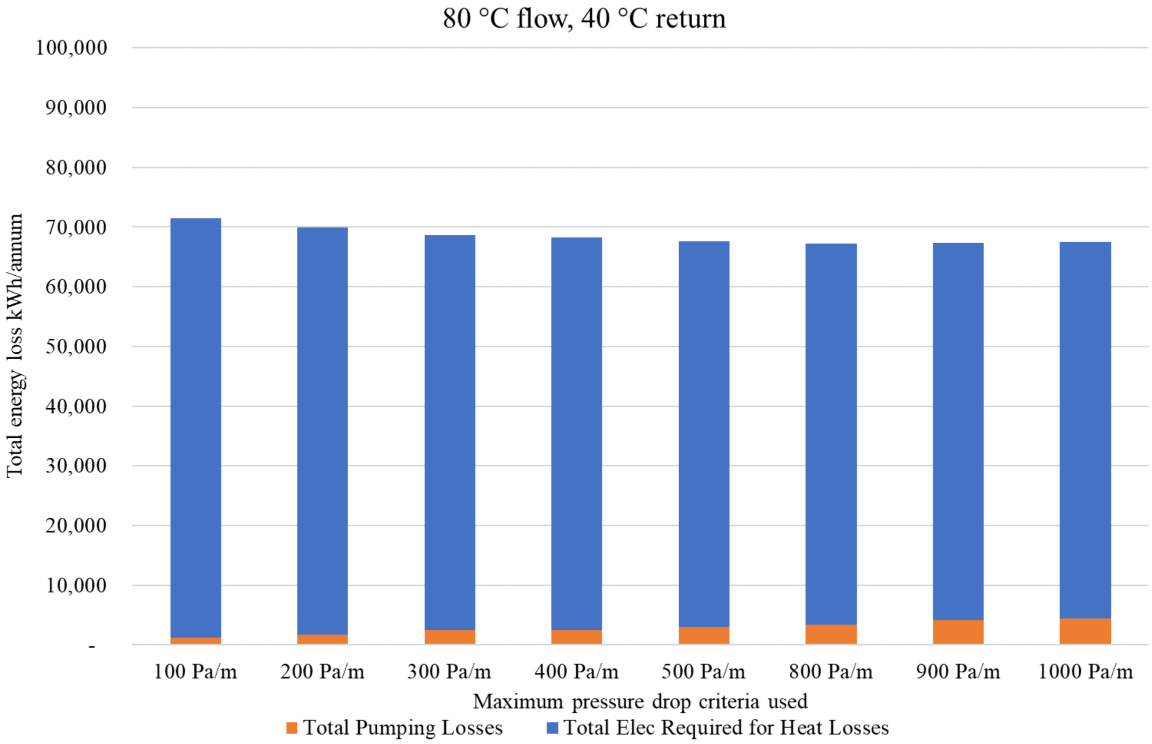


Figure 12 80/40 commercial pipe sizing results

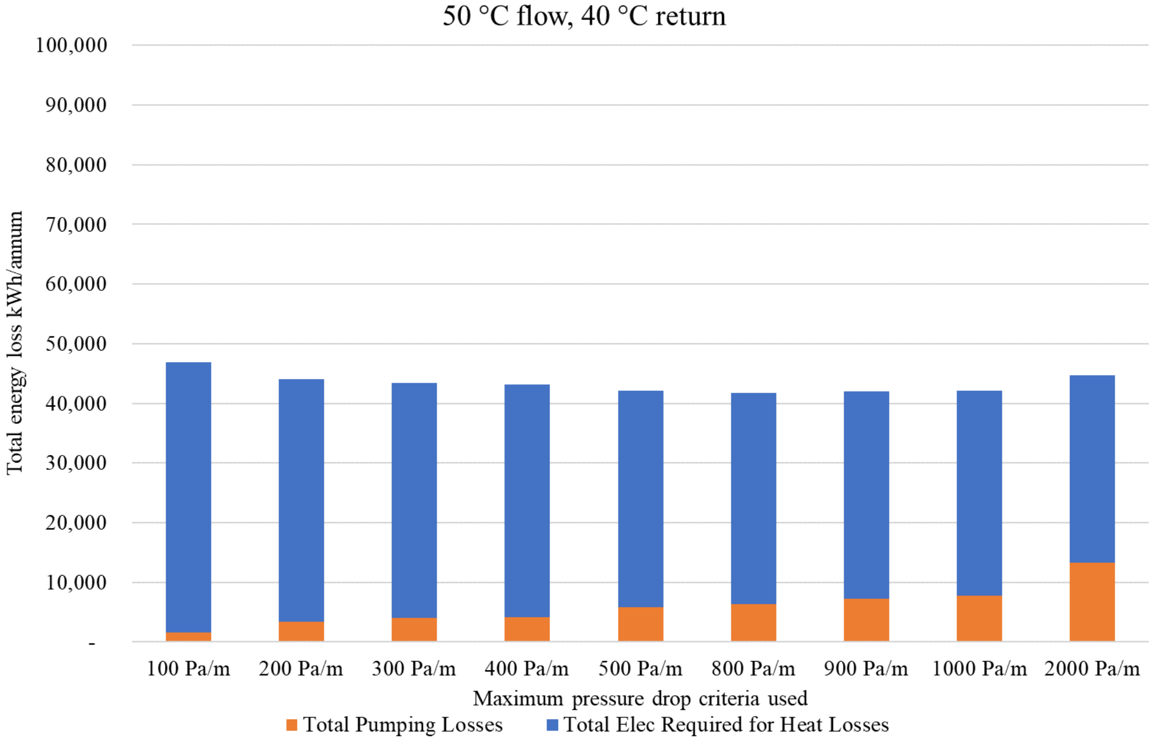


Figure 13 50/40 commercial pipe sizing results

Table 7 Commercial pipe sizing optimum results

Temperature scenario	**Pipe sizing criteria**	**Total energy loss**
80/60	800 Pa/m	74.1 MWh
80v/60	300 Pa/m	51.4 MWh
80/40	800 Pa/m	67.2 MWh
50/40	800 Pa/m	41.7 MWh

The pipe sizing optimum results show the same trend as the theoretically 1 km pipe (2,000 MWh) study presented earlier. The 50/40 network provides the lowest energy loss, with the 80v/60 providing comparatively low energy losses.

The decreasing heat loss and increasing pumping energy as the Pa/m criteria increases can be seen in the Figures 10 to 13 in line with the expected trends.

The results of the pipe sizing have variation in optimum diameters between the different design operating scenarios. The 50/40 and 80v/60 pipe diameters are all larger than the 80/60 and 80/40 pipes. Many of the pipes in the 80/60 and 80/40 optimum set are the same diameter, with some in the 80/40 set being one size smaller diameter than in the 80/60 set. The 50/40 and 80v/60 sets have the same diameter for many of the pipes, with some in the 50/40 set being one size larger.

6.1.3 Network future proofing

To determine the most future proofed pipe sizing, the optimum pipes for each temperature scenario have been used in the other scenarios. This allows the energy loss to be calculated for pipes operating outside of design conditions.

Note that the demand of all scenarios is 2,521 MWh/a of heat to the same connections.

Table 8 Total energy loss for operation at non-design temperatures

		Design temperature			
		80/60	**80/40**	**50/40**	**80v/60**
Operation temperature	**80/60**	74.1 MWh/a	74.4 MWh/a	83.9 MWh/a	78.3 MWh/a
	80/40	67.6 MWh/a	67.2 MWh/a	78.9 MWh/a	73.4 MWh/a
	50/40	66.0 MWh/a	71.3 MWh/a	41.7 MWh/a	43.0 MWh/a
	80v/60	56.6 MWh/a	59.1 MWh/a	53.7 MWh/a	51.4 MWh/a

Table 9 Difference in total energy loss compared to design temperature

		Design temperature			
		80/60	**80/40**	**50/40**	**80v/60**
Operation temperature	**80/60**	0 MWh/a	7.2 MWh/a	42.2 MWh/a	26.9 MWh/a
	80/40	-6.5 MWh/a	0 MWh/a	37.2 MWh/a	22 MWh/a
	50/40	-8.1 MWh/a	4.1 MWh/a	0 MWh/a	-8.4 MWh/a
	80v/60	-17.5 MWh/a	-8.1 MWh/a	12 MWh/a	0 MWh/a

The results show that for the 80/60 design temperature network, energy savings can be found by operating this network at any other temperature. These are significant when the operation is changed to 80v/60. This is expected as the maximum flow conditions are the same, however during 80v/60 annual operation the flow temperature is reduced and therefore heat losses are significantly lower.

For the 80/40 designed network, energy savings can be found by operating this network at 80v/60. Again, this is because the maximum flow conditions are suitable for the 80v/60 operation, but heat loss can be reduced at times of low demand. Operating at 80/60 or 50/40 gives minimal additional energy loss over the original design. Practically the operations would be to vary the flow temperature, whilst the return temperatures occur between 40 °C and 25 °C. This would be an even better performance.

For the 50/40 network, operating at any other temperature increases the energy losses. These are significant when operating at 80/60 or 80/40, as these are the operating temperatures at which the flow temperature is consistently higher.

For the 80v/60 network, operating at 50/40 gives lower energy losses. Operating at 80/40 or 80/60 gives significant additional energy losses.

The 50/40 designed network gives the lowest total energy losses (41.7 MWh/a), although the 80v/60 designed network gives comparatively low energy losses (51.4 MWh/a). This means that a network which will give the lowest energy losses in the future, the network should be designed at either 50/40 or 80v/60, depending on the flow temperatures acceptable for the connected buildings.

It is preferable to operate the 80/60 and 80/40 designed networks at variable flow temperatures. It is preferable to operate the 80v/60 and 50/40 designed networks at 50/40. This means that for the lowest energy losses the network should be operated at either 50/40 or with variable flow temperatures.

Therefore, for long term future proofing with regards to operating temperatures, the preferred network design and operating conditions are either 80v/60 and 50/40, dependent on the connected buildings.

If the temperature will change during the lifetime, then the lifetime energy losses, and therefore preferred scenario are dependent on the time operating under each state. If more than 2/3 of the time is spent operating at 50/40, then the network should be designed at 50/40. If less than 2/3 of the lifetime is spent operating at 50/40, then the network should be designed at 80v/40

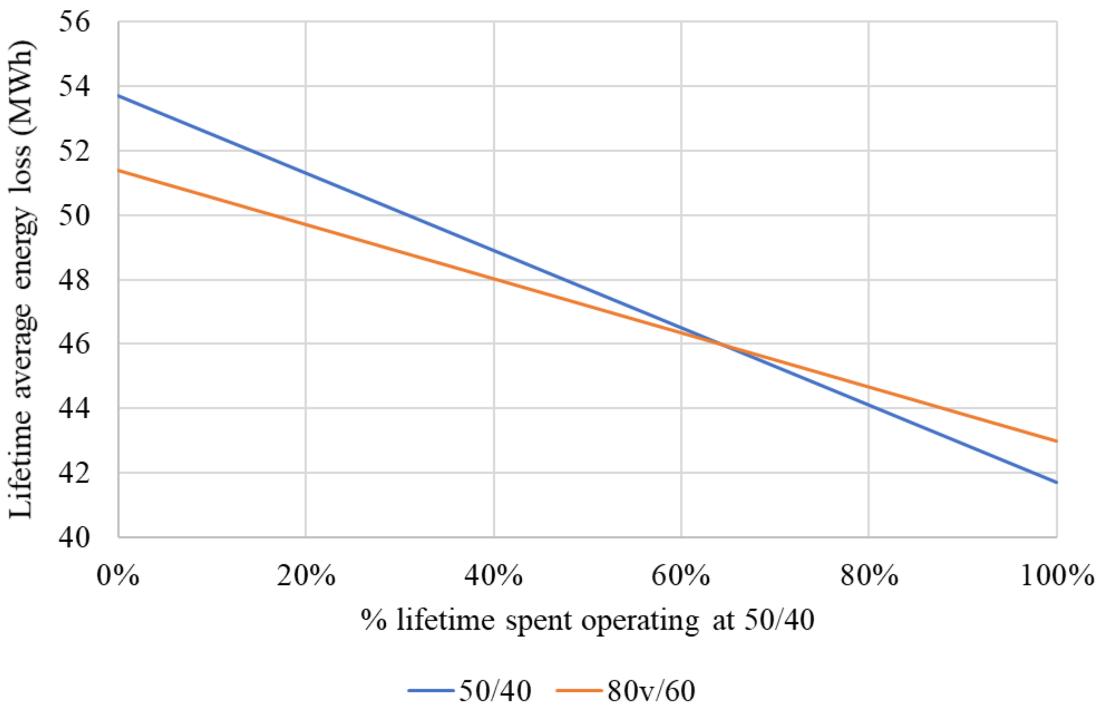


Figure 14 Total energy loss based on proportion of time spent at each operating temperature

6.2 Generation

6.2.1 Seasonal Heat Pump CoP

The seasonal CoP of the heat pump operating under the temperature scenarios for the Bury heat network are shown in the below figure. The CoP shown here is to raise the return to the flow temperature and does not account for any gas boiler ‘top up’, see following section. Note that 80/60 and 80/40 heat pumps have the same CoP of 3.0, due to the same constant flow temperature. The seasonal CoP for the 80v/60 operating condition is a CoP of 3.7. As expected this is between the CoP of 4.4 for 50 °C flow temperature, and CoP of 3.0 for the 80°C flow temperature.

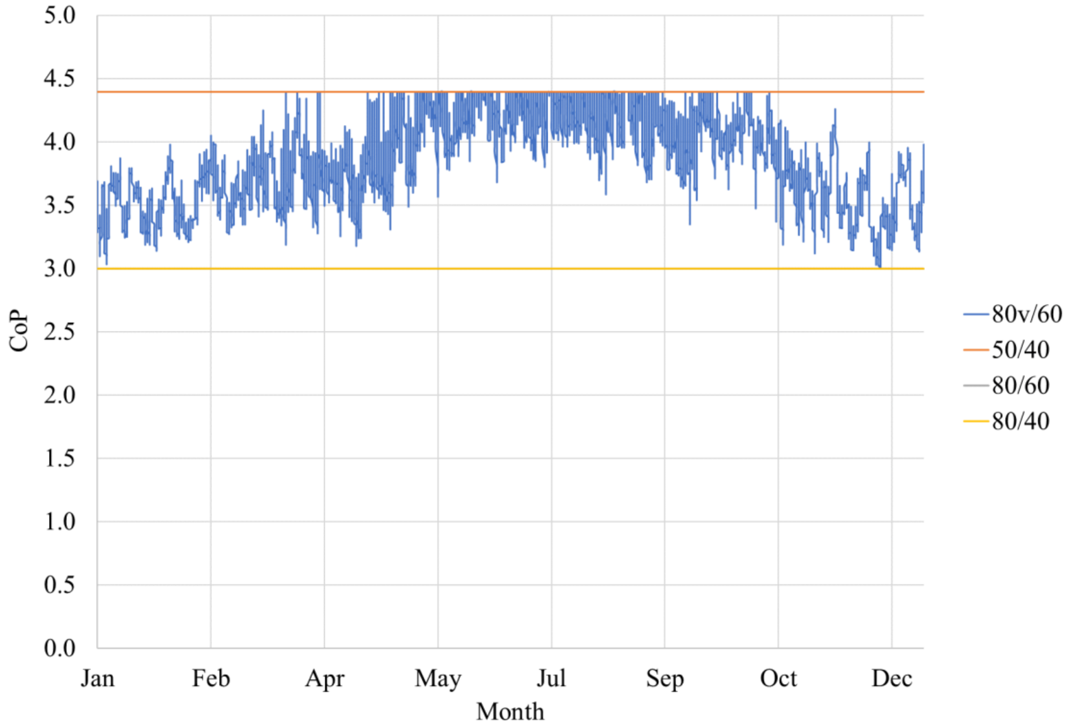


Figure 15 Heat pump CoP

6.2.2 Gas generation

The degree of gas generation for the Bury network is dependent on the switch on maximum output temperature of the heat pump (i.e. the input temperature of the boiler). This is a decision based on heat pump CoP and the cost ratio as described in earlier sections.

The following figure demonstrates the effect changing the switch on temperature has on annual output of the boiler as a proportion of annual network demand. Note that annual demand for all scenarios is 2,521 MWh/a.

This shows that the 50/40 network uses the least gas generation for a given heat pump temperature output. 80v/60 in general uses less gas generation for a given heat pump temperature output than the 80/60 and 80/40 networks.

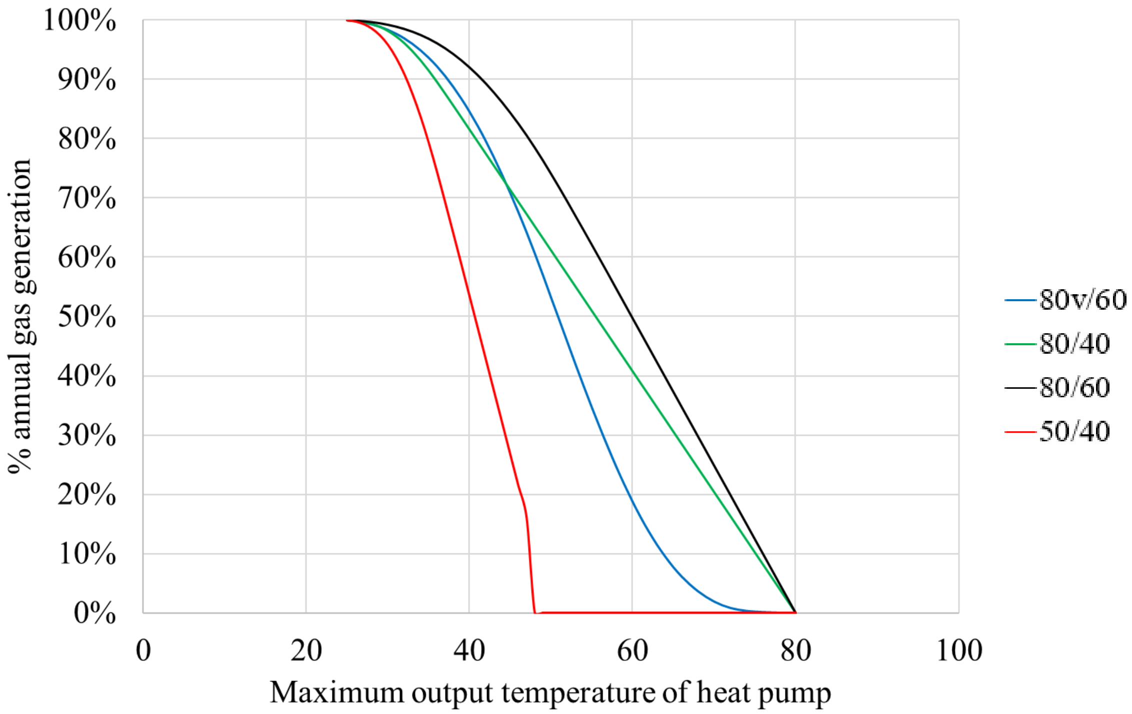


Figure 16 Gas generation based on output temperature of heat pump

7 Conclusion

The report has shown due to building requirements there are the viable temperature limits to network operation. These are between 80 °C and 50 °C flow temperatures, required due to domestic hot water delivery temperatures, and between 60 °C and 40 °C, dependent on the level of alteration to existing building space heating systems.

From the four variations for flow and return temperatures considered viable, a network designed and operated at 50 °C flow and 40 °C return gives the lowest annual energy losses. This network also requires the lowest annual energy input to provide the heat as the heat pumps have the highest Coefficient of Performance (CoP 4.4) at the flow temperatures. This means that the opportunity for low carbon electricity to supply the network is higher, as the cost of gas compared to electricity must be much lower to be viable. Hence, in terms of operation and capital costs, this network design and operating temperatures is the lowest cost option.

The key issue with implementing a 50/40 network is the extensive retrofitting to the building heating systems will be required. This is likely to be significant and far outweigh the cost savings attributed to the construction and operation of the network.

This network may be operated under non-design temperature conditions. If this is to be done, the using an 80 °C to 50 °C variable flow and 60 °C return gives approximately 0.5% additional energy loss per annum over the design operating conditions. This could be used to delay the immediate need to retrofit a large number of buildings.

The comparison between operating a 50/40 network at 80v/60, and an 80v/60 network at 50/40, depends on the amount of time at the non-design conditions. If the network is to be run for over 2/3 of its lifetime at 50/40, then the analysis has shown that lower lifetime energy losses can be achieved through the 50/40 design conditions.

A network designed and operated at 80 °C to 50 °C variable flow and 60 °C return gives the comparatively low energy losses. To operate at these return temperatures, only minimal retrofitting to the building heating systems will be required. The scenario utilises the lower supply temperatures, providing a variable Coefficient of Performance which improves the CoP of the heat pump. This option would achieve a seasonal heat pump CoP of 3.7. This means less energy is needed supplied this network when compared to the constant flow 80 °C networks (with CoP 3.0).

Additionally, operating this network at a lower temperature (50 °C flow and 40 °C return) will reduce the energy losses across the network, although require the full retrofitting of buildings.

As the 80v/60 network would require similar capital costs when compared to the 50/40 network, the decision on which provides the lowest long term costs is dependent on the additional energy savings achievable for the 50/40 network (through minimising heat loss and more efficient generation), and the costs associated with retrofitting the buildings.

If a network has already been installed, and has been designed at 80/60 or 80/40, it can be more efficiently run with a variable flow temperature.