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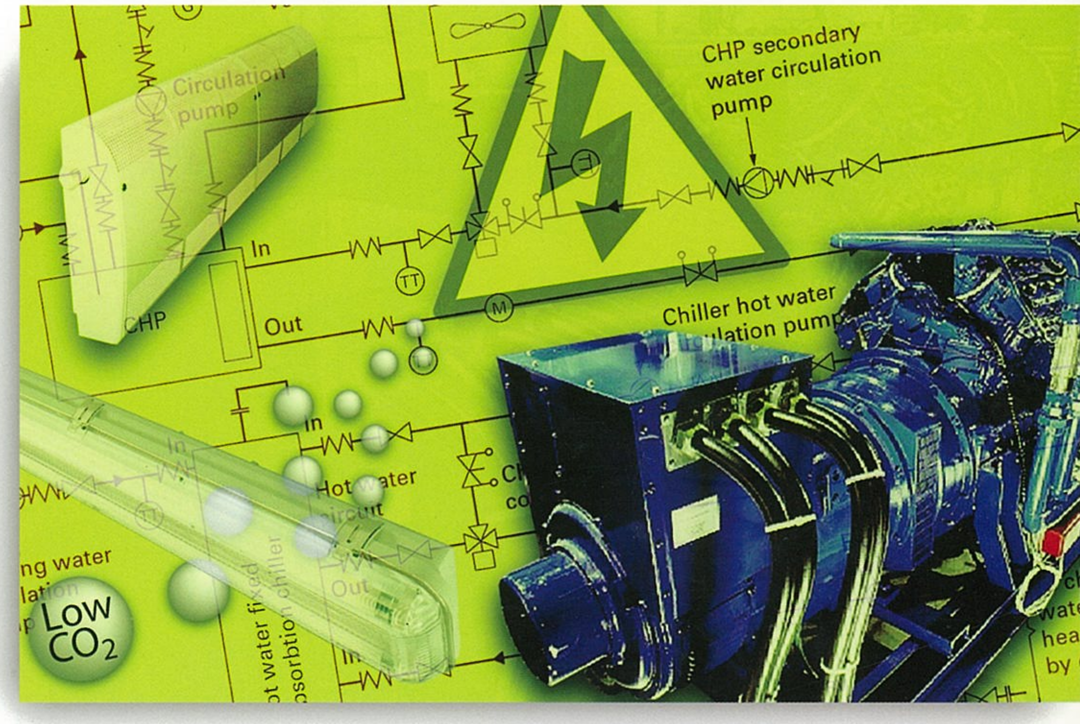
CHP for Existing Buildings



Guidance on design and installation

By Arnold Teekaram, Anu Palmer and James Parker

BG 2/2007

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SYMBOLS

IV	Isolating valve	3PMV	Three port motorised valve
DRV	Double regulating valve	2PMV	Two port motorised valve
OP	Orifice plate	P	Pressure gauge
VOCS	Variable orifice commissioning station	STR	Strainer
FORV	Fixed orifice regulating valve	NRV	Non-return valve
–	Pump	FC	Flexible couplng
FOCS	Fixed orifice commissioning station	TP	Test point
4PMV	Four port motorised valve	DPCV	Differential pressure valve
CFC	Constant flow controller	CV	Combination valve
DOC	Drain off cock	MAV	Manual air vent
TRV	Angled thermostatic valve		

Introduction

The main purpose of this section is to make designers aware of the issues which need to be addressed when considering their own designs for the installation of CHP. It is not to provide the designer with a design or circuit that could be installed.

Characteristics of common heating systems

To maximise heat recovery from CHP, attention should be given to the design of the heating system it serves as well as its interface with other heat-producing appliances. Designers should not merely consider CHP to be a lead boiler, as CHP and its heat rejection circuits have quite different operational characteristics to a boiler.

Problems associated with CHP often arise due to the lack of understanding by designers in relationship to the part load operation of the CHP, matching of the CHP to the part load performance to heating systems and constraints on return and flow temperatures for heat recovery from the CHP unit's different heat rejection circuits.

The characteristics of the most common heating systems are illustrated in Figure 2. These are:

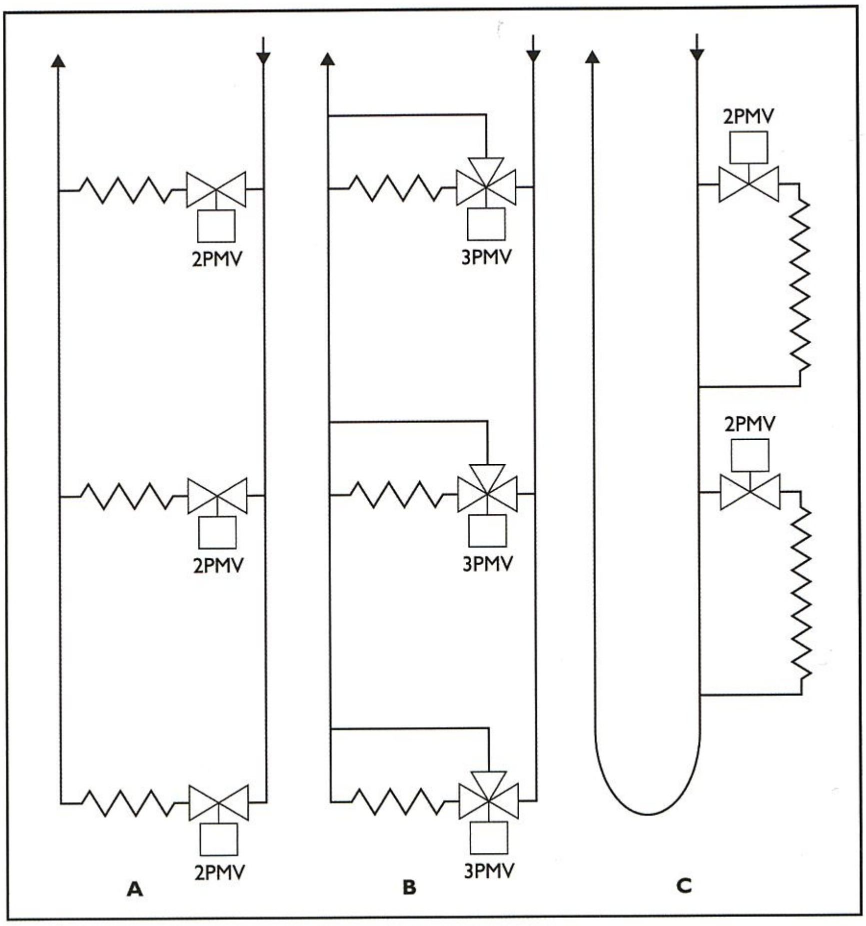
- One-pipe systems
- two-pipe systems using three-way valve control
- two-pipe systems using two-way valve control.

Several systems can be a mixture of two-way and three-way valve controls. Typically, two-way control is used on radiators while three-way valve control is used on air handling units and calorifiers.

When a CHP unit is fed with return water from a one-pipe system or two-pipe systems using three-way valve control, the effect is similar to a heating system with a constant flow temperature running under part load: return temperature rises as the load reduces.

The effect on the CHP unit of circuit A (the two-port fully modulating valve control) is to provide falling return temperatures on part load and lower flow rates.

Figure 2: Examples of the most common types of heating systems.



2 PMV - two port modulating valve
3 PMV - three port modulating valve.

A - two-pipe system with a two-way valve
B - two-pipe system with a three-way valve
C - one pipe system with two-way valves on heat emitters.

Table 2 illustrates the return temperature from a modulated three-way valve controlled heat emitter (system B) and from the modulating two-way valve emitter system (system A).

Table 2: Effect of two-way and three-way valve control.

	For flow temperature of 80°C	**Heat emitter load**				
	Load	100%	75%	50%	25%	0%
Three- Way	Return temperature full flow three-way	60°C	65°C	70°C	75°C	80°C
Three- Way	Flow (three-way control)	100%	100%	100%	100%	100%
Two-Way	With a variable-flow system using a two-way modulating valve, the return temperature will be approximate depending on the emitter	60°C	45°C	32°C	25°C	20°C
Two-Way	Flow (two-way modulating control)	100%	42·86%	20·83%	9·09%	0%

Under a zero load situation when domestic hot water calorifiers use three-way valve control and the heat demand is satisfied, the three-way valve diverts the flow directly to the return and the return water temperature rises to the flow temperature. This can affect the operation of a CHP unit that has been engineered to shut down or operate its dump cooling circuit when a return temperature is higher than that anticipated during design.

If a CHP unit is to provide its full output to a circuit using three-way valves then the designer should consider how (if the CHP has a constant flow through it and a constant temperature rise) the CHP will raise its flow temperature as the return temperature rises so as not to reduce the CHP's heat output. The designer should also consider whether the raised return temperature will affect the quantity and temperature of heat picked up from circuits, such as the intercooler or oil cooler.

Three-port valves normally cause sub-optimum CHP operation and should be avoided in heat distribution circuits wherever possible. The control system for heat emitters most suited to either condensing boilers or CHP (on account of the falling return temperature characteristic on part load) is a fully modulating, two-port valve control system.

For an example of a fully variable-flow two-way valve heating circuit design, see the Bodle Orchard circuit described in the BSRIA application guide AG 16/2002: *Variable Flow Water Systems-Design, Installation and Commissioning Guidance.*

Designers should also consider how to maximise the heat picked up from the CHP unit on part load using the lower return water temperatures while maintaining the required flow temperature from the CHP unit. This may be used to signal sequencing of other heat input devices. The solution is to vary the flow rate through the engine's heat exchange circuits, but this needs care to ensure that other functions are not impaired, such as oil cooling. (This solution is being considered by CHP package manufacturers.)

Many engines use engine-driven pumps for their normal jacket cooling system. As pump speed will vary with engine speed the flow rate around the jacket will fluctuate. Little attention may have been given to this aspect of CHP engine design, as only power, and not power plus heat are usually the basis of most discounted energy purchase contracts.

Most engines used for CHP are adapted from engines designed for other purposes where the heat is not used. The main objective is to minimise the cost of the heat rejection circuits and protect the engine.

Designers should note that some package suppliers of CHP are reluctant to consider variable flow.

3.1 HEAT REJECTION FROM CHP

Most packaged CHP units are designed on the basis of a constant flow rate. Depending on the quality of maintenance, the heat output from the various heat rejection circuits will change. Output will also fluctuate depending on the power load being met by the CHP unit.

Table 3 indicates how the performance and heat rejection for a CHP unit varies with the load. This table is based on net or lower calorific values of the energy in the fuel.

A higher overall efficiency value is given when using the net calorific value of the fuel in the efficiency calculations. Care should be taken when comparing efficiencies to ensure that the comparison is like-for-like, for example, that both calculations are based on net gross calorific value.

Depending on its source and composition, natural gas has different calorific values. North Sea natural gas typically has a net calorific value of 34·82 MJ/m^3 and a gross calorific value 38·62 MJ/m^3.

The ratio of the net over gross is therefore 0·902 and the gross over net 1·109. A power plant with a quoted efficiency of 50 percent based on the net calorific value would therefore have an efficiency of 45·08 percent based on the gross calorific value.

It is recommended that all efficiencies be based on the gross or higher calorific value of the fuel, whether for boilers or power.

For a CHP unit running on full load, the heat outputs will change as the engine performance changes. Efficiency will also drop between servicing and tuning.

Usual arrangements by package suppliers are to maintain the return temperatures to a low enough value to recover heat from the oil cooler and other circuits recovering lower grade heat from the cooling water. This is with a constant flow-rate, and with output and engine protection temperature set to specific values. A radiator or air-cooler is used to discharge heat to atmosphere if the return temperature is too high.

A common problem is CHP units tripping out on high jacket temperature where there is no provision for additional cooling of the return water.

Rejecting heat to atmosphere is undesirable but may be necessary in some circumstances to keep the CHP operating at low load.

Table 3: An example of effect on CHP performance and heat rejection as load varies.

Speed	**1500 rev/min**	Brake mean effective pressure (BMEP) at maximum continuous rating	**17·70 bar**
Fuel Gas	**Natural gas**	Compression ratio	**11·8**
NOx-emission	**500 mg/m³**	Co-emission (approximate value)	**950 mg/m³**
Jacket water outlet maximum	**90°C**	Intercooler water temperature maximum	**40°C**
Minimum methane number	**70**	Exhaust gas manifold	**Un-cooled**
Maximum oil temperature	**0°C**	Intercooler flow rate; low-temperature	**0 m³/h**
Return temperature high-temperature	**0°C**	Intercooler flow rate; high-temperature	**0 m³/h**
Standard rating conditions and tolerances. See general specification; volume values at normal conditions; exhaust flow at silencer; Exhaust pressure (Pe) = Standard performance for constant level speed of rotation and constant level of loads (ICFN) *(ISO 3046/I)*			

Energy balance					
Engine load	%	100	75	25	0·00
Engine rating	kW	646	485	162	0
BMEP	bar	17·70	13·28	4·43	0·00
Heat consumption	kWh/kWh	2·43	2·49	3·03	0
Energy balance absolute					
Input	kWh	1568	1208	489	130
Mechanical	kWh	646	485	162	0
Jacket water	kWh	196	189	112	51
Oil-cooler	kWh	67	55	39	22
Exhaust gas total	kWh	479	373	141	34
Exhaust gas 180°C	*kWh*	*333*	*262*	*101*	*25*
Exhaust gas 120°C	*kWh*	*390*	*305*	*116*	*28*
Exhaust gas 100°C	*kWh*	*409*	*320*	*121*	*29*
High temperature intercooler	kWh	93	41	0	0
Low temperature intercooler	kWh	32	24	0	0
Surface heat	kWh	20	18	27	19
Balance	kWh	31	23	10	4
Energy balance relative (%)					
Mechanical	%	41·2	40·1	33·0	0·0
Jacket water	%	12·5	15·6	22·8	39·6
Oil-cooler	%	4·3	4·6	8·0	16·8
Exhaust gas total	%	30·6	30·9	28·7	26·0
Exhaust gas 180°C	%	*21·2*	*21·6*	*20·6*	*19·0*
Exhaust gas 120°C	%	*24·9*	*25·2*	*23·8*	*21·8*
Exhaust gas 100°C	%	*26·1*	*26·4*	*24·8*	*22·7*
High temperature intercooler	%	6·0	3·4	0·0	0·1
Low temperature intercooler	%	2·1	2·0	0·0	0·0
Surface heat	%	1·3	1·5	5·5	14·5
Balance	%	2·0	1·9	2·0	3·0
Exhaust gas temperature	°C	485	494	516	534
Fuel/Air ratio	l	1·67	1·65	1·44	1·23
Exhaust gas mass flow rate, wet	kg/h	3358	2558	909	208
Exhaust gas mass flow rate, dry	kg/h	3104	2363	830	187
Exhaust gas volume, wet	m3/h	2656	2024	721	166
Exhaust gas volume, dry	m3/h	2352	1789	626	141
Combustion air mass flow rate	kg/h	3245	2471	874	199
Combustion air volume	m3/h	2510	1912	676	154

Supplied by Clarke Energy and used with permission of Jenbacher.

3.2 INTERFACING CHP WITH BOILER CIRCUITS

For most boiler circuits, the issue of higher return temperatures is not a problem as many arrangements include circuits to maintain a minimum return temperature to the boilers.

The control of the boilers is simple, with the heat output usually being controlled by a thermostat in the flow from the boiler and the thermostat controlling the input from the burner to meet the required flow temperature. Burners are either modulating, staged high/low or on/off.

If a CHP unit is treated as a lead boiler, this would mean engineering the boiler control so that the CHP engine runs to full load with the other boilers not being allowed to put heat into the system until the CHP unit is unable to meet the heat load. This approach allows the CHP unit to run for a longer period at full power, thus providing the best efficiency.

A heat-raising system based on a set of boiler modules will usually be run to balance the hours-run for each boiler module. Each boiler will get the chance to act as the lead boiler. If this method of control is applied to a heat-raising system that includes a CHP unit, the CHP engine will not run for as many hours as anticipated during design. This would reduce the contribution of the CHP unit, and therefore reduce the potential energy efficiency of the heating system.

It is therefore important to consider how the CHP unit is connected to the boilers and how that connection is controlled.

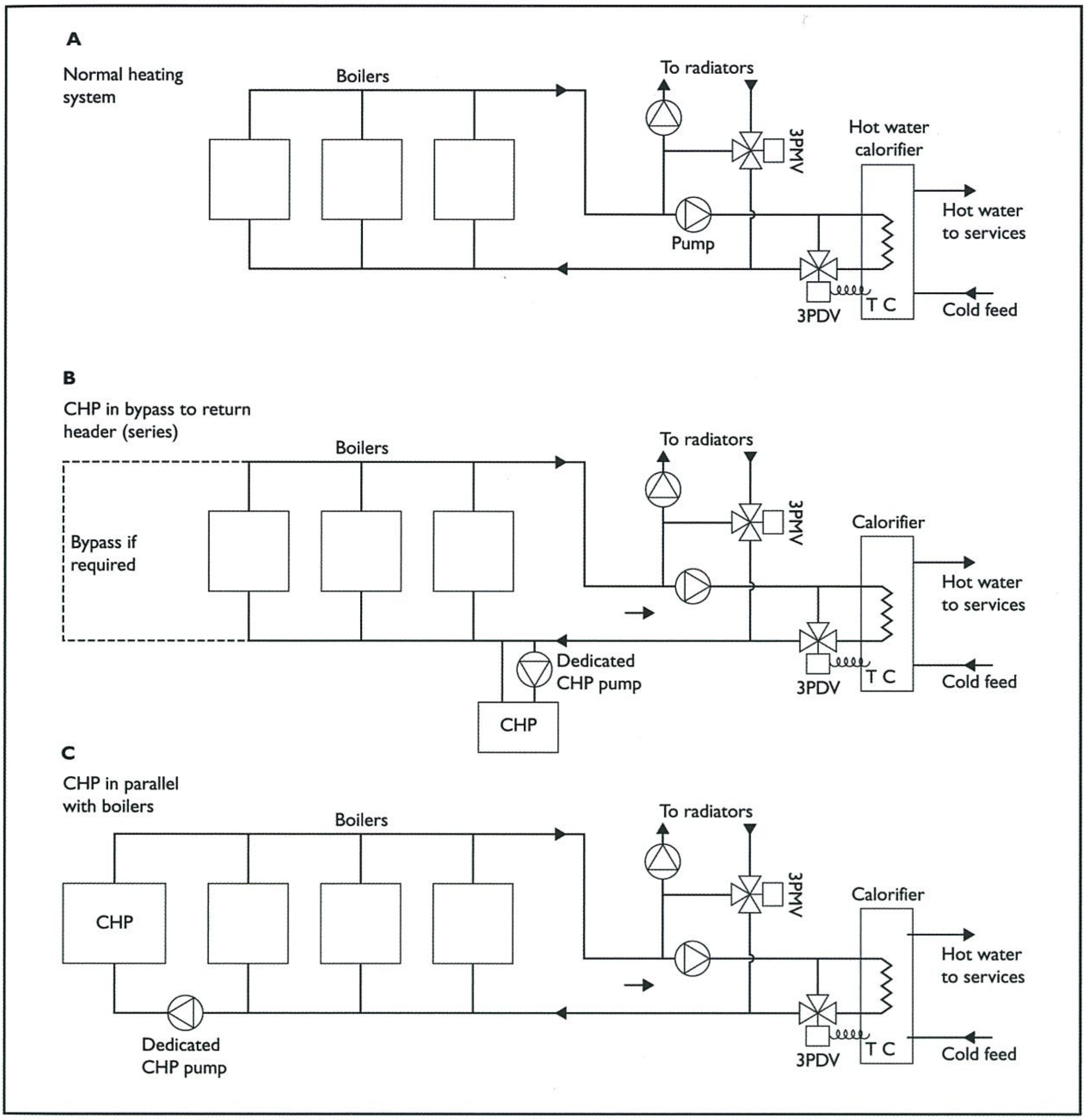
Where a CHP system is require to pre-heat water with a constant return temperature, much simpler, connection arrangements and controls are likely to be required than where the heat load demands modulate and return temperatures vary.

It is essential for a CHP unit to take the heat load at all times in preference to the boilers if carbon savings from the technology are to be maximised.

Examples of heating system designs likely to cause a high return temperature problem are shown in Figure 3.

Designers should consider the suitability of the circuits outlined in Figure 3 and how they can be controlled to ensure the CHP unit across the lead heat-raising device.

Figure 3: Examples of heating system connections deemed unsuitable for CHP.



3.3 THE EFFECT OF MIXING ON FLOW TEMPERATURE SETTINGS FOR BOILERS AND CHP

Circuit B in Figure 3 shows a common arrangement of a CHP unit acting as a return water pre-heater.

Designers should consider the flow rate in the return and through the CHP in determining how the CHP will work with the boilers and their flow temperature settings.

Figure 4 illustrates a boiler circuit where a CHP unit is connected as in Figure 3, diagram B.

Figure 4: Boiler circuit to illustrate the effect of mixing on flow temperatures.

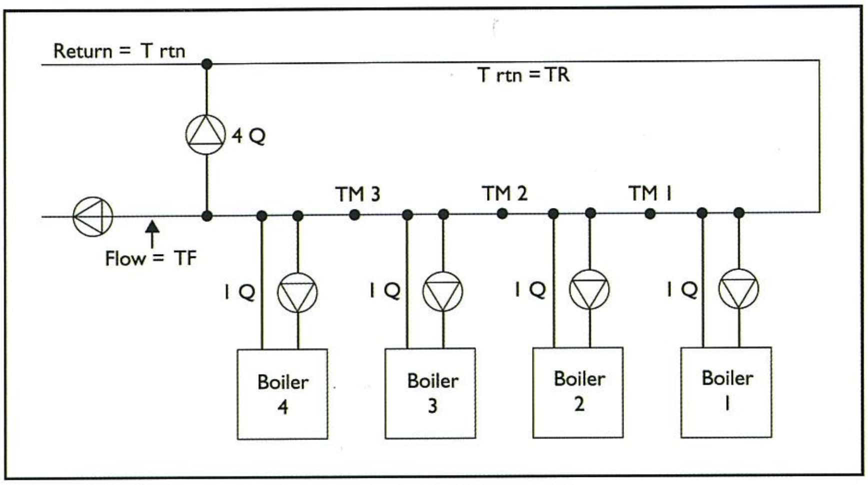


Table 4 and Table 5 consider the settings required for thermostats on boilers in order to achieve a flow temperature of 82·2°C for the heating system for up to nine boilers, as illustrated in Figure 4.

The circuits shown above are not recommended. They are shown so that designers can examine circuits from the literature and think about flow rates and flow and return temperatures.

Many such systems are designed for the boiler thermostats to be set at the same temperature as the required flow temperature. An analysis of the circuit hydraulics shows that if the full installed boiler output is to be achieved, then the thermostats need to be set significantly higher than 82·2°C.

Table 4 calculates the rise in temperature after a boiler has put its heat into the common flow. With only one boiler, the return temperature for the next boiler would have to be 82·2°C to meet the requirement.

From the return temperatures the required flow temperature of the boiler can be calculated to achieve an 82·2°C flow.

Table 4: Calculation of boiler temperature rise.

	Temperature rise after boiler of combined flow °C	Return from system °C	Return for next boiler °C								Required return temperature for each successive boiler °C
Boiler 1	11·10	71·10	82·20								
Boiler 2	5·55	71·10	76·65	82·20							
Boiler 3	3·70	71·10	74·80	78·50	82·20						
Boiler 4	2·78	71·10	73·88	76·65	79·43	82·20					
Boiler 5	2·22	71·10	73·32	75·54	77·76	79·98	82·20				
Boiler 6	1·85	71·10	72·95	74·80	76·65	78·50	80·35	82·20			
Boiler 7	1·59	71·10	72·69	74·27	75·86	77·44	79·03	80·61	82·20		
Boiler 8	1·39	71·10	72·49	73·88	75·26	76·65	78·04	79·43	80·81	82·20	
Boiler 9	1·23	71·10	72·33	73·57	74·80	76·03	77·27	78·50	79·73	80·97	82·20

Table 5: Required flow temperature off the boiler to achieve a flow temp of 82·2°C.

	Temperature rise after boiler of combined flow °C	Return from system °C	Flow temperature setting required °C								
Boiler 1	11·10	71·1	82·20								
Boiler 2	5·55	71·1	82·20	87·75							
Boiler 3	3·70	71·1	82·20	85·90	89·60						
Boiler 4	2·78	71·1	82·20	84·98	87·75	90·53					
Boiler 5	2·22	71·1	82·20	84·42	86·64	88·86	91·08				
Boiler 6	1·85	71·1	82·20	84·05	85·90	87·75	89.60	91·45			
Boiler 7	1·59	71·1	82·20	83·79	85·37	86·96	88·54	90·13	91·71		
Boiler 8	1·39	71·1	82·20	83·59	84·98	86·36	87·75	89·14	90·53	91·91	
Boiler 9	1·23	71·1	82·20	83·43	84·67	85·90	87·13	88·37	89·60	90·83	92·07

Table 5 shows that where a mixed flow circuit is generated by multiple boilers and a CHP unit, the output flow temperatures from successive boilers in the arrangement have to be progressively higher to meet the target flow temperatures of 82·2°C.

A CHP unit shown in configuration B has a heat load that may well be less than one quarter of the total heat load. This would be equivalent to a four-boiler arrangement, illustrated in Table 5. To reach a mixed flow temperature of 82°C the CHP unit would need to be producing a flow temperature of 90·5°C.

In the case of boilers in parallel but with the pumping conditions illustrated in Figure 3 circuit C, mixed-flow conditions need to be considered, as the circuit as drawn shows flow through the boilers at all times.

For circuit C, designers may wish to consider the effect of stopping the flow through some boilers where the CHP pump shown in the diagram is operating in series with the other pumps.

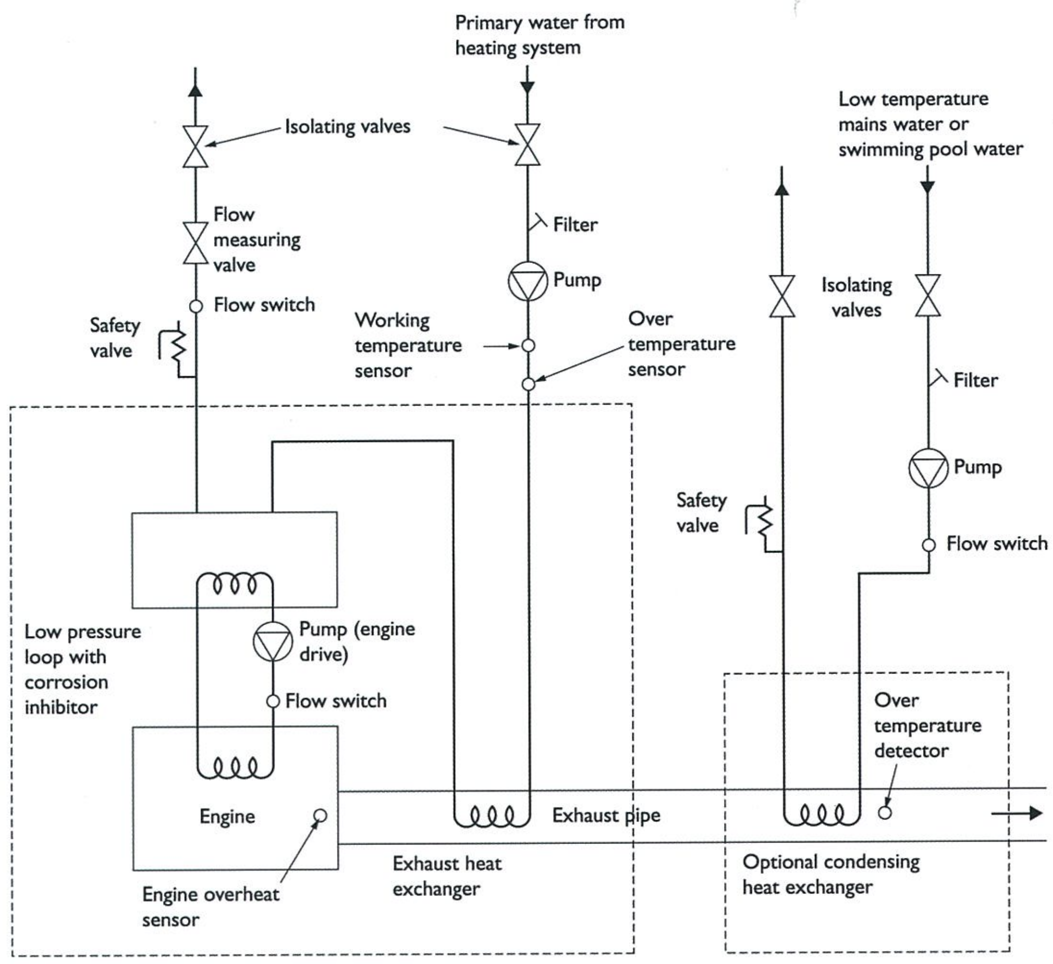
Examples of CHP circuits deemed unsuitable

Figure 5 and Figure 6 illustrate examples from the literature of how heat should be recovered from CHP. These circuits are not considered satisfactory, as they are likely to result in overheating of the jacket.

The purpose of showing the circuits in Figure 5 and Figure 6 is to encourage the designer to try to work out how the CHP could be controlled to act as a lead boiler given the fundamental hydraulic design and the pumping arrangements of existing systems.

Many examples from CHP literature (even some government publications) should be treated with caution. However reputable the source of the information, the designer must always analyse the actual circuit.

Figure 5: CHP circuit taken from literature which is deemed unsuitable – see text.



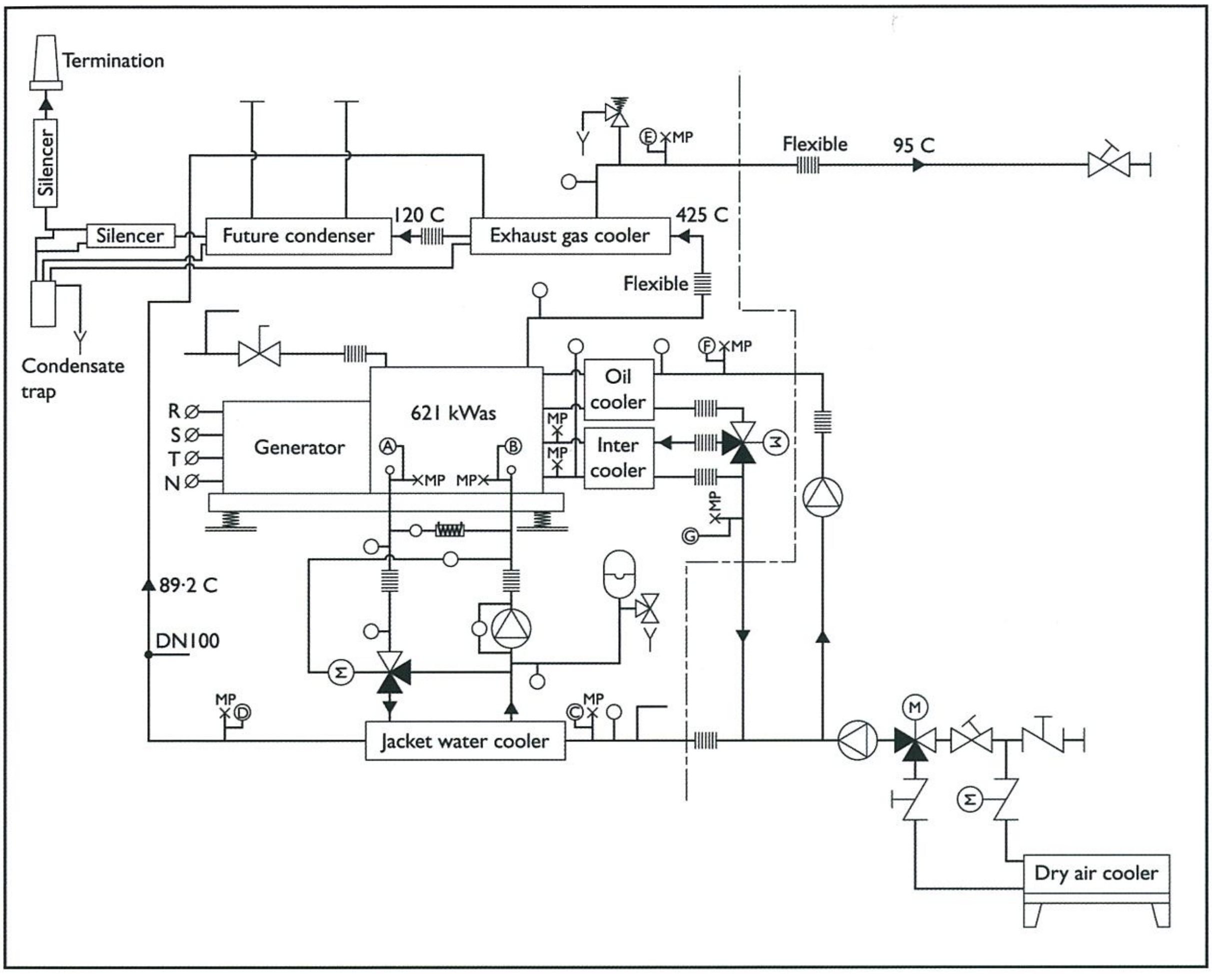
Note: Some units may also include oil cooling, generator cooling and/or cabinet cooling, heat recovery circuits

In Figure 5 a circuit is drawn showing the return water passing through a filter and then pumping through an exhaust heat exchanger, then through the engine jacket heat exchanger and possibly the oil cooling circuit. In the authors' opinion, the flows of water and the heat pick up is in the wrong direction.

Low-grade heat from the condensing exhaust heat exchanger and the intercooler would normally be picked up first followed by the oil cooler, the jacket and finally the exhaust heat exchanger.

The circuit in Figure 5 would not normally be used, but still finds its way into CHP literature.

Figure 6: Example of a heat recovery system for CHP engine linked to a dry air-cooler.



The circuit in Figure 6 reflects a standard circuit offered by CHP package manufacturers. The circuit is designed to protect the engine and is not intended to maximise the utility of the heat produced from the engine for heating purposes.

The dry air-cooler is normally designed to maintain a return water temperature to prevent overheating of the jacket. Many discounted energy purchase contracts for CHP guarantee the power output within defined parameters, but not the useful heat output. Designers considering this circuit should consider how to control the dry air-cooler in relation to the CHP unit and then consider how to control and connect the circuit to any boiler circuits. This should include adding top-up heat to the system to ensure that the dry air-cooler does not run when a boiler is running.

3.4 ITEMS FOR CONSIDERATION FOR EFFECTIVE CONTROL OF CHP

The following details should be considered with regards to finding the most effective control of CHP:

- Whether or not to operate with a supplementary boiler. Where electrically efficient gas fired CHP displaces coal or oil fired central generation, CO_2 savings arise even when the supplementary boiler is running due to the higher carbon content of the centrally generated electricity and the carbon losses in transmission and distribution of electricity.
- Whether to operate the CHP unit on a heat-led basis or an electricity-led basis.
- How to decide when to run, and at what level of output.
- How to calculate when it is economic or uneconomic to run the CHP and over what hours.
- Consider and recognise that the financial arrangements and form of contract for electricity, fuel, maintenance and financing will change operating decisions. Once the capital investment in CHP has been made, operating decisions should be made on the contribution that can be achieved from its operation. A common mistake is not to run the CHP system if the contribution is less than the capital repayment.
- How to achieve control and differentiate between different sources of heat supply and heat loads under conditions of inevitable fluctuations in flow temperatures from boilers and the CHP unit, and system return water temperatures.
- The sensitivity and accuracy of the controls.
- The capabilities of different burner/boiler combinations depending on whether the burner is on/off or modulating and its turn down limitations.
- The choice between on off control and partially modulating or fully modulating systems.
- Consideration of the advantages and disadvantages of two-way and three-way valve control systems and the hydraulic and temperature effects of these different systems.