HIGHLY EFFICIENT WATER HEATING SYSTEM

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ABSTRACT

The paper describes an application of a packaged, transcritical carbon dioxide water chilling system. The system is arranged to extract heat from the return water of an existing chilled water system. The transcritical system assists the refrigerating effect of the existing system with the result that, when the transcritical system is used to produce hot water, the power consumed can be discounted against power required for the chilled water system. The package described is capable of providing about 50kW of heating effect when heating mains water to 80°C. The paper describes the challenges presented in designing such a system and explains how they were overcome.

The paper shows that provided there is an existing need for chilled water, high temperature hot water can be provided for minimal input of extra power.

Key words :
Efficiency, transcritical, carbon dioxide, water heating.

1. INTRODUCTION

In recent years there has been a welcome change in attitude to energy efficiency. Twenty years ago the effect of increasing concentration of atmospheric carbon dioxide was not generally appreciated. Energy was cheap relative to cost of using it. Purchasers of equipment tended to consider first cost and value added to product more important than efficiency. The production of highly efficient refrigerating machinery was therefore an expensive and rather unrewarding hobby. In 1974 Pearson (1) showed that it was possible to produce a continuous air blast freezer using about 1/3 of the fan power that, at that time, was required for conventional freezers. The market was not particularly impressed because, at the time, energy efficiency was not commercially important.

At present energy is still too cheap but governments are beginning to realise that our prodigal use of energy must be curbed. A recent paper by Wade (2) in Auckland indicated that policies of penalising the inefficient and rewarding the efficient were starting to be applied. Such policies which are fiscally neutral are important in democracies where it is politically inexpedient to cause sharp rises in energy prices.

The device described in this paper is capable of producing high temperature hot water for practically no input of energy but the capital cost of the device is much greater than the cost of a simple electric heater or even of a gas fired boiler. A few years ago such a device would have been unsaleable. Under present conditions it is very probable that the benefits in terms of government assistance associated with the use of the device will give a very short payback and might avoid need for much greater capital expenditure on main air conditioning systems. Figure 1 shows the circuit diagram of the complete system as produced for the original patent specification (3). The device which is subject of this paper, is shown at the right hand side of the diagram and is labelled heat pump. The heat pump is associated with an existing chilled water system having its own separate refrigeration circuit shown at the left hand side of the diagram. The heat pump is used to extract heat from chilled water returning to the cooler of the separate refrigeration circuit. The heat pump therefore reduces the load on the separate refrigeration system. In so far as the heat pump and the separate refrigeration system operate at comparable efficiencies the power input to the heat pump is being used to produce required chilled water without any increase in required power provided that the separate refrigeration circuit is efficiently unloaded. In the illustrative diagram the full chilled water flow is shown being refrigerated by the heat pump but in most practical cases only a proportion of the chilled water would be refrigerated by the heat pump which would usually be of much lower capacity than the main chilled water system. The hot water is shown flowing from mains to an open tank. Again this is purely diagrammatic in a practical embodiment there would almost certainly be a closed storage tank.

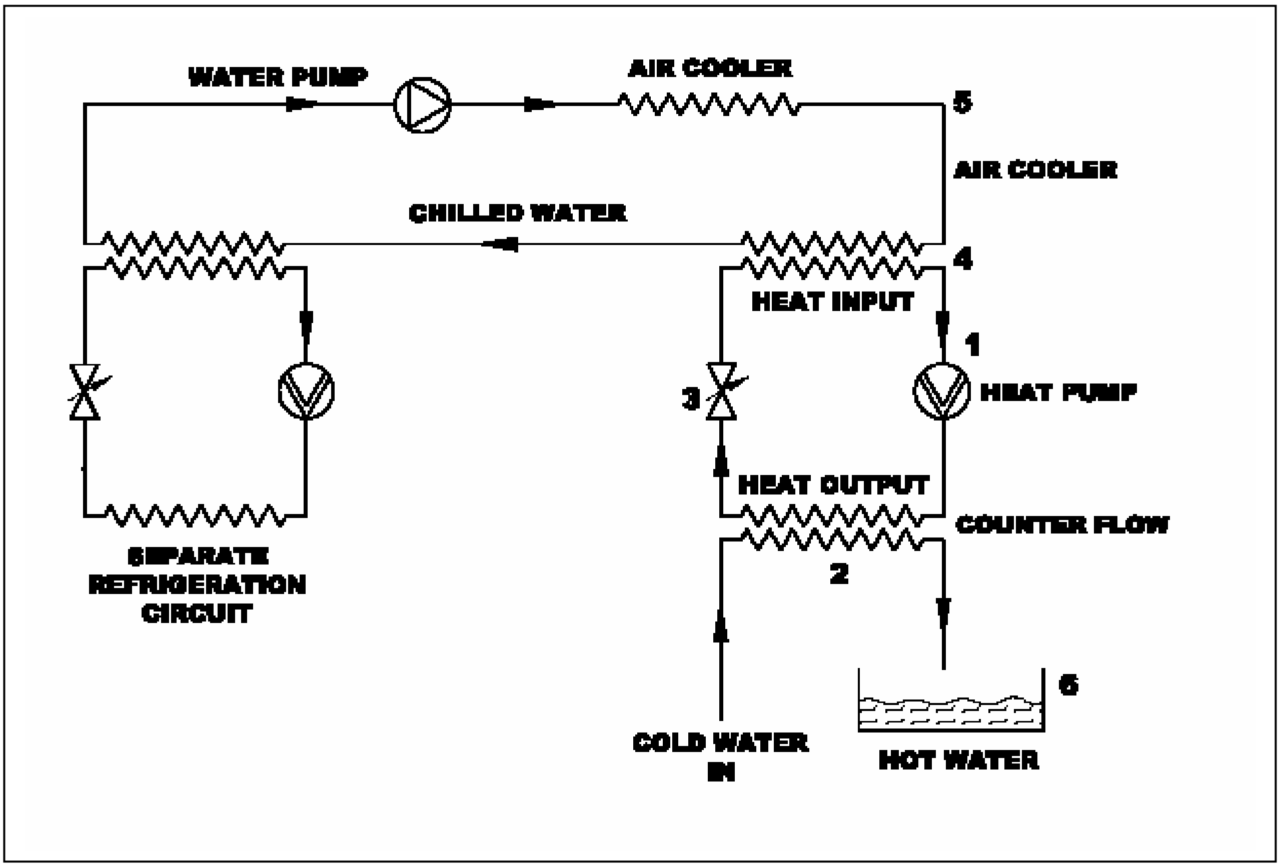

Figure 1 Circuit Diagram

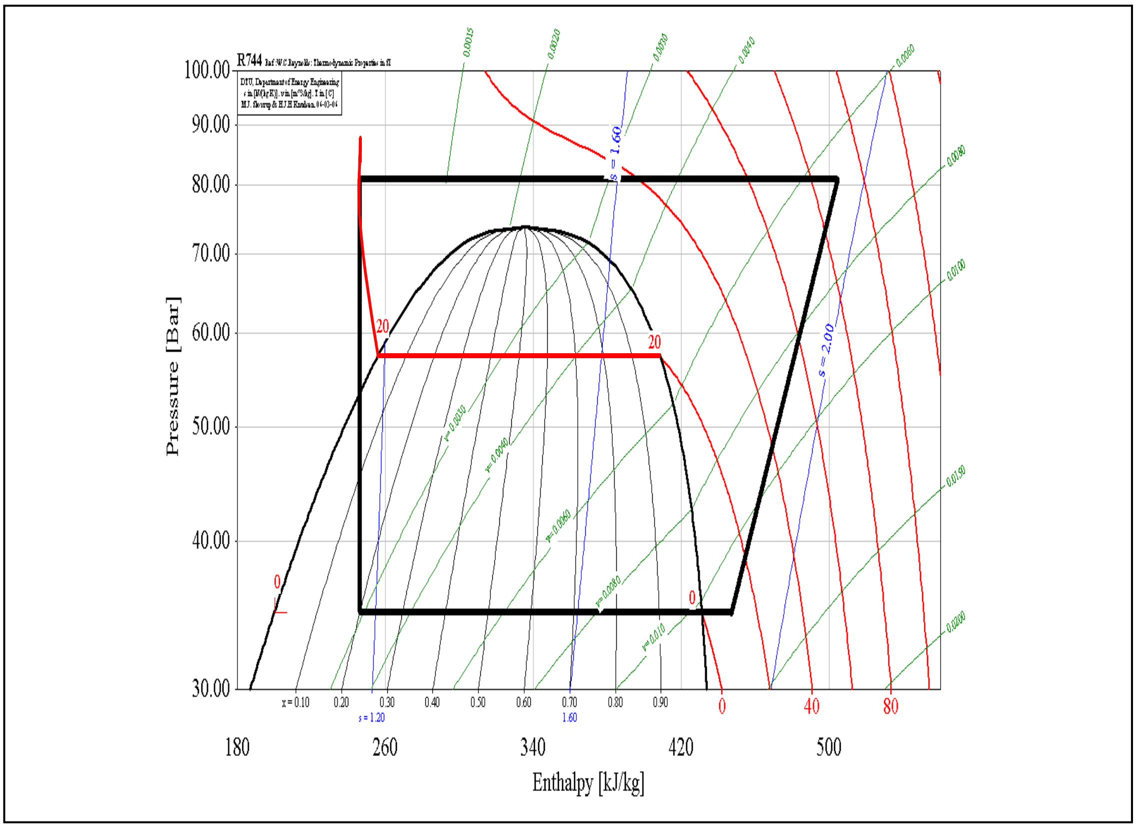

Figure 2 Mollier Diagram

For efficient heating of water through a large temperature range it is very beneficial to use carbon dioxide in a transcritical cycle not only because of the high discharge temperatures that are achieved but because cooling super-critical fluid in counterflow with mains water allows the water to be heated through a large temperature range [4]. It is a feature of the transcritical cycle that efficiency is much improved if the transcritical fluid can be cooled to a relatively low temperature. Figure 2 shows a pressure enthalpy diagram for a theoretical cycle of the type envisaged.

The application is to supply heat to raise water from 15°C to 50°C using carbon dioxide in a transcritical cycle as the working fluid. The evaporating temperature is to be 0°C. The heat is to be rejected at a discharge pressure of 80bar absolute. The super critical fluid is to be cooled to 20°C. The isentropic efficiency of the carbon dioxide pressure is taken as 0.6. The COP of the existing water chiller system is assumed to be 3.0. From the pressure enthalpy diagram (Fig 2) work input equals 60kJ per kg.

Heating effect equals 235kJ per kg. Refrigerating effect equals 175kJ per kg.

Work required to produce equivalent refrigerating effect from existing air-conditioning system (COP equals 3) is 175 ÷ 3 = 58kJ per kg.

Therefore the net work required to produce hot water is 60 – 58 = 2kJ per kg.

The electrical power cost of producing hot water using this system would appear to be negligible provided there is a simultaneous requirement for chilled water.

3. PRACTICAL EMBODIMENT OF THE SYSTEM

Figure 3 shows the refrigerant circuit diagram of the packaged unit. The compressor is a Bitzer semihermetic 4HC4-20K single stage 4 cylinder compressor of displacement $12m^3$ at 50 hertz. Maximum allowable pressures are 120bar on the high-pressure side and 75bar (standstill) on the low-pressure side.

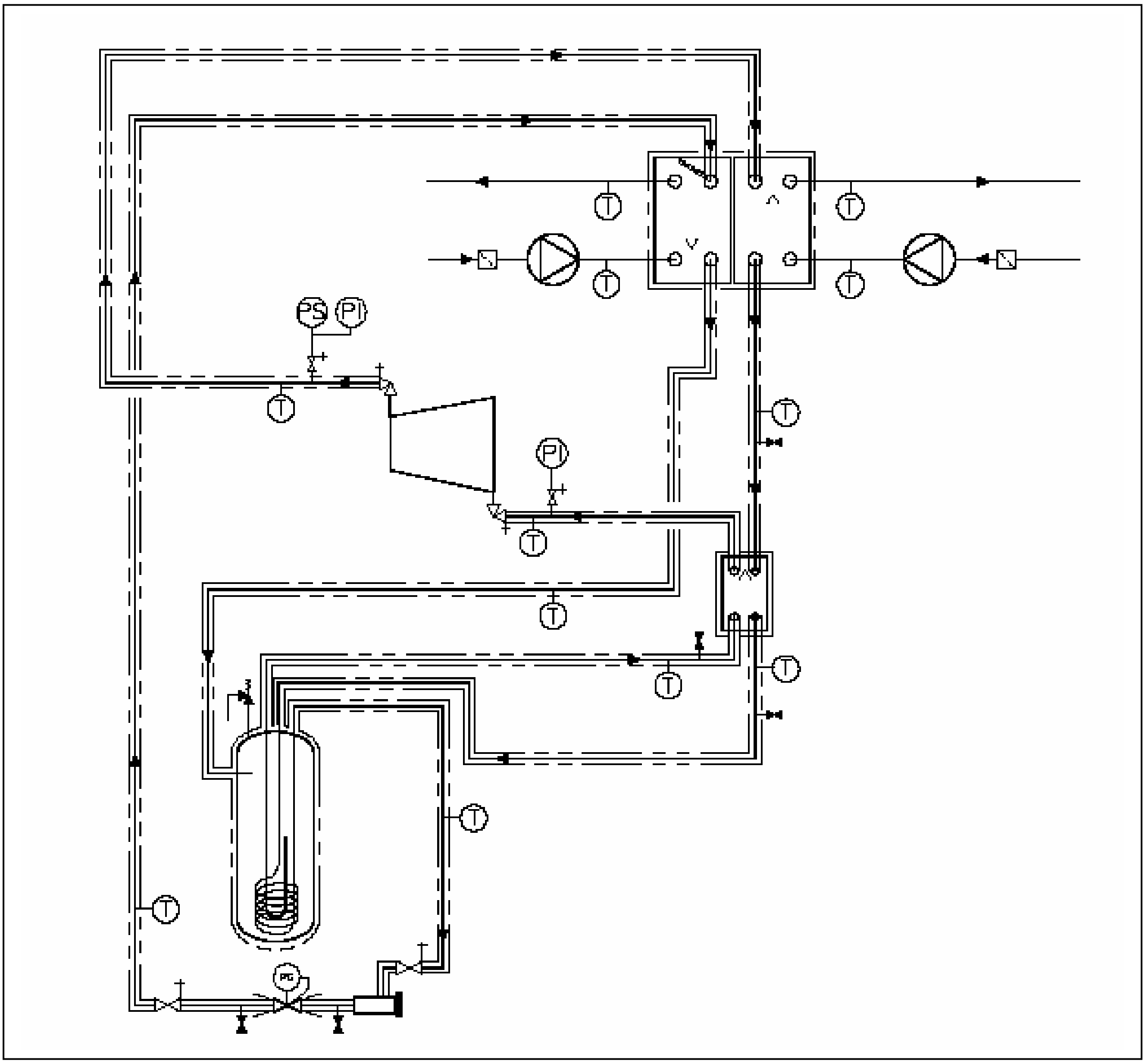

Figure 3 Refrigerant Circuit Diagram of Packaged Unit

Actual performance as calculated by Bitzer for a cycle where the refrigerant evaporates at 0^oC and the heat rejection takes place at 80bar absolute with the super critical fluid being cooled to 25^oC is 40.8kW refrigerating effect and 12.8kW absorbed power. The calculation assumes 10K useful superheat.

A miscible lubricant BSA85K is used and there is no oil separator. The evaporator and the gas cooler are identical brazed plate heat exchangers mounted back to back with high strength insulating material between. The heat exchangers are reinforced so that they are suitable for a maximum working pressure of 100bar gauge. After passing through the gas cooler the high pressure carbon dioxide is further cooled by heat exchange with the vapour going to compressor suction. The purpose of the heat exchanger is mainly to protect the compressor by providing some super heat. After the suction super heater the high-pressure carbon dioxide passes through a heat exchange coil in the low-pressure receiver. The low-pressure receiver first described by Pearson (4) (5) is a vessel in the suction line to the compressor that allows overfeeding from the evaporator followed by evaporation of the overfed liquid to dryness before it goes to the compressor. The low pressure receiver provides multiple functions 1) it protects the compressor from flooding; 2) it allows for storage of excess low pressure liquid refrigeration; 3) it provides additional cooling of the high pressure refrigerant, 4) it automatically overfeeds the evaporator provided the expansion device can open wide enough. 5) it acts as a fade out vessel to limit the pressure to which the carbon dioxide may rise during standstill. This is necessary because the maximum allowable pressure of the low side of the compresser is 75bar. It is conceivable that this pressure might be exceeded during standstill in warm weather if a fade out vessel were not provided. After the low-pressure receiver the high-pressure refrigerant passes through a filter drier and through an expansion device of the type that maintains a constant upstream pressure. After the expansion device the low-pressure refrigerant passes to the plate type evaporator, from which it is piped to the low-pressure side of the low-pressure receiver and from there through a suction pipe and suction gas superheater to the compresser.

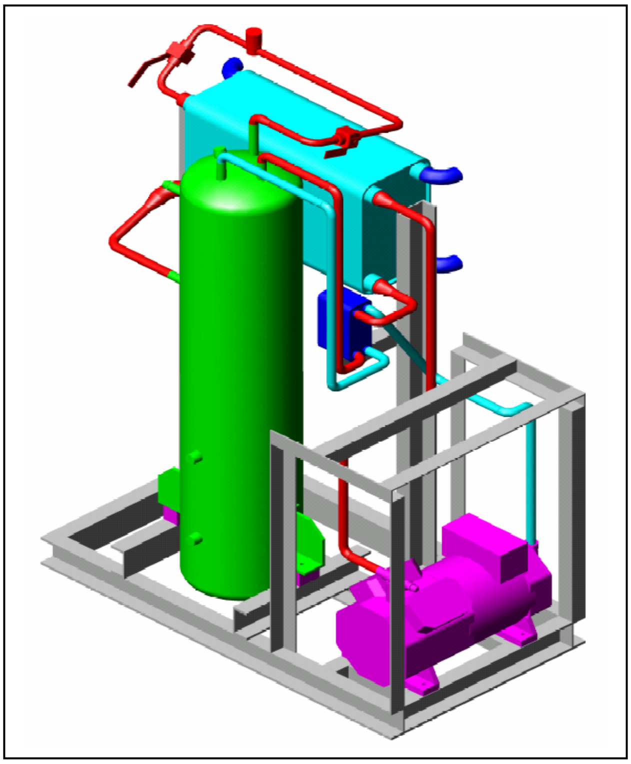

Figure 4 - Packaged Unit

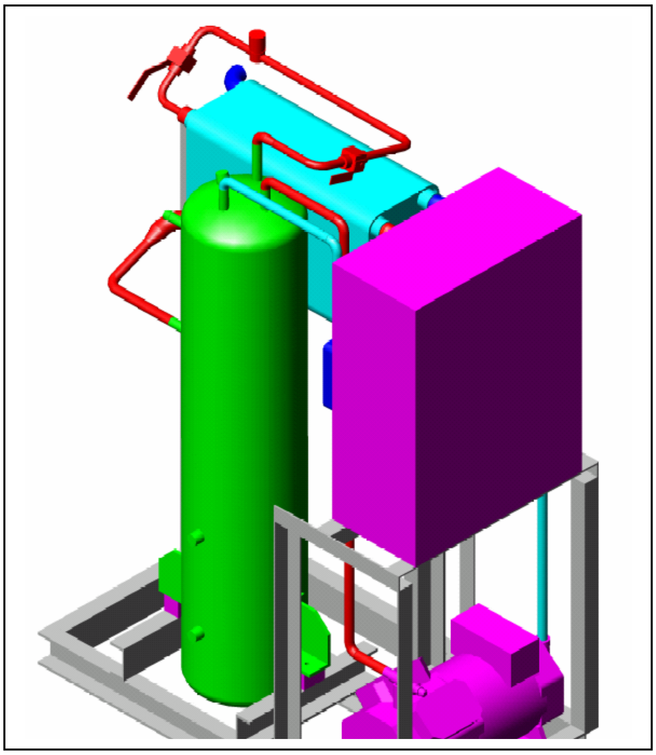

Figure 5 - Packaged Unit with Starter

Cold water pumps and hot water pumps are provided on the packaged unit. The cold water pump is much larger than the hot water pump because the temperature reduction in the cold water is much less than the temperature increase of the hot water.

Figure 4 is an isometric arrangement of the packaged unit showing the compressor the low pressure receiver is green, the reinforced heat exchanger in light blue and the suction line heat exchanger in dark blue.

Figure 5 shows the packaged unit with starter panel in place. Figure 6 shows the circuit diagram for the water circuit for the test rig. There are 2 water tanks one above the other to promote stratification. A recent paper by Cleland in Auckland (6) indicated that stratification might be achieved without need for this precaution. In practical application the standard unit would be connected to the chilled water return of the air conditioning circuit and the water would be pumped through the evaporator using the larger of the two pumps on the unit. Water would be pumped through the gas cooler from the hot water storage tank circulating at low rate so that hot water stratified in the top of the tank. On hot water being drawn from the top of the tank it would be replaced by cold mains water coming in to the bottom of the tank. In the test rig circuit a 3-way mixing valve is introduced to allow continuous operation in absence of a separate air conditioning system.

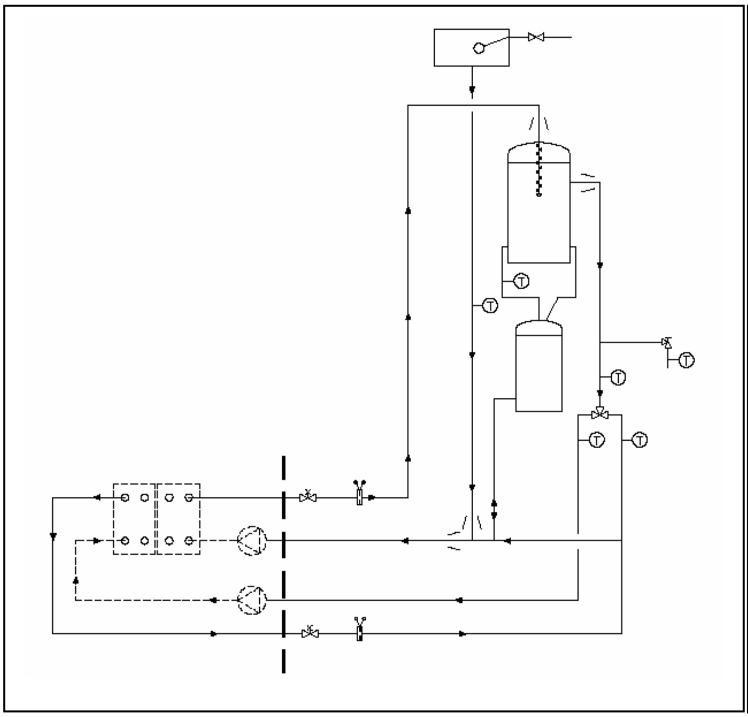

Figure 6 Water Test Circuit

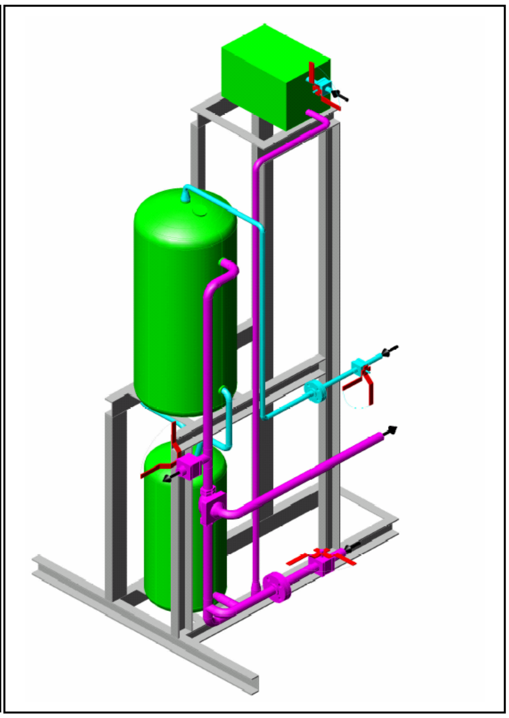

Figure 7 Isometric Arrangement

From their test results the compressor manufacturer was able to predict the performance of the compressor under the operating conditions that were specified. With the refrigerant evaporating at 0^oC and being discharged to a pressure of 80bar absolute with the supercritical fluid being cooled to 25^oC. The refrigerating effect was 40.8kW for a 12.8kW input with 10K useful superheat. This gives a coefficient of performance in the refrigerating mode of 3.19. The COP in heating mode is 4.19 if account is not taken of the useful refrigerating effect produced.

If the power required is allocated to chilled water duty the additional power for hot water production is practically zero.

CONCLUSIONS

A water heating package has been designed that will provide high temperature hot water for practically zero additional power requirement.

Such a capital intensive article would not have been saleable even a few years ago despite its high efficiency.

Such an article could not have been produced a few years ago because efficient transcritical carbon dioxide compressors were not available.

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