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SECOND GENERATION INTEGRATED MICROTURBINE, ABSORPTION CHILLER, AND SOLID WHEEL DESICCANT SYSTEM

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ABSTRACT

This paper describes a second generation Cooling, Heating, and Power (CHP) for Buildings system. CHP systems have the potential to achieve thermal efficiencies twice that of conventional remote power plants by utilizing waste heat and eliminating losses due to transmission and distribution. However, system integration issues need to be dealt with before these objectives can be realized. The CHP System being evaluated consists of a microturbine generator that provides on-site electrical power, using natural gas as fuel, for a medium-sized commercial office building. The waste heat of the microturbine is used in a single-effect absorption chiller, which provides cooling, and a solid wheel desiccant system, which provides dehumidification. Technical issues related to modifications of the original CHP system are described and improvements to the original system design and performance are evaluated including a reduction in the number of components, a reduction of parasitic power consumption, and improvements in controls for the heat recovery process. Overall system performance as well as individual component performance is compared between the first generation CHP system and the current system. Recommendations are presented for further improvements on integrated CHP systems for commercial applications.

INTRODUCTION

The Chesapeake building on the University of Maryland, College Park campus has been the site of Distributed Generation (DG) and CHP research and demonstration for the last three years. Both Air Conditioning (AC) Zones in the building have their own CHP system acting to reduce the cooling workload of the Roof Top Unit (RTU) serving that zone during the cooling season by supplying both sensible and latent cooling produced by thermally activated equipment recovering waste heat from their respective prime movers. As new equipment becomes available and tests are completed on existing equipment the systems will continue to evolve. The installed systems have had two complete cooling seasons of testing in 2001 and 2002 and the CHP System described in this report, consisting of a microturbine, absorption chiller and a solid desiccant system has had a different microturbine in each of these periods. The changes that accompanied this change of prime mover were guided in part by the particular characteristics of each generator and partly from the knowledge that was gained during the first cooling season about system performance.

1 SYSTEM DESCRIPTION

The second generation CHP System 2 consists of three components, a 60kW microturbine, a single effect LiBr absorption chiller and a Solid Desiccant Unit (SDU). The waste heat from the microturbine is directed through the regenerator of the absorption chiller directly, without an intermediate steam generation step. The exhaust gases are then directed through an insulated duct to the roof where they are used to regenerate the desiccant wheel of the SDU. The equipment has been split between the ground floor of the building and its roof due to the structural weight restrictions of the building design.

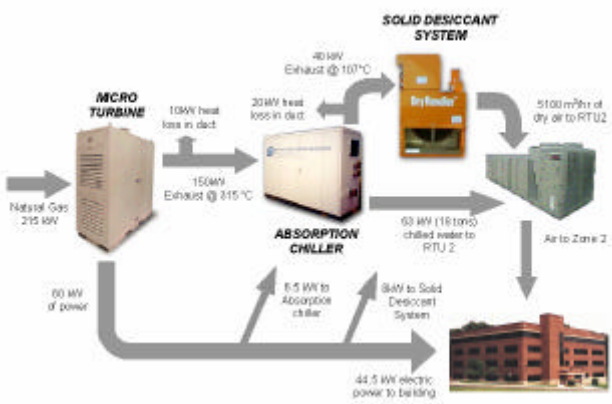

Figure 1. CHP system 2 schematic layout

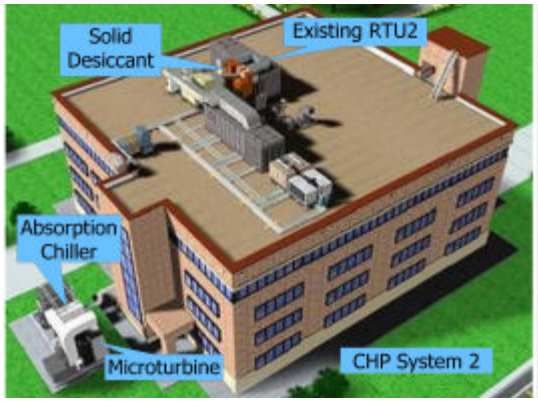

Figure 2. Location of the CHP equipment around the Chesapeake building

1.1 Microturbine

The cooling season of 2002 saw the installation of a new 60kW microturbine as the previous 75kW model had been de-commissioned by its manufacturer at the close of the 2001 cooling season. A number of changes needed to be made (summarized in Table 1) as a result of this change because the electrical and thermal characteristics of the two units differed. On the electrical side the 75kW unit was a synchronous machine that used an external transformer to step the voltage up to the buildings supply, whereas the new 60kW model uses power electronics to adjust its supply voltage without the additional parasitic power consumption. On the thermal side, the characteristics of the exhaust gases were quite different as well. The 60kW unit is a smaller capacity machine with greater thermal efficiency (rated 25% (HHV) ± 2% at ISO conditions) but produced exhaust gases at a higher temperature. The absorption chiller was designed for the exhaust gas characteristics of the previous unit and this meant that the 2002 installation required a fan to add outdoor air into the exhaust stream to bring the Mixed Air (MA) temperature down to a level that the absorption chiller was rated to withstand. The next major change was the removal of the high temperature fan that had previously been used to draw the exhaust gases away from the microturbine and push them through the absorption chiller and up onto the roof to the SDU. This had been installed to prevent backpressure on the 75kW microturbine. The new 60kW microturbine was rated by the manufacturer as capable of withstanding up to 20cm of water column backpressure on the exhaust and approximately 4cm of this capability was used in CHP System 2. As the exhaust ducting after the absorption chiller was routed directly up the side of the building there was a considerable stack effect as the warm exhaust gases were drawn up the duct, which assisted in keeping the backpressure to a minimum. This backpressure imposed a power output reduction in the microturbine of 3kW at elevated compressor inlet temperatures (Figure 3), which could also be considered parasitic power.

Since the original high temperature fan consumed 5kW of electrical power the power savings as a result of its removal was 2kW. Since CHP can be considered a strategy particularly appropriate for generators with lower efficiency since they leave a significant fraction of their heat input in their exhaust, the low power production relative to the thermal output from the microturbine should indicate that all power produced should be conserved as much as possible. Since the thermal efficiency of the electrical grid in the commercial sector is 31% (HHV) [1], the energy intensity of electricity generated from a microturbine is greater with its efficiency at 24% (HHV). The energy savings in microturbine-based CHP systems arise from the appropriate utilization of its exhaust heat, not as a result of the electrical efficiency of the generator in isolation [2], [3].

Table 1. Summary of the changes as a result of new microturbine

2001 – 75kW MT	**2002 – 60 kW MT**
High temp fan needed (5 kW)	No high temp fan (0 kW)
External high voltage transformer (4 kW)	No external transformer needed (0 kW)
Piston compressor (4 kW & loud)	Scroll compressor w/VFD on DC bus (4 kW & quiet)
260° C Exhaust temp, no MA fan needed	315° C Exhaust temp, need MA fan* (<1 kW)
No pressure drop imposed (0 kW loss on MT)	4 – 6 cm w.c. backpressure (3 kW loss on MT)
Total 13 kW loss	**Total 8 kW loss**

* Mix Air (MA) fan needed to lower exhaust temp to 270º F to meet chiller requirements.

The microturbine power output is dependant upon compressor air inlet temperature as the controller electronically limits the rotational speed, the turbine speed is adjusted automatically by the controller adjusted from 88,000 rpm to 96,000 rpm. This limit is reached at an inlet air temperature of 30°C and after this point the air mass flow rate required to deliver 60kW of power output cannot be met and the power output declines (Figure 3). The impact of 5cm water column backpressure is to move the de-rate air temperature back 4°C and reduce the power output by up to 3kW at temperatures above this point.

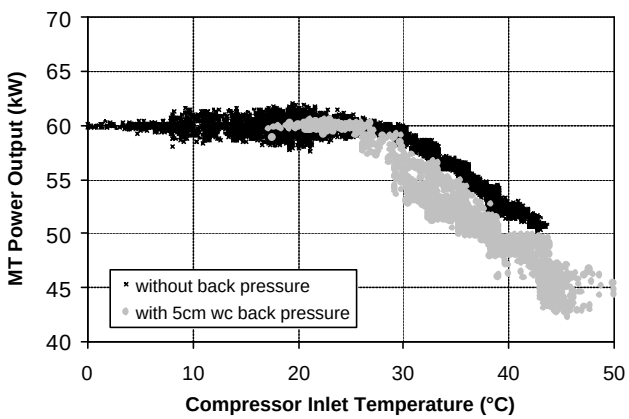

Figure 3. Performance of the microturbine across a range of compressor inlet temperatures

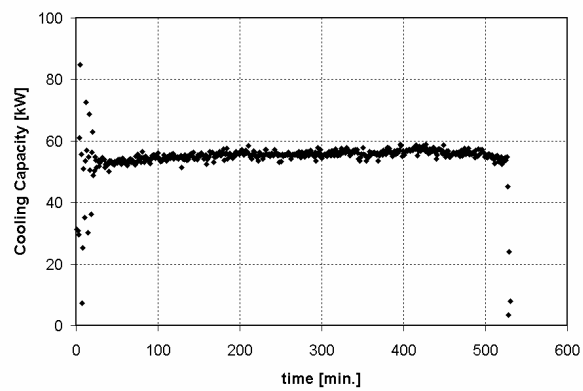

Figure 4. Cooling performance of the absorption chiller from startup through to shutdown on the 29th July 2002

1.2 Absorption Chiller

The absorption chiller in the system is a single-effect, direct fired, LiBr unit modified from its original design specifically for the original 75kW microturbine in CHP System 2. During the cooling season of 2002 the impact of the change in exhaust mass flow rate and temperature altered the performance of the absorption chiller and required internal setpoints to be adjusted to obtain the anticipated cooling capacity. The exhaust of the 2002 microturbine was hotter (315 compared to 260°C) and had a lower mass flow rate (0.49 compared to 0.69 kg/sec) relative to the microturbine that was originally installed. Initially this meant that the chiller was unstable, with cooling capacity swinging from over 70kW of cooling right down to 5kW of cooling as the cooling tower fan cycled on and off to try and stabilize the absorption cycle. The solution was to lower the cooling tower temperature setpoints for both the cooling tower fan on and off command by 3°C. This had the effect of shortening the cooling tower cycles and stabilizing out startup transients inside the absorption chiller to the point where the large cooling swings were eliminated after less than 30 minutes of operation. In Figure 4 the start up variation in cooling capacity stabilized within thirty minutes and remains constant at 55kW throughout the day until the system was shut down. Since the chilled water piping had to be run up the side of the building to the coil retrofitted into the MA chamber of the RTU almost 200m of steel pipe had to be cooled as well to reach steady state operating conditions.

1.3 Solid Desiccant Unit

The SDU contains both an enthalpy wheel designed to recover cooling from the building exhaust air and a desiccant wheel that utilizes the waste heat remaining in the exhaust stream after passing through the absorption chiller. The desiccant wheel rotates at 12 rotations per hour whereas the enthalpy wheel spins much faster at 45 rpm.

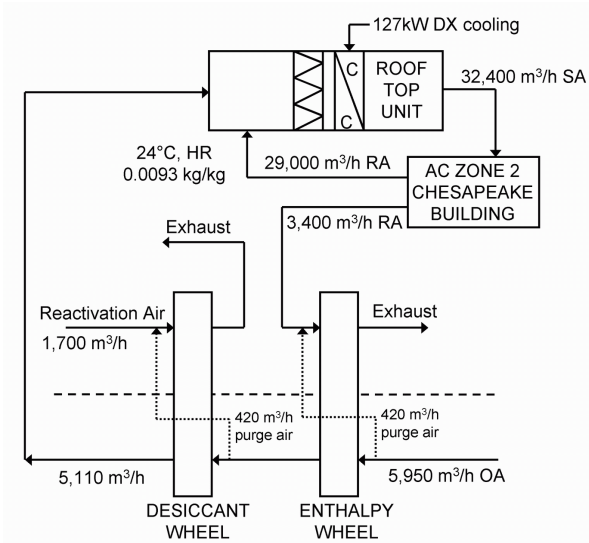

Figure 5. Schematic layout of the SDU and its integration into the RTU

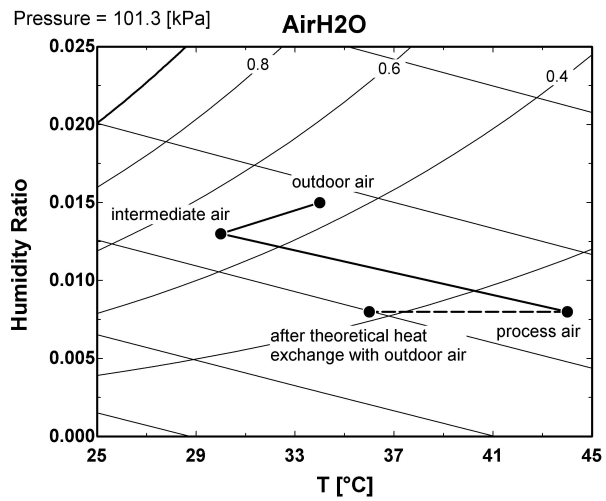

Figure 6. Psychrometric processes occurring inside the SDU

Figure 5 shows the layout of the two wheels and Figure 6 details the psychrometric processes occurring to the process air as it passes from being outdoor air through the enthalpy wheel to become intermediate air, through the desiccant wheel to become process air to be added into the RTU. Also shown in Figure 6 is the likely result of adding an 80% effective heat exchange step after the solid desiccant wheel to remove some of the sensible heat added during the process of desiccation. The desiccant step is almost an isenthalpic process where latent heat is transformed into sensible, except for the influence of the binding heat of water as it is absorbed onto the desiccant material and the heat carried over from the regeneration side as the desiccant wheel rotates – both effects that slightly increase the enthalpy of the process air. What the SDU is not equipped with, but would really benefit from is a heat exchanger, either another wheel or a conventional model so that the enthalpy of the process air is further reduced and the cooling load that the RTU needs to carry out to process this air is reduced. The performance of desiccant units is often quoted in terms of latent performance alone, but in this application the desiccant has been included in the system for energy savings, not just humidity control and an enthalpy reduction measure is useful.

$$COP_{therm,total} = \frac{cooling\,capacity}{heat\,input}$$

Equation 1. Thermal (electrical input excluded) COP for the total cooling of the LDU

$$COP_{therm,latent} = \frac{moisture\,removal \times latent\,heat\,of\,evaporation}{heat\,input}$$

Equation 2. Thermal (electrical input excluded) COP for the latent cooling of the LDU

Table 2. COP of the SDU at the Chesapeake building

Description	Value	Value w/ additional heat exchange step
COP thermal, total	11kW/22kW = 0.5	22kW/22kW = 1
COP thermal, latent	29kW/22kW = 1.3	29kW/22kW = 1.3

The values listed as thermal COP's in Table 2 are given this title because of the omission of any reference to their electrical power consumption, which is an equally important input whose importance is discussed later in the paper. These heat additions and cooling outputs quoted in Table 2 are time-averaged values produced by analyzing temperatures, humidity's and flowrates of the process streams inside the SDU. The total (enthalpy reduction) cooling COP can be seen to double under the influence of the additional heat exchange step while the latent COP remains unchanged.

2 PERFORMANCE DISCUSSION

CHP systems can be installed to try and achieve a wide range of goals, generally separated into the three categories of saving energy, saving cost or making environmental improvements. There is some conflict between these three goals, for example the combustion of coal typically produces 60% more CO_2 and 200% more NO_x per unit of energy, but costs 70% less than natural gas. Diesel engines are cheaper and more efficient than microturbines, but are more polluting. Using grid electricity may be less expensive than the CHP alternative but can have much greater Primary Energy Consumption (PEC) when providing the same services. Comparing the Chesapeake building with other CHP systems is made more complicated by the lack of defined standards for rating the performance of diverse CHP networks in any of these three areas. This makes it difficult to demonstrate where the benefits of a particular system lie in either a scientific or commercial forum. All three targets, economic, environmental and energy can probably best be dealt with by comparing them against their alternatives.

2.1 Comparison with Vapor Compression Systems

The main competitor for CHP cooling systems is electric vapor compression cooling which serves the majority of the cooling market. A Vapor Compression (VC) system has one input, electrical power and one output, kilowatts (or tons) of cooling capacity at a certain temperature, making the derivation of a generic COP straightforward (kilowatts cooling output/kilowatts electrical power input). In a CHP system there is typically one input, a combustible fuel, but many possible outputs: cooling, heating and power. In Figure 7 and Figure 8 schematics of the energy consumption of two systems that produce cooling plus power outputs are shown.

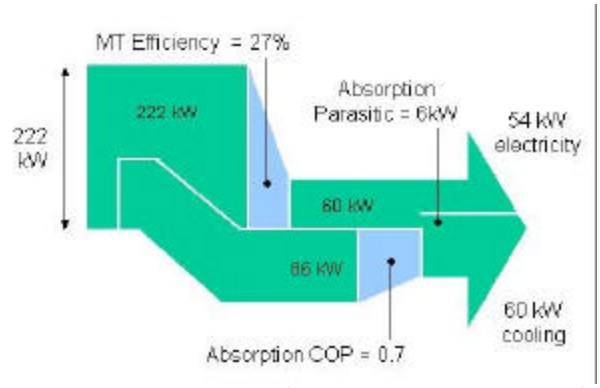

Figure 7. Representation of the energy consumption of a microturbine-based cooling + power system

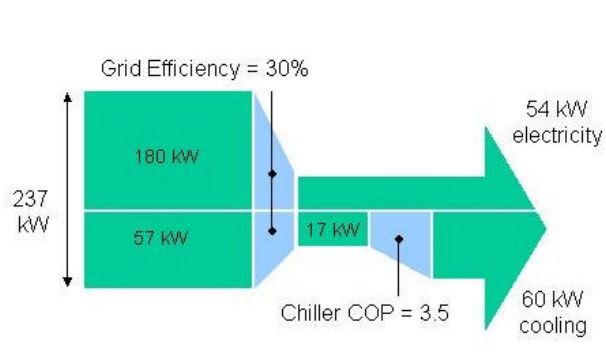

Figure 8. Representation of the outputs of a conventional power + cooling system

The energy flows indicate that the energy savings from a CHP cooling system when compared to conventional grid based system is 6 %. Figure 7 & Figure 8 are intended to represent small-scale units in both cases – efficiencies and COP's can be improved on both systems if the components were much larger.

Table 3. Impact of potential changes to CHP System 2 and impact upon energy savings achievable

Changes to CHP System	Energy savings relative to grid electric alternative
Base case as installed	6%
Absorption Chiller requires half current parasitic power (3kW)	11%
Double effect absorption chiller with thermal COP of 1.2	20%
Double effect chiller (1.2) with half the current parasitic power (3kW)	23%
Microturbine with 30% (HHV) efficiency + double effect chiller (1.2) with half the current parasitic power (3kW)	31%

The pathway to substantial energy savings with microturbine-based CHP cooling systems can be seen in Table 3. Changes and improvements can be made within the CHP system to compete with the relatively static targets of grid electrical efficiency and VC chiller COP. For large centrifugal VC units that use cooling towers, higher COP's are achieved than the 3.5 used as a comparison in this example but for air-cooled units at the scale of 60-100kWR this is a reasonable number. The use of Variable Frequency Drives (VFD) and potential improvements in control may bring about the parasitic power reductions suggested as possible in Table 3. There are microturbines under development at the moment that are expected to achieve 35% (LHV) efficiency [4]. It should be highlighted that the targets of energy savings and economic savings may involve very different challenges and metrics and as such energy savings do not directly translate into cost savings and economics tend to be much more site specific and a much more dynamic target with demand charges, ratchet clauses and net metering all contributing.

2.2 Microturbine Performance

The characteristics of the microturbine are critical to the performance of the overall system as it provides all of the electrical power and waste heat for the remaining components of the CHP system. However in heating applications the efficiency of the generator can become less significant from an energy perspective since any heat not provided by the CHP application may simply need to be made up by burning fuel elsewhere in the heating plant, effectively making investments in improved microturbine efficiency less valuable. In CHP cooling applications on the other hand where the waste heat utilizing components such as absorption chillers or desiccant units consume significant quantities of electrical power, the electrical efficiency of the generator in the system will affect the energy savings obtained. In addition, the parameter matching between the prime mover and its downstream components is also more critical to the system performance in cooling applications. In cooling CHP applications reductions in waste heat temperature will not only reduce the temperature difference available for heat transfer but can also affect the stability and COP's of the downstream waste heat utilizing equipment.

2.3 Absorption Chiller Performance

There are a number of different perspectives that can be taken when evaluating the benefits of an absorption chiller in a CHP system. The first is from a purely COP perspective where the heat input is related to the cooling output. The COP of the absorption chiller in CHP System 2 was measured at 0.65; slightly below the COP achieved operating with its design heat input during the first season of operation where it reached 0.7. The second perspective is to look at its electrical consumption and compare it to an electrical VC system. Consuming 6kW of electricity and producing 60kW of cooling, the output is much greater than the 3.5 COP displayed by the RTU installed at the Chesapeake building and also even much better than a large centrifugal chiller could be expected to achieve with the same electrical input. This perspective highlights the benefits of absorption chillers when the waste heat to fire them is available. Reductions in peak electrical demand particularly can have significant economic benefits.

2.4 SDU Performance

Determining the impact of the desiccant unit on the building is not as straight forward as the microturbine and the absorption chiller where the outputs can be clearly defined [5]. As mentioned in the system description the SDU can be rated in terms of its performance as a dehumidifier (latent COP) or as a tool to reduce the cooling load of the RTU that it serves (total, or enthalpy reduction COP). Desiccant units can provide humidity control that would traditionally be very energy intensive to obtain by suppressing the dew point of the air through over cooling and reheating. It can be rated on its electrical consumption as well, the 8.5 kW of electrical power consumption results in 11kW of total cooling as it is set-up now, 22kW if the heat exchange step were to be added. A desiccant unit may help avoid more electrical power than the direct cooling provided by avoiding RTU overcooling, but this will also depend upon the characteristics of the equipment that the outdoor air is delivered to for processing [6].

2.5 System Performance

Figure 9 shows the exhaust temperature and energy distribution along CHP System on a typical test day. The microturbine produces 162 kW waste heat, in which 48% (92 kW) is used by the absorption chiller, 14% (22 kW) is used by the solid desiccant unit, another 23% (18 + 20 kW) is lost to the environment through the insulation cladding of the ductwork or the low grade heat dumped to the environment after the SDU. Including the power output of the microturbine, the total heat utilization of the CHP system is (60+92+22) kW/222 kW = 79%.

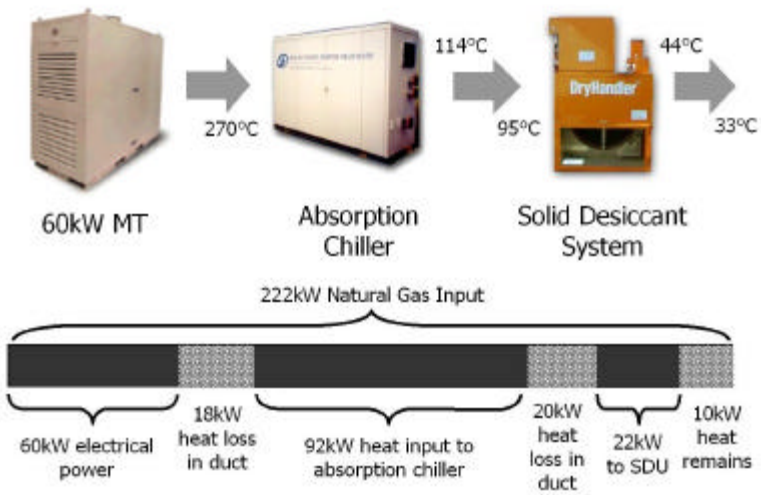

Figure 9. The temperature profile and energy distribution of exhaust gas stream in CHP System on a sample day July 29, 2002

CONCLUSION

The changes made to the microturbine based CHP system for the 2002 cooling season improved most aspects of the performance of the system, improving the electrical efficiency of the generator and reducing the parasitic electric power consumption in the system. As the absorption chiller was operating with off-design exhaust conditions its thermal COP was slightly lower, at 0.65 than the 0.7 measured during the 2001 cooling season. The system demonstrated stable operation and while energy savings is not a parameter that can be measured directly from instrumentation, the system improved on the energy consumption of the traditional grid electric based alternative. Future generations of CHP systems can improve on the performance of this system by using double effect chillers, reducing the parasitic power of the cooling components, reducing the exhaust heat losses, matching parameters between components, and utilizing future generations of microturbines with higher electrical efficiencies.

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