

Numerical and Experimental Study of 120 °C Heat Pump Using a Scroll Compressor



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Abstract Numerous industrial process require hot air or hot water at temperatures below 120 °C and diesel or LPG-fired boilers are often used to generate it. The use of a high-temperature heat pump can help reduce emissions and lower the CO₂ footprint by eliminating these fired sources. This study reports on the development and continuous operation of a high temperature heat pump with a heating capacity of 60 kW that generates hot water at 120 °C at a heating COP of 2.27. Using this hot water, hot air at 100 °C is generated using a heat exchanger. The high temperature heat pump increased carbon emissions by 20.3% compared to diesel, based on India's emission factor of 790 gCO₂/kWh. However, considering the emission factor of the European Union, the high temperature heat pump has the potential to reduce carbon emissions by 57.8%.

Keywords High temperature heat pumps · Decarbonization · Hot air dryer · Energy conservation

1 Introduction

In recent years, the issue of energy conservation and environmental concerns has received a lot of attention, leading to the implementation of various government policies aimed at lowering carbon emissions. With many countries committing to achieving carbon emission reduction targets at international events such as COP26 [1], industries are also exploring technologies to reduce their energy consumption and carbon footprint.

Process heating, which primarily uses hot water, hot air and steam, is a common energy requirement in many industrial processes and is typically met by the use of fossil-fired boilers [2]. In India, the industrial sector consumes 55.85% of total primary energy, with 16–20% used for process heating at temperatures ranging from

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50 to 250 °C [3]. The use of alternative technologies to replace fossil-fueled boilers for process utility could result in significant reductions in carbon emissions. High-temperature heat pumps (HTHP) have emerged as a promising technology in this regard, receiving significant research attention over the last decade, with efforts to achieve operating temperatures of up to 160 °C. Ongoing research aims to increase operating temperatures even further by utilising different refrigerants and HTHP architectures.

The choice of a suitable refrigerant is an important aspect of HTHP design. For optimal performance, it is preferable for the critical temperature of the refrigerant to be slightly higher than the desired operating temperature range. The potential refrigerants for HTHP applications are listed in Table 1.

Many studies have been conducted in recent years on HTHP, particularly on the use of natural refrigerants such as CO₂ and hydrocarbons, as well as synthetic refrigerants such as R245fa, R1233zd(E), R1224yd(Z), R1234ze(E), R1234ze(Z), R717, R365mfc, R1336mzz(Z), and R600. Fukuda et al. [5] investigated the thermodynamic, numerical, and experimental properties of R1234ze(E) and R1234ze(Z) for HTHP and discovered that the COP was maximised when the operating temperature was 20 K below the critical temperature. The simulations revealed that R1234ze(E) had lower pressure drop losses at higher temperatures of 105 and 125 °C, resulting in a higher COP. Kondu and Koyama [6] evaluated the thermodynamic properties of R717, R365mfc, R1234ze(E), and R1234ze(Z) using four different cycle configurations and sink temperatures of 160 °C and source temperatures of 80 °C. Among the refrigerants studied, the Cascade cycle of R1234ze(Z) and R365mfc had the highest COP. Nilsson et al. [7] created an HTHP by modifying a reciprocating compressor to use R1336mzz(Z) as the working fluid, achieving a heating COP of 2.5 at 60 °C source temperature and 120 °C sink temperature.

Studies have demonstrated that when utilized appropriately, natural refrigerants have exhibited comparable or superior performance compared to synthetic refrigerants. White et al. [8] built a transcritical CO₂ heat pump with a heating COP of 2.46

Table 1 Properties of HTHP refrigerants [2, 4]

ASHRAE number	Boiling point (°C)	Mass composition (%)	Critical temperature (°C)	Critical pressure (bar)	GWP 100 yr	ODP	Safety group
R1234ze(E)	-18.97	Pure fluid	109.36	36.34	6	0	A2L
R1336mzz(Z)	33.45	Pure fluid	171.35	29	2	0	A1
R1224yd(Z)	14.62	Pure fluid	155.54	33.4	1	0.00012	A1
R1233zd(E)	18.26	Pure fluid	166.45	36.2	1	0.00034	A1
R245fa	15.05	Pure fluid	153.86	36.5	858	0	B1
R600a	-11.75	Pure fluid	134.66	36.29	4	0	A3
R600	-0.49	Pure fluid	151.98	37.96	4	0	A3
R1234ze(Z)	9.728	Pure fluid	150.12	35.31	6	0	A2L
R365mfc	40.193	Pure fluid	186.85	32.66	782	0	A2

and hot water temperature of 120 °C. Stavset et al. [9] studied HTHP using hydrocarbons at temperatures up to 115 °C, whereas Bamigbetan [10] studied HTHP using ammonia, hydrocarbons, and their mixtures in a simple cycle, double stage with intercooler, and cascade cycle. The study found that the cascade cycle of hydrocarbon mixtures showed an 11.8% increase in COP compared to pure fluids at a sink temperature of 110 °C.

The literature on scroll compressor based HTHP is scarce, with only a handful of articles exploring laboratory-based studies. Despite its potential, there has been a lack of field-level application of HTHP. This study aims to address this gap by presenting the development of an HTHP using a scroll compressor to generate hot water utilizing condensate water that is available in few industries and also to generate hot air at a temperature of 100 °C using the developed HTHP, which has many industrial applications. The choice of R245fa as the refrigerant for this research project was motivated by its well-established performance in high-temperature cycles, as evidenced by previous studies. By opting for R245fa, the intention is to minimize result variability and conduct a comprehensive assessment of the HTHP.

2 System Simulation

The heat pump system uses vapour compression technology with a scroll compressor that can withstand temperatures of up to 150 °C. Brazed-type plate heat exchangers are used in the evaporator, economizer, and condenser. The economizer is used to heat the vapour injection line as it travels through the main line. To reduce the pressure from the condenser to the intermediate vapour injection line pressure, an expansion valve is used. The temperatures of the source and sink water are 60 °C and 120 °C, respectively, DWSIM [11], an open-source sequential modular steady-state simulator that produces results comparable to commercial software [12], was used to simulate the system. The simulation was run under steady-state conditions, with adiabatic pipes and zero pressure drop, and the heat exchangers were assumed to be 100% efficient with no fouling. The Peng Robinson equation of state was used to calculate the thermodynamic properties of the refrigerant. Figure 1 depicts the DWSIM system simulation model.

3 Pilot Study

Due to the continuous availability of waste heat, a pilot-scale experiment was conducted at a sugar processing facility in Hariawan, Uttar Pradesh, India. The source for the HTHP was hot water condensate at 60 °C. The HTHP was tested for 24 h of uninterrupted continuous operation. Figure 2 depicts the commissioning of HTHP.

The experimental setup was built using the numerical model's specifications. With the exception of the compressor, which was a pre-market sample, all components

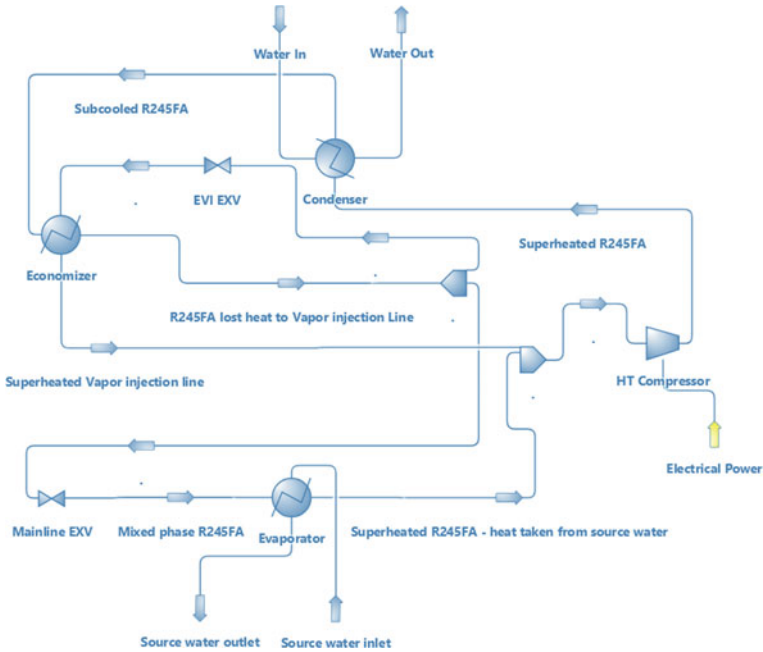


Fig. 1 Theoretical model of HTHP in DWSIM



Fig. 2 Pilot study experimental setup

were commercially available (provided as an application engineering prototype to facilitate faster technology penetration into the market). The hermetically sealed scroll compressor with a displacement of 444.5 cc/rev was used. Scroll compressors are known for its high efficiency, less number of moving parts, smooth operation and enhanced reliability. The compressor has a two pole three phase induction motor. The maximum allowable winding temperature is 135 °C. The maximum continuous current (MCC) and locked rotor amperage (LRA) are 62.3 A and 290 A, respectively. The maximum allowable discharge pressure is 25 barg. The manufacturer suggests

using RFL68FA oil to ensure the long life of the compressor. This specific lubricant is compatible with R245fa refrigerant, guaranteeing proper lubrication while minimizing the risk of any negative interaction between the lubricant and refrigerant. By using RFL68FA, the compressor receives the necessary protection against corrosion, oxidation, and other forms of degradation, thereby enhancing its overall durability and reliability. SWEP heat exchangers with 60 plates were chosen for the condenser and evaporator. The circulating water pump in the sink was linked in a closed-loop configuration. The circulating water pump was chosen for its high head and special seals that can withstand temperatures of up to 120 °C. To ensure the safety of the components, the HTHP was outfitted with high and low-pressure switches. The PLC was programmed with high and low pressure limits that would automatically shut down the machine if they were exceeded. To prevent pump dry runs, flow switches were installed in both the source and sink water loops. To protect the compressor and other electrical components, the machine was also outfitted with protection devices such as a phase reversal protection device, phase loss protection device, 3-phase protection device, over voltage protection device, and high current protection device. An internal protection switch was also installed to prevent compressor winding failure.

Preliminary experiments determined that the optimum charge for the HTHP was 15 kg. The circulating water volume flow rates of the condenser and evaporator ranged from 6 m³/h to 8 m³/h and 2 m³/h to 5 m³/h, respectively. The source temperature, which varied between 50 and 60 °C depending on waste heat in the industry, was monitored to ensure it did not exceed 60 °C for proper operation within the compressor envelope.

Six calibrated thermistors, four PT-100 temperature sensors, and six pressure transmitters were used to measure temperature and pressure at various state points. The uncertainty of the temperature and pressure sensors is ± 1.1 K and $\pm 0.5\%$ FS, respectively. Three endress and hauser vortex mass flow metres were used to measure the flow rates of source and condenser water, with an uncertainty of 0.5% of the measured value. An energy metre was used to calculate total electricity consumption. Data from these measuring devices was collected by a PLC (IPG215D) and sent to the cloud via a gateway using RS485 communication and a cellular network, with data being logged and pushed every 5 s.

4 Performance Parameters

The two important performance parameters studied in this paper are compressor efficiency and coefficient of performance (COP). The research focuses on two compressor efficiency measures: isentropic efficiency and total compressor efficiency. The ratio of the theoretical enthalpy change during an ideal, isentropic compression process to the actual enthalpy change during the compression process is defined as isentropic efficiency. It is calculated as the ratio of isentropic to actual

enthalpy change. Total compressor efficiency is defined as the ratio of the theoretical work input required for an ideal, isentropic compression process to the actual electrical power input required for the compression process. It is defined as the ratio of isentropic work input to actual electrical power input. Equations 1 and 2 give the mathematical expressions for isentropic efficiency and total compressor efficiency, respectively.

$$\eta_{isen} = \frac{h_{2,isen} - h_1}{h_2 - h_1} \quad (1)$$

$$\eta_{total} = \frac{m_{ref}(h_{2,isen} - h_1)}{E_{comp}} \quad (2)$$

The mass flow rate of the refrigerant is calculated using the manufacturer's AHRI polynomial [2]. REFPROP 10 [4] was used to calculate the enthalpies for the measured temperatures and pressures. The heating COP is calculated by dividing the amount of heat delivered by the amount of electrical work done. The system COP, on the other hand, is calculated by dividing the total electrical work by the sum of heat delivered in the condenser and heat absorbed in the evaporator. The heating and system COP is given by Eq. 5 and 6, respectively, which uses Eq. 4 and 3, respectively.

$$Q_{evap,ref} = m_{ref}(h_1 - h_4) \quad (3)$$

$$Q_{cond,ref} = m_{ref}(h_2 - h_3) \quad (4)$$

$$COP_{heating} = \frac{Q_{cond,ref}}{E_{total}} \quad (5)$$

$$COP_{sys} = \frac{Q_{cond,ref} + Q_{evap,ref}}{E_{total}} \quad (6)$$

5 Hot Air Generation

Numerous industries require hot air for drying applications. An experimental setup was built to generate hot air and was supplied to a chamber having a dimension of $1.31 \times 1.53 \times 1.5$ m at a temperature of around 100°C . Hot water at 120°C was first generated using the developed HTHP. This hot water was passed through a fin and tube heat exchanger which transferred the heat from water to air. The heat exchanger has 40 number of tubes with 6 rows. The developed HTHP requires a source temperature of 60°C . Hence, another low temperature heat pump (LTHP) of



Fig. 3 Experimental setup of hot air generation using HTHP

40 kW of heating capacity with a heating COP of 4.3 was used to generate water at 60 °C. R417A refrigerant was used in the LTHP. An air blower at a rated flow rate of 1.1 m³/s was installed along the duct of cross section 0.4 m × 0.4 m. Figure 3 shows the experimental setup for hot air generation. Calibrated k type thermocouple was installed in the chamber which was measured and recorded using a KEYSIGHT data acquisition system (DAQ 970A). A detailed experiments and analysis is yet to be performed to obtain the drying characteristics of various materials and the energy, exergy, economic and environmental benefits of using the HTHP in hot air generation.

6 Results and Discussion

The performance of the compressor is assessed by the compressor discharge temperature. The discharge temperature affects the solubility and viscosity of the lubricant. The manufacturer has set a maximum limit of 150 °C and 60 °C for the maximum discharge and suction temperature, respectively. Suction temperature is a crucial parameter that affects the compressor's performance. RFL68FA oil was used in the compressor, and its average temperature remained below 100 °C. Although the oil's color changed slightly after a few months of operation, the compressor's performance was not affected. However, detailed analysis of the oil must be performed in the future. The experiment's average suction and discharge temperatures were 48.14 °C and 124.4 °C, respectively, and the simulations' temperatures were 50.17 and 131.1 °C. Both were within the safe limits specified by the compressor manufacturer. The maximum hot water outlet temperature observed in the experiments was 120.83 °C. Pressure sensors were installed in both the low and high-pressure sides of the compressors to measure the pressure. For added security, low and high-pressure switches were installed. The high-pressure values, on the other hand, were found to be sensitive to small fluctuations in the flow rate of the source and sink water, and occasionally exceeded the safety limits, causing the compressor to trip. As a result, maintaining a consistent flow rate of the source and sink is critical for ensuring the

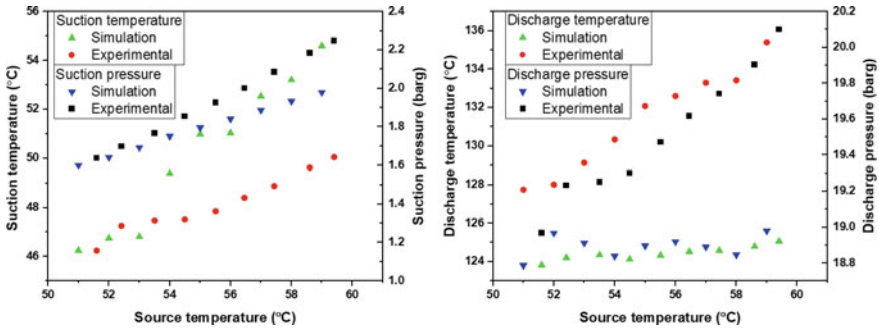


Fig. 4 shows the suction and discharge temperature and pressure values

heat pump’s safe and efficient operation. Figure 4 shows the suction and discharge temperature and pressure values. The experimental and simulation values were found to be consistent.

There is some variation in the pressure values compared to the simulation due to the position of the pressure sensor and pressure drop in downstream components. The maximum discharge pressure measured during operation was 20 barg, which was within the manufacturer’s maximum limit of 25 barg. During the experiments, the maximum pressure ratio recorded was 11.74, which was less than the manufacturer’s specified maximum of 17. The consistency of experimental and simulation pressure and temperature values suggests that simulations can accurately predict compressor performance over a wide range of operating conditions beyond the limits of the experimental setup.

Based on the experimental results of pressure and temperature, the total compressor efficiency and isentropic efficiency have been calculated. On average, the total compressor efficiency was found to be 63%, while the isentropic efficiency was 71.7%. The difference between these values is due to losses in the electric motor, frequency converter, transmission, and heat. Figure 5 illustrates the compressor efficiencies heating COP and compressor power for various source temperature. Notably, the high-temperature premarket scroll compressor efficiencies were found to be comparable to those of existing industrial standard heat pump compressors. The heating COP of the heat pump is influenced by the source temperature. When the source temperature is lowered, the temperature lift increases, causing the compressor to consume more power. Consequently, the heating COP of the heat pump decreases. Note that the study did not take into account the power consumption of the circulating water pumps, as the same pumps used in the existing process were utilized for the pilot demonstration. The focus of the study was on the performance of the compressor. During the pilot testing, the HTHP was tested for 24 h of continuous operation using the available condensate water as the source. Figure 6 shows the variation of the source and sink temperatures during the continuous operation.

During the 24-h continuous operation of HTHP the available condensate water was used as the source. The system’s performance was stable, with an average

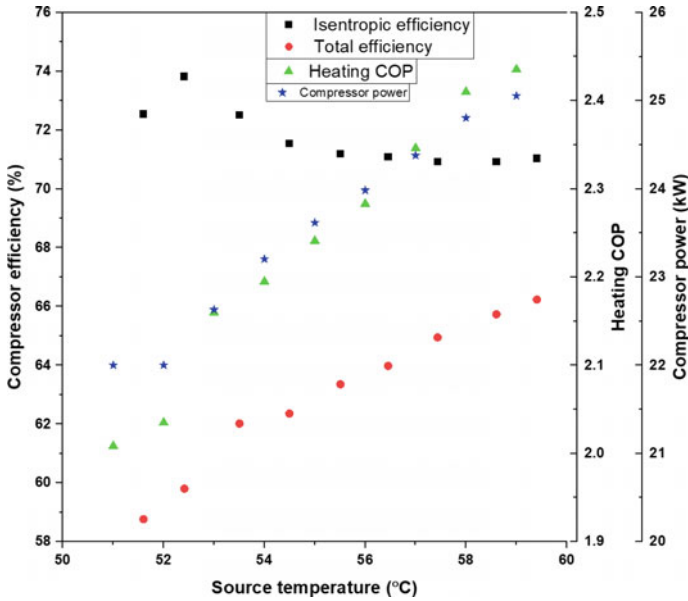
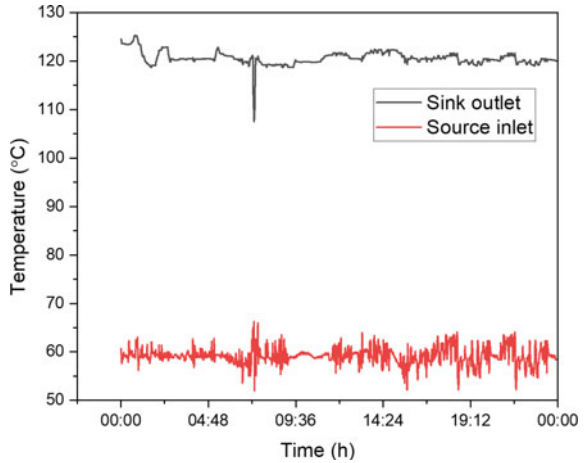


Fig. 5 Experimental results of compressor efficiencies, heating COP and compressor power

Fig. 6 Variation of source and sink temperature during continuous operation

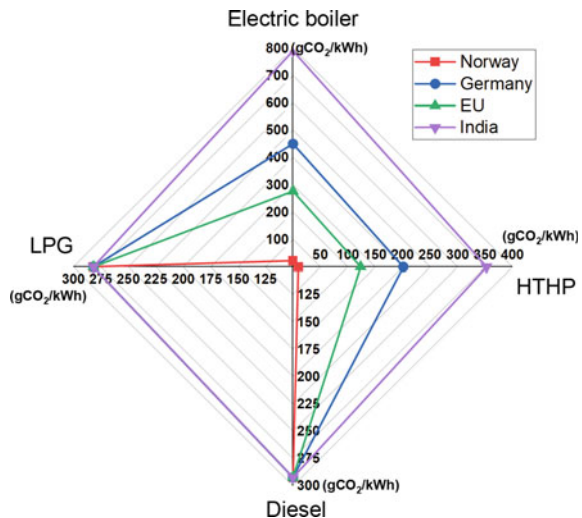


heating COP of 2.27 and an average compressor power consumption of 26.8 kW was recorded. However, there were some fluctuations in the system’s performance due to interruptions in the supply of condensate water at the desired temperature.

6.1 Emission Study

Heat pumps are recognised as a potential decarbonization solution, but their effectiveness is dependent on the emission factor of the country where they are installed or whether the location of installation has a green source of electricity production. Regardless, electrification is the first step towards decarbonization, and heat pumps can be a viable option. To compare CO₂ emissions, HTHP with waste heat of 60 °C as source were compared to 85% efficient fossil-fired boilers. CO₂ emissions were discovered to be dependent on the country’s emission factor. The European Union, Germany, Norway, and India were found to have emission factors of 276 gCO₂/kWh, 450 gCO₂/kWh, 22 gCO₂/kWh, and 790 gCO₂/kWh, respectively [13, 14]. Figure 7 depicts CO₂ emissions per kW of heating capacity for various countries and fuels. In all countries studied, the HTHP emitted 55.3% less CO₂ than electric boilers. Due to the current high emission factor, replacing LPG would increase carbon emissions by 49% in India. The HTHP was found to emit 55.3% less CO₂ than electric boilers in all countries studied. In countries like Norway, where electricity generation is mainly from hydropower, HTHP would reduce emissions by at least 95, 14, and 47% in Norway, Germany, and the European Union (EU), respectively, by replacing LPG. However, in India, replacing LPG with HTHP would increase carbon emissions by 49% due to the high emission factor in the country. It is important to note that “greening the grid” is a more feasible solution compared to finding alternatives for combustion sources. Despite the slightly higher emissions from HTHP in India, it still has potential for the future as the country moves towards cleaner sources of electricity. Additionally, the benefits of HTHP, such as reduced energy consumption and lower operating costs, make it an attractive option for many applications.

Fig. 7 gCO₂/kWh for various fuels



7 Conclusion

Finally, this study highlights the design, numerical and experimental evaluation of a 60 kW HTHP capable of producing hot water at 120 or hot air at 100 °C. The pilot scale experiments and numerical studies yielded similar results, with an average heating COP of 2.27 when using waste condensate water at 60 °C. Furthermore, the carbon emissions of any technology are heavily influenced by the emission factor of the country or location of installation. Due to its high emission factor of 790 gCO₂/kWh, the HTHP was found to emit 20.3% more carbon than diesel in India, while it has the potential to reduce carbon emissions by 57.8% when compared to the EU emission factor. When calculating the carbon emissions of HTHPs, it is critical to consider the emission factor of the country or installation location. Future research can look into the potential of HTHP for applications such as steam generation and drying. The drying properties of various products must be investigated because they have a significant potential for energy conservation and decarbonization in industries. Furthermore, additional experiments with other potential natural and synthetic refrigerants can be conducted to investigate the possibilities of performance improvement, which will increase the economic and environmental benefits of HTHP technology.

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Nomenclature

<i>Comp</i>	Compressor
<i>Cond</i>	Condenser
<i>COP</i>	Coefficient of performance
<i>E</i>	Electrical power (kW)
EU	European Union
<i>Evap</i>	Evaporator
FS	Full scale
<i>h</i>	Specific enthalpy (kJ/kg)
<i>isen</i>	Isentropic
<i>m</i>	Mass flow rate (kg/s)
<i>Q</i>	Rate of heat transfer (kW)
<i>ref</i>	Refrigerant
<i>sys</i>	System

- 1 Compressor suction
- 2 Compressor discharge

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